THE EFFECT OF ALTERNATIVE HEAVY TRUCK SUSPENSIONS

ON FLEXIBLE PAVEMENT RESPONSE

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Abstract

In this paper, the dynamic effects of heavy vehicle suspensions are investigated by analyzing parametric studies performed using previously developed simulation tools. In particular, the truck simulation package , VESYM, and the flexible pavement simulation package, VESYS, are used to look at the influence of alternative suspension types, e.g., walking beam, leaf spring/short rocker, air spring and semi-active shock absorbers on pavement response.

The intent of the study is to determine differences between suspension types as well as to look at the sensitivity to parameter changes and optimization within a particular suspension type.

The simulation results indicate that substantial improvements in pavement longevity may be possible by careful suspension selection and optimization.

1. Introduction

It has long been recognized that pavement damage due to vehicle loading is largely due to heavy vehicles. The "fourth power" regression equations relating axle load to pavement damage was primarily due to the AASHO Road Tests [1]. In order to determine how important the dynamic component of the total tire force is when considering pavement damage a number of analytical and experimental studies have been recently performed [2-10]. Results have been published indicating that the dynamic component can account for anywhere from 2 - 50 % [11] of the road damage.

This paper examines the dynamic effects of heavy truck suspensions including active and semi-active suspensions on pavement response. The motivation for this research is to assist in the design of advanced heavy truck suspensions in order to extend the life of the interstate highway system, and to reduce its maintenance cost.
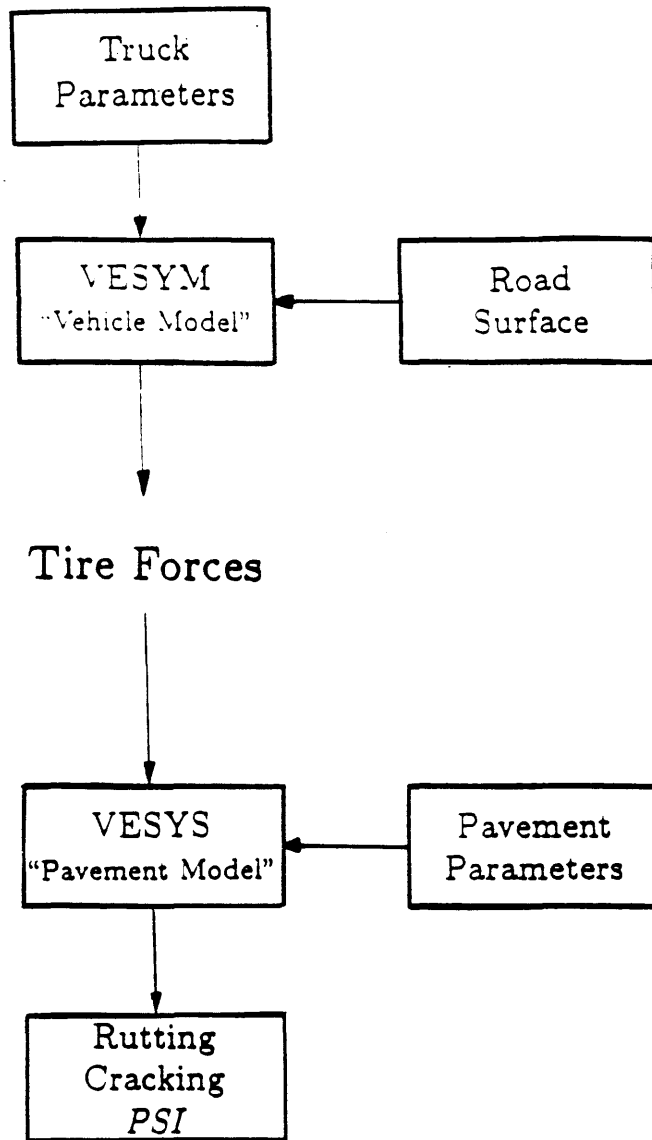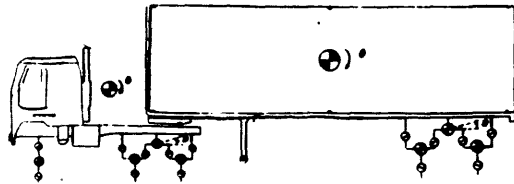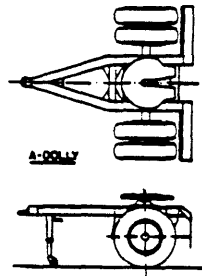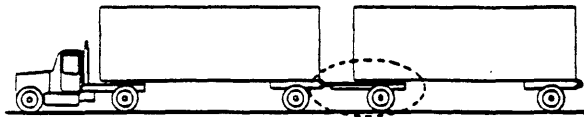
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        Tire Forces

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        ┌─────────────┐              ┌─────────────┐
        │   VESYS      │◄─────────────│  Pavement   │
        │"Pavement Model"│             │ Parameters  │
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        ┌─────────────┐
        │   Rutting    │
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Fig.1   Flow Chart for Vehicle Studies

(a)

A-DOLLY

(b)

Fig.2   Vehicle Types
    a) Tractor semi-trailer
    b) Multiple Trailer

Our study consists of two parts: vehicle analysis and the evaluation of the dynamic effects on pavement damage. Figure 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 the tire load impresses on the pavement, then predicts the deterioration that results from years of repeated load application.

2. Vehicle Models

The truck models are created to describe a vehicle's dynamic response to a road input. These simulations calculate and store a vehicle's tire force history which is used by VESYS for calculating road damage. Since the accuracy in predicting this road damage directly depends on the tire force, accurate modeling of the dynamics is crucial. The vehicle simulation package VESYM is 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.

Fig.2 shows the vehicle types modeled in VESYM.

Tire Model

Since tires are the only part of a vehicle that are in direct contact with the road, the dynamics of all other vehicle components must pass through the tires before being felt by the pavement. Thus, the tire is a critical part of a vehicle's dynamics and must be accurately modeled to obtain reliable road damage results.

