

UC Berkeley

Earlier Faculty Research

Title

Active and Semi-Active Heavy Truck Suspensions to Reduce Pavement Damage

Permalink

<https://escholarship.org/uc/item/573562sj>

Authors

Yi, K.

Hedrick, J.K.

Publication Date

1989-11-01



**Active and Semi-Active Heavy Truck
Suspensions to Reduce Pavement Damage**

K. Yi
J.K. Hedrick

November 1989
Reprint, No. 27

**The University of California
Transportation Center**

University of California
Berkeley, CA 94720

**The University of California
Transportation Center**

The University of California Transportation Center (UCTC) is one of ten regional units mandated by Congress and established in Fall 1988 to support research, education, and training in surface transportation. The UC Center serves federal Region IX and is supported by matching grants from the U.S. Department of Transportation, the California State Department of Transportation (Caltrans), and the University.

Based on the Berkeley Campus, UCTC draws upon existing capabilities and resources of the Institutes of Transportation Studies at Berkeley, Davis, and Irvine; the Institute of Urban and Regional Development at Berkeley; the Graduate School of Architecture and Urban Planning at Los Angeles; and several academic departments at the Berkeley, Davis, Irvine, and Los Angeles campuses. Faculty and students on other University of California campuses may participate in

Center activities. Researchers at other universities within the region also have opportunities to collaborate on selected studies. Currently faculty at California State University, Long Beach, and at Arizona State University, Tempe, are active participants.

UCTC's educational and research programs are focused on strategic planning for improving metropolitan accessibility, with emphasis on the special conditions in Region IX. Particular attention is directed to strategies for using transportation as an instrument of economic development, while also accommodating to the region's persistent expansion and while maintaining and enhancing the quality of life there.

The Center distributes reports on its research in working papers, monographs, and in reprints of published articles. For a list of publications in print, write to the address below.



**University of California
Transportation Center**

108 Naval Architecture Building
Berkeley, California 94720
Tel: 415/643-7378
FAX: 415/643-5456

Authors of papers reporting on UCTC-sponsored research are solely responsible for their content. This research was supported by the U.S. Department of Transportation and the California State Department of Transportation, neither of which assumes liability for its content or use.

**Active and Semi-Active Heavy Truck
Suspensions to Reduce Pavement Damage**

K. Yi
J.K. Hedrick

Department of Mechanical Engineering
University of California at Berkeley

Reprint, No. 27

SAE Technical Paper Series
November 1989

The University of California Transportation Center
University of California at Berkeley

Active and Semi-Active Heavy Truck Suspensions to Reduce Pavement Damage

K. YI and J. K. Hedrick
 Dept. of Mech. Engrg.
 University of California at Berkeley
 Berkeley, CA

Abstract

Active and semi-active suspensions have been evaluated for application on tractor/semi-trailer trucks using a pitch plane simulation model called "VESYM". VESYM is a fully non-linear time domain simulation model. This paper reviews the effect of alternative heavy truck suspensions on pavement damage by using the flexible pavement simulation program "VESYS". It is shown that by estimating axle tire force and using this signal to modulate a semi-active shock absorber that significant reduction in pavement degradation can be achieved.

1. Introduction

In order to improve ride quality and handling performance, a number of analytical and experimental studies on active and semi-active suspensions have been recently performed, showing that performance of the vehicle can be significantly improved when compared to passive suspensions by using either active or semi-active suspensions[2]. Relatively little research has been performed on the reduction by active/semi-active suspensions of pavement damage due to heavy vehicles' dynamic loads. This paper investigates the dynamic effects of heavy truck suspensions on pavement response and the dynamic characteristics of active/semi-active suspensions when considering pavement damage. The motivation for this research is to assist in the design of advanced heavy truck suspensions in order to both extend the life of the interstate highway system and reduce its maintenance cost.

Our study consists of two parts: vehicle analysis and the evaluation of suspension dynamic effects on pavement damage. Fig.1 shows our procedure for combined vehicle/pavement studies. VESYM is a pitch/heave truck simulation package[10]. VESYS[12] is a viscoelastic flexible pavement simulation program developed at a number of institutions. Coupled to a data file describing a particular pavement structure and seasonal testing cycle, VESYS calculates the stresses and strains that tire load impresses on the pavement, then predicts the deterioration that results from years of repeated load application.

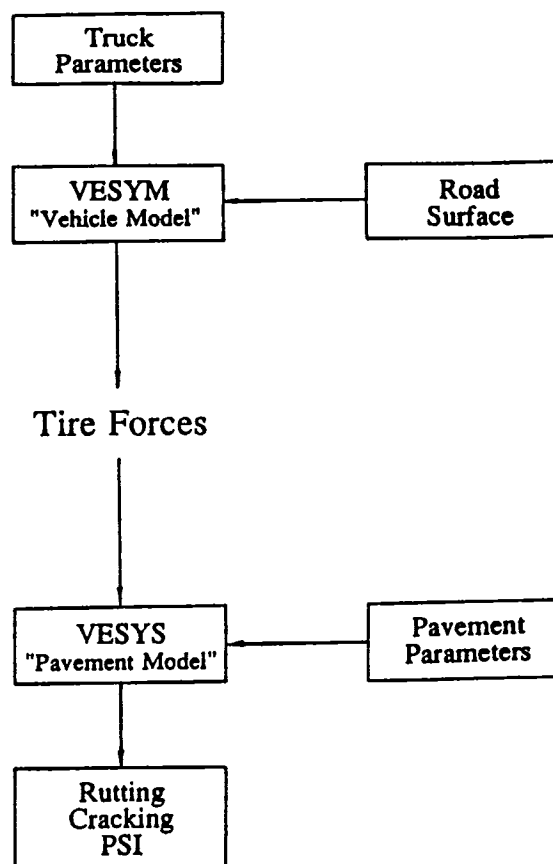


Fig.1 Flow Chart for Vehicle Studies

2. Vehicle Models and Road Input

2.1. Vehicle Models

The truck models are created to describe a vehicle's dynamic response to road inputs. These simulations calculate and store a vehicle's tire force history, which is then used by

VESYS to calculate road damage. Since the accuracy in predicting road damage directly depends on the tire force, accurate modeling of the dynamics is crucial. The vehicle simulation package VESYM was created based on the modeling characteristics shown below.

