Numerical Investigation of Continuous Damping of The Semi-Active Suspension System for Passenger Car

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Abstract. The suspension of the car is considered an important element in the vehicle. The primary function of the suspension system is to isolate the vehicle structure from shocks and vibration due to irregularities of the road surface. There are two main objectives need to be satisfied which are: ride comfort and road handling. Ride comfort is inversely proportional to the absolute acceleration of the vehicle body, while the road handling is linked to the relative displacement between the vehicle body and the tires. This paper presented an attempted to enhance the performance of the shock absorber by developing a model of continuously variable damping (CVD). To evaluate the effect of the developed semi-active shock absorber on the dynamic behaviour of the vehicle, the model was analyzed and compared with the passive and On/Off sky-hook control strategy in the quarter car using two different types of road (random excitation, bumpy) as input to the quarter car model. Force hysteresis loop with different sets of orifice diameter was generated. The result indicates the CVD shows a reduction in both body acceleration and vertical displacement contrasting with passive and On/Off sky-hook 73.4% and 53.8% respectively and also the selling time by 79% and 59% for a bumpy road. This considered an improvement toward the ride comfort and vehicle stability. The simulated results for the quarter car model are shows similar trends and within range when compared with reference research paper.

Keywords: Vehicle Suspension System; CVD; controllable; absorbers; orifice.

1. Introduction
The concentration of researchers and automobile manufactures on suspension system back to decades of time as it improves the car stability, handling characteristics and provide more comfort to both the car and the passenger. The suspension system considered one of the most important parts of the vehicle toward the dynamic behaviour [1-5]. The history drawback that the passenger cars are the main land transportation ever used by mankind. Base on the controllability, the suspension can be divided into three types: passive or conventional, semi-active and active suspension [3]. The passive or conventional suspension comes with a fixed damping ratio by the manufactures where the users cannot change or adjust its dynamics properties, this will draw up a conflict between ride handling and comfort [4-8]. Active suspension is very convenient in term of performance toward the enhancement of vehicle dynamic behaviour and mostly used in luxury vehicles, despite that, it requires high power consumption, control units and high cost due to the complexity, and required hardware and software [7-8]. Lastly, the
semi-active suspension system dispenses better performance once compared with the conventional, while with the active in term of cost reduction and simplicity [6-7].

In general, the suspension system is a combination of three parts: springs, shock absorbers and linkages, each one of them have its own contribution to the stability of the car [9]. Shock absorber considered an important element of the suspension system, function as energy dissipation and vertical displacement reduction, it uses mostly by land transportation. The shock absorber evaluated by hysteretic diagram also called force-velocity diagram. The shock absorber can be categorized into two based on their way of dissipating energy: hydraulic and dry friction. Hydraulic shock absorbers are commonly used in current automobiles [10].

Tremendous researchers give their effort toward the development of a numerical model that can observe the damping behavior. Identification methods can be categorized into: parametric and non-parametric also called physical perspective. Parametric modeling regards the output (as for example, the typical force-displacement plot from which the hysteresis phenomenon is evident) more or less like a black box [11]. Despite the fact that this technique is powered by the capacity of machine learning in perceiving conditions among the inputs and outputs, there are practical and theoretical downsides. There is another was proposed called gray box, artificial neural network technique was adopted to overcome the hysteresis in the shock absorber, the hydraulic and friction forces were considered within the parametric model [12]. Parameter identification of non-linear hysteretic model of shock absorber was discussed, the model considered three passes of the fluid flow in monotube charged with gas: flow through the piston orifice, bleed orifice and piston leakage due to gap between the piston and cylinder, beside that the stiffness properties of stock shims and the moving piston that separates the gas the oil in the chamber were included [9] [13]. In a few decades back, Lang made first attempts to formulate the characteristic of shock absorber numerically, the complexity of evaluation of the physical model was occurred due to the extremely number of parameters reached up to 80 [14]. A few years later, Morman used Lang’s model to validate with experimental testing [15]. Later on, a great achievement was occurred in simplifying the physical model by reducing the number of parameters [16]. Chavan carried out a study of the dynamic response of hydraulic damper and its influence by force-displacement hysteresis, and road input to ride comfort and vehicle stability [5].