For many computer applications, the point contact tire model, a linear spring and damper combination, will suitably describe the tire (Fig.3 a)). This is true because the typical force pattern of a rotating tire is close to linear through small displacements, as shown in Fig.3 b).
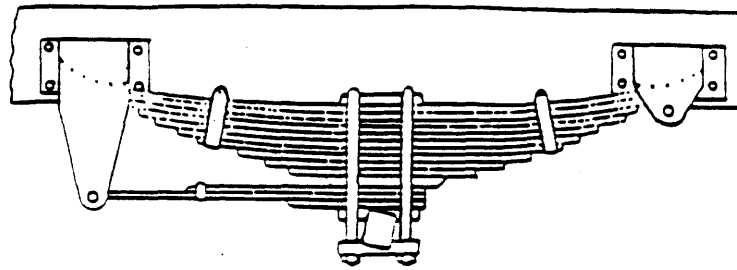
(a)
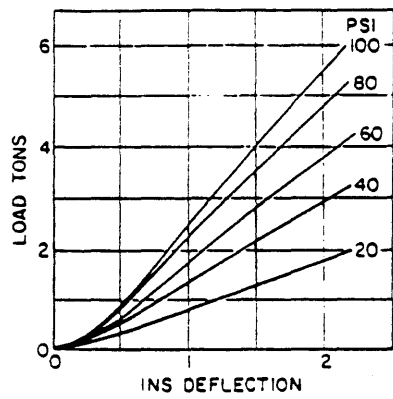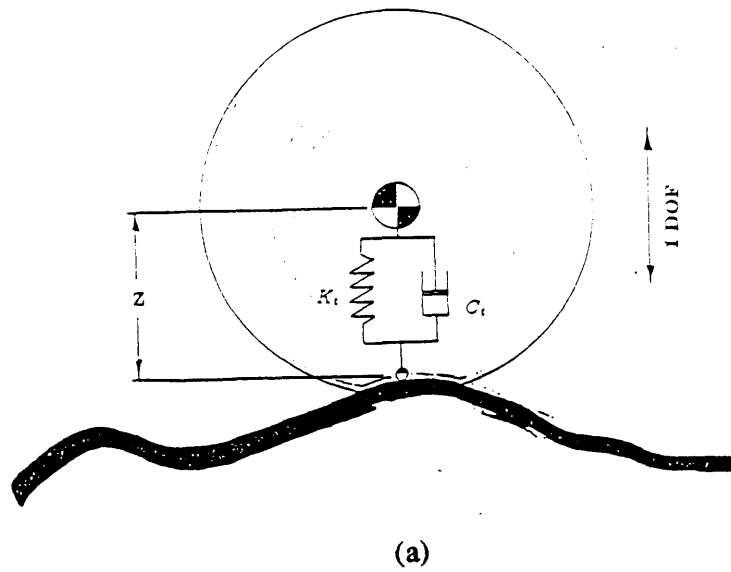
11.00 × 20 tire deflection data over a range of pressures.
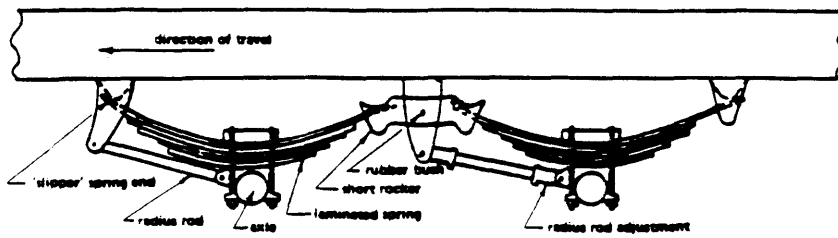
(b)

Fig.3   Point Contact Model and Force Deflection Curve [10]
        a) The tire modelled as a linear spring and a point follower
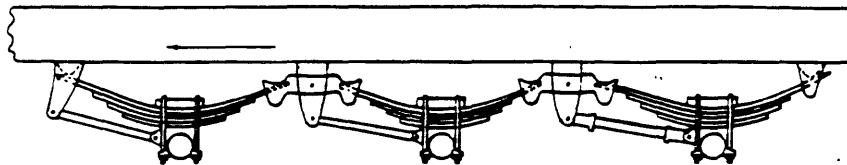   b) The normal deflection curve of an 11.0 × 20 tire

(a)

(b)

(c)

Fig.4   Leaf Spring Suspension [2].
a) Single Axle Leaf Spring
b) Tandem Leaf Spring/Short Rocker
c) Tridem Leaf Spring/Short Rocker

The tire model used for this research is the point contact model. Simplicity and computing speed are its main advantages. For the flexible pavements (broad-band input) examined in this study, the simple point contact model was found to be adequate. It is assumed that there are no sharp discontinuities such as joints or potholes. The manner in which the dynamic pavement loads are distributed at the road surface is also determined by the tire. In this study, the tire force is assumed to be uniformly distributed over a circular contact patch.

Suspension Model

All suspension systems contain two main ingredients, a spring component and a damper component. The suspension's main purpose is to filter out the axle excitation before these disturbances reach the chassis.

There are a variety of different suspensions used on heavy vehicles. However, some types of suspensions have grown more popular than others. In the truck industry the overwhelming majority are leaf springs. Leaf springs are less expensive, simpler and more reliable than any other common suspension. In addition they act as both spring and damper simultaneously, thus, reducing or eliminating the need for independent shock absorbers. The second most common type is the air-spring suspension.

The suspensions modeled in VESYM are:

Leaf Spring Suspension.
a) Single axle leaf spring.
b) Tandem Leaf Spring/Short rocker.
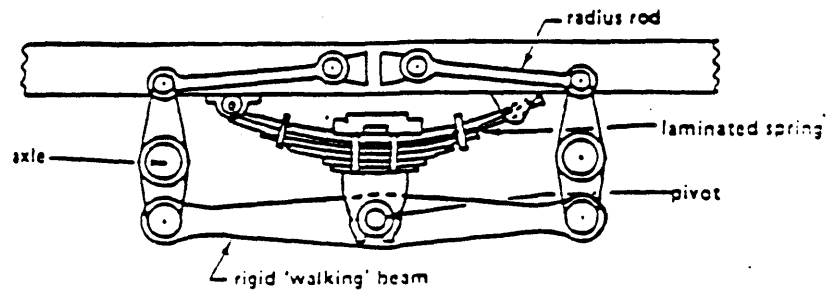c) Tridem Leaf Spring/Short Rocker.

Tandem Walking Beam Suspension.

Air Spring Suspension.
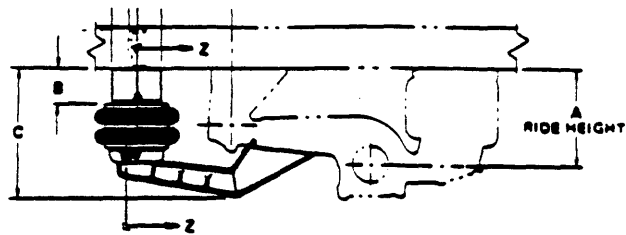a) Single Axle Air Spring Suspension.
b) Tandem Air Spring suspension.

Fig.4 and Fig.5 shows the suspension geometries.

Leaf Spring Model

A leaf is made up of laminated strips of curved steel or leaves. The two ends are supported by the chassis and 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.6. The mathmatical leaf-spring model used in this study is a modified version of the model developed by Fancher[13].