Modeling characteristics

- 1) The vehicle bodies are considered to be rigid bodies.
- 2) The vertical and pitch motions of the vehicle are small and roll is neglected.
- 3) The tires are assumed to behave as a linear spring-damper combination.
- 4) The tires are allowed to leave the road surface.
- 5.a) A nonlinear leaf spring model is used.
- 5.b) A nonlinear air spring model is developed and used.
- 5.c) A nonlinear shock absorber model is developed and used.

A five axle tractor/semi-trailer truck model with leaf spring suspensions is shown in Fig.2. The vehicle model in this study is broadly based upon the Navistar COF-9670 International Premium Tractor on a 142 inch wheelbase. Typical closed trailer and vehicle data are shown in Table 1.

Table 1 Vehicle Data
(five axle tractor semi-trailer)

Vehicle Speed	55	MPH
Sprung mass of Tractor	261.2	slugs
Tractor moment of inertia	5759.2	slugs ft ²
Sprung mass of Trailer	2223.0	slugs
Trailer moment of inertia	353457.0	slugs ft ²
Total vehicle length	48.7	ft
Tire stiffness		
single tire	61440.0	lbs/ft
dual tire	122880.0	lbs/ft
Average axle loadings		
steer axle	13273.	lbs
drive axle	18758.	lbs
trailer axle	18374.	lbs

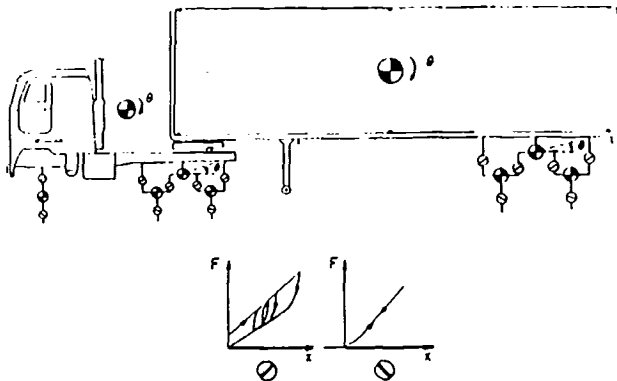


Fig.2 Five Axle Tractor/Semi-Trailer Model

Leaf Spring Suspension Model

All suspension systems contain two main elements: a spring component and a damper component. The suspension's main purpose is to filter out axle excitation before these disturbances reach the chassis. There are variety of different suspensions used on heavy vehicles; however, leaf spring suspension have grown more popular than others because leaf springs are less expensive, simpler and more reliable than other common suspensions.

A leaf is made up of laminated strips of curved steel or leaves. The two ends are supported by the chassis and the middle of the leaf spring is connected to the axle. As the leaf spring is compressed, the steel leaves bend acting as springs, and the leaves slide across each other dissipating energy through coulomb friction. A typical force-deflection curve is shown in Fig.3. The mathematical leaf-spring model used in this study is a modified version of the model developed by Fancher[13].

The equation used is

$$F_i = F_{env_i} + (F_{i-1} - F_{env_{i-1}}) e^{-|\delta_i - \delta_{i-1}| / \beta}$$

where

- F_i is the suspension force at the current simulation time step i
- F_{i-1} is the suspension force at the last simulation time step $i-1$
- δ_i is the suspension deflection at the current simulation time step
- δ_{i-1} is the suspension deflection at the last simulation time step
- F_{env_i} is the force corresponding to the upper boundary when δ is increasing (or the force corresponding to the lower boundary when deflection δ is decreasing) at δ_i , and
- β is the friction parameter which characterizes the rate at which the calculated force approaches the upper (or lower) boundary.

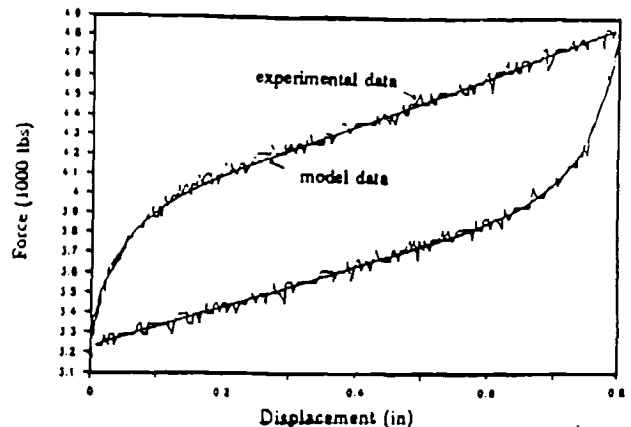


Fig.3 Leaf Spring Force-Deflection Curve [10].
(Experimental Versus Model Data)

Shock Absorber Model

The nonlinear shock absorber force pattern is shown in Fig.4. Its force characteristics are described in terms of velocity for compression and expansion. A mathematical model of nonlinear shock absorbers is represented as:

Compression

$$F_{sh,i} = \begin{cases} C_{com,l} \cdot v_{sh,i} & \text{for } 0 < v_{sh,i} < V_{thr,com} \\ C_{com,g} \cdot v_{sh,i} + C_{com,l} \cdot V_{thr,com} & \text{for } V_{thr,com} < v_{sh,i} \end{cases}$$

Rebound

$$F_{sh,i} = \begin{cases} C_{reb,l} \cdot v_{sh,i} & \text{for } V_{thr,reb} < v_{sh,i} < 0 \\ C_{reb,g} \cdot v_{sh,i} + C_{reb,l} \cdot V_{thr,reb} & \text{for } v_{sh,i} < V_{thr,reb} \end{cases}$$

where

$F_{sh,i}$ is the force created by the shock absorber at time step i

$v_{sh,i}$ is the velocity of the shock absorber at time step i

$C_{com,l}$ is the damping coefficient in compression for velocities smaller than $V_{thr,com}$

$C_{com,g}$ is the damping coefficient in compression for velocities greater than $V_{thr,com}$

$C_{reb,l}$ is the damping coefficient in rebound for velocities less than $V_{thr,reb}$

$C_{reb,g}$ is the damping coefficient in rebound for velocities greater than $V_{thr,reb}$

$V_{thr,com}$ is the threshold velocity in compression

$V_{thr,reb}$ is the threshold velocity in rebound.

