THE EFFECT OF 'ON/OFF' CONTROL STRATEGIES ON RIDE DYNAMICS

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RINGKASAN : *Model satu perempat sebuah kereta digunakan untuk mengkaji kesan strategi kawalan 'BUKA/TUTUP' ke atas dinamik tunggangan. Strategi ini dikenakan ke atas peredam untuk memacu sistem ampalan semi-aktif. Perbandingan dibuat di antara sistem pasif dan semi-aktif. Anjakan nisbi maksimum dan hentakan maksimum yang dialami oleh jisim terpegas adalah direkodkan ketika menjalankan simuilasi.*

ABSTRACT : A quater car model is used to study the effect of 'ON/OFF' control strategies on ride dynamics. The strategies are applied to the damper in order to drive a semi-active suspension system. Comparisons are made between the passive and semi-active system. Maximum relative displacement and maximum jerk experienced by the sprung mass are recorded during the simulations.

KEYWORDS : Semi-active, suspension, jerk, damping, displacement, velocity, acceleration, automobile, comfort.

NOMENCLATURE

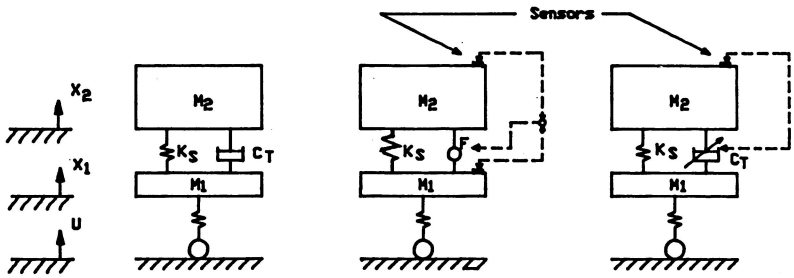
M_1	Unsprung mass of the whell and chasis
M_2	Sprung mass of the automobile body
X_1	Displacement of unsprung mass
X_2	Displacement of sprung mass
$\dot{X}_1$	Velocity of unsprung mass
$\dot{X}_2$	Velocity of sprung mass
$\ddot{X}_1$	Acceleration of unsprung mass
$\ddot{X}_2$	Acceleration of sprung mass
K_T	Tyre stiffness
K_S	Spring stiffness
C_T	Damping coefficient
U_o	Maximum amplitude for input displacement
ω_{ex}	Excitation frequency
ζ_2	Damping ratio
ω_{2n}	Natural frequency of the suspension system
Z_{max}	Maximum relative displacement between M_1 and M_2
$JERK2_{max}$	Maximum jerk experienced by the automobile body
Z_{max}/U_o	Ratio between the maximum relative displacement of the automobile body and the maximum amplitude of the input function

INTRODUCTION

In recent years, the importance of providing better ride comfort especially for compact cars has been emphasized. Ride comfort for road vehicle drivers depend largely on the vibration isolation characteristics of the suspension system.

Conventional passive automobile suspension consist of linkages of metal with springs and dampers, which may be pneumatic, hydraulic, or rubber devices. The elements are passive since no external power is required. The performance of an active suspension is far more superior than a passive one. In general, an active system is a complex system and is more costly to install and operate. Recognizing both the performance benefit as well as the limitations of active suspension systems, a semi-active suspension system has been developed (Karnopp *et al.*, 1974). The elements in a semi-active suspension system are the same as that in the passive suspension, but the damper parameter can be modulated. It can be visualised as a conventional hydraulic shock absorber with a variable orifice (Margolis, 1983). The three different types of automobile suspensions which are models of the real system are shown in Figure 1.