(a)

(b)
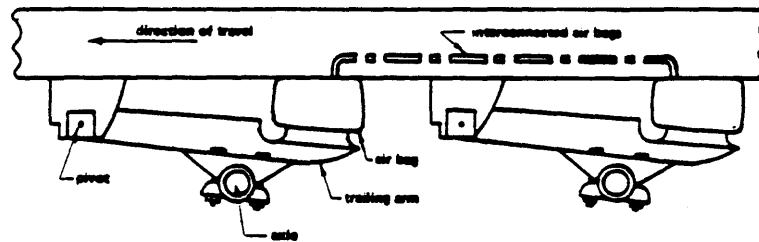
(c)

Fig.5   Walking Beam Suspension and Air Suspension [2].
         a) Tandem Walking Beam Suspension
         b) Single Axle Air Spring Suspension
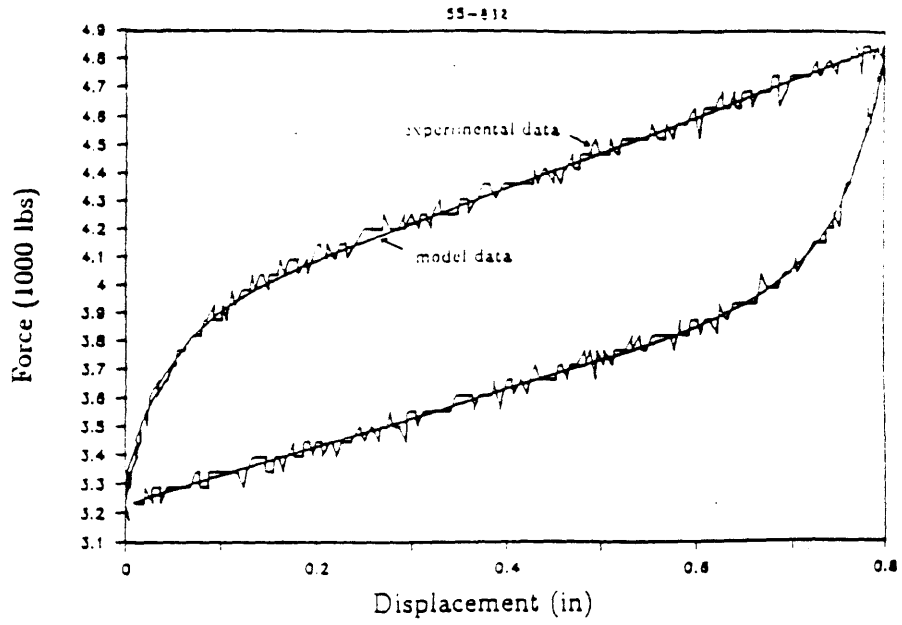   c) Tandem Air Spring Suspension

Fig.6   Leaf Spring Force-Deflection Curve [10].
(Experimental Versus Model Data)

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 increaseing (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.

Air Spring Suspension

The air spring is also a nonlinear spring, and works off the compression of air. However, unlike the leaf spring it does not develop large amounts of internal friction to release energy into the environment. In addition the force is much more of a determinate quantity than path traced by the compression and rebound force curves of a leaf spring.

Air springs are used in single, tandem or tridem axle configurations. In the latter two cases there is an air delivery system, which allows air to move between the air springs, faciliating load sharing between axles. While the air itself is a source of damping, vehicle vibrations are damped primarily by the shock absorbers, which are mounted in parallel with the air springs.

The air-bags are generally equipped with height control valves. The primary purpose of these valves is to maintain the proper spacing between the vehicle frame and the axle by adjusting the pressure in the air spring in response to vehicle loading. In order to achieve this result without using excessive amounts of air, a time delay is incorporated in the design, which prevents the valve from functioning during a momentary change of axle to frame spacing which may occur while the vehicle traverses rough pavements. The delay is longer than one second, which is slower than the slowest vehicle vibration mode. Thus in a dynamic analysis the air in the air-bags may be treated as a closed system. In a dynamic analysis the air is continually being compressed and expanded at rates corresponding to the natural frequencies of the vehicle. The air in the bag does not have time to exchange heat with the surroundings because the heat transfer process has a relatively long time constant, so the gas compression/expansion process may be assumed to be adiabatic.

Shock Absorber Model

The shock absorber is a relatively standard piece of equipment used on some trucks and many buses. Its force characteristics are described in terms of velocity for compression and expansion. Fig.2.6 shows the shock absorber force pattern. The non-linear shock absorber model is described below:

Compression

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

Rebound

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

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 $V_{thr,com}$

$C_{com,l}$ 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 threshhold velocity in compression

$V_{thr,reb}$ The threshhold velocity in rebound.

3. 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 the road irregularities. To assure that the vehicle models will generate accurate dynamics, the road model must describe all surface irregularities affecting the natural modes of a vehicle.

Second order filters are used to create the road models[10]. The transfer function of a second order filter can be written as follows:
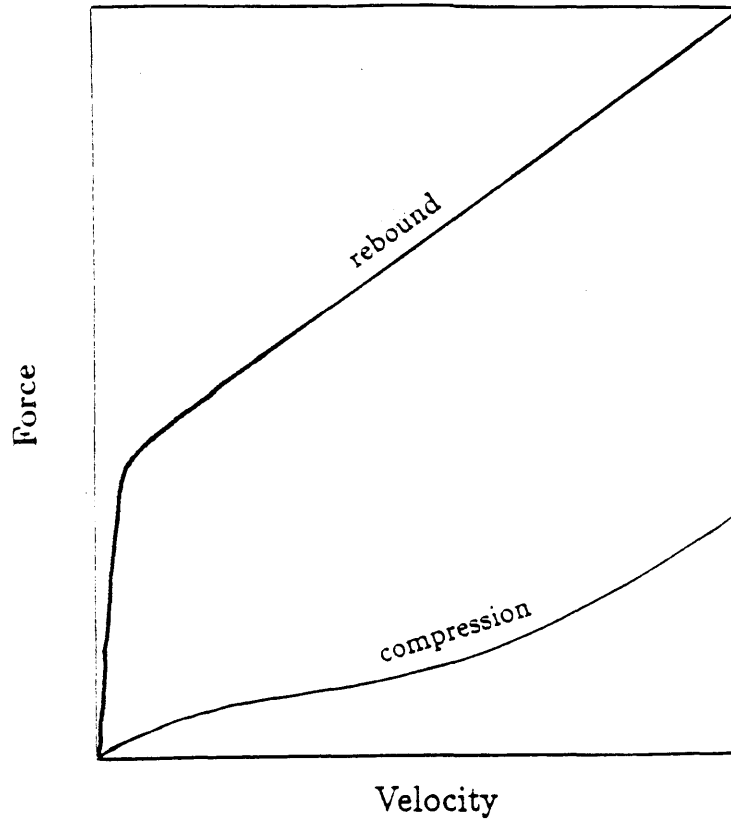
Fig.7   Shock Absorber Force Pattern [10].

$$G(s) = \frac{A\ F(s)}{s^2 + 2\,\delta\,\gamma\,s + \gamma^2}$$

Since the road is a stationary random process and assuming no initial conditions (for the road model), Laplace transform theory states that:

$$s^n\ G(s) = \frac{\partial^n}{\partial x^n}\ g(x)$$

By transforming the filter into the space domain the following equation is derived:

$$g_{fl}(x) = \ddot{y}(x) = A\ R(x) - 2\,\delta\,\gamma\,\dot{y}(x) - \gamma^2\,y(x)$$

$$R(x) = \sqrt{\frac{12}{\Delta x}}\ \left(r_n - \frac{1}{2}\right)$$

where

x is the horizontal road distance

$g_{fl}(x)$ is the road height for a flexible road

A is gain

δ is damping

γ is the cut off wavenumber(*rad/ft*)

$R(x)$ is a random function of uniform distribution

Δx is the horizontal displacement step size

r_n is a computer-generated random number of uniform probability density between 0 and 1, and a trapezoidal method is used for integration

The three parameters present in the filter A, δ, and ω determine the shape of the slope PSD curve. However, a single filter's effect on the slope PSD is limited to a specific wave number range independent of the shaping parameters. It is found that this range does not cover the entire wave number range important to the natural modes of heavy vehicles.