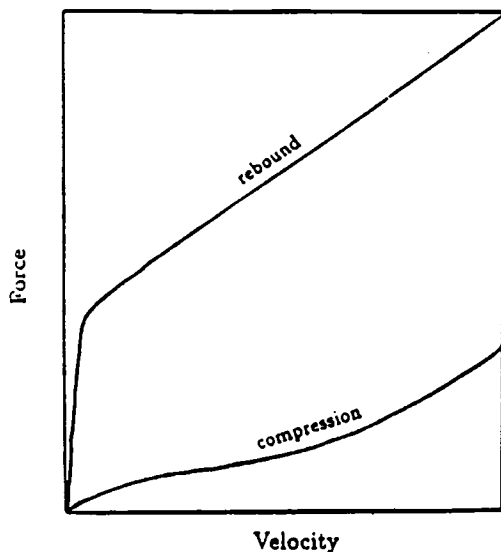


Fig.4 Shock Absorber Force Pattern [10].

2.2. Road Input Model

All roads have irregularities in the smoothness and flatness of their surface. As a vehicle traverses a road, it will be excited by these irregularities. To assure that the vehicle models generate accurate dynamics, the road model must describe all surface irregularities affecting the natural modes of a vehicle.

Second order filters and computer-generated random numbers are used to create the road models[10]. Fig.5 shows the slope PSD of the created road model.

The pavement used in this paper is a five inch thick, high grade asphalt surface course on a 15 inch base course of crushed stone. The material properties of the pavements were held constant and pavement data are shown in Table 2.

Table 2 Pavement Data
(the modulus of elasticity, psi)

Season	Asphalt	Base	Ground
Winter	1,604,300.	40,000.	4,500.
Spring	745,500.	30,000.	3,000.
Summer	152,400.	40,000.	4,500.
Fall	550,800.	40,000.	4,500.

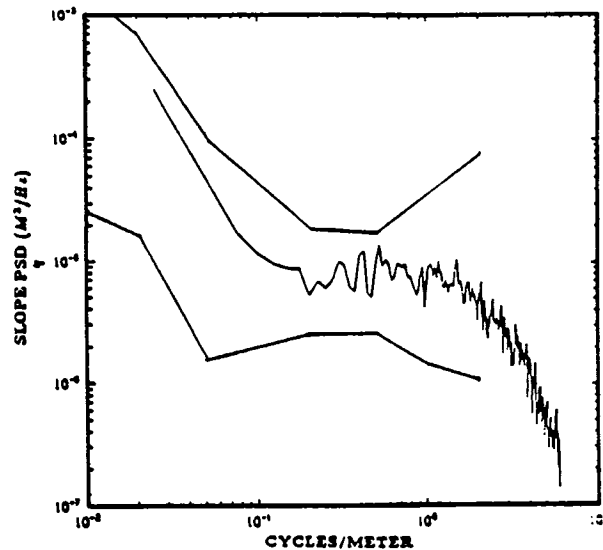


Fig.5 Slope PSD of generated roads[10]. Outlines show range of slope PSD values found in [14].

3. Results

The effects of heavy truck suspensions on pavement damage have been studied. In particular, the performance of active and semi-active suspensions are investigated. The dynamic effects of alternative suspensions are compared to an ideal suspension, i.e., one with no dynamic effect. The ideal suspension was taken to be a perfect suspension whose coefficient of variation of the tire force is zero. Pavement damage is calculated using the viscoelastic flexible pavement simulation package, VESYS.

In order to examine the dynamic characteristics of various suspensions, the Dynamic Load Coefficient(DLC, the coefficient of variation of the tire force) and the frequency distribution of the tire force are compared. The dynamic effect on pavement damage was compared by using dynamic equivalency factors, DEF.

The Dynamic Load Coefficient is defined by

$$DLC = \frac{\sigma}{L}$$

where σ is standard deviation of the tire forces about the mean and L is mean tire force.

The dynamic equivalency factors, DEF, are defined for cracking, rutting and present serviceability(PSI).

$$DEF = \frac{NF_{st}}{NF_{dy}}$$

where NF_{st} is the number of cycles to failure for a particular damage mode, with no vehicle dynamics, i.e. when the DLCs of the axles are held to zero. This case is referred to as the static case and NF_{dy} is the number of cycles to failure when the vehicle dynamics are considered, i.e. DLCs are not zero. This case is referred to as the dynamic case.

The failure criterion used are:

Rutting	Rutting Depth = 0.8 in.
Cracking	Area cracked = 50 %
Serviceability	PSI = 1.5

Present Serviceability Index (PSI) was developed by AASHO to quantify the pavement degradation[1].

The PSI is defined by:

$$PSI = 5.03 - 1.91 \log(1 + 10^6 \bar{SV}) - 0.01 \sqrt{C + P} - 1.38 \bar{RD}^2$$

in which

- \bar{SV} is the mean of the slope variance in the two wheel paths.
- $C + P$ is the area in square feet per thousand square feet of pavement, that exhibits cracking or surface patches.
- \bar{RD} is the mean rut depth in the two wheel paths.

It was found in the AASHO road tests that a new pavement had a PSI of about 4.2. This was equivalent to the mean slope variance of 1.7×10^{-6} with no rutting or cracking. Pavements were generally taken out of service at the road tests when the PSI dropped below 1.5.

