CONTROL OF ACTIVE VEHICLE SUSPENSION SYSTEMS

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ABSTRACT

There is currently interest in active vehicle suspension systems, in several different areas including; road-going passenger cars, commercial vehicles and buses, rail vehicles, military vehicles such as tanks and other all-terrain vehicles. Although there is common ground shared by these various applications, there are also significant differences. For examples, a rail vehicles has a totally different means of guidance as compared with a road-going vehicle; and the correction from wheel, to bogie, to body for a rail vehicle is also rather different from the wheel-body correction in a conventional road-going car. Even more extreme are the unique requirements encounter when examining the magnetic suspension systems, associated with so-called "MAGLEV" (magnetically levitated) mass-trans transport system vehicles. The subject matter to be treated in this paper will be essentially restricted to active vehicle systems in one area, namely road-going cars.

BACKGROUND-REQUIREMENTS FOR A ROAD GOING VEHICLE SUSPENSION SYSTEM

People working in this area tend to have their own views on what are the primary functions of a vehicle suspension system. It would seem that a reasonable statement of the requirements is as follows

1. Primary Requirement
   - To allow a wheeled vehicle, in motion, to keep all wheels in adequate contact with the road surface.
2. Secondary Requirements
   - To minimise the effects of shock and vibration on the vehicle and its components.
   - To minimise passenger discomfort.
   - To preserve the vehicle ability to obey directional commands, and maintain directional stability.

Other qualifications can be attached as well, for example; the vehicle will have to operate under a wide range of load, speed and road surface roughness conditions.

TRADITIONAL (PASSIVE) VEHICLE SUSPENSIONS

The conventional, passive car suspension can be simplified to the form shown in Figure 1. This is a two degree of freedom model for the vertical motion of one wheel only.

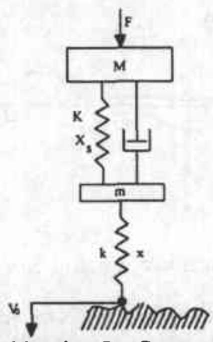

Figure 1. Model of Passive Car Suspension System.

The meanings of the symbols used are as indicated below

M :car body mass (sprung mass)
m : wheel axle mass (unsprung mass)
K : main spring constant
k :spring constant of tyre
B : damping coefficient of shock absorber
X_s: compression displacement of spring
x : compression displacement of tyre
F : disturbing force on body mass
V_0: velocity input disturbance at road contact point.

It should be noted that there is some damping associated with the tyre, but this is usually small and is commonly omitted from the model. This model (and an even simpler variant) will be examined in order to explore the potential and limitation of the traditional passive suspension system.

Prior to looking at this, it is relevant to the first define specifically the limitations of passive suspension systems.

PASSIVE SUSPENSIONS-LIMITATIONS

1. The spring can only generate force in response to a relative displacement between body and wheel.
2. The damper (shock absorber) can only generate forces related to the relative velocity of the wheel with respect to the body (ie relative velocity).
3. In general, the performance of the suspension is fundamentally limited by the stored energy in the body/wheel interconnection which is in turn related to its history.
4. The damper, generating forces in response to relative velocity, can only operate in the dissipate mode.

Having considered these limitations, we can now take account of the significant point of difference between active and passive suspensions.

FUNDAMENTAL PROPERTY OF ACTIVE SUSPENSIONS

An active suspension can continually supply and control the flow of energy to the body/wheel interconnection. By contrast, passive systems can only

1. Dissipate energy,
2. Temporarily store, and later return energy to the system.

An active system can generate forces which do not depend upon energy previously stored by the suspension.

ANALYSIS OF A GENERALISED ROAD VEHICLE SUSPENSION MODEL

As a starting-point in a generalised analysis, we can further simplify our model of Figure 1, into a single degree of freedom model. This is shown in Figure 2.

In this case, the wheel is taken as being in direct, constant contact with the road, without the flexible connection previously attributed to tyre.

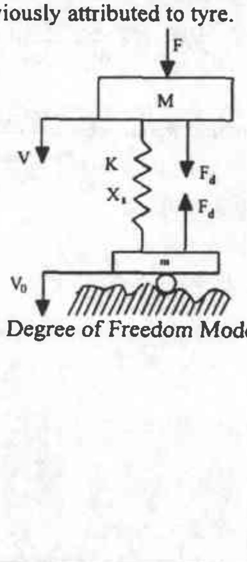

Figure 2. Single Degree of Freedom Model.