In order to create a road with a correct power distribution in the slope PSD plot, it is necessary to superimpose three seperate road profile with different cutoff frequencies and gains. Using this method allows much more freedom than with a single filter. Fig.8 shows the slope PSD and the profile of the created road model.
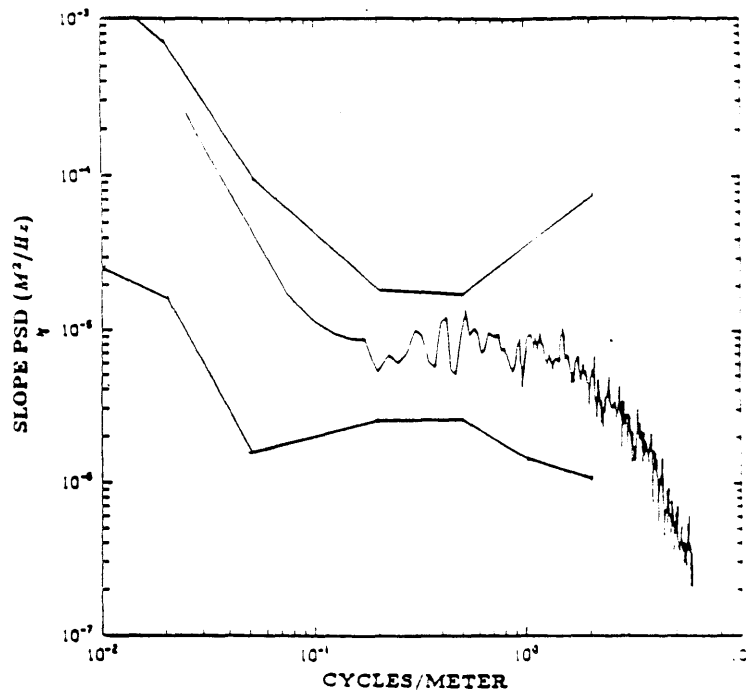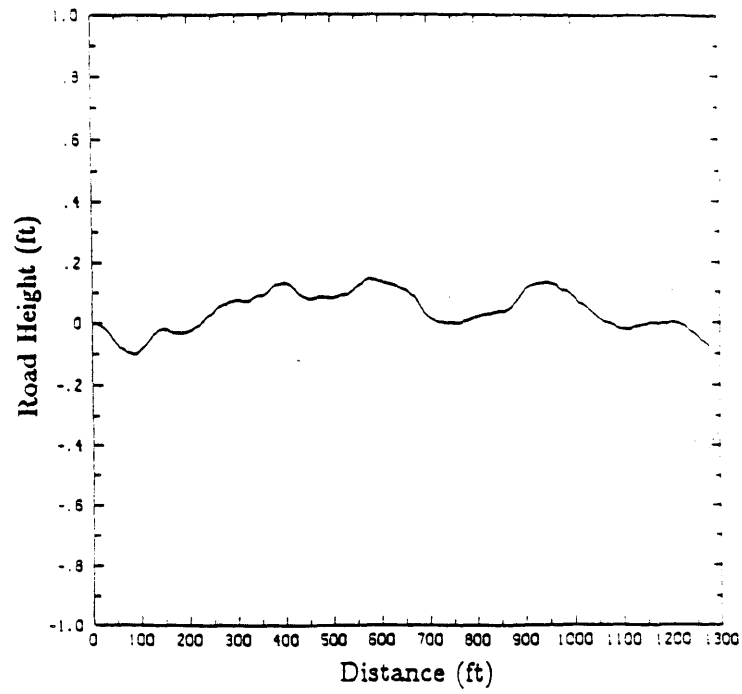
Fig.8   Road Plots [10].

a) Road profile, height versus length

b) Slope PSD of superimposed roads. Outlines show range of slope PSD values found in [14].

4. Results

The effect of the suspension stiffness,K, shock absorber damping and suspension type have been studied. In particular, active and semi-active suspensions are investigated.

The dynamic effects of alternative suspensions are compared to an ideal suspension, that is, 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.

The vehicle model in this study is broadly based upon the Navistar COF-9670 International Premium Tractor on a 142 inch wheelbase, and a typical closed trailer and vehicle data are shown in Table 1. 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 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.o	lbs/ft
dual tire	122880.0	lbs/ft
Average axle loadings		
steer axle	13273.	lbs
drive axle	18758.	lbs
trailer axle	18374.	lbs

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.

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 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 DLC's 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. DLC's 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:

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

in which

$\overline{SV}$ 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.

$\overline{RD}$ 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. Pavement were generally taken out of service at the road tests when the PSI dropped below 1.5.

Effect of Suspension Stiffness, K

The effect of varying the average leaf-spring stiffness,(K1+K2)/2, while holding K1/K2 constant on vehicle response and pavement damage is shown in Fig.9. Fig.9 (a) shows that increasing the spring constant increases the DLC. Fig.9 (b) shows that the tire mode (approximately 10 Hz) in the tire force is significantly increased as the spring constant increases and the body mode in the tire force (approximately 2 Hz) is dominant when the spring is soft.

Fig.9 (c) shows that the variation of the spring constant produced only mild changes in the PSI and the rutting and significant changes in the cracking, because cracking is highly dependent on the maximum of the tire force.

Effect of Shock Absorber Damping

The effect of shock absorber damping on leaf spring suspensions was examined. As expected, DLC's decrease with damping. Fig.10 (a) shows that there is an optimal damping for a minimum DLC and increasing damping more than this optimal value causes the DLC to increase. The DLC decreases 31.0% (from 0.184 to 0.127) at $\overline{SV} = 22 \times 10^{-6}$ with the optimal shock absorber damping. Fig.10 (b) shows that in both the body and the tire modes the tire forces are attenuated with damping. The PSI decreases by about 3.%, cracking by 4.% and rutting by 3.% with the optimal shock absorber damping.

Fig.10 (c) shows that the presence of shock absorbers on a leaf spring suspension reduced the pavement damage, but there is a limit in the improvement of life.

