Semi-active suspensions to reduce road damage: theoretical design and implementation

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Abstract

The peak dynamic tire loads, which are greatly in excess of static loads, are highly dependent on the dynamic characteristics of vehicle suspensions. Analytical research has shown that the vehicle-generated pavement damage can be reduced by using more advanced suspensions. This paper describes a proposed semi-active suspension for heavy trucks to reduce the dynamic tire forces.

Semi-active control laws to reduce dynamic tire forces are investigated and a state estimator for semi-active suspensions is formulated such that the dynamic tire force can be obtained with acceleration measurements.

An experimental study on the performance of a semi-active suspension to reduce the dynamic tire forces is made via a laboratory vehicle test rig. The semi-active suspension has been implemented by the use of a modulable damper, accelerometers and a personal computer. Experimental studies using the laboratory test rig show that the performance of the semi-active suspension is close to that of the best passive suspension for all frequency ranges in the sense of minimizing the dynamic tire forces and that the dynamic tire force can be replaced by the estimated one. The dynamic tire forces for both passive and semi-active control test cases are compared to show the potential of a semi-active suspension to reduce the dynamic tire forces.

1. Introduction

Active and semi-active suspensions for ground vehicles have been a very active subject of research in the past years due to their potential to improve vehicle performance[1,2,3], and they have recently been commercialized on high performance automobiles. Development of active suspensions had been started in the 1930's, but most of the significant developmental work has been done since 1950. Semi-active suspensions were proposed in the early 1970's, showing that performance comparable to that of fully active suspensions can be achieved by the use of semi-active suspensions[5]. Many analytical and experimental studies on active and semi-active suspensions to improve ride quality and handling performance have been recently performed. The conclusion is that active and semi-active suspensions can provide substantial performance improvements over optimized passive suspensions in general and semi-active suspensions can be nearly as effective as fully active suspensions in improving ride quality using state variable feedback[4,5,6,7].

Although an active suspension provides better performance than semi-active suspensions, it has major drawbacks such as the need for a large external power source, increased complexity and cost, and decreased reliability. A semi-active suspension combines the advantages of both active and passive suspensions, i.e., it provides good performance compared to passive suspensions and is economical, safe and does not require either higher power actuators or a large power supply.

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While considerable research on active and semi-active suspensions has been concentrated on the improvement of ride quality, little research has been made on the reduction by active/semi-active suspensions of the dynamic tire force of heavy vehicles to reduce pavement damage[8,9,10,13,14]. Semi-active suspensions to reduce dynamic tire force will be presented in this paper.

The motivation for the theory on the disturbance decoupled bilinear observer proposed in this paper comes from the state estimation problem in semi-active suspensions. A bilinear observer structure for bilinear systems with unknown disturbances is developed such that the estimation error is independent of the unknown external disturbances and the proposed observer is applied to estimate the tire force in a semi-active suspension.

2. A Semi-active Suspension Model and Control Laws

2.1 Bilinear Model of a Semi-active Suspension

This section describes a bilinear model of a semi-active suspension. A bilinear model of a semi-active suspension was introduced by Kimbrough[11] in 1986. Bilinear systems have structural properties that are useful for modeling semi-active suspensions.

Consider the quarter car semi-active suspension model shown in Fig.1. The equations of motion of this system can be written as follows:

$$m_s \ddot{z}_s = f_p + f_s \quad (2.1)$$

$$m_u \ddot{z}_u = -f_p - f_s + f_t \quad (2.2)$$

Heavy vehicles and roads: technology, safety and policy. Thomas Telford, London, 1992.

where

$$f_p \equiv -k_s(z_s - z_u),$$
$$f_t \equiv -k_t(z_u - z_r),$$
$$f_s \equiv \textit{a semi-active force}$$

and k_s and k_t are the stiffness of the spring and the tire respectively.

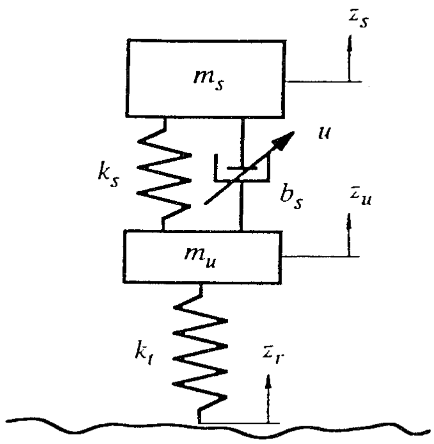


Fig.1 Quarter Car Semi-active Suspension Model

The semi-active force, f_s, may be represented as a nonlinear function of the damping valve area of the semi-active damper, suspension velocity and the material properties of the fluid in the damper.

$$f_s = f_s(c, a_d, (\dot{z}_s - \dot{z}_u)) \tag{2.3}$$

where c is a constant dependent on the properties of the fluid, a_d the damping valve area and $(\dot{z}_s - \dot{z}_u)$ the suspension velocity. Since the damping valve area is controlled by an electromagnetic device such as a stepper motor, the equation of motion may be be written as follows:

$$\frac{d}{dt}a_d = f_a(v) \tag{2.4}$$

where v is the control input for the electromagnetic device.