The other important difference in Figure 2 is that we have replaced the damper of Figure 1 with a more general velocity-law force generator.

We will consider two possible approaches for the generation of the force F_d

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$$F_d = -B(V - V_0) \qquad (1)$$

This is the relationship governing a conventional damper, with $(V - V_0)$ representing the relative velocity between body and wheel. Alternatively, we could consider

$$F_d = -B_1 V \qquad (2)$$

This represents a possible strategy for an active velocity-dependent force generator (We note that in this case V is the actual (vertical) velocity of the car body with respect to the road).

This strategy would release us from the constraint of F_d being a function of relative velocity between vehicle and wheel.

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Regardless of the law we choose for generating F_d, the state-space formulation can be represented

$$\begin{bmatrix} \dot{X}_s \\ \dot{P} \end{bmatrix} = \begin{bmatrix} 0 & -\frac{1}{M} \\ K & 0 \end{bmatrix} \begin{bmatrix} X_s \\ P \end{bmatrix} + \begin{bmatrix} 0 \\ 1 \end{bmatrix} [F_d] + \begin{bmatrix} 0 & 1 \\ 1 & 0 \end{bmatrix} \begin{bmatrix} F \\ V_0 \end{bmatrix}$$

Noting that we have chosen
X_s : spring deflection,
P : body momentum as our state variables.

Alternatively, if we carry out a transfer function analysis, the relationship between body velocity and disturbance velocity reduces to

$$\frac{V}{V_0} = \frac{2\xi_0 s + 1}{s^2 + 2\xi s + 1} \quad \text{for passive case} \qquad (3)$$

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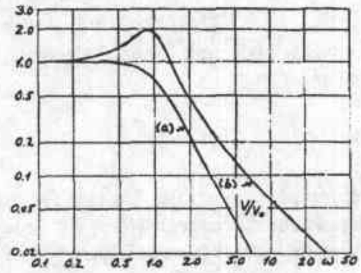
$$\frac{V}{V_0} = \frac{1}{s^2 + 2\xi_1 s + 1} \quad \text{for active case} \qquad (4)$$

where;

$$\xi = \frac{B}{2\sqrt{MK}}, \quad \xi_1 = \frac{B_1}{2\sqrt{MK}}$$

Equations 3 and 4 can be represented in frequency-response plot form, as shown in Figure 3.

The important point to note in Figure 3 is that, while curve (a) (the active system) shows a good isolation above the break frequency with roll-off 40 db/decade, the passive case (curve (b)), with the $2\xi s$ numerator term will have a high-frequency roll-off characteristic that becomes less (ie reduced degree of roll-off) as ξ is increased. So in the passive case, a compromise has to be struck between insufficient roll-off (large ξ) and excessive resonance around the break frequency (small ξ). The value of $\xi = 0.313$ shown in Figure 3 represents this compromise.



a. Active 1 dof System ξ_1=0.707.
b. Passive 1 dof System ξ_1=0.313.

Figure 3. Frequency Response Plots.

We also note in Figure 3 that for the active case (curve (a)), we have chosen $\xi_1 = 0.707$, which gives us a rapid transition into the roll-off region, without showing any degree of underdamping (this being, of course, the value of ξ_1 for critical damping).

TWO DEGREE OF FREEDOM MODEL

The approach we have followed above can be repeated for the two degree of freedom model, where we now include the elasticity of the tyre in our analysis.

The model is shown in Figure 4, which is similar to Figure 1 except for the replacement of the damper by a generalised force generator.

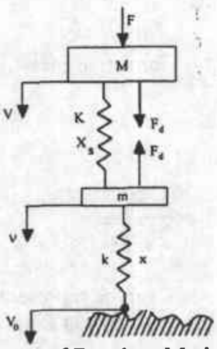


Figure 4. Two degree of Freedom Model.