The concept of a semi-active damper which requires small external power has been developed by Crosby of Lord Corporation (Krasnicki, 1979) and Karnopp of the University of California (Karnopp *et al.*, 1974). It was first introduced by Parker (1978) and has since been developed into a more sophisticated system. The first prototype employing a "skyhook" damper was



(a) Passive (b) Active (c) Semi-active

Figure 1. *Idealised model of various suspension*

successfully tested in 1972 at Lord Corporation (Krasnicki, 1979). Sharp and Hassan (1986) showed that the semi-active suspension system gave better ride comfort compared to the passive systems. In this paper, maximum relative displacement between sprung and unsprung mass is recorded during simulations to study the performance of the suspension system. These parameters are defined in equations 1 and 2 respectively.

$$Z = X_2 - X_1 \quad (1)$$

$$JERK2 = d^3X_2/dt^3 \quad (2)$$

The mathematical model for the semi-active suspension system is developed from a conventional passive system after computer simulations for the passive suspension system has been tested with step, ramp and sinusoidal inputs. Limited working space is one of the constraints that should be considered while designing an automobile suspension. Three types of semi-active suspensions were studied. Type I semi-active suspension uses both the absolute and relative velocity of the mass as the condition parameters to switch the damper 'ON' or 'OFF'. Type II semi-active suspension uses both the relative velocity and relative displacement of the mass, while Type III uses both the relative velocity and relative acceleration of the mass. Similar semi-active suspension systems have been analysed by Rakheja and Sankar (1986).

MATHEMATICAL MODEL

The two degrees of freedom system as shown in (a) and (c) of Figure 1 are used to develop the computer program for simulation. The governing equations can be presented as follows:

$$M_1\ddot{X}_1 + C_T(\dot{X}_1 - \dot{X}_2) + K_T(X_1 - U) + K_S(X_1 - X_2) = 0 \quad (3)$$

$$M_2\ddot{X}_2 + C_T(\dot{X}_2 - \dot{X}_1) + K_S(X_2 - X_1) = 0 \quad (4)$$

Let

$$\dot{X}_1 = X_{11} \quad \text{and} \quad \dot{X}_2 = X_{22} \quad (5)$$

Substituting into equations (3) and (4) gives

$$\dot{X}_{11} = (1/M_1)[C_T(X_{11}-X_{22})-K_T(X_2-U)-K_S(X_1-X_2)] \quad (6)$$

$$\dot{X}_{22} = (1/M_2)[C_T(X_{11}-X_{22})-XK_T(X_2-U)-K_S(X_1-X_2)] \quad (7)$$

The Fourth Order Rungge Kutta algorithm is used in the simulations since it is in close

agreement with the analytical solution when tested with unit step and ramp inputs. The maximum relative displacement (Z_{max}) between sprung mass (M_2) and unsprung mass (M_1) represents the maximum working space requirement for the suspension while the maximum jerk ($JERK2_{max}$) is the third derivative of displacement for the sprung mass (M_2). The input function used to study the performance of the suspension system can be presented as follows:

$$U = \begin{cases} \dfrac{U_o\,(1 - \cos\omega_{ex} t)}{2}; & 0 \le t \le 2\pi/\omega_{ex} \\ 0 & ;\ t > 2\pi/\omega_{ex} \end{cases} \qquad (4)$$

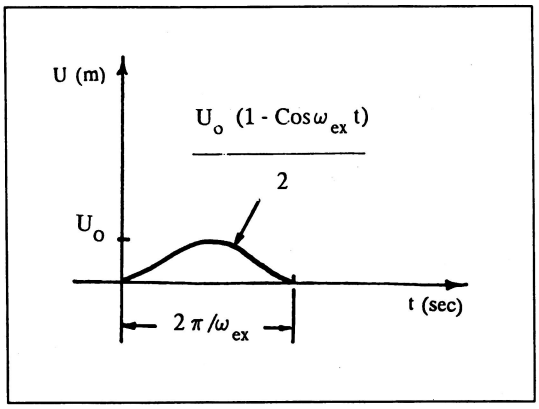


Figure 2. *Graphical representation of the input function*

In real world applications, the input function represents a "bump" on the road. The duration of the "bump" can be controlled by changing the excitation frequency ω_{ex}. Typical data for passenger cars are used for simulations. They are as shown in Table 1.