Effect of Suspension Type

The effect of suspension type with typical parameters is shown in fig.11. The walking beam leaf-spring suspension has the highest DLCs, the air spring suspension the lowest, and leaf spring/short rocker suspension is inbetween. The DLC for the walking beam suspension is 35.% greater and the DLC for the air spring suspension 20.% smaller when compared to the leaf spring suspension.
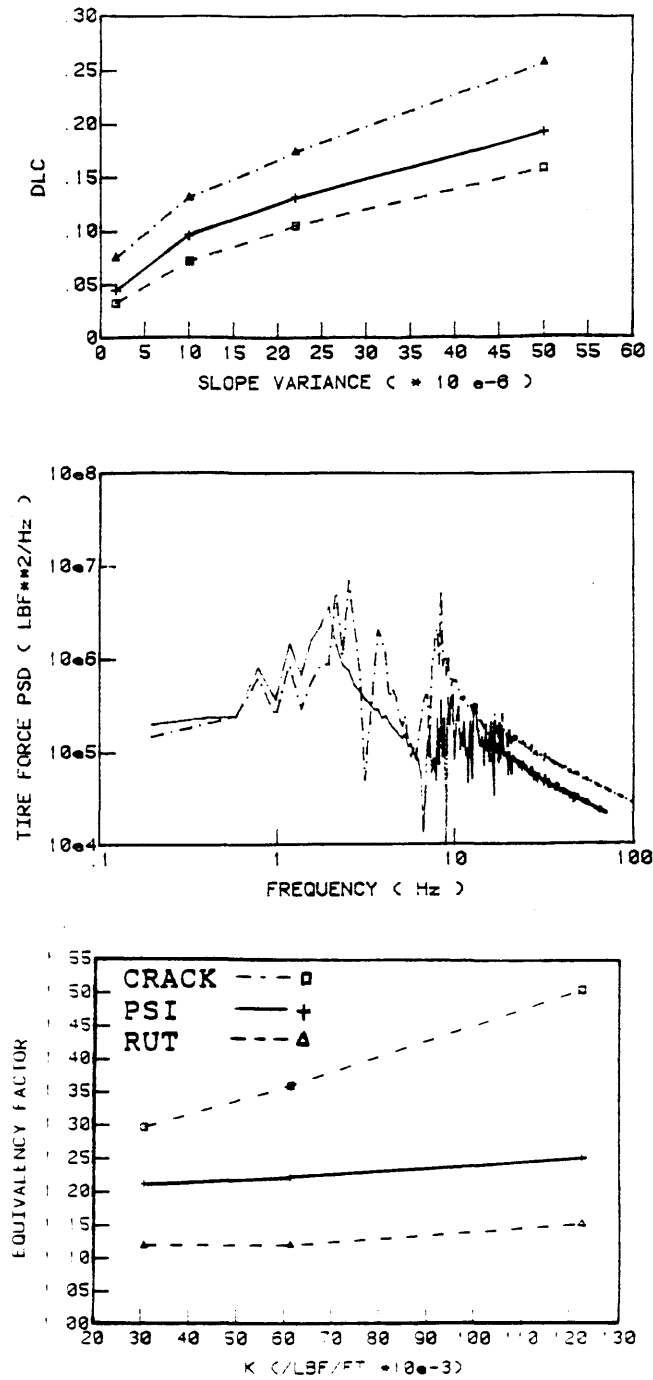
Fig.9 Effect of Spring Stiffness on Vehicle Response and Road Damage
a) Leading Drive Axle DLC versus SV
    □ — — — K = 30,600 lbs/ft
    + ————— K = 61,200 lbs/ft (standard)
    △ — · — K = 122,400 lbs/ft
b) Leading Drive Axle Tire Force PSD at SV = 22 × 10⁻⁶
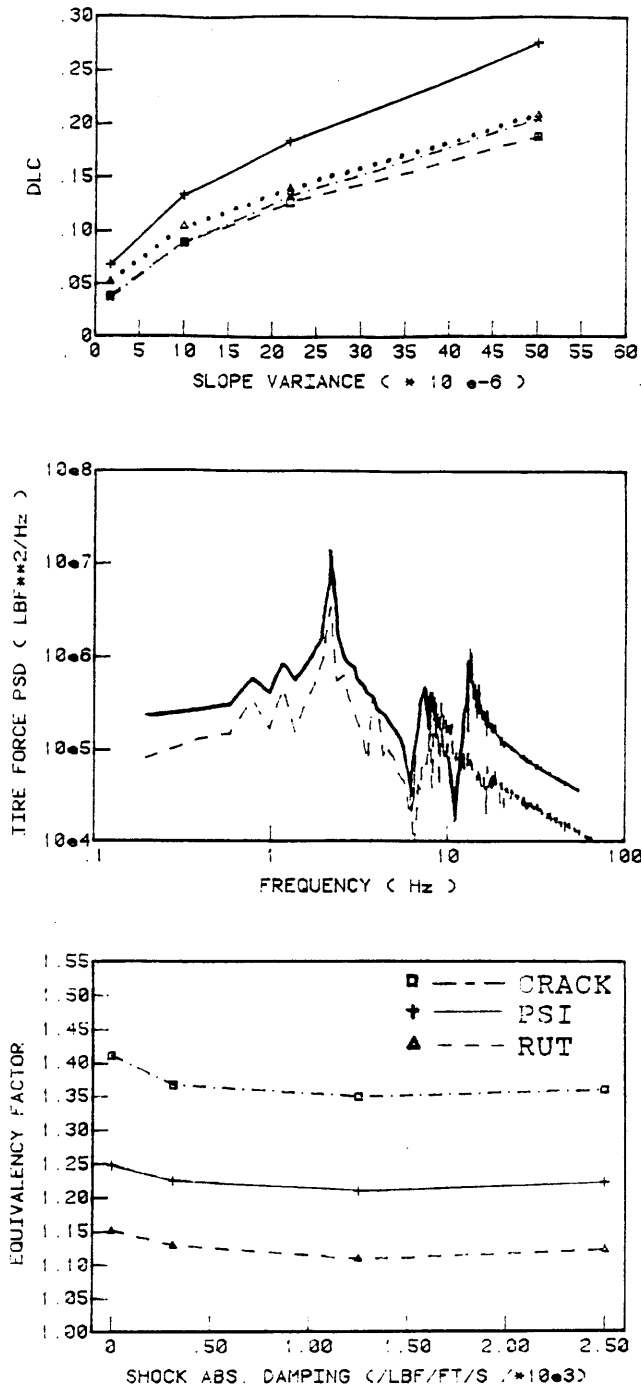c) Effect of Leaf Spring Stiffness on Pavement Damage

Fig.9 Effect of Spring Stiffness on Vehicle Response and Road Damage
a) Leading Drive Axle DLC versus SV
    □ — — — K = 30,600 lbs/ft
    + ————— K = 61,200 lbs/ft (standard)
    △ — · — K = 122,400 lbs/ft
b) Leading Drive Axle Tire Force PSD at SV = 22×10^{-6}
c) Effect of Leaf Spring Stiffness on Pavement Damage