Active and Semi-Active Suspensions

Recent research on the performance of active and semi-active suspensions for automotive vehicles[2,15] shows that

either ride quality or road holding i.e., tire force variations, can be improved significantly by using active/semi-active suspensions, and that semi-active suspensions can perform nearly as well as active suspensions. In this study, active and semi-active suspensions are investigated for application on five axle tractor/semi-trailer trucks with leaf spring suspensions to reduce pavement damage.

There are three modes, i.e., body mode, tire mode and short rocker mode in the tire forces of the leaf spring/short rocker suspensions. The body mode (near 2 Hz) and the tire modes (near 13 Hz) are dominant in the tire forces of the driving axles, and the body mode (near 2 Hz) and short rocker mode (near 9 Hz) are dominant in those of the trailer axles. Hence it is necessary to control both the axles and short rocker to reduce the tire force variations at all modes.

It was shown from the quarter car linear model study [2] that tire deflection, i.e., tire force, feedback control yields optimal road holding performance. Tire forces and the difference of the tire forces in the tandem axle are used as feedback signals for the active/semi-active suspensions in this study. Although measurements of tire forces are very difficult to make, they can be estimated from accelerometer and deflection sensors.

Fig.6 shows the leaf spring/short rocker suspension model for active/semi-active suspensions. Control forces for active and semi-active suspensions used in this paper are:

$$F_{act1} = -G1 \cdot (F_t - L)$$

$$F_{act2} = -G2 \cdot \Delta F_t$$

$$F_{semi-acti} = \begin{cases} F_{acti} & \text{if power} < 0 \text{ i.e., power dissipation} \\ 0 & \text{if power} > 0 \text{ i.e., power supply} \end{cases}$$

where

- F_{act1} is the control force for axles
- F_{act2} is the control force for short rockers
- $G1, G2$ are the feedback gains
- F_t is the tire force
- L is the mean tire force
- ΔF_t is the tire force difference in tandem axle

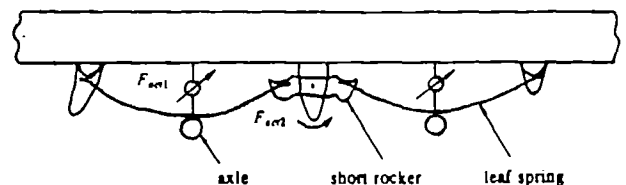


Fig.6 Leaf Spring/Short Rocker Suspension Model for Active and Semi-Active Suspensions.

The semi-active force is implemented by a shock absorber with a variable orifice. Thus, the semi-active forces are represented as:

$$F_{semi-act1} = -V1 \cdot \Delta \dot{d}$$

$$F_{semi-act2} = -V2 \cdot (\dot{\theta}_{sr} + \dot{\theta}_B)$$

where $V1, V2$ are damping rates, $\Delta \dot{d}$ is the suspension deflection rate, $\dot{\theta}_{sr}$ is the short rocker angular velocity and $\dot{\theta}_B$ is the body angular velocity.

For this representation, the semi-active force can be generated for $V_i \geq 0$. Thus V_i is modulated for $V_i \geq 0$.

Fig.7 shows the comparison of tire force power spectral densities. Significant improvements in reducing tire force variations at both the tire mode (13.5 Hz in driving axle) and the short rocker mode (7 Hz in driving axles, 9 Hz in trailer axles) can be achieved by using passive dampers on both the axles and short rocker. But the tire force variations at the body mode (near 2 Hz) can not be reduced by the passive dampers. As Fig.7 a), b) show, significant reduction of tire force variations at the body mode frequency can be obtained by using semi-active suspensions. A comparison of the tire force PSD of active and semi-active suspensions is shown in Fig.8. Fig.8 indicates that the spectral densities at low frequency range are similar in both cases. The high frequency components in active suspensions at above 20 Hz are due to active force. It is interesting to note the similarity between the two cases in Fig.8.

