MILITARY TECHNICAL COLLEGE

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THE EFFECT OF SUSPENSION GEOMETRY ON A VEHICLE RIDE CHARACTERISTICS DURING DECELERATION AND ACCELERATION

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ABSTRACT

In designing a modern vehicle, some factors have to be considered. The most significant one, in so far as its long term effects are most profound and widespread such as the transverse-engined(front-wheel-drive) concept. Concomitant with this concept was the realization that the resulting low polar moment of inertia would give, when fitted with conventional suspensions, a high pitch frequency or "small-vehicle ride". The object of this paper is to demonstrate the effects of the suspension geometry on the vehicle pitch and bounce attitudes during braking or acceleration. Two degree of freedom models of a vehicle are employed herein. The steady-state positions in both pitch and bounce are developed for linear systems of typical suspensions that may be either conventional or coupled systems. The results indicate that the rear suspension system makes very significant contribution to the anti-dive properties of either conventional or coupled suspension systems.

INTRODUCTION

Many commercial vehicle simulations have been developed. However, most of these limited by design the component performance characteristics. The result of these on-going studies is a sophisticated simulation [1-2] which given the appropriate input can model a specific commercial vehicle performance.

While the simulation has been validated for a new specific examples, whether or not the simulation works in a commercial atmosphere depends upon obtaining accurate and confident results within time/cost limitations governing a specific operation. Each user could have a different defination of whether or not the simulation works depends upon his defination of how the simulation is to be used, how accurate the results must be, what level of resources are available, and how much confidence in the results is required to meet program objectives.

In order to use the simulation, it is firstly necessary to assemble massive amounts of input data. Some of these data have not be defined in the terms

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required by the simulation. Many of the component performance levels are required in absolute values whereas they have been traditionally defined in relative terms. A prime example is tire traction. Other data has a generated nature not readily available, but obtainable, such as the vehicle's sprung mass pitch moment of inertia [3-4]. The simulation of a vehicle during its braking or accelerating should include the bounce, pitch and rolling movements of the body. It is well-known that the magnitude of the movements is affected by the design of the suspension units, but a concise and satisfactory explanation does not appear to have been published. An analysis based on the concept of suspension derivatives [5] and employing the well-known principle of virtual work has been developed. It can be shown that the bounce and pitch movements during braking or accelerating are dependent upon the suspension derivative($\partial x/\partial z$) relating fore and aft movement to vertical movement of the suspension and the movements during braking are also modified by the front and rear distribution of braking effort. Suspension compliance is also shown to affect the behaviour of the vehicle.

The anti-dive characteristics of suspension are considered in [6] by determining an intantaneous pitch centre for the suspension and then assuming that the rear suspension can be replaced by an equivalent linkage. The explanation, principles and analysis performance of coupled suspension as hydrogas and hydrolastic means are described in [7-8]. In the present work the effects of suspension geometry on the vehicle pitch and bounce attitudes during braking or accelerating is studied by using two-degree-of-model for conventional and coupled suspension systems of a vehicle.

ANALYSIS METHOD

Vehicle Model For Conventional Suspension System

The vehicle model has two-degree-of-fredom, pitch and bounce, and is contained springs that have the same effective stiffness relative to the ground as do the suspension and wheel assemblies. Damping is included in the equations. The equations of motion are developed and the steady-state attitudes

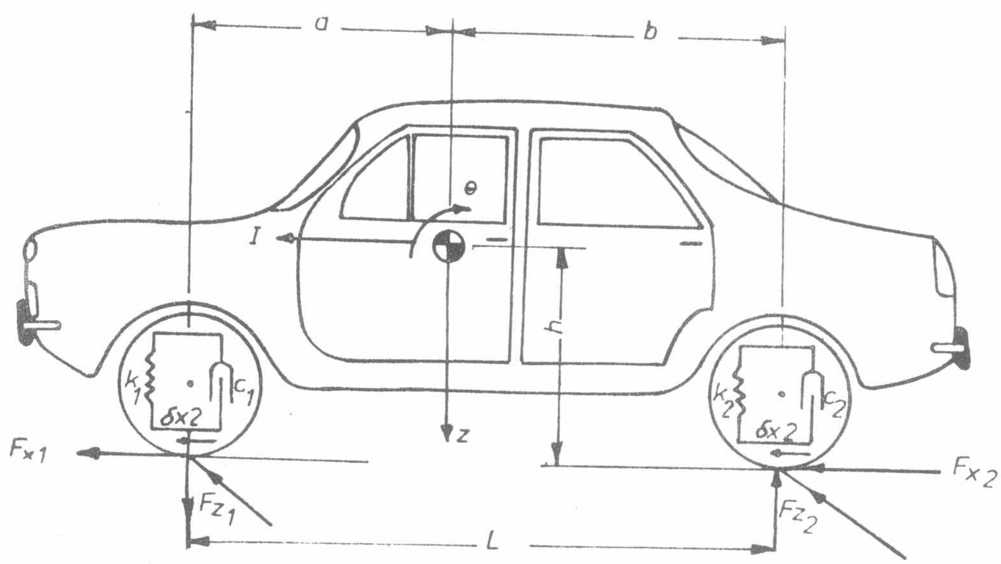

Fig. 1. Vehicle model for conventional suspension system

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are obtained from these equations. The purpose of developing the equations in this manner is to enable the effects of braking or accelerating and of road irregularities to be calculated.