Fig.10   Effect of Shock Absorber Damping on Vehicle Response and Road Damage
         a) Leading Drive Axle DLC versus SV
            + ————— No damper except steer wheel
            △ · · · · 375 lbs/(ft/s)
            □ — — — 1250 lbs/(ft/s)
            ✕ — · — 2500 lbs/(ft/s)
         b) Leading Drive Axle Tire Force PSD at SV = 22×10^{-6}
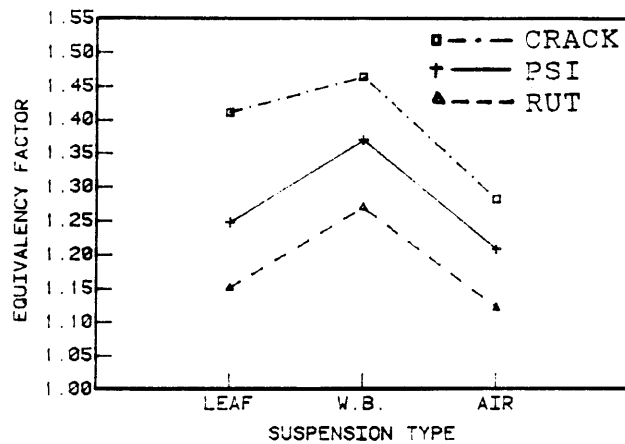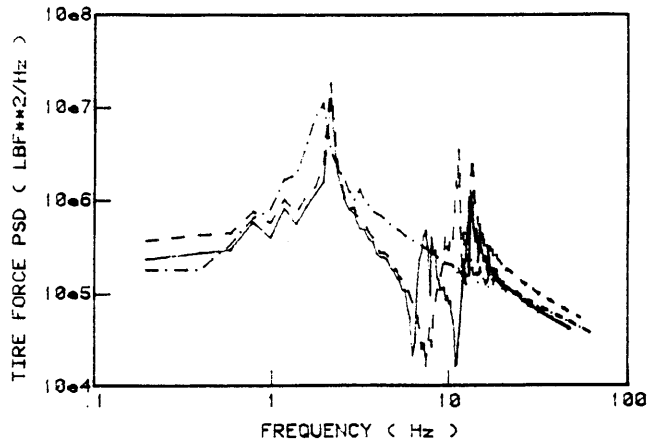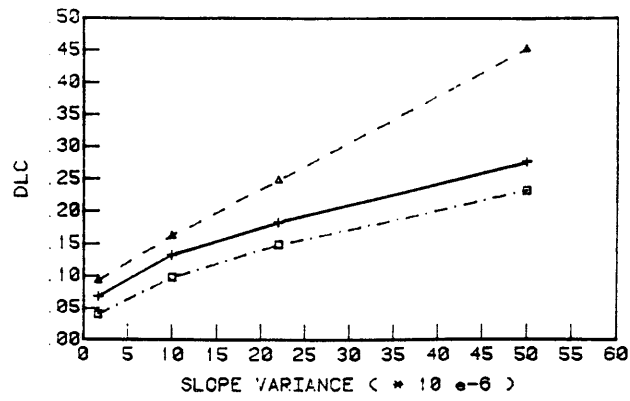   c) Effect of Shock Absorber Damping on Pavement Damage

Fig.11 Effect of Suspension Type on Vehicle Response and Road Damage
a) Leading Drive Axle DLC versus SV
    $+$ ———— Leaf Spring/Short Rocker Tandem
    $\triangle$ — — — Walking Beam Tandem
    $\square$ —— · — Air Spring Tandem
b) Leading Drive Axle Tire Force PSD at SV = 22×10^{-6}
c) Effect of Suspension Type on Pavement damage

Fig.11 (b) shows that the walking-beam suspension has significant power at about 10 Hz, corresponding to the bogie pitch mode which has very little damping[4]. The shock absorbers on the air spring suspension damp out the tire mode oscillation.

Fig.11 (c) shows that pavement damage due to walking beam suspensions are severe compared to the other suspensions. This result is expected from the DLC analysis. The walking beam suspension causes a 5.3% increase in the cracking equivalency factor, a 11.9% increase in the rutting equivalency factor and a 12.2% increase in the PSI equivalency factor, when compared to the leaf spring suspension. The air spring suspension, on the other hand, causes a 12.9% decrease in cracking equivalency factor, a 3.% decrease in rutting and 4.% decrease in the PSI equivalency factor.

Active and Semi-active Suspensions

Active suspensions,i.e., computer controlled suspensions that can put energy into the system and semi-active suspension,i.e., one that are designed only to dissipate energy are investigated as techniques to reduce the pavement damage.

Recent research on the design of active and semi-active suspensions for automotive vehicles[15] showed that significant improvements in either ride quality (near 1 Hz) or road holding can be achieved by using active/semi-active suspensions. The study on the performance of active suspensions for quater car models showed that the tire force variations at the body-mode and the tire-mode frequency can be reduced. Fig.12 shows the quater car model for an active suspension and Fig.13 shows the tire deflection frequency response for the quater car model.

In this study, tire force feedback control is used for the active and semi-active suspensions in order to reduce the pavement damage due to vehicle dynamics. Measurements of tire forces are very difficult to make, however, it can be estimated from accelerometer and deflection sensors.

The comparison of tire forces at $SV = 22 \times 10^{-6}$ with optimal passive damping, active suspensions and semi-active suspensions is shown in Fig.14. Fig.14 (b) and (c) indicate that the fluctuations of tire forces can be reduced significantly with either active or semi-active suspensions and the peak tire force is similar in both active and semi-active cases.

Fig.15 shows the control forces, i.e., actuator forces. Fig.15 indicates that the maximum value of control force is similar in both cases. The active suspensions require power all the time while the semi-active damper can only dissipate energy. It is interesting to note the similarity between the two forces in Fig.15.

Control forces for active and semi-active suspensions used in this paper are:

$$F_{act} = -G(F_t - L)$$

$$F_{semi-act} = F_{act} \quad \text{if } power < 0 \quad i.e. \quad power \ dissipation$$
$$F_{semi-act} = 0 \quad \text{if } power > 0$$

where G is the feedback gain, F_t is tire force and L is the mean tire force.

Fig.16 indicates that significant improvements towards reducing pavement damage can be achieved by using either active or semi-active suspensions. Fig.16 (a) and Fig.16 (c) illustrate that semi-active suspensions can provide substantial improvements over the optimal passive suspensions and nearly as good as active suspensions.

Fig.16 (a) indicates that the active and semi-active suspensions cause a 48.% decrease in the DLC compared to the standard leaf spring suspension and a 25.% decrease compared to the optimal passive leaf spring suspension. Fig.16 (b) shows that the tire force variations both at the body-mode and tire-mode are reduced by active suspension. The tire mode frequency is increased because of the active control force. Fig.16 (c) indicates that the semi-active suspension causes an 11.5 % decrease in the PSI equivalency factor, this translates to a 11.5 % increase in the pavement life, a 20.1 % decrease in the cracking equivalency factor and a 3.1 % decrease in the rutting equivalency factor, when compared to the optimal passive suspension.