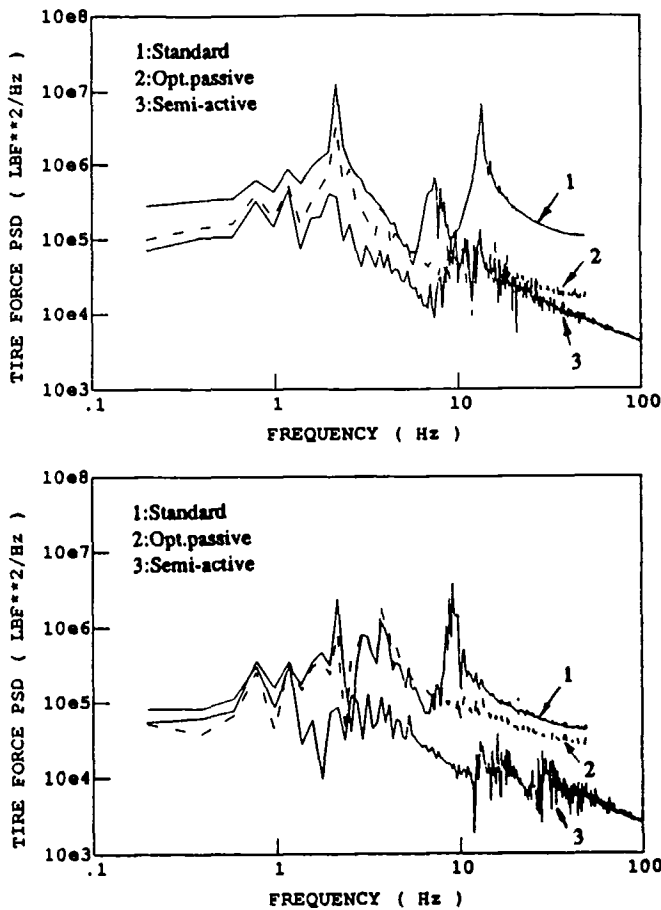
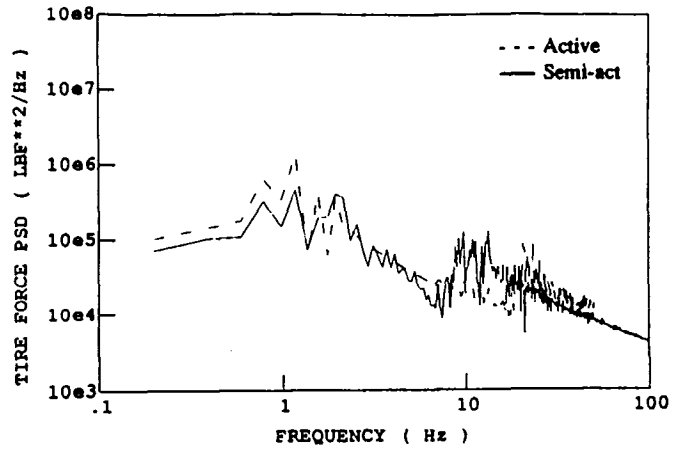
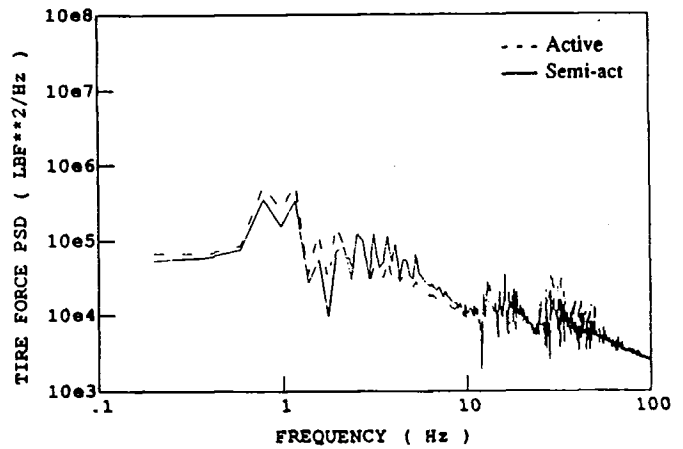


Fig.7 Tire Force PSD of Tandem Axle at 55 mph ($sv = 22 \times 10^{-6}$)
 — 1 : Passive Dampers on Axles (small damping)
 - - - 2 : Optimal Passive Dampers on both Axles and Short Rockers
 — 3 : Semi-Active Damper on both Axles and Short Rockers
 a) Comparison of Driving Axle Tire Force PSD
 b) Comparison of Trailer Axle Tire Force PSD



a) Driving Axle



b) Trailer Axle

Fig.8 Comparison of Tire Force PSD of Active and Semi-Active Suspensions at 55 mph ($sv = 22 \times 10^{-6}$)

Fig.9 indicates that excellent load equalization performance can be achieved by using a semi-active short rocker damper. The tire force variations at the short rocker mode (near 9 Hz in trailer axle) decreased significantly due to the decrease of the short rocker angle PSD at 9 Hz by short rocker control.

Fig.10 shows the comparison of the tractor sprung mass center vertical acceleration PSD. Low frequency components are similar in both semi-active and passive suspensions and are significantly decreased in the active case when compared to passive suspensions. High frequency components are increased in both active and semi-active suspensions when compared to passive suspensions. This is due to the sharp peaks and troughs in acceleration time histories due to active control forces.

Fig.11 indicates that suspension deflection of active suspensions is much larger than that of the semi-active and passive suspensions. This implies that the decrease of the tire force variations can be achieved at the expense of greatly increasing the suspension deflection in active suspensions. This is a drawback of an active suspension based on tire force feedback control. The suspension deflection of semi-active suspensions is relatively small compared to active suspensions and the peaks of the sprung mass acceleration is higher than that of active suspensions because of the small suspension deflection in semi-active suspensions.

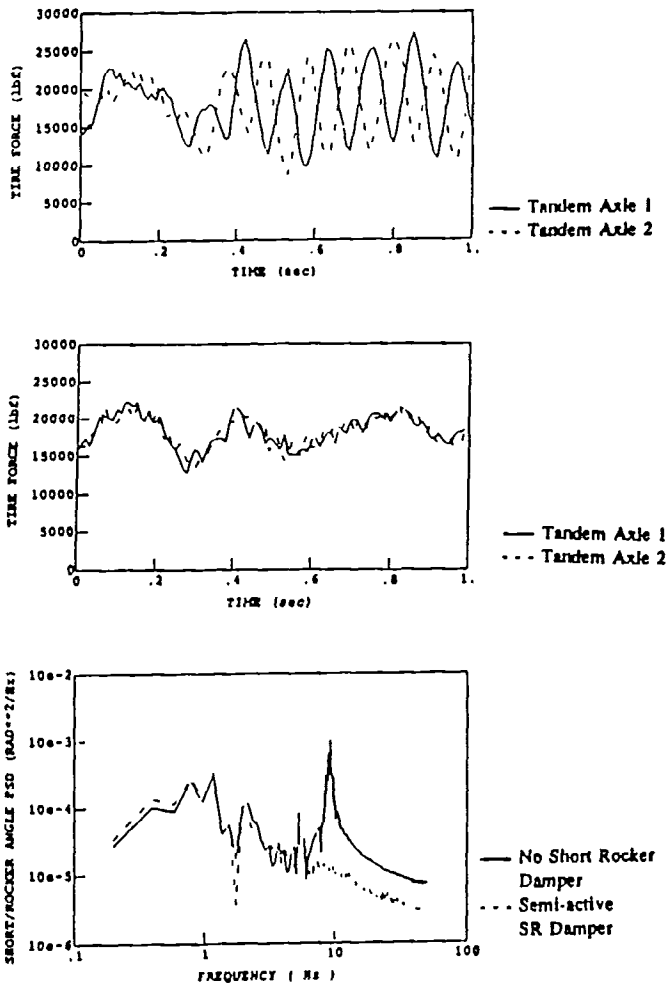


Fig.9 The Effect of Semi-Active Short Rocker Damper at 55 mph ($sv = 22 \times 10^{-6}$)

a) No Short Rocker Damper
 — Trailer Tandem Axle 1
 - - - Trailer Tandem Axle 2

b) Semi-Active Short Rocker Damper

c) Short Rocker Angle PSD (Trailer Axle)
 — Standard Dampers on Axles and No Short Rocker Damper
 - - - Semi-Active Short Rocker Damper