From inspection of Fig. 1, the equations of motion are seen to be the well-known pitch and bounce equations. From Fig. 1, it can be seen that the effect of displacing the vehicle body by an amount of Z, while simultaneously pitching by θ is to deflect the front and rear suspensions respectively by

$$\delta Z_1 = Z - a.\theta \quad , \quad \delta Z_2 = Z + b.\theta \tag{1}$$

The forces between the vehicle and the ground are:

$$F_{z_1} = K_1(Z - a.\theta) \quad , \quad F_{z_2} = K_2(Z + b.\theta)$$

Similar terms can be developed for the velocities across the dampers and hence the damper forces can be obtained. The equations of motion are then as follows:

Summing vertical forces

$$M\ddot{Z} + C_1(\dot{Z}-a.\dot{\theta}) + C_2(\dot{Z}+b.\dot{\theta}) + K_1(Z-a.\theta) + K_2(Z+b.\theta) = Q_Z \tag{2}$$

and taking moments about the origin

$$I_y\ddot{\theta} - a.C_1(\dot{Z} - a.\dot{\theta}) + b.C_2(\dot{Z} + b.\dot{\theta}) - a.K_1(Z - a.\theta) + b.K_2(Z + b.\theta) = Q_\theta \tag{3}$$

Where Q_Z and Q_θ are the inputs to the system from the ground. These may be road surface irregularities or forces and moments generated in same other manner. The particular case of the inputs for braking or accelerating is developed below.

Considering the inputs to the system due to only to fore and aft forces acting at ground level and ignoring vertical disturbances from the ground then the vitual work is:

$$\delta W = F_{x_1}.\delta X_1 + F_{x_2}.\delta X_2 + (F_{x_1}+F_{x_2})\, h.\delta\theta \tag{4}$$

The ratio of fore and aft movement to vertical movement of the suspension is the suspension derivative $\partial X/\partial Z$. Substituting for δX_1 and δX_2, the appropriate suspension derivative, the virtual work equation becomes,

$$\delta W = F_{x_1}.\partial X/\partial Z\big|_1.\delta Z_1 + F_{x_2}.\partial X/\partial Z\big|_2.\delta Z_2 + (F_{x_1}+F_{x_2})\, h.\delta\theta \tag{5}$$

where $\delta Z_1 = \delta Z - a.\delta\theta$ and $\delta Z_2 = \delta Z + b.\delta\theta$

Therefore,

$$\delta W = F_{x_1}.\partial X/\partial Z\big|_1(\delta Z - a.\delta\theta) + F_{x_2}.\partial X/\partial Z\big|_2(\delta Z + b.\delta\theta) + (F_{x_1}+F_{x_2})h\,\delta\theta \tag{6}$$

The Z input disturbance is

$$\partial(\delta W)/\partial(\delta Z) = Q_Z = F_{x_1}.\partial X/\partial Z\big|_1 + F_{x_2}.\partial X/\partial Z\big|_2 \tag{7}$$

and the θ input disturbance is

$$\partial(\delta W)/\partial(\delta\theta) = Q_\theta = a.F_{x_1}.\partial X/\partial Z\big|_1 + b.F_{x_2}.\partial X/\partial Z\big|_2 + (F_{x_1}+F_{x_2})h \tag{8}$$

$$\begin{bmatrix} MD^2+(C_1+C_2)D +(K_1+K_2) & (-aC_1+bC_2)D +(-aK_1+bK_2) \\ (-aC_1+bC_2) +(-aK_1+bK_2) & I_yD^2+(a^2C_1+b^2C_2)D +(a^2K_1+b^2K_2) \end{bmatrix} \begin{bmatrix} Z \\ \theta \end{bmatrix} = \begin{bmatrix} F_{x1}\partial X/\partial Z|_1 +F_{x2}\partial X/\partial Z|_2 \\ -aF_{x1}\partial X/\partial Z|_1 +bF_{x2}\partial X/\partial Z|_2 +(F_{x1}+F_{x2})h \end{bmatrix} \quad (9)$$

where D is the operator d( )/ dt

These general equations can be simplified for specific cases. For example when the effective springs stiffnesses K_1 and K_2 are linear, the steady-state attitudes of the vehicle may be obtained by setting the acceleration and velocity terms to zero. However, equation(9) can be rewritten as

$$\begin{bmatrix} K_1+K_2 & -aK_1+bK_2 \\ -aK_1+bK_2 & a^2K_1+b^2K_2 \end{bmatrix} \begin{bmatrix} Z \\ \theta \end{bmatrix} = \begin{bmatrix} F_{x1}\partial X/\partial Z|_1 + F_{x2}\partial X/\partial Z|_2 \\ -aF_{x1}\partial X/\partial Z|_1 + bF_{x2}\partial X/\partial Z|_2 +(F_{x1}+F_{x2})h \end{bmatrix} \quad (10)$$