2. Variable damping and control strategies

Previously, semi-active shock absorbers were manually adjustable by opening or closing a bypass valve. Eventually, electronically adjustable dampers were introduced by researches such as the hydraulic damper [17], the magneto-rheological fluid damper (MR) [2] and the electro-rheological fluid damper (ER) [18]. Mostly the attention is given to the hydraulic damper, due to the simple structure, lower cost, and reliability in a term for performance [9]. The appropriate control technique is delineated as the one that reduces the value of the body vertical displacement, tyre vertical displacement, and the vehicle body acceleration. typically, these quantities are depending on the mass of the sprung and unsprung masses. Different types of control hypothesis have been implemented by many researchers in determining the most appropriate control strategy that can reduce the elevation factors [19].

A few decades back Karnopp et al. as the first to initiate the idea of the semi-active suspension system. An intelligent controller has been utilized to produce an active actuating force and, he introduced sky-hook strategy which is about fixing the adjustable damper between the car body and imagining hook and ground -hook [8] [19]. Swevers et al. have developed a parameterized, model-free control structure for a passenger car equipped with an electro-hydraulic semi-active suspension system [20]. Valašek et al and Ahmadian et al. examined control algorithm called hybrid control that homogenized the advantages of sky-hook and ground-hook. Furthermore, an integrated preference of both the sky-hook (SH) and the acceleration was driven damper control (ADD), in their respective ranges of the best behaviour. As a result, SH-ADD can provide the quasi-optimal performance of ride qualities [21-22]. A continuously variable damper (CVD) was presented by some researchers, based on skyhook and ground-hook control strategy, damper force in the semi-active case must be equal to the damper force in the Skyhook case [23-24]. Used the advantage of mechanical power without hardware or energy consumption to construct a set of four valves that can realize the performance semi-active damper from the passive damper base on the velocity and displacement [25]. Zhao et al. Adapted reversible technique
to identifying the roughness of road by using adjustable semi-active suspension, which can lead cost reduction of the production [26].

In addition to the above controlling algorithms and techniques, there are also a large number of another control techniques been originated for the seek of enhancing the ride quality. For example, clipped optimal control [27-28], sliding mode control [29], LQR/LQG [30-31] and so on.

In this paper, an attempt is made to develop a mathematical model to realize semi-continuous variable damping (CVD) characteristics from commercially available monotube shock absorber. This model inspired by the model presented in [4-5] [14]. Therefore, the force-displacement and force-velocity characteristics curves are determined and presented, and furthermore to apprehend the variable damping properties a time response of quarter car simulation is carried out and compared with passive and on/Off sky-hook control to prove the effectiveness.

3. Damper description

The damper consists of several main parts as illustrated in Table 1. The tube of the damper houses the standard internal piston. Once assembled, the tube is divided into three chambers: gas, rebound, and compression as the typical monotube damper, shown in figure 1. The compression and rebound chambers are connected to each other through a restricted piston orifice and a bypass valve which is controlled by a stepper motor. When in compression, the fluid flows through two flow paths, through the internal piston and bypass valve. The damping through the internal piston is dictated by the area of the orifices and the stiffness of the shim stacks whereas the flow through the bypass valve depends upon the displacement adjustable pin that is controlled by application of voltage. This takes place the same way but in the opposite direction of the rebound stroke. An electronically controlled semi-active damper mechanism to be implemented. The size of the single semi-active damper was kept as the universal damper size to utilize the advantage of the use of the standard parts that to obtain equivalent results when tested manually and with a controller. Table 1 presents the parts of the damper.