Fig.12 indicates that significant improvements towards reducing pavement damage can be achieved by using either active or semi-active suspensions, and that semi-active suspensions can provide substantial improvements over an optimal passive suspension. The significant reduction in tire force variations, i.e., the reduction in pavement damage, has been achieved at the expense of increasing the high frequency sprung mass acceleration. This is shown in Fig.10, where low frequency components in the sprung mass acceleration are decreased and high frequency components are increased.

As the feedback gains $G1, G2$ increase, DLCs, i.e., tire force variations decrease, and the peaks in sprung mass acceleration increase in semi-active suspensions. Hence an optimized compromise between tire force variations and ride quality is necessary in the design of semi-active suspensions.

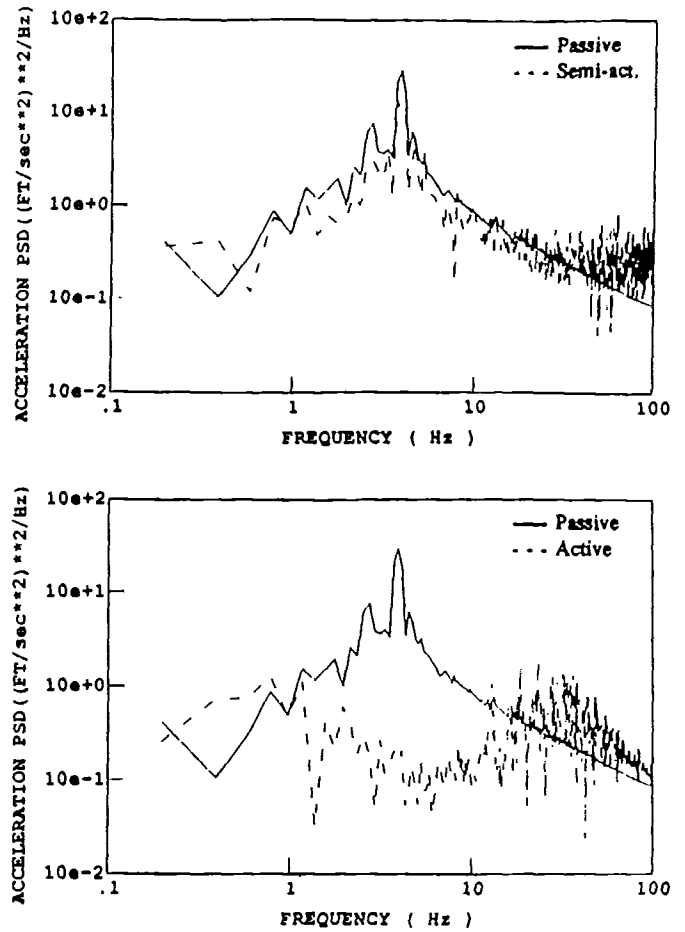


Fig.10 Tractor Spring Mass CG Vertical Acceleration at 55 mph ($sv = 22 \times 10^{-6}$)

a) Comparison between Semi-Active and Passive Suspension

b) Comparison between Active and Passive Suspension

Active suspensions require sensors such as accelerometers, deflection sensors and velocity transducers, as well as a controller and high power actuators with good frequency response at high frequency, requiring a relatively large power supply. However, semi-active suspensions do not require high power actuators or a large power supply because the sensors and actuators operate at low power when modulating the damping rate of shock absorbers. Semi-active suspensions are less expensive than active suspensions and relatively robust to gain variations when compared to active suspensions.

Effect of Damping Rate Limitation and ON-OFF Control in Semi-Active Suspensions

It was shown previously in this paper that semi-active suspensions with continuously modulated dampers can provide substantial improvements over passive suspensions. The effect of the maximum damping rate in semi active suspensions was investigated and the the performance of ON-OFF semi-active suspensions was compared with that of a continuously modulated semi-active suspension.

If the maximum damping ratio of a commercially available heavy truck shock absorber is assumed to be 0.8, this is

approximately equivalent to a damping rate of 6630 lbs/(ft/sec). When there is a limit in the damping rate, the semi-active forces of a continuously modulable damper are represented as:

$$F_{semi-act1} = \begin{cases} -V1_{max} \cdot \Delta d & \text{if } G1 \cdot (F_t - L)/\Delta d \geq V1_{max} \\ -V1 \cdot \Delta d & \text{if } 0 < G1 \cdot (F_t - L)/\Delta d < V1_{max} \\ 0 & \text{if } G1 \cdot (F_t - L)/\Delta d \leq 0 \end{cases}$$

For this representation, the semi-active forces are generated for $0 \leq V1 \leq V1_{max}$.

The control forces of ON-OFF semi active suspensions used in this paper are represented as:

$$F_{semi-act1} = \begin{cases} -V1_{max} \cdot \Delta d & \text{if } G1 \cdot (F_t - L)/\Delta d \geq V1_{crit} \\ 0 & \text{if } G1 \cdot (F_t - L)/\Delta d < V1_{crit} \end{cases}$$

i.e., $V1$ takes 0 or $V1_{max}$. For this control logic, $V1_{crit}$ has an effect on both the DLC, i.e., tire force variations, and the sprung mass acceleration. As $V1_{crit}$ becomes smaller for a given $V1_{max}$, DLCs decrease and the peaks in sprung mass acceleration increase. Thus a trade-off between tire force varia-

tions and ride quality is necessary in the design of ON-OFF semi-active suspensions. A "conventional" ON-OFF system is when $V1_{crit}$ is 0.