The values of F_{x1} and F_{x2}, which occur during braking, are not independent, but are in a fixed ratio $\lambda = F_{x1}/F_{x2}$. It is convenient to rewrite equation (10) in terms of this ratio and also of the total effect F_x, then $F_{x1}= F_x/(1+\lambda)$ and $F_{x2}=\lambda F_x/(1+\lambda)$. Then the steady-state equations becomes

$$\begin{bmatrix} K_1+K_2 & -aK_1+bK_2 \\ -aK_1+bK_2 & a^2K_1+b^2K_2 \end{bmatrix} \begin{bmatrix} Z \\ \theta \end{bmatrix} = \begin{bmatrix} \partial X/\partial Z|_1+\lambda\partial X/\partial Z|_2 \\ -a\partial X/\partial Z|_1+b\lambda\partial X/\partial Z|_2 + h(1+\lambda) \end{bmatrix} F_x/(1+\lambda) \quad (11)$$

hence

$$Z|_{ss} = F_x\left\{\frac{a\lambda K_1\partial X/\partial Z|_2+ bK_2\partial X/\partial Z|_1}{(1+\lambda)\,K_1K_2L} + \frac{h(aK_1-bK_2)}{K_1K_2L^2}\right\}, \text{and} \quad (12)$$

$$\theta|_{ss} = Fx\left\{\frac{\lambda K_1\partial X/\partial Z|_2- K_2\partial X/\partial Z|_1}{(1+\lambda)\,K_1K_2L} + \frac{h(K_1+K_2)}{K_1K_2L^2}\right\} \quad (13)$$

Equation(13) can be written as

$$\theta_{ss}= \theta_0 + \theta_1 + \theta_2$$

where

$$\theta_0 = \frac{F_x\,h(K_1+K_2)}{K_1K_2L^2}, \quad \theta_1 = \frac{-F_x\,K_2\partial X/\partial Z|_1}{(1+\lambda)K_1\,K_2L}, \text{ and}$$

$$\theta_2 = \frac{F_x\lambda K_1\partial X/\partial Z|_2}{(1+\lambda)\,K_1K_2L}$$

Not that during braking the sum of F_x is negative.

The relative effectiveness of the fore and aft suspension derivative $(\partial X/\partial Z)$ in controlling the vehicle attitude is seen to be dependent upon the braking ratio selected as well as the effective stiffness of the springs of the vehicle.

The relative effectiveness of front suspension:

$$\text{Effect.}\Big|_1 = (\theta_1/\theta_0).100 = \left\{ \frac{-K_2.\partial X/\partial Z|_1}{(1+\lambda)(K_1+K_2)h} \right\} .100 \qquad (14)$$

The relative effectiveness of rear suspension:

$$\text{Effect.}\Big|_2 = (\theta_2/\theta_0).100 = \left\{ \frac{\lambda K_1 \partial X/\partial Z|_2}{(1+\lambda)(K_1+K_2)h} \right\} .100 \qquad (15)$$

Vehicle Model For Coupled Suspension System

The equations of motion for the suspension design may be derived from the idealized system shown in Fig. 2, where the coupling between front and rear suspensions is represented by the symboles ${}_1K_2$ and ${}_2K_1$. These symboles include the mechanical characteristics of the coupling and is not necessarily a simple spring. The damping is assumed, in this situation, to be acting between the axles and the vehicle body and no damping is present in the coupling unit.

The analysis of the coupled suspension ride modes is similar to that of the previous system except that the force at each axle is dependent on the deflection at that axle and also the deflection of the other axle.

$$F_{z_1} = {}_1K_1\delta Z_1 + {}_1K_2\delta Z_2 \quad , \quad F_{z_2} = {}_2K_2\delta Z_2 + {}_2K_1\delta Z_1$$