CONTROL STRATEGIES FOR THE SEMI-ACTIVE SUSPENSION

The objective of the control strategy is to select suitable damper force so that better ride comfort can be achieved from the suspension system. During simulations the 'ON' condition of the damper switch corresponds to a high damping factor value of $\zeta_2 = 0.7$, where

Table 1. Data for Computer Simulation

Sprung Mass (M_2)	260.0 kg
Unsprung Mass (M_1)	37.0 kg
Tyre Stiffness (K_T)	130,000 N m^{-1}
Spring Stiffness (K_S)	13,000 N m^{-1}
Damping Coefficient (C_T)	2574 N.s m^{-1} (ON)
	1103 N.s m^{-1} (OFF)

$$\zeta_2 = \frac{C_T}{2\sqrt{K_S \times M_2}} \text{ and } \omega_{2n} = \sqrt{K_S/M_2} \qquad (9)$$

and the 'OFF' condition corresponds to a low damping factor value of $\zeta_2 = 0.3$. The passive suspension simulation, however, is done with damping factor $\zeta_2 = 0.7$. The Type I strategy uses both absolute velocity and relative velocity of the sprung mass as the controlled para-meters. This strategy is shown in Figure 3.

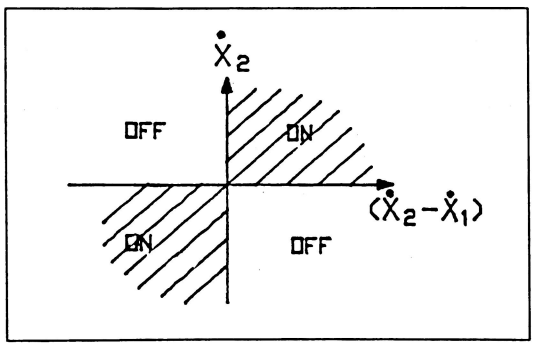


Figure 3. *Control logic for Type I semi-active suspension*

$\dot{X}_2$ = absolute velocity of the sprung mass (M_2)

$(\dot{X}_2 - \dot{X}_1)$ = relative velocity between sprung mass (M_2) and unsprung mass (M_1)

From equation (4), the actual damper force applied to the sprung mass (M_2) can be presented as follows:

$$F_{damper} = -C_T (\dot{X}_2 - \dot{X}_1) \quad (10)$$

The ideal damper force that is desired to restrict the motion of sprung mass (M_2) can be written as follows:

$$F_{desired} = -C_T \dot{X}_2 \quad (11)$$

The control strategy for modified Type I system is applied to the damper in order to avoid sudden changes in the damper force caused by the change in sign in the sprung mass velocity. This is achieved by the following rule:

1. If $(\dot{X}_2-\dot{X}_1)$ changes sign, use Figure 3 to control the switch.
2. If only $\dot{X}_2$ changes sign :

(a) If the damper is 'ON' and the magnitude of the damper force is less than the magnitude of the spring force, use control logic as in Figure 3.

(b) or else, do not switch the damper.

The control logic for the Type II semi-active suspension is derived and shown in Figure 4. In this case, both the relative displacement and the relative velocity of the automobile body are taken as the controlled parameters.

The control strategy for modified Type II system is applied to the damper in order to suppress the condition that may initiate a sudden change in the spring force exerted on the sprung mass (M_2). Thus, the following rule is derived:

1. If $(\dot{X}_2-\dot{X}_1)$ changes sign, use control logic in Figure 4.

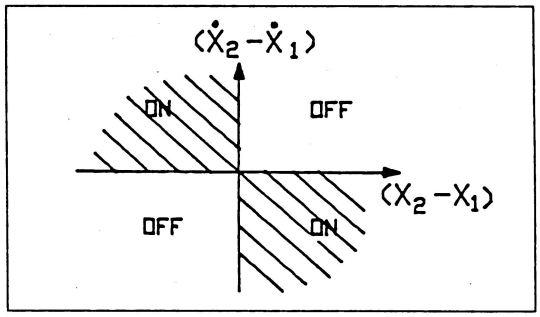


Figure 4. *Control logic for Type II semi-active suspension*

$(\dot{X}_2-\dot{X}_1)$ = relative velocity between sprung mass (M_2) and unsprung mass (M_1)

(X_2-X_1) = relative displacement between sprung mass (M_2) and unsprung mass (M_1)

2. If (X_2-X_1) changes sign, do not change the switch condition.

The Type III system uses Figure 5 as the control strategy in which the damper is 'ON' when the relative velocity is in the opposite direction to the relative acceleration of the automobile body and vice-versa.