Fig.13 shows the DLC ratio and PSI equivalency factor ratio comparing the continuous and ON-OFF semi-active suspensions. The results for ON-OFF semi-active suspensions were obtained for $V1_{crit} = V1_{max}/2$. The DLC ratio is defined by

$$DLC \text{ ratio} = \frac{DLC \text{ of semi-active suspensions}}{DLC \text{ of passive suspensions}}$$

The equivalency factor ratio is calculated by the same method. As can be seen in Fig.13, both ratios decrease as the maximum damping rate increases but the difference is not significant. It is interesting to note that the performance of ON-OFF semi-active suspensions is similar to that of continuously modulated semi-active suspensions and is even better when the maximum damping rate is large. It should be noted that the better performance of ON-OFF semi-active suspensions was achieved at the expense of increasing the peaks in sprung mass acceleration.

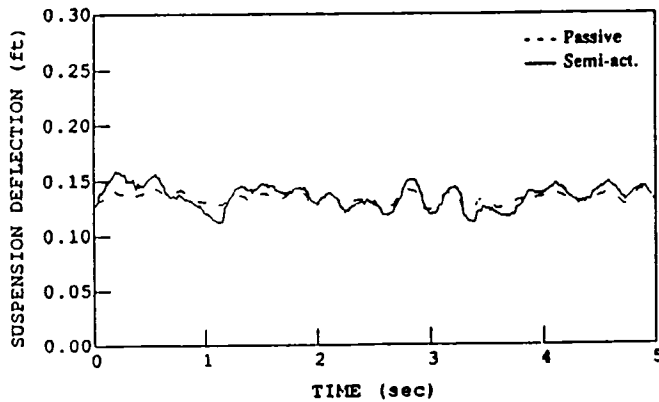
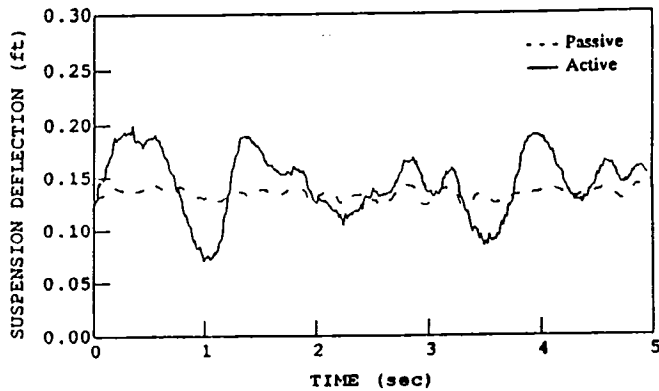


Fig.11 Suspension Deflections at 55 mph ($sv = 22 \times 10^{-6}$)
a) Comparison between Active and Passive Suspension
b) Comparison between Semi-Active and Passive Suspension

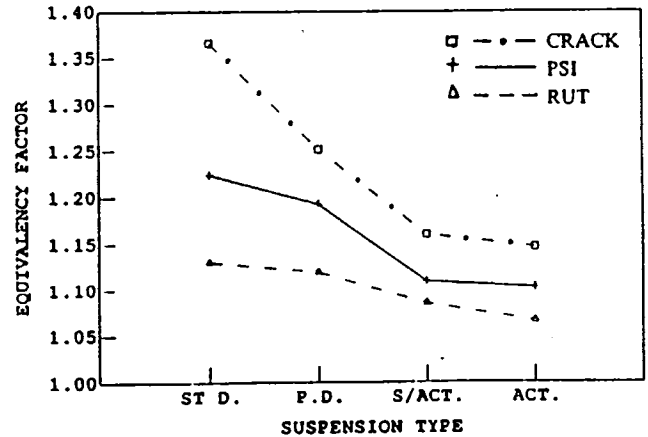
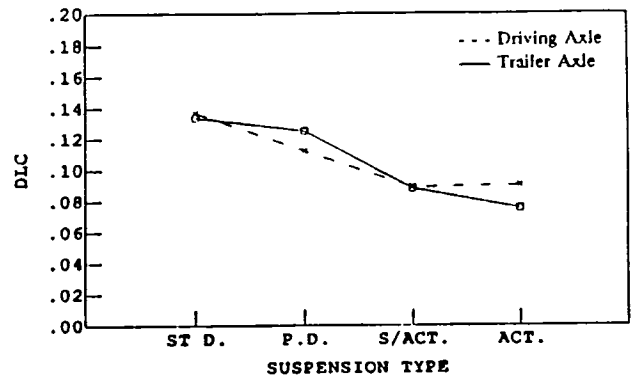


Fig.12 Effect of Active and Semi-Active Suspension on Vehicle Response and Road Damage
a) DLC versus Suspension Type (55 mph, $SV = 22 \times 10^{-6}$)
b) Equivalency Factor versus Suspension Type
ST D. : Standard Damper
P.D. : Optimal Passive Damper
S/ACT.: Semi-Active Damper
ACT. : Active Suspension