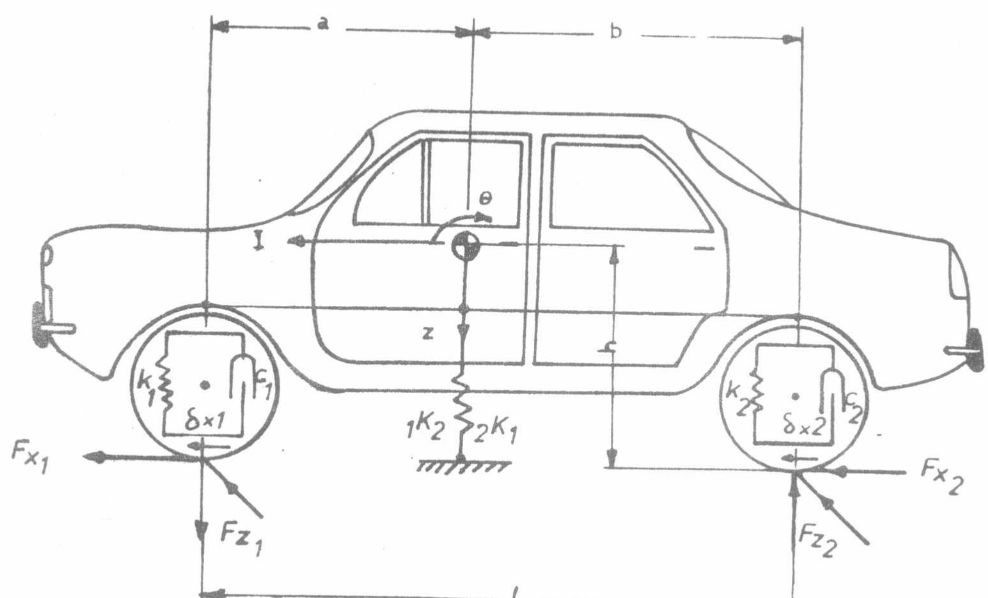

Fig. 2 Vehicle model for coupled suspension system, all the terms from Fig. 1 apply to this system

The "reciprocal" theorm of clerk maxwell leads to the fact that ${}_1K_2 = {}_2K_1$. Substituting for δZ_1 and δZ_2, then,

$$F_{z1} = {}_1K_1(Z - a.\theta) + {}_1K_2(Z + b.\theta)$$
$$F_{z2} = {}_2K_2(Z + a.\theta) + {}_1K_2(Z - a.\theta) \quad (16)$$

Thus the equations of motion become:

$$M\ddot{Z} + C_1(\dot{Z}-a\dot{\theta}) + C_2(\dot{Z}+b\dot{\theta}) + ({}_1K_1 + {}_1K_2)(Z-a\theta) + ({}_2K_2 + {}_1K_2)(Z+b\theta) = Q_z \quad (17)$$

$$I_y\ddot{\theta} + aC_1(\dot{Z}-a\dot{\theta}) + bC_2(\dot{Z}+b\dot{\theta}) - a\;{}_1K_1(Z-a\theta) + {}_1K_2(Z+b\theta) + b\;{}_2K_2(Z+b\theta) + {}_1K_2(Z-a\theta) = Q_\theta \quad (18)$$

$$\begin{bmatrix} MD^2+(C_1+C_2)D + K_1+K_2+2\,{}_1K_2 & (-aC_1+bC_2)D - aK_1+bK_2 \\ (-aC_1+bC_2)D - aK_1 + bK_2+(b-a)\,{}_1K_2 & I_yD^2+(a^2C_1+bC_2)D + a^2K_1+b^2K_2-2ab\,{}_1K_2 \end{bmatrix} \begin{bmatrix} Z \\ \theta \end{bmatrix} = \begin{bmatrix} \partial X/\partial Z|_1 + \partial X/\partial Z|_2 \\ -a\partial X/\partial Z|_1 + b\,\partial X/\partial Z|_2 + h(1+\lambda) \end{bmatrix} \frac{F_x}{(1+\lambda)}$$

(19)

proceeding as in the previous section, the steady-state pitch and bounce positions are determined assuming that the springs are linear. Hence,

$$Z|_{ss} = F_x\left[\frac{(aK_1 - b_1K_2)\,\partial X/\partial Z|_2 + (bK_2 - a_1K_2)\,\partial X/\partial Z|_1}{(1+\lambda)\;L(K_1K_2 - {}_1K_2^2)} + \frac{h}{L^2}\left\{\frac{aK_1 - bK_2 - (b-a)\,{}_1K_2}{K_1K_2 - {}_1K_2^2}\right\}\right] \quad (20)$$

and

$$\theta|_{ss} = F_x\left[\frac{(K_1 + {}_1K_2)\,\partial X/\partial Z|_2 - (K_2 - {}_1K_2)\,\partial X/\partial Z|_1}{(1+\lambda)\;L(K_1K_2 - {}_1K_2^2)} + \frac{h}{L^2}\left\{\frac{K_1 + K_2 + 2\,{}_1K_2}{K_1K_2 - {}_1K_2^2}\right\}\right] \quad (21)$$

Equation(21) can be written as

$$\theta|_{ss} = \theta_0 + \theta_1 + \theta_2 \quad (22)$$

where

$$\theta_0 = \frac{F_x h}{L^2}\left(\frac{K_1+K_2+2\,{}_1K_2}{K_1K_2 - {}_1K_2^2}\right), \qquad \theta_1 = \frac{-F_x(K_2 - {}_1K_2)\,\partial X/\partial Z|_1}{(1+\lambda)L(K_1K_2 - {}_1K_2^2)}, \text{ and}$$

$$\theta_2 = \frac{F_x(K_1 + {}_1K_2)\,\partial X/\partial Z|_2}{(1+\lambda)L(K_1K_2 - {}_1K_2^2)}$$

The relative effectiveness of the fore and aft suspension derivative $(\partial X/\partial Z)$ for the vehicle accuped by coupled suspension system are:

Front suspension,

$$\text{Effect.}\Big|_1 = (\theta_1/\theta_0).100 = \frac{-(K_1+{}_1K_2)\; L\partial X/\partial Z\big|_1}{h(1+\lambda)(K_1+K_2-2\,{}_1K_2)} \;.\; 100 \qquad (23)$$

Rear suspension,

$$\text{Effect.}\Big|_2 = (\theta_2/\theta_0).100 = \frac{(K_1+{}_1K_2)\partial X/\partial Z\big|_2}{h(1+\lambda)(K_1+K_2-2\,{}_1K_2)} \;.\; 100 \qquad (24)$$

APPLICATION

Data Preparation and Calculation Procedure

A set of steady-state calculations has been prepared for a typical medium saloon vehicle. Details of the vehicle when occupied by either conventional or coupled suspension system are listed in Table 1. A braking force sufficient to develop a deceleration of 0.6 g is assumed. The values of braking force ratio(λ) are presented in Table 2.

Table 1. Details of the vehicle systems required for application

Symbol	Conventional system	Coupled system
W, N	15000	15000
I_y, N/m^2	25400	25400
C_1, N/ms	8000	8000
C_2, N/ms	9000	9000
K_1, N/m	42000	30000
K_2, N/m	60000	40000
${}_1K_2$, N/m	-	16000
${}_2K_1$, N/m	-	16000
L, m	2.6	2.6
a, m	1.2	1.2
b, m	1.4	1.4
h, m	0.6	0.6

Table 2. The values of braking force ratio

No.	F_{x_1},N	F_{x_2},N	$= F_{x_1}/F_{x_2}$
1	500	500	500/500=1
2	550	450	550/450=1.2
3	580	410	580/410=1.4
4	600	400	600/400=1.5
5	630	370	630/370=1.7

The calculation procedure started first for the conventional suspension system by substituting the values presented in Tables 1 and 2 into equations (12) and (13). The values of $\partial X/\partial Z|_1$ and $\partial X/\partial Z|_2$ were taken as -0.5, 0.0, 0.5, while the range of braking force ratio was from 1.0 to 1.7. The calculation procedure was repeated for the coupled suspension system at the same conditions by using equations (20) and (21). The effectiveness of each system was calculated based on equations (14) and (15)(conventional) and equations (23) and (24)(coupled).

Results of Conventional Suspension System

The total vehicle bounce attitudes during braking are shown in Figs. 3,4 and 5 against the values of braking force ratio(λ) stated in Table 2. Fig. 3