By defining state variables for this system as follows:

$x_1 = z_s - z_u$	*suspension deflection*
$x_2 = \dot{z}_s$	*sprung mass velocity*
$x_3 = z_u - z_r$	*tire deflection*
$x_4 = \dot{z}_u$	*unsprung mass velocity*
$x_5 = a_d$	*damping valve area*

we can rewrite the equation of motion as follows:

$$\dot{x}_1 = x_2 - x_4$$
$$\dot{x}_2 = -\frac{1}{m_s}k_s x_1 + \frac{1}{m_s}f_s(x_5, (x_2 - x_4))$$
$$\dot{x}_3 = x_4 - w \tag{2.5}$$
$$\dot{x}_4 = \frac{1}{m_u}k_s x_1 - \frac{1}{m_u}k_t x_3 - \frac{1}{m_u}f_s(x_5, (x_2 - x_4))$$
$$\dot{x}_5 = f_a(v)$$

where the unknown disturbance, w $(=\dot{z}_r)$, is the rate of change of road elevation.

Since the electromagnetic device has much a faster response than mechanical systems, the differential equation for the damping valve area dynamics may be replaced by an algebraic equation as follows:

$$x_5(t) = c_1 v(t) \tag{2.6}$$

where c_1 is a constant.

Fig. 2 shows the time response of the semi-active damper for a step input with a constant suspension velocity of 0.18 m/sec. This experimental result was obtained for the semi-active damper with twenty different damping rate settings* using the half car test rig at the Vehicle Dynamics Laboratory at UC Berkeley. A step command was used to modulate the damping rate of the semi-active damper by a stepper motor and a load cell was used to measure the force generated by the damper. Fig.2 illustrates that the response of the semi-active force may be approximated for constant suspension velocity as a first order dynamic equation as follows:

$$\frac{d}{dt}f_s = \frac{1}{T_{fs}}(b_1 v(t) - f_s) \tag{2.7}$$

where T_{fs} is 0.005 second.

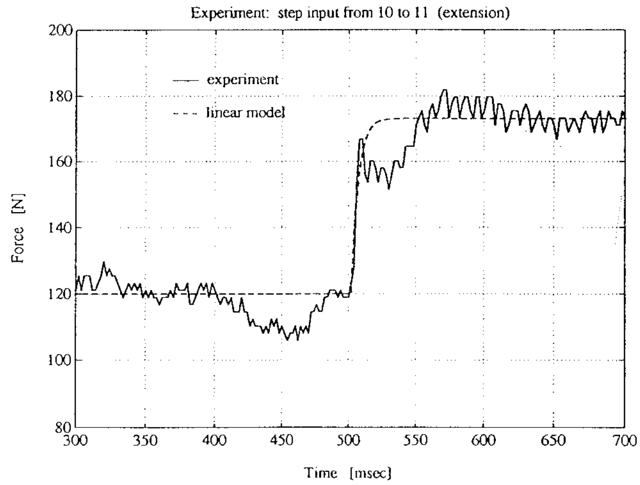


Fig.2 Step Input Response of the Semi-active Damper for Constant Suspension Velocity of 0.18 *m/sec* and its Linear Approximation

Since the time constant, T_{fs}, is small enough, the relation between the semi-active force, $f_s(t)$, and the control input, $v(t)$, may be represented by an algebraic equation for constant suspension velocity as follows:

$$f_s(t) \approx k\, v(t) \tag{2.8}$$

where k is a constant dependent on the suspension velocity. This validates the equation (2.6). Thus the semi-active force may be written as a function of the input, $v(t)$, and the suspension velocity, $(x_2 - x_4)$, as follows:

$$f_s(t) = f_s(v(t), (x_2 - x_4))$$

Typical force-velocity curves and their linear approximations for the different control inputs, v, are shown in Fig.3. This experimental data is also for the semi-active damper with twenty states. It is illustrated that the semi-active force-suspension velocity curves can be represented by a bilinear equation, i.e.,

$$f_s(t) = \alpha\, v(t)\,(x_2 - x_4)$$

Since αv is equivalent to the damping rate for the control input, v, for the electromagnetic device, we can define new

*The semi-active dampers were provided by the Lord Corp. of Erie, Pa.

input, $u(t)$, as follows:

$$u(t) = \alpha v(t) - b_s$$

where b_s is the passive damping rate and the state equation (2.5) can be rewritten as the following bilinear state equation:

$$\dot{x} = A\,x + D\,x\,u + F\,w \tag{2.9}$$

where the unknown disturbance $w\ (=\dot{z}_r)$ is the rate of change of road elevation and

$$x = [\ z_s - z_u\ \ \dot{z}_s\ \ z_u - z_r\ \ \dot{z}_u\ ]^T\ ,$$

$$A = \begin{bmatrix} 0 & 1 & 0 & -1 \\ -\dfrac{k_s}{m_s} & -\dfrac{b_s}{m_s} & 0 & \dfrac{b_s}{m_s} \\ 0 & 0 & 0 & 1 \\ \dfrac{k_s}{m_u} & \dfrac{b_s}{m_u} & -\dfrac{k_t}{m_u} & -\dfrac{b_s}{m_u} \end{bmatrix},$$

$$D = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & -\dfrac{1}{m_s} & 0 & \dfrac{1}{m_s} \\ 0 & 0 & 0 & 0 \\ 0 & \dfrac{1}{m_u} & 0 & -\dfrac{1}{m_u} \end{bmatrix},\quad F = \begin{bmatrix} 0 \\ 0 \\ -1 \\ 0 \end{bmatrix}.$$