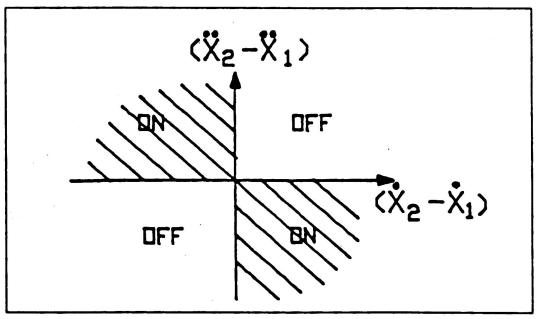


Figure 5. *Control logic for Type III semi-active suspension*

$(\dot{X}_2-\dot{X}_1)$ = relative velocity between sprung mass (M_2) and unsprung mass (M_1)

$(\ddot{X}_2-\ddot{X}_1)$ = relative acceleration between sprung mass (M_2) and unsprung mass (M_1)

The control strategy for modified Type III system is applied to the damper in order to suppress the condition that can initiate sudden changes in the inertia force exerted on the sprung mass (M_2). Thus, the following rule is derived:

1. If $(\dot{X}_2-\dot{X}_1)$ changes sign use Figure 5 to control the switch.
2. If $(\ddot{X}_2-\ddot{X}_1)$ changes sign, do not change the switch condition.

Therefore, six control strategies are used for the simulation, namely Type I, Type II, Type III, modified Type I, modified Type II and modified Type III.

DISCUSSION

Initially, the three types of simple 'ON/OFF' control rules are applied to the suspension system during simulations. The results obtained from the three simple 'ON/OFF' control rules to the damper are shown in Figures 6 and 7. At high excitation frequencies, the working space required by the Type II control rule is smaller than the passive suspension. However, all the three types of 'ON/OFF" control rules can cause discomfort to the rider because the magnitude of $JERK2_{max}$ obtained is greater than the results from the passive suspension system.

Next, the three modified control strategies are introduced and the results obtained are presented in Figures 8 and 9. It is clear that the working space required by the modified control rules are larger compared to the passive suspension. However, modified Type III control rules applied to the damper can provide better ride comfort when compared to the passive suspension. The results obtained from modified Type I and modified Type II control rules are identical since both of them depend largely on the relative velocity of M_2 and M_1 to control the damper.

At high input frequencies the simple 'ON/OFF' control switch applied to the damper will increase the magnitude of maximum jerk

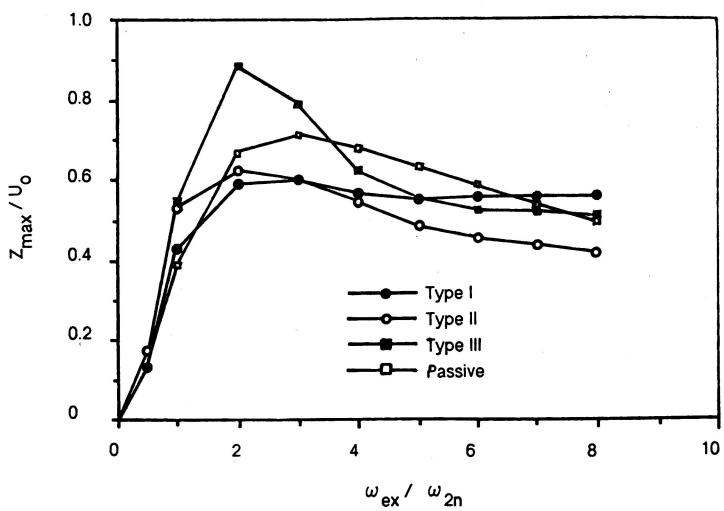


Figure 6. *Maximum relative displacement for simple 'ON/OFF' semi-active suspension (U_o = 1m)*