![Figure 1. The controllable monotube hydraulic shock absorber](image)

### Table 1. Shock absorber parts description.

| No | Description                      | No | Description                  | No | Description              |
|----|----------------------------------|----|------------------------------|----|--------------------------|
| 1  | Stepper Motor                    | 9  | Compression Chamber          | 17 | Nut                      |
| 2  | Stepper Motor Mounting           | 10 | Dust Cover                   | 18 | Shim Stock               |
| 3  | Piston Rod                       | 11 | Spring                      | 19 | Piston and Valve         |
| 4  | Outer Cylinder                   | 12 | Cap Bolt                    | 20 | Road Guide               |
| 5  | Rebound Chamber                  | 13 | Upper Mount                 | 21 | Inner Cylinder Cover     |
| 6  | Bleed Orifice                    | 14 | Gas Chamber                 | 22 | Adjustable Rod           |
| 7  | Adjustable Spring Holder         | 15 | Floating Piston             | 23 | Stepper Motor Shaft      |
| 8  | Supporting Cover of outer Cylinder | 16 | Oil Cylinder               |    |                          |

4. Fluid flow modelling

The damping force is generated due to the resistance of hydraulic fluid flow between the different side of the shock absorber through a set of paths. The major assumption for summation of flow is that the
damper oil is incompressible and therefore has a constant density. This assumption allows consideration of volumetric rather than mass flow rates [4]. This is expressed in equation 1 and shown in figure 2 is the total volumetric flow rate of the damper, $Q$ in m$^2$/sec.

$$Q = Q_v + Q_b + Q_{lp}$$  \hspace{1cm} (1)

where $Q$ is the overall fluid flow, $Q_v$ flow rate through the valves, $Q_b$ is the fluid flow through the bleed orifice and $Q_{lp}$ fluid flow through the piston boundary. Where boundary can be seen in figure 2. Equivalent flow across boundary due to rod insertion, overall fluid flow, $Q$, is related to velocity regardless of the compression or rebound. This is shown in equation 2.

$$Q = (A_c - A_r) \dot{x}$$  \hspace{1cm} (2)

where, $A_c$ is the area of the piston on the compression side, while $A_r$ is the area of the piston on the rebound side and $\dot{x}$ is the piston velocity.

\begin{figure}
\centering
\includegraphics[width=\textwidth]{damper_diagram}
\caption{The total volumetric flow rate of the damper diagram.}
\end{figure}

The pressure difference, $\Delta p = p_c - p_r$, between both side of chambers is considered to establish the partial flow rate for each part. A Bernoulli’s equation can be used to model unsteady flow through a passage of area, $A$. It has the form:

$$Q = AC_D \left( \frac{2\Delta p}{\rho} \right)^{\frac{1}{2}}$$  \hspace{1cm} (3)

where the $C_D$ is a coefficient of discharge at steady state and $\rho$ is the density. $C_D$ is a dimensionless parameter including acceleration number. Reynolds number, Cauchy number, and thickness to length ratio have experimentally modified this term by defining a dynamic discharge coefficient, $C_D$ [14].

$$C_D = f \left\{ \frac{\alpha l}{v^2}, \frac{\mu}{\rho vl}, \beta v^2, \frac{s}{l} \right\}$$  \hspace{1cm} (4)

where, the Reynolds number, $Re = \frac{\mu l}{\rho v}$ and Cauchy number, $Ca = \beta v^2$, $\alpha$ is a piston acceleration, $l$ is a piston length, $\mu$ is a fluid viscosity, $\beta$ is a viscous friction coefficient, $v$ is a flow velocity and $s$ is a thickness. The leakage of oil between the piston seal and the cylinder results in a certain amount of fluid flow. Lang modelled this flow using laminar flow through parallel plates [14].

$$Q_{lp} = \left( \frac{(p_c - p_r)b^3}{12\mu l} + \frac{b}{2} \dot{x} \right) \pi D_p$$  \hspace{1cm} (5)
where, \( b \) is the height of the piston, \( D_p \) is the diameter of the piston, \( \mu \) is a fluid viscosity, \( l \) is a piston length, \( p_c \) and \( p_r \) are compressions and rebound pressure respectively.