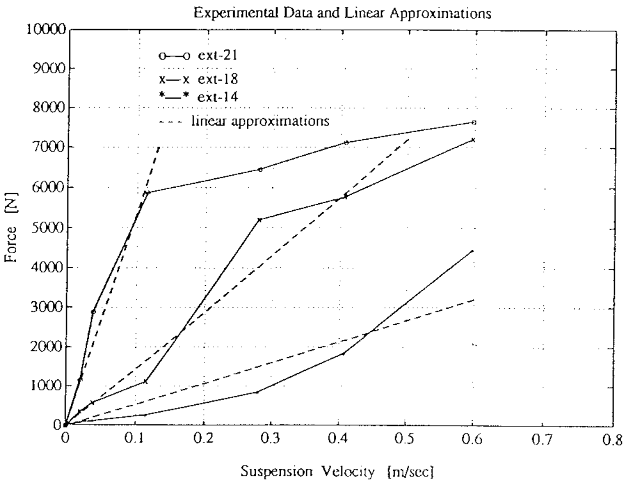


Fig.3 Force-Velocity Relations and Their Linear Approximations (experimental data for a semi-active damper with 21 settings)

2.2 Semi-active Control Laws to Reduce Dynamic Tire Force

Consider the quarter semi-active suspension model shown in Fig.1. The equation of motion of this system is represented by a bilinear form:

$$\begin{aligned} \dot{x} &= A\,x + B\,f_s + F\,\dot{z}_r \\ &= A\,x + D\,x\,u + F\,w \end{aligned}$$

The desired force, f_s, is found for the deterministic case by solving a typical LQ problem with the following performance index:

$$J = \lim_{T\to\infty} \int_0^T \begin{bmatrix} \rho_1\ddot{z}_s^2 + \rho_2(z_s - z_u)^2 + \rho_3\dot{z}_s^2 \\ +\rho_4(z_u - z_r)^2 + \rho_5\dot{z}_u^2 + r f_s^2 \end{bmatrix} dt \tag{2.10}$$

where ρ_1 is weighting factors for sprung mass acceleration, ρ_2 through ρ_5 for states of the suspension system, and r for the input, i.e., control force. The performance index given by equation (2.10) can be rewritten as follows:

$$J = \lim_{T\to\infty} \int_0^T [\ x^T Q\,x + 2\,x^T M\,f_s + r\,f_s^2\ ]\ dt$$

$$Q \ge 0\ ,\ \ Q - M\,r^{-1}\,M^T \ge 0\ ,\ \ r > 0\ .$$

The optimal force which minimizes the performance index is given as following constant gain state feedback control law:

$$f_{s,opt} = -\,r^{-1}\,(\,B^T H + M^T\,)\,x \tag{2.11}$$

where H is determined by solving the following algebraic Riccati equation:

$$-(A - Br^{-1}M^T)^T H - H(A - Br^{-1}M^T) - (Q - Mr^{-1}M^T) + HBr^{-1}B^T H = 0$$

The frequency responses for passive and active suspensions were computed to show the improvement over the passive suspensions and to compare the difference between the full state feedback case, the sprung mass velocity feedback case and the tire force feedback case. A comparison of frequency responses between passive, active suspension with state feedback (State fdbk) and active suspension with tire force feedback (TF fdbk) cases is shown in Fig.4. The tire force feedback control law is implemented by setting all feedback gains in the state feedback control law (2.11) equal to zero except the tire force feedback gain. The tire force and sprung mass acceleration are shown in Fig.4. There are two modal frequen-