Fig.17 shows the comparison of leading drive axle tire force PSD at $\overline{SV} = 22 \times 10^{-6}$. The semi-active suspension damps out the body mode oscillation significantly when compared with the optimal passive suspensions. Fig.17 (b) illustrates that the tire forces in the body mode are attenuated with either the active or the semi-active suspensions.

5. Conclusions

The dynamic effects of heavy vehicle suspensions have been investigated. In particular, the effects of suspension types and suspension stiffness and shock absorber damping have been studied and active and semi-active suspensions are investigated to reduce the pavement damage due to vehicle dynamics.

The results of this paper have shown that semi-active suspensions can significantly reduce the pavement damage when compared with optimal passive suspensions and perform nearly as well as active suspensions. Substantial improvements in pavement longevity may therefore be possible via suspension optimization and semi-active suspensions.
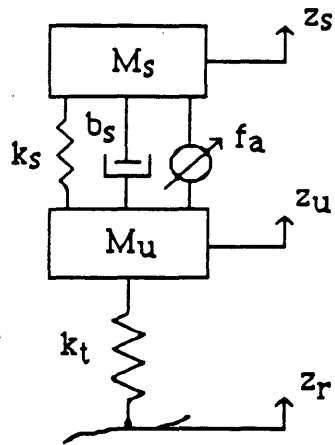
$$m_s \ddot{z}_s = f_p + f_a$$

$$m_u \ddot{z}_u = -f_p - f_a + k_t(z_r - z_u)$$

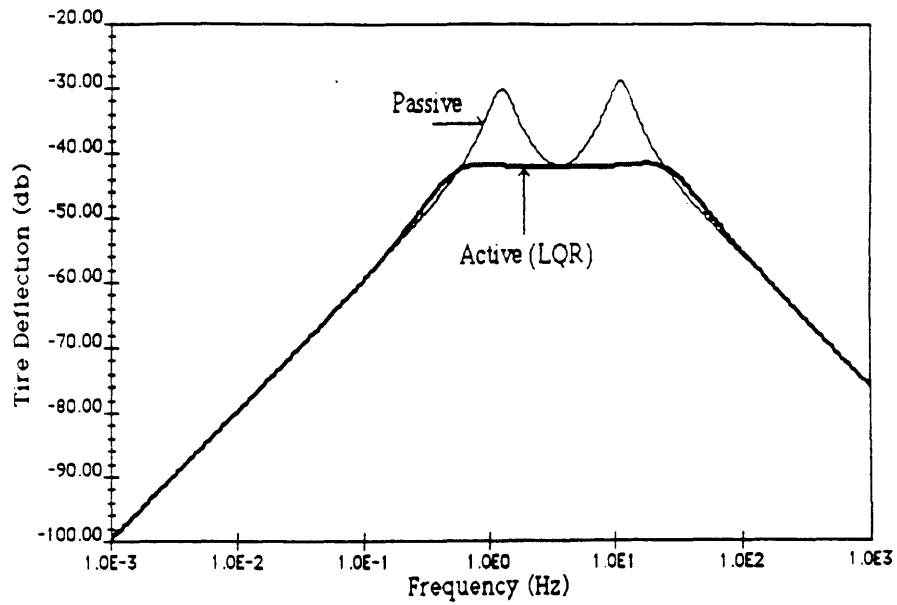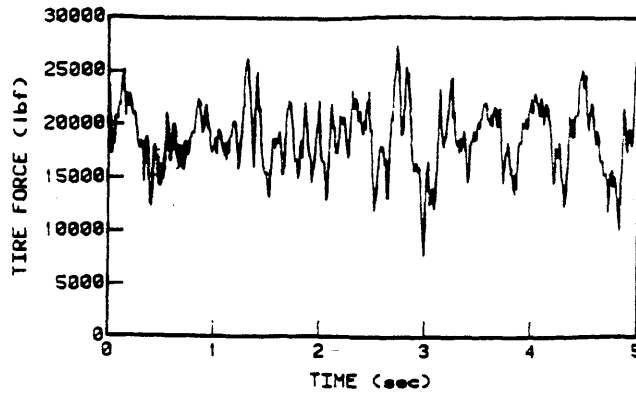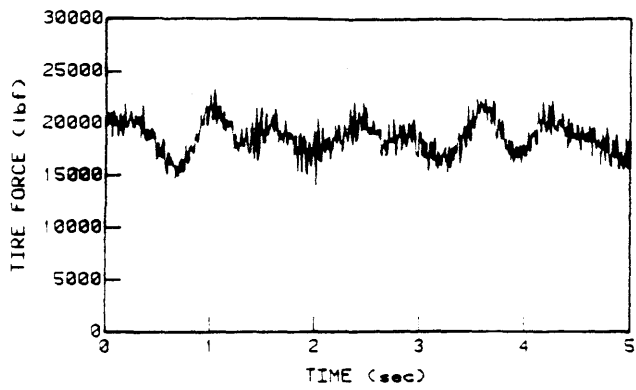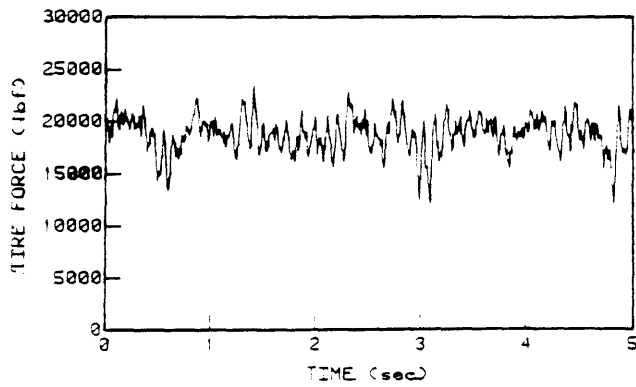Fig.12   Quarter Car Model for the Active Suspension [15]

Fig.13   Actiove Suspension for Road Holding   [15]
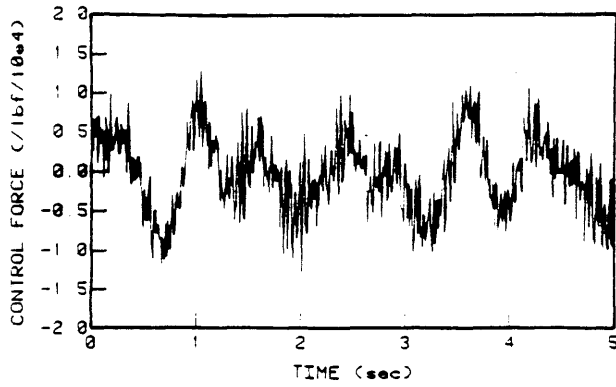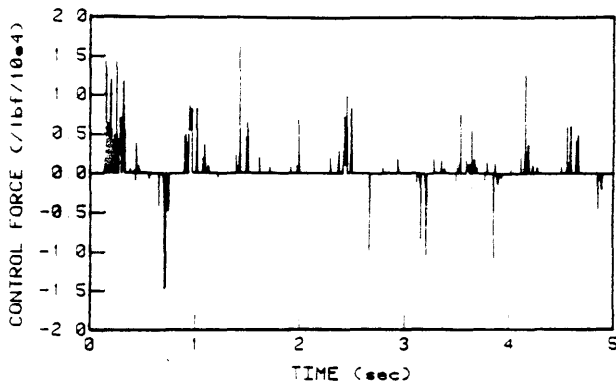
(a)