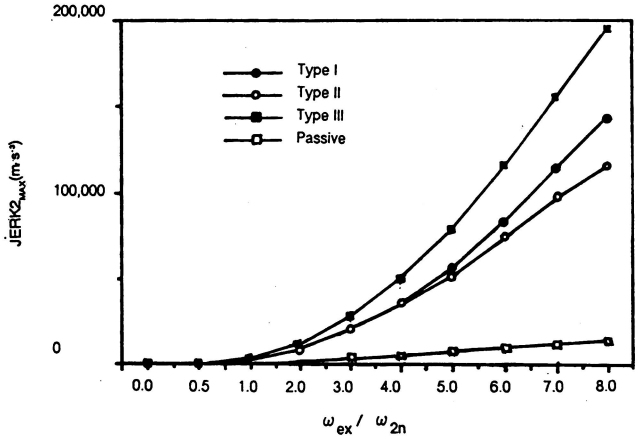


Figure 7. *Maximum jerk experienced by M_2 for semi-active suspension (U_o = 1m)*

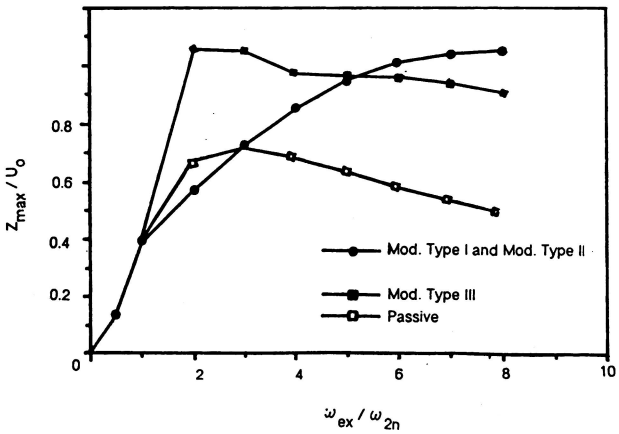


Figure 8. *Maximum relative displacement for modified 'ON/OFF' semi-active suspension (U_o = 1m)*

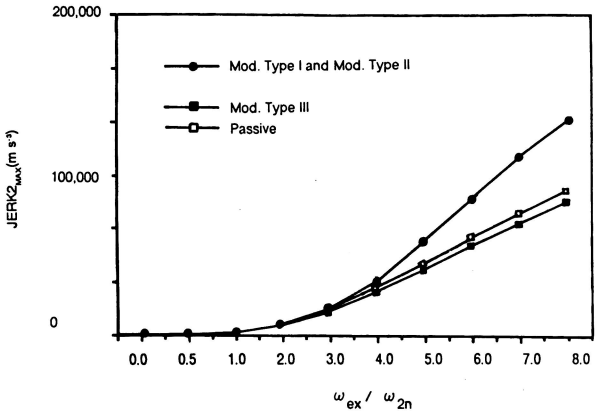


Figure 9. *Maximum jerk experienced by M_2 for modified' 'ON/OFF' semi-active suspension (U_o = 1m)*

experienced by the automobile body significantly, while slightly reduce the maximum working space requirement. In general, modified control logics are able to reduce the magnitude of maximum jerk experienced by the automobile body as compared to the original simple 'ON/OFF' control rules for all the three types of semi-active suspensions. Moreover, modified Type III control rules are able to reduce the magnitude of maximum jerk, experienced by the automobile body to a value that is lower than that obtained from passive suspensions. However, working space required by the modified control logic is about twice the amount required for passive suspensions. The two criteria that have been selected, namely the working space and the jerk, always contradict each other. For instance, better ride comfort can only be achieved at the expense of having a larger working space. Ride comfort is inversely proportional to the jerk experienced by the sprung mass (M_2). It has been shown that when humans are exposed to whole body vibrations for long periods of time, back disorders can arise (Griffin and Griffin, 1989). Further studies are required to be carried out in order to obtain a control strategy that can provide better ride comfort whilst utilizing a smaller working space than that required by a passive suspension system.

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