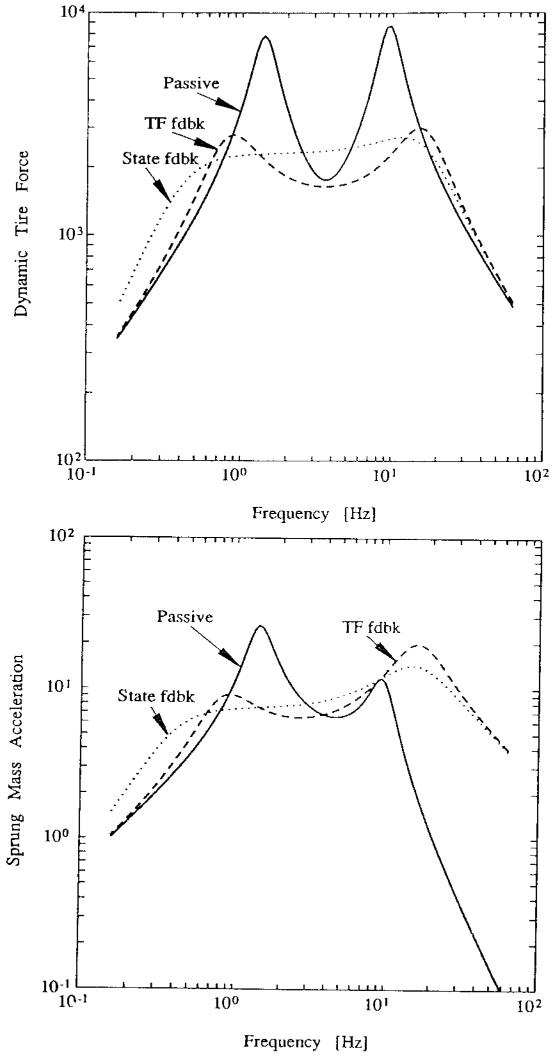


Fig.4 Frequency Responses of Quarter Car Model (A Comparison between State Feedback and Tire Force Feedback Cases)

cies in the tire force case, i.e., a 2 Hz body mode and a 10 Hz axle mode. It shows that the peaks at all the modal frequencies are significantly reduced by the active suspensions and the performance of the tire force feedback case is close to that of the full state feedback case. This shows the intuitive result that the most important state variable in the implementation of the active suspension control law to reduce the tire force, i.e., the axle load, is the tire force. It is illustrated that the higher frequency components in the sprung mass acceleration are increased in the active suspension cases.

Useful insight on the selection of a control law can be obtained from the above results. Since the full state feedback control law is very difficult to implement in real systems, the tire force feedback control law may be practical. Since measurements of the tire forces are very difficult to make, the bilinear observer is proposed to estimate the dynamic tire force from accelerometers in section 3.

A reasonable semi-active control law applicable to a suspension with continuously modulable dampers can be obtained from the active control law, i.e., it is given as follows[6]:

$$u(t) = \begin{cases} u_{\min} & \text{if } u^*(t) \le u_{\min} \\ u^*(t) & \text{if } u_{\min} \le u^*(t) \le u_{\max} \\ u_{\max} & \text{if } u_{\max} \le u^*(t) \end{cases} \quad (2.12)$$

where $u(t)$ is the damping rate of the modulable damper and

$$u^*(t) = -\frac{f_{s,opt}}{(\ suspension\ velocity\ )}$$

$f_{s,opt}$ is the desired control force which can be determined by some active control law, i.e., state feedback or the tire force feedback, or the sprung mass velocity feedback.

3. The Design of State Estimator for a Semi-active Suspension

An observer structure for bilinear systems proposed in [3] is applied to estimate the tire force in a vehicle semi-active suspension problem.

A number of studies on active and semi-active suspension control laws have been recently performed assuming that all states are available, showing that the performance of the vehicle can be significantly improved when compared to passive suspensions by using either active or semi-active suspensions [4,12]. Also, it has been shown that the most important state variable for the suspension controls to improve ride quality is absolute sprung mass velocity and the most important one to reduce pavement damage is the dynamic tire force [4,12]. Although measurements of the sprung mass velocity may be made by integrating the output of an accelerometer, measurement of the tire forces is very difficult to make for real time control because of the unknown road input. Thus it is necessary in semi-active suspension control to design an observer which estimates necessary states, whose estimation error due to initial conditions converges to zero sufficiently quickly and whose error is independent of the unknown road input.

Based on the design method proposed in [3] an observer is designed to estimate the dynamic tire force, which is difficult to measure in real time control. Assume that axle acceleration and sprung mass acceleration are measured. Thus the measurement y is

$$y = \begin{bmatrix} y_1 \\ y_2 \end{bmatrix} = \begin{bmatrix} \dot{x}_2 \\ \dot{x}_4 \end{bmatrix} \quad (3.1)$$

Select tire deflection as v, which is the disturbance related state. i.e.,

$$v = \hat{x}_3 \quad (3.2)$$

From the relation between v ($= \hat{x}_3$) and the measurement y_2, z is determined. i.e.,

$$\begin{aligned} v &= \hat{x}_3 = f_1(\ y_2, \hat{x}_1, (\ x_2 \hat{-} x_4\ )\ ) \\ &= f_2(\ y_2, z\ ) \end{aligned} \quad (3.3)$$

and

$$z = W\hat{x} = \begin{bmatrix} \hat{x}_1 \\ (\ x_2 \hat{-} x_4\ ) \end{bmatrix} \quad (3.4)$$

$$W = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & -1 \end{bmatrix}$$

In this case a state estimator for a semi-active suspension is expressed as

$$v = \hat{x_3} = [\ \frac{k}{k_t}\ \ \frac{b}{k_t}\ ]\ z + [\ 0\ \ \frac{1}{k_t}\ ]\ z\ u + [\ 0\ \ -\frac{m_u}{k_t}\ ]\ y \quad (3.5)$$

$$\dot{z} = \begin{bmatrix} 0 & 1 \\ -\frac{k}{m_s} & -\frac{b}{m_s} \end{bmatrix} z + \begin{bmatrix} 0 & 0 \\ 0 & -\frac{1}{m_s} \end{bmatrix} z\ u + \begin{bmatrix} h_1 \\ h_2 \end{bmatrix} [\ y_1 - y_{1z}\ ] + \begin{bmatrix} 0 \\ 1 \end{bmatrix} y_2$$

where

$$y_{1z} = -\frac{k}{m_s} z_1 - \frac{b}{m_s} z_2 - \frac{1}{m_s} z_2\ u \quad (3.6)$$