5. Variable damper system concept

5.1. Semi-active valve flow (\( C_b \))

The valve is composed of a circular ball detent that allows a change in area from 0 to 5 mm\(^2\). The valve is to be modeled as a bleed hole that allowed a constant amount of fluid at any moment. Equation 6 is formulated to find the flow through the adjustable bleed orifice in the piston.

\[
Q_b = A_b(\theta) C_b \left( \frac{2(p_c - p_r)}{\rho} \right)^{\frac{1}{2}}
\]  

(6)

Where, \( A_b \), Area of the bleed, was determined through measurement of the damper bleed orifice. Area of the variable bleed orifice as a function of displacement angle can be calculated from equation 7 [27].

\[
A_b(\theta) = 2 \left[ \tan \frac{\theta}{2} \left( \left( r^2 - r^2 \tan \frac{\theta}{2} \right)^{\frac{1}{2}} - \frac{r}{2} \cos^{-1} \left( \frac{(r^2 - r^2 \tan \frac{\theta}{2})^{\frac{1}{2}}}{r} \right) \right) \right] \frac{r}{2} \sin \left( \sin^{-1}\frac{r - \theta}{2} \right)
\]

(7)

where, \( R \) is the radius of a rotary pin, \( r \) is the radius of the bleed orifice and \( \theta \) is the displacement angle of a rotary pin can be calculated according to equation 8.

\[
\theta = N_s \times 1.8^\circ
\]

(8)

where, \( N_s \) is the number of steps generated by the stepper motor

6. The mathematical model of quarter car suspension

Quarter car simulation is considered an effective method to demonstrate the influence of damping force on vehicle dynamic behavior due to input waves from irregularity road surface. This simulation can give us chance to evaluate the effect of the controllable bleed orifice in the damper and the vertical acceleration of the vehicle body by the shock absorber. The simulation was done using the SIMULINK toolbox of MATLAB R2015a. The governing equation for a quarter-car model is expressed in equation 9 according to figure 3.

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**Figure 3.** Continuos variable semi active damper.
\[
\begin{align*}
M_s \ddot{X}_s + C_s(t)(\dot{X}_s - \dot{X}_u) + K_s(X_s - X_u) &= 0 \\
M_u \ddot{X}_u + K_t(X_u - X_r) + C_s(t)(\dot{X}_u - \dot{X}_s) + K_s(X_u - X_s) &= 0
\end{align*}
\]

(9)

The sprung mass can be defined as \( M_s \) and unsprung mass is represented by \( M_u \). This includes any mass that moves with the wheel and tire. For an independent suspension, a portion of the suspension links and half shafts should be included. \( K_s \) and \( C_s(t) \) are the spring rate and damper coefficient of the suspension, and \( K_u \) is the tire spring properties. The displacement of the sprung and unsprung masses and the road excitation are \( X_r, X_u, \) and \( X_r \), respectively. Some assumptions are considered in this model such as neglecting the damping coefficient of the tire and constant spring rate. Vehicle parameters used in this simulation are presented in table 2 according to the exciting literature [33].

Table 2 Properties of quarter car model [33].

| Parameter                  | Symbol | Value     | Unit     |
|----------------------------|--------|-----------|----------|
| Sprung mass                | \( m_s \) | 280       | kg       |
| Unsprung mass              | \( m_u \) | 35        | kg       |
| Shock absorber spring constant | \( K_s \) | 17600     | N/m      |
| Shock absorber damping coefficient | \( C_s \) | 892-6203  | Ns/m     |
| Tire stiffness             | \( K_u \) | 190,000   | KN/m     |

7. Design of Control algorithm

In order to improve the feeling of riding for irregular variation of road surface, an input for a road surface as a disturbance is set and the control algorithm is designed as per the requirements. While designing control algorithm, an ideal skyhook and continuous variable damping (CVD) have been correlated and a following control policy is defined.