We can choose a general velocity-control law for F_d

$$F_d = -B_1 V + B_2 v \tag{5}$$

We can see that passive damping would be represented by the case

$$B_1 = B_2 > 0$$

which would reduce Equation 5 to the form

$$F_d = -B_1 (V - v)$$

This being essentially the same as Equation 1. If however $B_1 \neq B_2$, then F_d will contain on active force generating component.

We can again formulate a state-space representation for this system

$$\begin{bmatrix} \dot{x} \\ \dot{X}_s \\ \dot{p} \\ \dot{P} \end{bmatrix} = \begin{bmatrix} 0 & 0 & -\frac{1}{M} & 0 \\ 0 & 0 & \frac{1}{m} & -\frac{1}{M} \\ k & -K & -\frac{B_2}{m} & \frac{B_1}{M} \\ 0 & K & \frac{B_2}{m} & -\frac{B_1}{M} \end{bmatrix} \begin{bmatrix} x \\ X_s \\ p \\ P \end{bmatrix} + \begin{bmatrix} 0 & 1 \\ 0 & 0 \\ 0 & 0 \\ 1 & 0 \end{bmatrix} \begin{bmatrix} F \\ V_0 \end{bmatrix}$$

In this case, our state variables are the two spring deflections, x and X_s, and the two momentum variables p (wheel) and P (body).

As before, we can choose to carry out an analysis in transfer function form, which will yield relationships for wheel velocity and body velocity in response to the disturbing velocity input V_0

$$\frac{V}{V_0} = \frac{2ne\xi_2 s + 1}{D} \tag{6}$$

$$\frac{v}{V_0} = \frac{s^2 + 2\xi_1 s + 1}{D} \tag{7}$$

where;

$$D = \frac{s^4}{n^2} + \left(\frac{2\xi_2}{n} + \frac{2\xi_1}{n^2}\right)s^3 + \left(1 + \frac{1}{n^2} + \frac{1}{n^2 e}\right)s^2 + 2\xi_1 s + 1$$

n : a ratio of natural frequencies $= \frac{\omega_{wheel}}{\omega_{body}}$,

e : the mass ratio =m/M

$$\xi_1 = \frac{B_1}{2\sqrt{MK}}\ ;\ \xi_2 = \frac{B_2}{2\sqrt{mk}}$$

For the active case (ie $B_1 \neq B_2$), this analysis can yield result which are a little better than the passive case. Figure 5 shows frequency plot results, where;

(a) represents body response (active case)
(b) represents body response (passive case)
(c) represents wheel response (active case)
(d) represents wheel response (passive case)

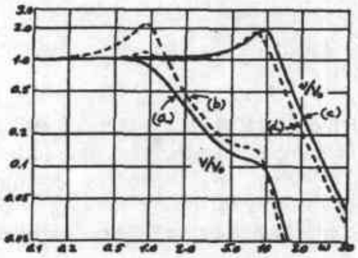


Figure 5. Frequency Response Plots.

It can be seen that, for the active case, the body velocity shows a good following characteristic with respect to the disturbance velocity input, up to $\omega \cong 1$ rad/sec, and a reasonable roll-off performance thereafter, (curve (a) good isolation performance for $\omega > 1$). Similar comments can be made about the wheel characteristic (curve (c)), although some resonance is exhibited near $\omega = 10$ rad/sec.

These results are not spectacular and it should also be pointed out that a fairly judicious selection of parameters (such as e, the wheel-to-body mass ratio) had to be made to achieve these results.

In fact, this approach to active suspension is fairly limited in its achievable performance, and it is necessary to incorporate an additional concept in our approach in order to realise a worthwhile system.

The additional step we will take is to incorporate a load levelling force generator, in addition to the active damping control that we have already explored. Figure 6 shows such a system, with the centre element being connected in series with an actuator. We still have the active damping control generating the force F_d, as previously.

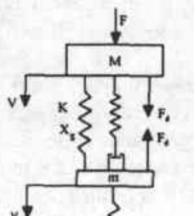
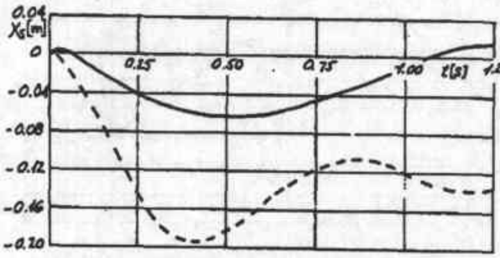


Figure 6. Two degree of Freedom Model.