Define the estimation error, e_z, as follows:

$$e_z = \begin{bmatrix} x_1 - z_1 \\ (\ x_2 - x_4\ ) - z_2 \end{bmatrix} \quad (3.7)$$

Then, the error dynamics are expressed as

$$\dot{e}_z = \begin{bmatrix} h_1 \frac{k}{m_s} & 1 + h_1 \frac{(\ b + u\ )}{m_s} \\ (\ h_2 - 1\ ) \frac{k}{m_s} & (\ h_2 - 1\ ) \frac{(\ b + u\ )}{m_s} \end{bmatrix} e_z \quad (3.8)$$

If the error dynamics are stable, the dynamic tire force estimation error tends to zero by Theorem 1 and Theorem 2 presented in [3]. The stability of the bilinear observer depends on the observer feedback gains (h_1 , h_2). The stability region of the gains (h_1 ,h_2) can be found by applying LEMMA 1 in [3].

LEMMA 1. The error dynamics (3.8) is asymptotically stable if the observer feedback gains (h_1 ,h_2) satisfy the following conditions:

$$\alpha < h_1 < 0$$

$$h_2 < 1$$

where

$$\alpha = -\frac{4m_s}{u_{\max}} \left[1 + 2\frac{(b+u_{\min})}{u_{\max}} + 2\sqrt{\frac{(b+u_{\min})}{u_{\max}}(1+\frac{(b+u_{\min})}{u_{\max}})} \right]$$

Proof: See Ref. [3].

The state estimator discussed in this section for a semi-active suspension estimates the tire deflection, i.e., the dynamic tire force, the spring deflection and the spring deflection rate with the axle acceleration and the sprung mass acceleration measurements. The suspension velocity should be known in order to determine the damping rate of the modulable shock absorber. The measurements of acceleration may be made with ease compared to velocity or deflection measurements. As mentioned in the introduction, this study has been motivated by a state estimation problem in semi-active suspension control to reduce the dynamic axle load where the dynamic tire force and the spring deflection rate are the most important states in the control law [4,8]. Therefore the state estimator designed in this section may be very effective in semi-active suspension control to reduce the dynamic tire force.

4. Laboratory Experiments

Experimental studies using a half car test rig were conducted to test the proposed semi-active suspension with the bilinear observer-controller. The objectives of this experiment were:

(*i*) to determine the potential of the semi-active suspension to reduce dynamic tire loading.

(*ii*) to verify the performance of the proposed disturbance decoupled bilinear observer under a realistic semi-active suspension system where real implementation problems such as nonlinearity of the semi-active dampers, parameter uncertainty and the effect of unmodeled dynamics etc. may arise.

(*iii*) to investigate the feasibility of the observer-controller from a real time perspective.

The semi-active suspension was implemented on a half car model. Though the half car model is different from the tractor/semi-trailer heavy truck model, experimental verification of the performance of the semi-active suspension using the half car model is important and may be helpful in real vehicle implementation.

4.1 Half Car Test Rig

The laboratory half car model used in experimental study is shown in Fig.5. The experimental setup of the U.C. Berkeley Active/Semi-active Suspension consists of a hydraulic power system, a road profile generating system, a vehicle dynamics simulating system , sensors and an electronic control system. The simulated vehicle, i.e., laboratory half car model, consists of four parts:

- Sprung mass(Vehicle body/chassis)
- Unsprung mass(tires and axle)
- Suspensions, and
- Guide rails.

The vehicle parameters for the half car test rig are given in Table 1.

Load cells are used to measure the tire force and the force provided by the semi-active damper. The suspension load cells have a range of 4,448 N and the tire force load cells can measure up to 22,240 N.

A semi-active damper with 20 states has been used to generate the desired semi-active forces. The semi-active damper force versus suspension velocity curves are shown in Fig.6.

The damper position is controlled by a stepper motor. The stepping time of this stepper motor is 1 msec and the force response of the semi-active damper for one step change of the damper position from position 10 to 11 for constant suspension velocity is shown in Fig.2. There exist oscillations in the damper force after a step change due to the vibration of the stepper motor axle.

4.2 Experimental Results

A continuous semi-active control law was implemented using a continuously modulable damper with 20 damping rate settings. The damping rate was modulated between setting 1 and setting 14 which is approximately equivalent to 1058 *N*/(*m*/*sec*) to 5436 *N*/(*m*/*sec*) range.