\[
F_{\text{sky}} = C_{\text{sky}} \dot{X}_s
\]

(10)

where, \( F_{\text{sky}} \) is skyhook force next is to determine if the semi-active damper is able to provide the same force. If the sprung and unsprung masses are separating, then the semi-active damper is in tension. Thus, the force applied to the sprung mass is:

\[
F_{\text{damper}} = C_s(t)(\dot{X}_s - \dot{X}_u)
\]

(11)

where, \( F_{\text{damper}} \) is the force applied to the sprung mass. Since we are able to generate a force in the proper direction, the only requirement to match the skyhook suspension is,

\[
C_s(t) = C_{\text{sky}} \frac{\dot{X}_s}{(\dot{X}_s - \dot{X}_u)}
\]

(12)

The on-off skyhook control strategy switches between high and low damping coefficient in order to achieve body comfort specifications. This control law consists in changing the damping factor of the damper (its fluid flow, fluid viscosity, air resistance, etc) according to the body velocity and the suspension relative velocity by using a logical rule as stated in equation 13[8] [35].

\[
C_{\text{sky}} = \begin{cases} 
C_{\text{max}} & \dot{X}_s(\dot{X}_s - \dot{X}_u) \geq 0 \\
C_{\text{min}} & \dot{X}_s(\dot{X}_s - \dot{X}_u) < 0 
\end{cases}
\]

(13)
It is worth emphasizing that when the product of the two velocities is positive that the semi-active damping force is proportional to the velocity of the sprung mass. Otherwise, the semi-active damping force is at a minimum. Where, \( C_{\text{max}} \) and \( C_{\text{min}} \) are the maximum and minimum damping coefficients of a damper, respectively (and usually \( C_{\text{max}} = C_{\text{sky}} \)). Equation 13 implies that when the relative velocity across the suspension and the sprung mass absolute velocity have the same sign, a damping force proportional to \( C_{\text{max}} \) is desired. Otherwise, this control law deactivates the controlled damper when the body velocity and relative velocity have opposite sign.

8. Road profile generation

Two types of road disturbance are assumed as the input for the system. The road profile shown in Figure 5 random excitation based on road roughness and a single bump are frequently used by many researchers [1]. The road surfaces are classified into A-H according to the degree of roughness as described in International Organization for Standardization (ISO) 8608 [34] based on power spectral density (PSD) function. Basically, the A-D classes are considered to be paved roads where the vehicle can reach the higher speed and they have a minor degree of roughness. Table 3 presents the road roughness standard deviations for various types of roads.

The random road signal excitation of PSD [33] and the sinusoidal bump can be expressed as in equations 14 and 15 respectively.

\[
\dot{X}_r(t) = -\alpha VX_r(t) + \eta(t) \tag{14}
\]

\[
X_r(t) = \begin{cases} \frac{a}{2} \left( 1 - \cos \left( \frac{2\pi V t}{\lambda} \right) \right), & 1.0 \leq t \leq 1.25 \\ 0, & \text{else} \end{cases} \tag{15}
\]

The solution to the above equation for \( Z_0(t) \) is used as road input to quarter-car simulation as presented in Figure 5.

where, \( \alpha \) for the shaping filter and the ISO standard are shown in table 3, \( V \) is the car forward velocity on this paper 120 km/h is to be examined and \( \eta(t) \) white noise, \( a \) is the bump height, \( \lambda \) is the half wavelength of the sinusoidal road undulation.

| Road Category   | \( 10^3 \) m | \( G_0(n_0)(10^6 \text{ m}^3) \), \( G_0 = 1 \) | \( \alpha \) (rad/m) |
|-----------------|---------------|---------------------------------|-------------------|
| A (very good)   | 2             | 1                               | 0.127             |
| B (good)        | 4             | 4                               | 0.127             |
| C (average)     | 8             | 16                              | 0.127             |
| D (poor)        | 16            | 64                              | 0.127             |
| E (very poor)   | 32            | 256                             | 0.127             |

9. Graphic representation using Matlab library blocks

In Figure 6 is illustrated the graphical presentation of the semi-active damping based on equations 9-13. Solving was performed using Simulink library blocks computing. In order to analyze the behavior of the quarter car suspension system, parameters were used as input as shown in Table 2.
Figure 4. A segment of the road profile as input for quarter car model (a) input excitation signal, (b) bumpy road input.