(b)

(c)

Fig.14 Tire forces at SV = 22×10^{-6}
   a) Optimal passive Damping
   b) Active Suspension (Tire Force Feedback)
   c) Semi-Active Suspension (Tire Force Feedback)

(a)

(b)

Fig.15   Control Forces at SV = 22×10^{-6}
(Tire Force Feedback)
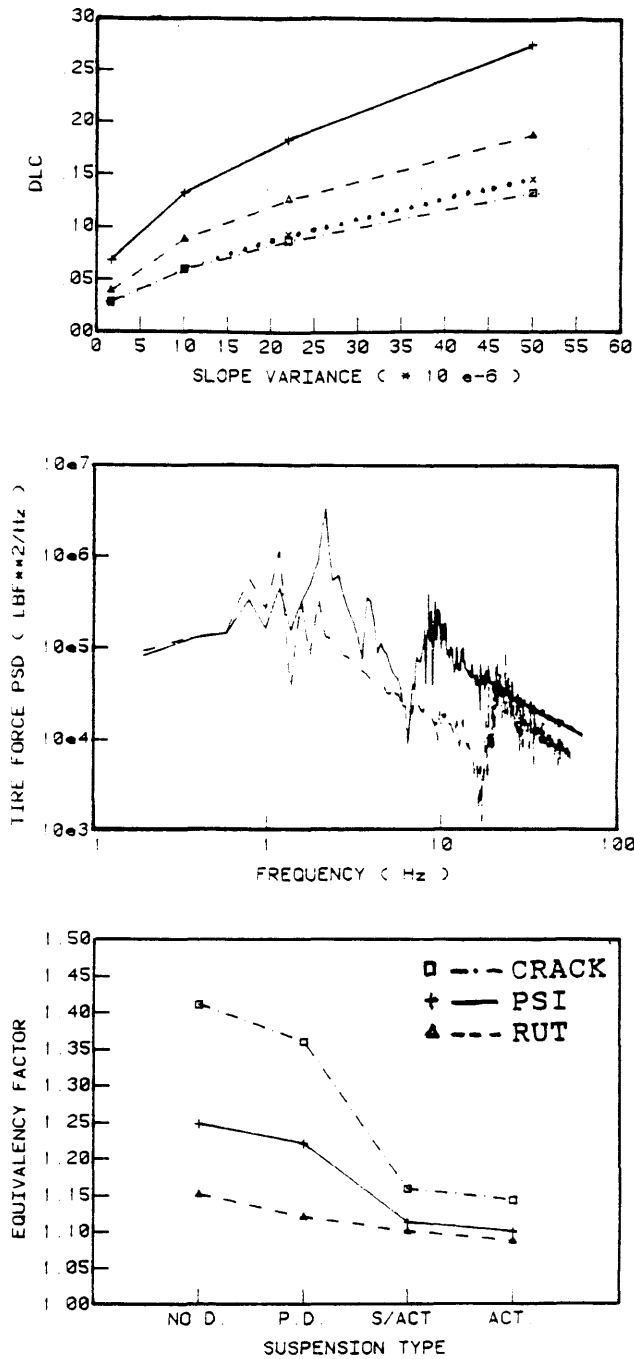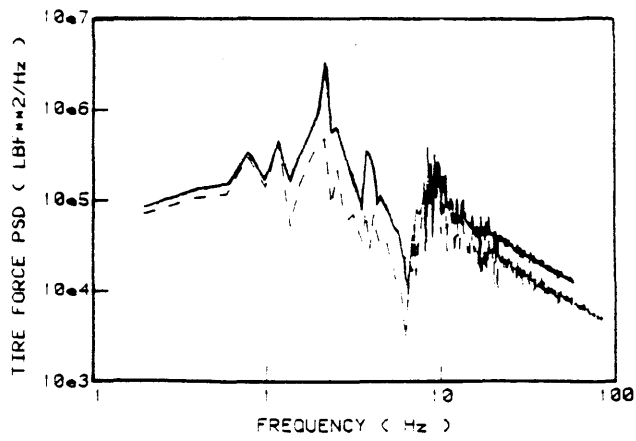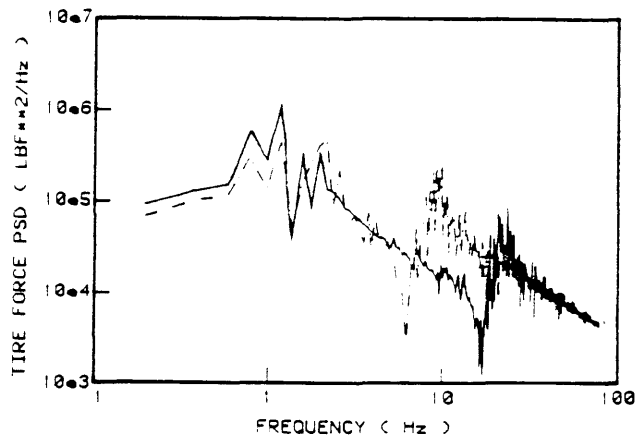a) Active Suspension
b) Semi-Active Suspension

Fig.16 Effect of Active Suspension on Vehicle Response and Road Damage (Leaf Spring Tandem)

a) Leading Drive Axle DLC versus SV
  + ———————— No Damping
  △ — — — Optimal Passive Damping
  □ — · — Active Suspension
  ✗ · · · · · Semi-Active Suspension
b) Leading Drive Axle Tire Force PSD at SV = 22×10^{-6}
  ———————— Optimal Passive Damping
  — · — Active Suspension
c) Effect of Active Suspension on Pavement damage

(a)

(b)

Fig.17   Comparison of Leading Drive Axle Tire Force PSD at SV = 22×10^{-6}
a)   ——————   Optimal Passive Damping
          — — —   Semi-Active Suspension
b)   ——————   Active Suspension
          — – —   Semi-Active Suspension

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. Sweatman,P.F.,"A Study of Dynamic Wheel forces in Axle Group Suspensions of Heavy Vehicles", Australian Road Research Board, Special Report No.27.

3. R.R.Addis, A.R.Halliday, C.G.B.Mitchell, "Dynamic Loading of Road Pavements",Proceedings,International Symposium on Heavy Vehicle Weights and Dimensions,Kelowna,British Columbia,Canada,June,1986.

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. D.Cebon,"Road Damaging Effects of Dynamic Axle Loads",Proceedings, International Symposium on Heavy vehicle Weights and Dimensions, Kelowna, British Columbia, Canada, June,1986.

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.