Firstly, dynamic tire force measured from a load cell was used to implement a semi-active control law to reduce dynamic tire force. Frequency responses for passive and semi-active suspensions were obtained between 0.5 to 18 Hz road inputs. In addition, responses for superpositioned six sine wave road inputs were compared for passive and semi-active cases.

The dynamic tire forces were estimated using sprung and

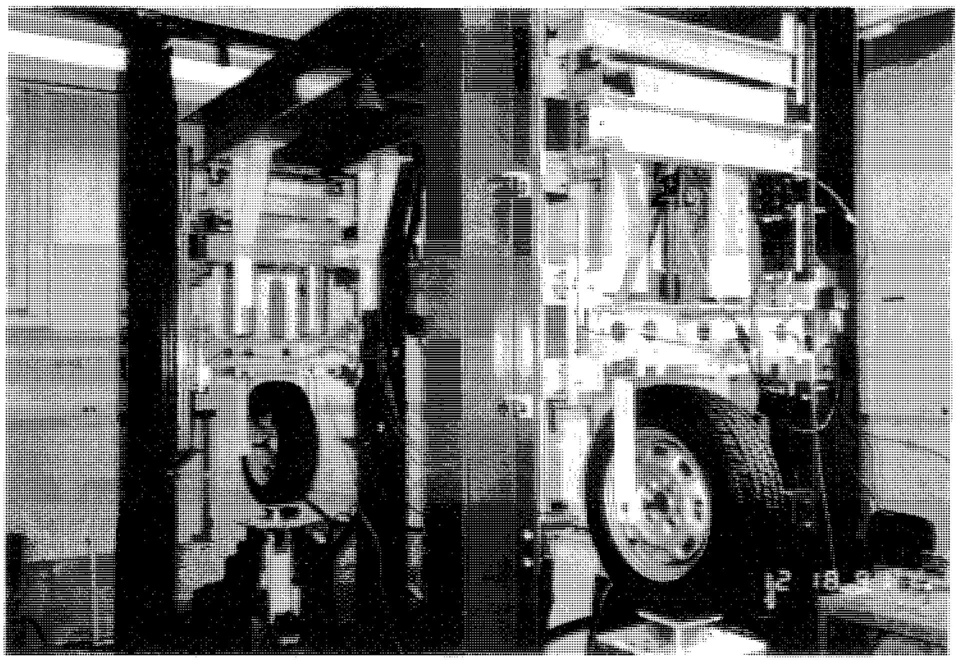


Fig.5 Berkeley Half Car Suspension Test Laboratory

unsprung mass acceleration measurements by a disturbance decoupled bilinear observer. Then the estimated dynamic tire force was used for the semi-active control law.

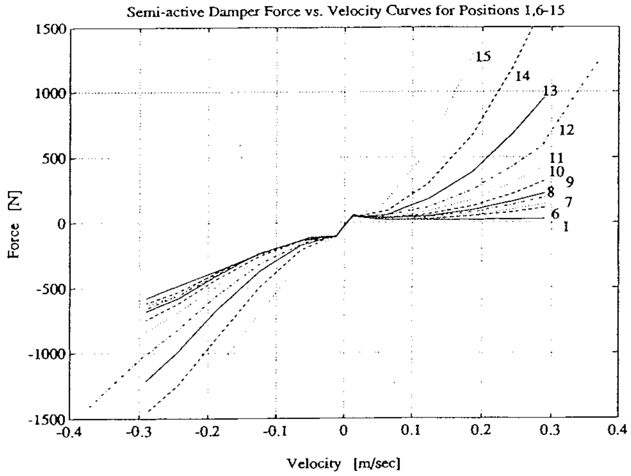


Fig.6 Semi-active Damper force vs. Velocity Curves

4.2.1 Dynamic Tire Force Feedback

Fig.7 shows a comparison of the frequency response of the passive and semi-active suspensions for dynamic tire force and sprung mass acceleration. The control law was designed to minimize the dynamic tire force. For small dynamic tire force, there are three frequency ranges:

0.5 - 1.5 Hz : the hard passive is better than the soft passive

1.5 - 9.0 Hz : the soft passive is better than the hard passive

9.0 - 17. Hz : the hard passive is better than the soft passive

The comparison of the frequency responses indicates that the performance of the semi-active suspension is close to the best passive suspension for all frequency ranges in the sense of minimizing the dynamic tire force.

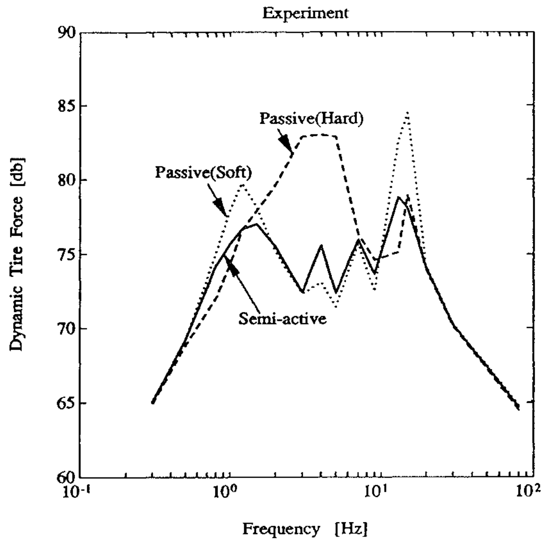
At 1.2 Hz, the sprung mass acceleration is lower than that of the soft passive case, whereas between 1.5 and 9 Hz, the sprung mass acceleration is close to the soft passive case. Sprung mass acceleration increases at the axle bounce mode frequency ,i.e., at 13 Hz to 15 Hz range.