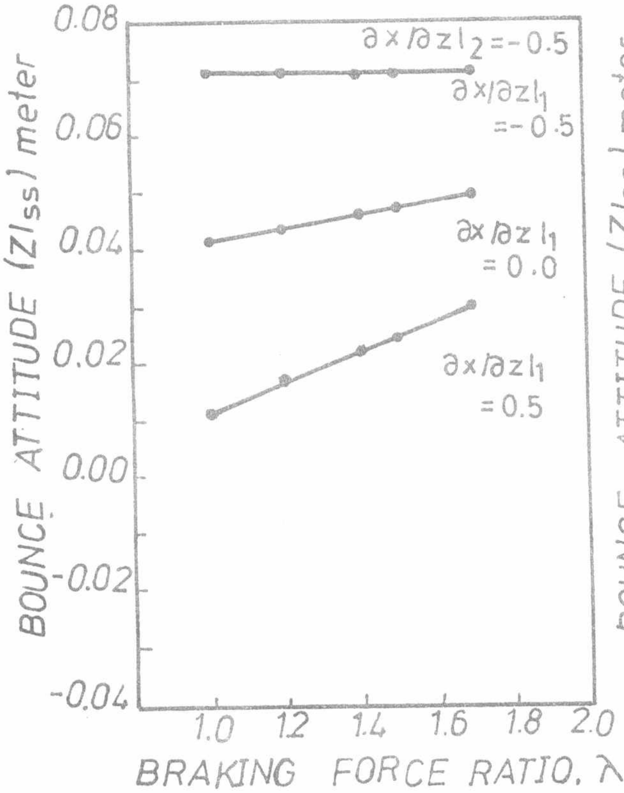

Fig. 3 The steady-state bounce attitude for conventional system

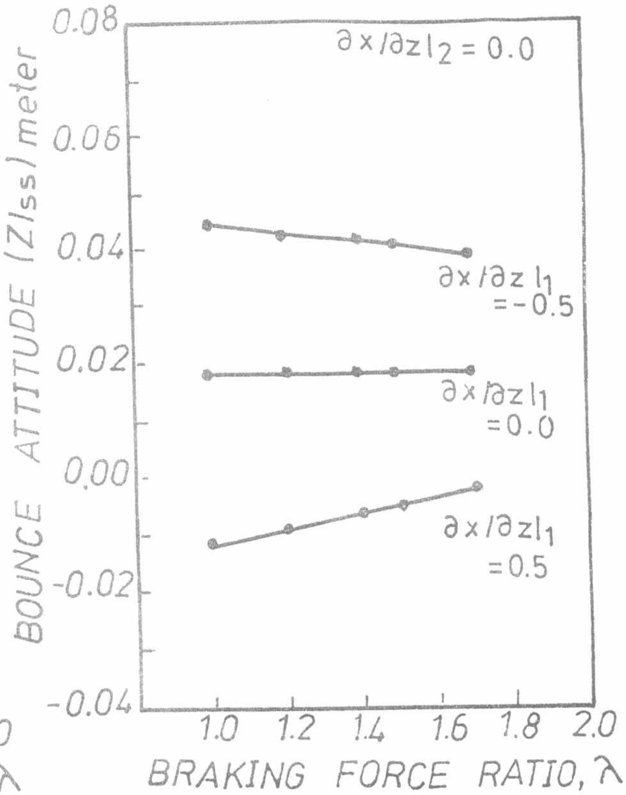

Fig. 4 The steady-state bounce attitude for conventional system

shows this attitude when the front suspension derivative($\partial X/\partial Z|_1$) = -0.5, while the rear suspension derivative($\partial X/\partial Z|_2$) varies from -0.5,0.0,0.5. The results in this figure indicate that the increase in rear suspension derivative will result as an increase in the vehicle bounce attitude. The comments can be noticed in Figs. 4 and 5, despite of the values of bounce attitude at $\partial X/\partial Z|_1$=0.0 (Fig. 4) and $\partial X/\partial Z|_1$ =0.5(Fig. 5) are much lower than that at $\partial X/\partial Z|_1$= -0.5. In all cases, the bounce attitude remains constant when the front suspension derivative is equal to the rear suspension derivative.

The total pitch attitudes during braking are shown in Figs. 6,7 and 8 against the values of braking force ratio(λ) stated in Table 2. Fig. 6 shows this attitude when the front suspension derivative($\partial X/\partial Z|_1$) = -0.5, while the rear suspension derivative($\partial X/\partial Z|_2$) varies from -0.5,0.0,0.5. In all cases calculated here the reduction of total pitch