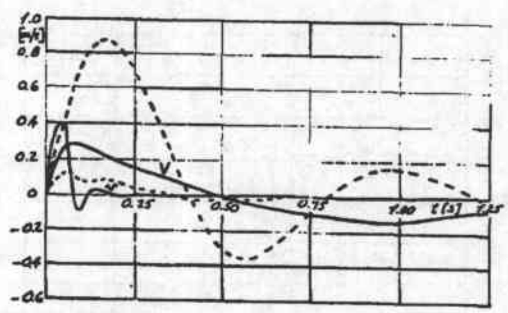
The actuator is used in a load-levelling mode, such as to maintain zero wheel to body displacement error.

The mathematical analyses could again be carried out and shown in similar form to our earlier cases. In this instance, the state-space analysis yields an 'A matrix that is [5x5], and the transfer function analysis yields fifth order denominators. Performance results for such a system are given in Figure 7 shows the response to a step load disturbance input, with the active system response shown as solid lines, and the passive system response as dashed lines.

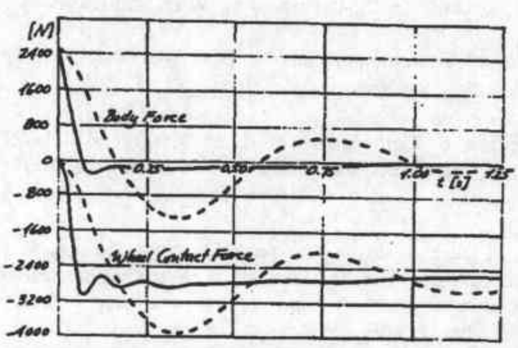
It can be seen that, expect for wheel velocity, the peak excursions are far more effectively contained in the active system.



a. Main Spring Deflection, X_s.



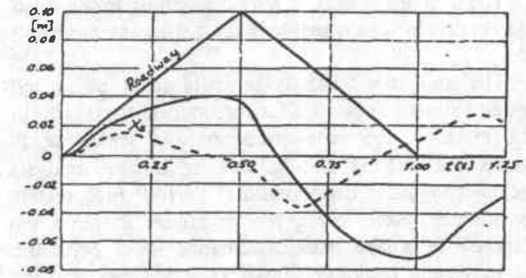
b. Velocity V and v.



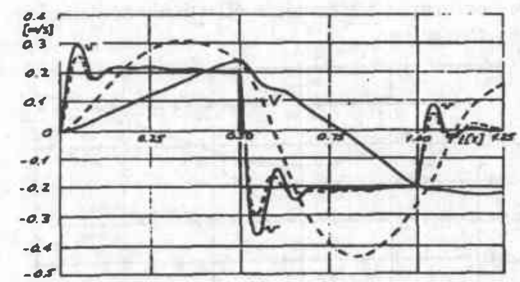
c. Forces on Body and Between Wheel and Roadway.

Input: Suddenly Applied Load F=Mg.

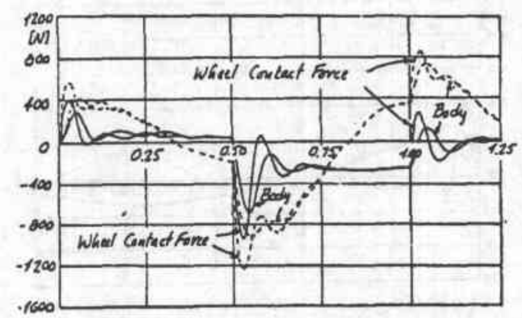
Figure 7. Comparison of Time Response Plots of Basic Passive System (dashed curve) and the Active System.



a. Main Spring Deflection, X_s, with Roadway Profile.



b. Velocity V, v, and V_0.



c. Forces on Body and Between Wheel and Roadway.

Input: Triangular Roadway Bump or Double Step Roadway Velocity.

Figure 8. Comparison of Time Response Plots of Basic Passive System (dashed curve) and the Active System.

Figure 8 shows the response to a road surface irregularity. The disturbance is a triangular cross-section obstacle, which results in a step velocity disturbance input.