Figure 5. Matlab representation of f physical quarter car (a) passive (b) Skyhook and CVD control strategies.

10. Simulation results and discussion
To investigate the effectiveness of the bleed orifice on the generated force, A mathematical modelling for semi-active damper was created and with different setting mode of bleed orifice diameters were examined. The diameter varies from 0-2.5 mm, when the setting is zero have the same response as the passive shock absorber. The damping force decreases with increases of orifice damper and vice versa, and this will affect the performance of the shock absorber by varying its softness and stiffness. A force-displacement and force-velocity plots are shown in figure 6.
Figure 6. Details of the predicted damper force from the sine wave excitation with various bleed orifice diameter, (a) Force-displacement curve and (b) force-velocity curve.

The proposed system along with skyhook logic and passive suspension system are simulated to obtain the body acceleration, vertical displacement and suspension deflection for a particular type of road condition. Figure 7(a) shows the sprung mass acceleration response of the passive, On/Off sky-hook and continuous variable damping (CVD) in time domain against the input signal stated in figure 4. CVD has a great decrement in the maximum level of the acceleration and has the best response. The response characteristics of CVD has accomplished a significant reduction in vertical acceleration of the vehicle when compares with both passive and sky-hook control which are measured to be 73.4% and 53.8% respectively. Also, the bumpy road the response of CVD has less overshoot and settling time by 79% and 59% as illustrated in figure 7(b). Therefore, it can be concluded that the ride comfort characteristics of CVD were improved compared with the ones of the passive shock absorber.

Figure 8(a) shows the analysis result of the vertical displacement semi-active damper model in the time domain. Figure 8(b) shows the analysis results of vertical displacement in bumpy road condition. Both these figures provide comparisons between passive, On/Off Sky-Hook and CVD. As shown in figure 8, the response characteristic of the CVD has achieved the lowest amplitude and settling time. CVD shows slightly improved characteristics of ride comfort compares passive shock absorber.

The simulation results also reflect that, the peak to peak suspension deflection of semi active suspension by CVD controller has been decreased tremendously with early dies out of 1.5 sec as illustrated in figure 9. Similarly, the reduction in peak-to-peak velocities and accelerations are also occurred in case of Sky-Hook suspension as compare to passive suspension of sprung mass. However, the benefits achieved in reducing peak-to-peak accelerations are rather low as compared to benefits observed in the peak-to-peak displacements and velocities.

Figure 7. Sprung mass acceleration response of the semi-active damper model: (a) with input excitation signal and (b) bumpy road as input.
Figure 8. Sprung mass displacement response of the semi-active damper model: (a) with input excitation signal and (b) bumpy road as input.

Figure 9. Suspension deflection of the semi-active damper model: (a) with input excitation signal and (b) bumpy road as input.

11. Conclusion
The research presented in this paper demonstrates a mathematical model of an adjustable semi-active damper. The fluid flow rate and the damping force of a semi-active shock absorber of an automotive system were theoretically estimated. The obtained result of the mathematical model was analyzed. It is shown that various damping force can be calculated by adjusting the opening area of the bleed orifice. To ensure the vehicle dynamics performance under the continues variable damping (CVD) semi-active damper, a quarter car model used for analysing. The simulation result of time response was compared with the response of passive shock absorber and the skyhook control strategy on both paved and bumpy road driving conditions. According to the simulation result can be observed the vertical acceleration and displacement are decreased, which inverse proportional with ride comfort. The reported result here is to contribute to an understanding of effect controllable orifice in providing a continuous variable damping (CVD) of the shock absorber.

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