Fig.7 illustrates that the semi-active suspension with the dynamic tire force minimizing control law improves both the dynamic tire force and the sprung mass acceleration within a 0.8 to 7 Hz range and aggravates both the sprung mass acceleration and the suspension deflection at the axle bounce mode frequency, i.e., at 13 to 15 Hz.

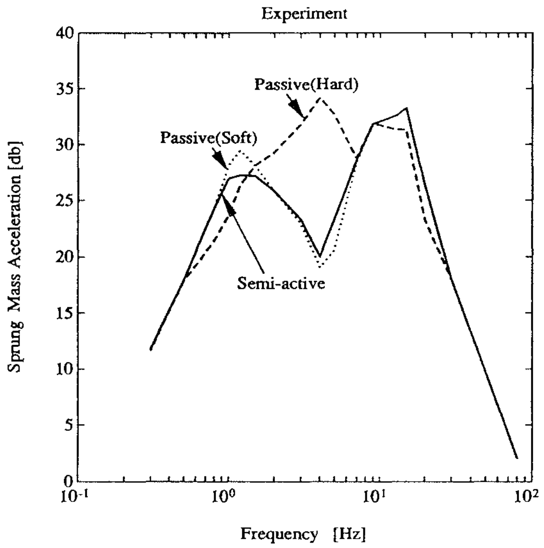
In order to compare the performance of the semi-active and passive suspensions for a more realistic road input case, experiments were executed for a sum of six sine waves road input. The amplitude of each sine wave was chosen to generate a similar spectral density to that of a real road. Comparison of the passive and semi-active cases are shown in Fig.8. It can be seen that peak dynamic tire force is reduced by 40 % in this case.

4.2.2 Dynamic Tire Force Estimation and Estimated Dynamic Tire Force Feedback

It was shown in section 4.2.1 that the dynamic tire force feedback semi-active control law provides good performance compared to passive suspensions at all frequency ranges and knowledge of the dynamic tire force and suspension velocity is essential in semi-active control to reduce the tire force variation. Although measurements of the tire force are very difficult to make for real time control, they can be estimated from



(a) Dynamic Tire Force



(b) Sprung Mass Acceleration

Fig. 7 Comparison of Frequency Responses of Passive and Semi-active Suspensions

accelerometers by the disturbance decoupled bilinear observer proposed in section 3.

The dynamic tire force was estimated from two acceleration measurements , i.e., sprung and unsprung mass accelerations, by a bilinear observer, and then the estimated dynamic tire force was used to implement a semi-active control law without the measurement of the tire force.

Fig.9 shows a comparison of the dynamic tire forces measured by the load cell and estimated by the observer for the realistic road input case described in section 4.2.1. The observer started to work after 0.2 seconds and the estimated dynamic tire force is very close to the real one. It was shown that the estimation error quickly dies out. The estimation error is due to parametric errors such as sprung mass error, spring stiffness error, and tire stiffness error and modeling error such as nonlinear damping characteristics and ignored friction.

Frequency characteristics of the semi-active suspension with the estimated DTF feedback are compared to those of the semi-active suspension with the measured DTF feedback and those of passive suspensions in Fig.10. While the dynamic tire force of the observer-controller case is similar to that of the DTF feedback case in a 2.0 to 15 Hz range, it was indicated that the dynamic tire force of the observer-controller case is greater than that of the measured DTF feedback case at body mode frequency range, i.e., 1.0 to 1.5 Hz range.

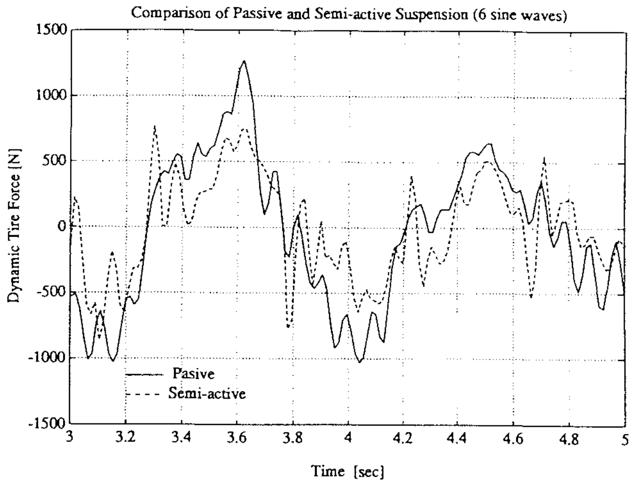


Fig. 8 Comparison of Dynamic Tire Force for Sum of Six Sine Waves Road Input Case