Again, the active system displays generally superior results. For example, Figure 8 b shows considerably greater velocity overshoot in the passive case (in both directions). Also very noteworthy are the reduced peak forces experienced by both body and wheel in the active case (Figure 8 c).

CURRENT DIRECTIONS FOR ACTIVE VEHICLE SUSPENSION DEVELOPMENT

In order to examine the current directions being pursued in this field, it is useful to clarify the definitions of the various approaches in categorising suspension types. This is unimportant because some of the literature tends to use various terms rather loosely. The main suspension categories can be regarded

<u>Passive-conventional</u> parallel spring-damper arrangement.

<u>Semi-active:</u> parallel spring damper arrangement, but with the damper characteristics able to be rapidly modified. This type of system has essentially no requirement for external power (as district from the two active approaches below). The system can only operate in the dissipate mode.

<u>Fully active:</u> the springs and dampers of the conventional passive suspension are replaced by hydraulic actuators, which generate forces in accordance with some control law. In this case, an external power source is required in order to provide energy for generation of the forces.

<u>Slow-active</u> is an active system requiring external power, but uses an actuator, commonly in series with a spring element. In this case, the actuator response speed requirements are not as demanding as in the fully active case, and actuators with lower bandwidth can be employed. Actuators for use in full active systems can require response times of 30 msec, whereas the slow-active system can utilise actuators that are 5-10 times slower.

SYSTEMS CURRENTLY UNDER DEVELOPMENT

<u>Semi-active:</u> there is much work taking place currently in semi-active control. This type of system is very attractive in cost terms, since a power source and actuators are not required. Several companies are actively involved in such development work and not surprisingly, many of these companies are long-standing manufacturers of automotive shock absorbers. Companies engaged in this work include Armstrong, Fichtel and Sachs, Bilsten and Delco.

<u>Fully-active:</u> development work in fully-active suspensions appears to be most advanced at Volvo. The work done at Volvo is supported by Lotus, who carried out much pioneering work in this field for use in various areas, including their Formula I racing cars. Lotus has offered external consultancy arrangements to several manufacturers, including Volvo.

The Volvo system is well-advanced in development, and prototype systems have been built and demonstrated.

The Volvo control algorithm utilises 66 independent variables, with control being via an on-board microcomputer.

Capabilities of the Volvo system includes

- Provision of negative roll (ie leaning inwards during cornering).
- Ability to reverse normal braking dive, or achieve level braking.
- Provision for variable front/rear roll stiffness ratio, giving selectable understeer/oversteer characteristics.

Forward Sensing Action

As a result of the response speed required, the earliest possible registration of inputs is necessary. Consequently, developers of active vehicle suspension systems are incorporating some of the following transducers;

- Steering wheel angle/velocity transducer, to give forward warning of direction change (actual direction change in some systems is derived from a yaw gyro).
- Brake-pedal actuation detector and throttle actuation detector, to give advances indication of deceleration or acceleration.
- Some systems include sonar/radar road surface scanners ahead of the front wheels, in order to provide advance warning of road surface irregularities.

RECOGNISED ADVANTAGES AND DISADVANTAGES OF ACTIVE SUSPENSION SYSTEMS

A summary highlighting the main advantages and disadvantages in the use of active suspensions is presented below;

Advantages	Disadvantages
Low static deflection	Potentially high cost
Low natural frequencies Passenger comfort	Complexity in servicing
Consistent characteristics with varying loads	Concerns over reliability due to complexity
Possibility of zero or negative body roll, elimination of braking dive	Extreme demands on high-pressure hydraulic seals to operate from –40 ^{0}C to +300 ^{o}C
Improved viability of simple suspension geometries, such as trailing arm types, due to ability to design for zero body roll.	Significant power consumption, high peak power requirement.

FUTURE

The high cost of implementing active suspension systems is likely to prevent their widespread use during the next decade.

However, the semi-active control strategy, where damper rates are constantly and automatically adjusted to suit instantaneous requirements is far lower in cost and some manufacturers are in the final stages of implementation. Nissan and Toyota are two such manufacturers.

There seems to be little doubt that progression towards active suspensions will prove to be viable in high performance vehicles by the turn of the century, but cost factors will make the transfer to the low-end market very difficult.

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