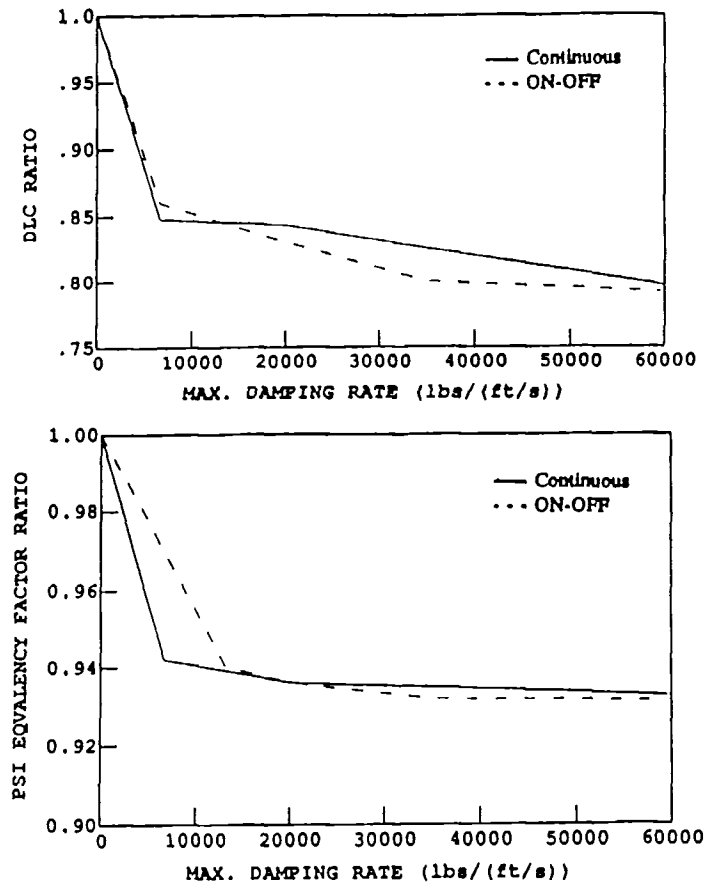


Fig.13 Effect of Maximum Damping Rate
 — continuous semi-active suspension
 - - - ON-OFF semi-active suspension
 a) DLC RATIO versus Maximum Damping Rate (55 mph, $SV = 22 \times 10^{-6}$)
 b) Equivalency Factor RATIO versus Maximum Damping Rate

4. Conclusions

The performance characteristics of active and semi-active suspensions have been investigated to reduce pavement degradation due to vehicle dynamics. In addition, the effects of the maximum damping rate on the performance of semi-active suspensions was investigated.

It was shown that tire force variations at all modes, i.e., the low frequency body mode, the high frequency tire mode and the short rocker mode can be reduced by semi-active suspensions. Active and semi-active suspensions can significantly reduce pavement damage when compared with optimal passive suspensions and the performance of semi-active suspensions is nearly as good as active suspensions. The improvements in the reduction of pavement damage increase as the maximum damping rate increases but the difference is not significant. The performance of ON-OFF semi-active suspensions was similar to that of continuously modulated ones. A compromise between the improvements in the reduction of tire force variations and ride quality is necessary in the design of semi-active suspensions because the reduction of pavement damage has been achieved at the expense of increasing the high frequency components in sprung mass acceleration.

References

1. The AASHO Road Test, Report 5, "Pavement Research", Highway Research Board Special Report 61E, National Academy of Science, National Research Council Publication 954, 1962.
2. T. Butsuen, "The Design of Semi-Active Suspensions for Automotive Vehicles", Ph.D. Thesis submitted to the Dept. of Mechanical Engineering, Massachusetts Institute of Technology, June, 1989.
3. Sweatman, P.F., "A Study of Dynamic Wheel forces in Axle Group Suspensions of Heavy Vehicles", Australian Road Research Board, Special Report No.27.
4. Sayers, M. and Gillespie, T.D., "Dynamic Pavement/Wheel Loading for Trucks with Tandem Suspensions", 8th IAVSD Symposium on the Dynamics of Vehicles, Cambridge, MA, USA, August, 1983.
5. S.O'Connell, E. Abbo, K. Hedrick, "Analyses of Moving Dynamic Loads on Highway Pavements: Part 1: Vehicle Response", Proceedings, International Symposium on Heavy Vehicle Weights and Dimensions, Kelowna, British Columbia, Canada, June, 1986.
6. J.K. Hedrick and K. Yi, "The Effect of Alternative Heavy Truck Suspensions On Flexible Pavement Response", Proceedings, Second International Symposium on Heavy vehicle Weights and Dimensions, Kelowna, British Columbia, Canada, June, 1989.
7. E. Abbo, K. Hedrick, M. Markow, B. Brademeyer, "Analysis of Moving Dynamic Loads on Rigid Pavements", Conference Proceedings, Heavy Truck Suspension Design, Canberra, Australia, March 1987.
8. D. Cebon, "An Investigation of the Dynamic Interaction Between Wheeled Vehicles and Road surfaces", Ph.D. Thesis submitted to the department of Engineering, University of Cambridge, 1985.
9. Warburton, Dave J., "Parametric Studies of the Pavement Damage Caused By Tractor Semi-Trailer Trucks", S.M. Thesis, Dept. of Mechanical Engineering, Massachusetts Institute of Technology, January, 1988.
10. J.K. Hedrick, et al., "The Simulation of Vehicle Dynamic Effects on Road Pavements", Final Report, USDOT Office of University Research/FHWA, June 1989.
11. OECD Road Research Group, "Road Research: Impact of Heavy Freight Vehicles", Paris, France: OECD, 1983, pp 58-77, 154.
12. "Predictive Design Procedures VESYS Users Manual", Federal Highway Administration, Project 5C, January, 1978.
13. Fancher, P.S., et al., "Measurement and Representation of the Mechanical Properties of Truck Leaf Springs", SAE Paper No. 800905, August, 1980.
14. Sayers, M., "Characteristic Power Spectral Density Functions for Vertical and Roll Components of Road Roughness", Symposium on Simulation and Control of Ground Vehicles and Transportation Systems. Ed. L. Segal, et al. New York, ASME, 1986
15. C. Yue, T. Butsuen, and J.K. Hedrick, "Alternative Control Laws for Automotive Active Suspensions", Proceedings, American Control Conference, Atlanta, GA, June 15-17, 1988.

Positions and opinions advanced in this paper are those of the author(s) and not necessarily those of SAE. The author is solely responsible for the content of the paper. A process is available by which discussions will be printed with the paper if it is published in SAE Transactions. For permission to publish this paper in full or in part, contact the SAE Publications Division.

Persons wishing to submit papers to be considered for presentation or publication through SAE should send the manuscript or a 300 word abstract of a proposed manuscript to: Secretary, Engineering Activity Board, SAE.

Printed in U.S.A.