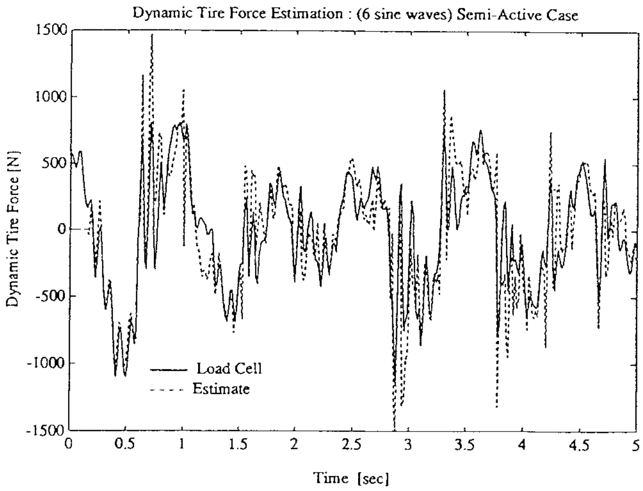


Fig. 9 Comparison of Measured and Estimated Dynamic Tire Force (sum of six sine wave input case)

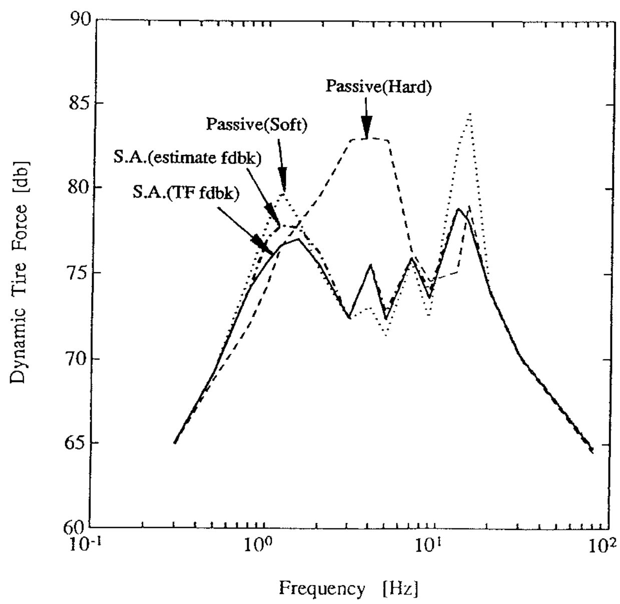


Fig. 10 Comparison of Frequency Responses of Passive and semi-active suspensions

Table 1 Half Car Parameters

Specifications	Value	Unit
Sprung Mass	574.7	*Kg*
Pitch Moment of Inertia of the Sprung Mass	768.9	$Kg{\cdot}m^2$
Unsprung Mass	59.5	*Kg*
Spring Constant	16812.	*N/m*
Tire Stiffness	190000.	*N/m*
Wheel Base	2.74	*m*
Distance from the C.G. to the Front Suspension	1.38	*m*
Distance from the C.G. to the Rear Suspension	1.36	*m*
Equivalent Sprung Mass		
Front	285.3	*Kg*
Rear	289.4	*Kg*

5. Conclusions

A bilinear model of a semi-active suspension was formulated and it was shown via experimental data that a bilinear model does represent a semi-active suspension with sufficient accuracy. The performance of tire force feedback and the sprung mass velocity feedback cases are compared to determine which state is most important in dynamic tire force control. It has been shown that the performance of the tire force feedback with optimal passive damping case is similar to that of the full state feedback case in the dynamic tire force control.

Experimental studies performed using a half car test rig were presented. The semi-active control law to reduce the dynamic tire force was implemented by the use of a semi-active damper with twenty states. The performances of passive and semi-active suspensions were compared for sinusoidal road and superpositioned sinusoidal road input cases. The superpositioned input was created to simulate a more realistic road input.

The dynamic tire force was estimated using sprung and unsprung mass acceleration measurements and the performances of the dynamic tire force feedback and the estimated tire force feedback cases have been compared to those of the passive suspensions.

Experimental results have shown that:

(i) the continuous semi-active control law can be implemented by a semi-active damper with twenty states.

(ii) the disturbance decoupled bilinear observer proposed in section 3 is effective for the estimation of the dynamic tire force.

(iii) the dynamic tire force can be reduced by the semi-active suspensions with dynamic tire force feedback and the dynamic tire force can be replaced by the estimated one.

Although the semi-active damper used in these experimental studies has a nonlinear force-velocity characteristic, the semi-active suspension can be described by a bilinear model and the bilinear observer-controller designed based on this bilinear model has shown good performance.

Experimental studies via the half car test rig have shown that the introduction of semi-active suspensions has the potential to dramatically reduce the dynamic tire force. The semi-active suspension has been implemented in the experimental studies by the use of a modulable damper, accelerometers and a personal computer. In order to determine the size of reductions in dynamic tire force in actual field conditions, the semi-active suspension has been implemented on a four-leaf spring tandem driving axle of a five axle tractor/semi-trailer using a modulable damper, accelerometers and a microprocessor. The actual vehicle tests are in progress.

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