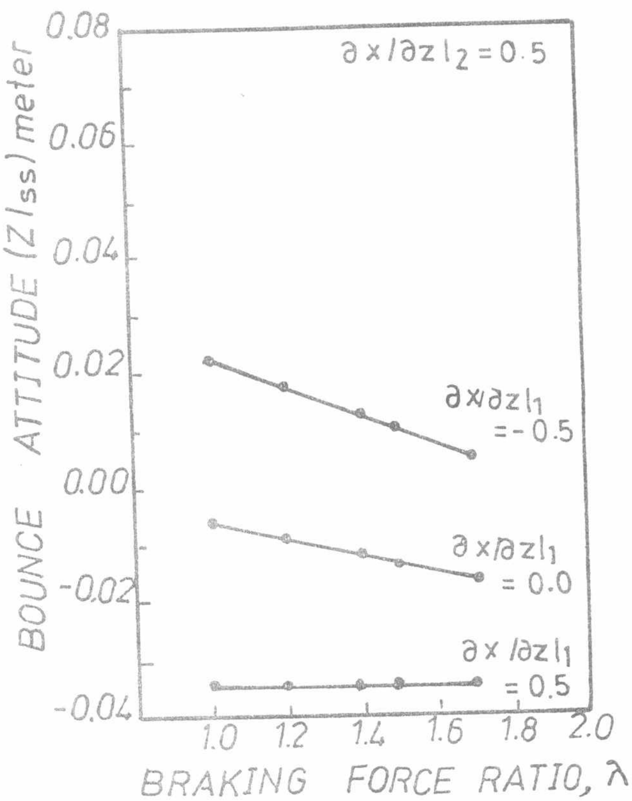

Fig. 5 The steady-state bounce attitude for conventional system

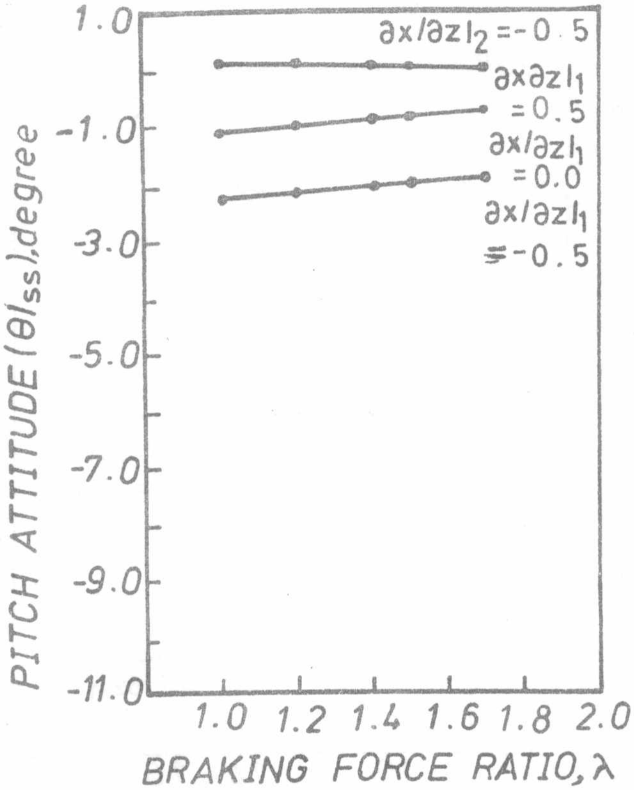

Fig. 6 The steady-state pitch attitude for conventional system

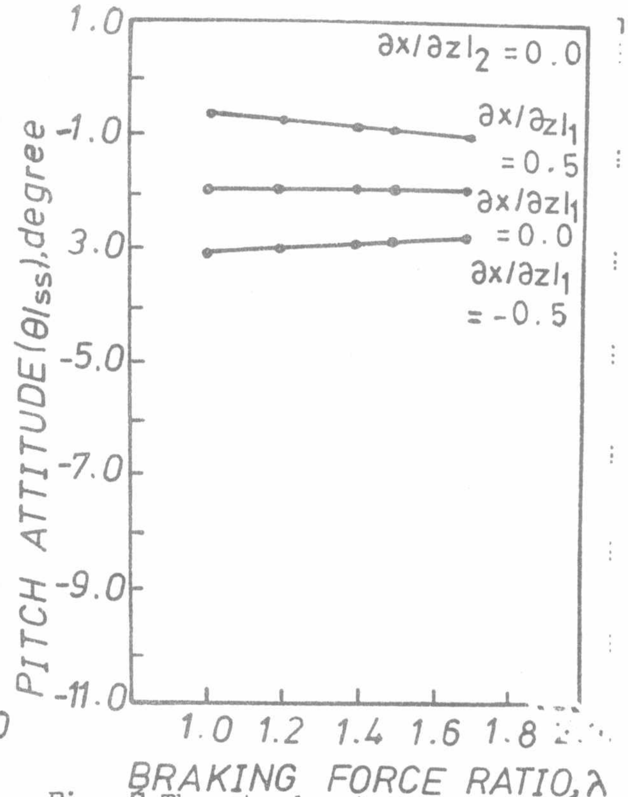

Fig. 7 The steady-state pitch attitude for conventional system

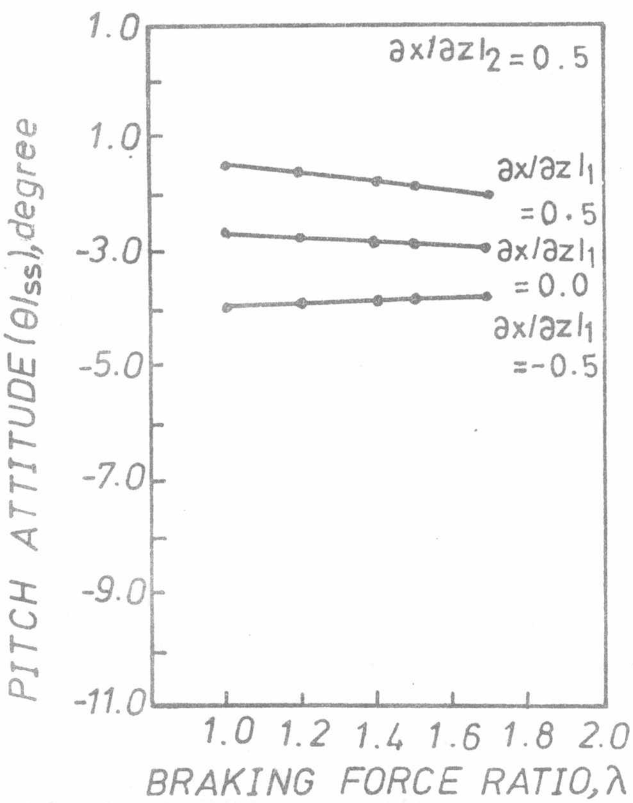

Fig. 8 The steady-state pitch attitude for conventional system

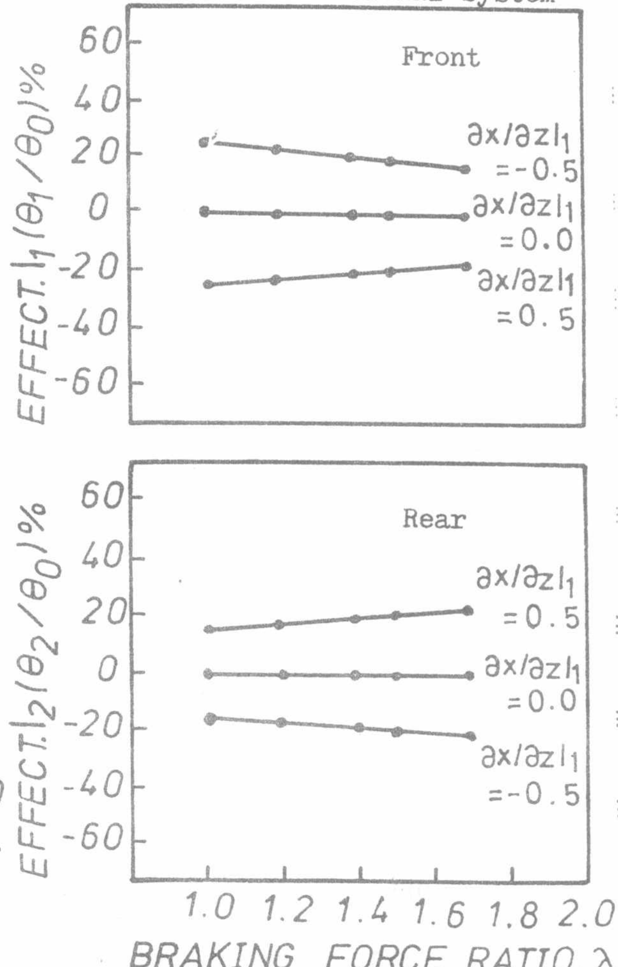

Fig. 9 Effectiveness of pitch attitude for conventional system

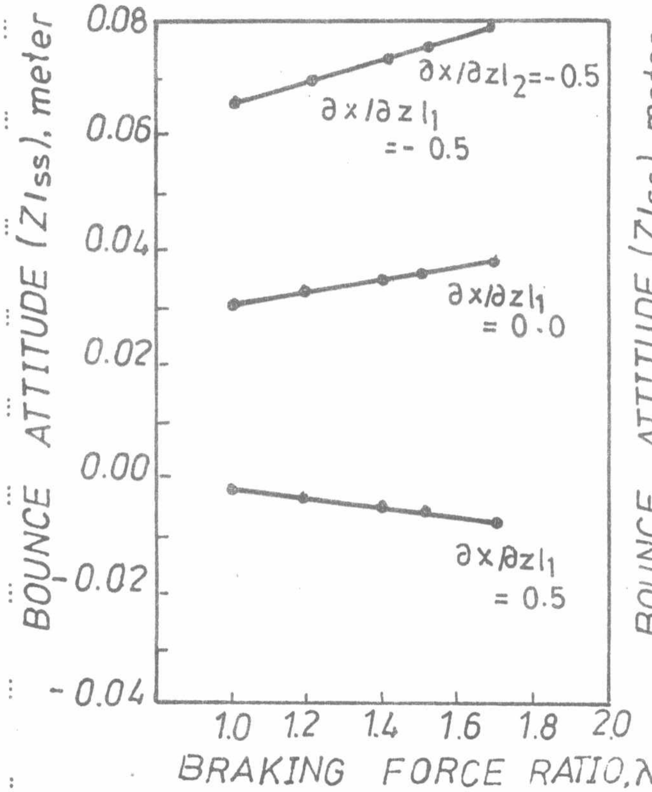

Fig. 10 The steady-state bounce attitude for coupled system

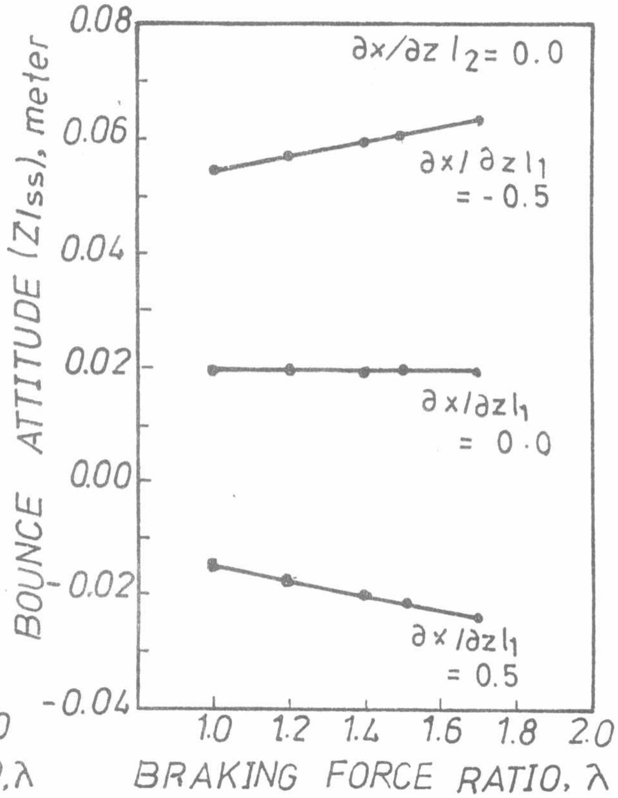

Fig. 11 The steady-state bounce attitude for coupled system

attitudes due to increase the front and rear suspension derivatives are smaller than the total bounce attitudes, but only when the case of $\partial x/\partial z|_1$ and $\partial x/\partial z|_2$ are equal to zero the bounce attitudes remain constant

Figure 9 shows the relative effectiveness of the front and rear suspensions. It can be seen that the front suspension derivative($\partial x/\partial z|_1$) is the controller for the rear suspension and versa vice.

From the above discussions it can be concluded that the rear suspension is important in controlling the total vehicle pitch attitudes and that its effect increases as the rear wheels become more heavily

Results of Coupled Suspension System

The total vehicle bounce attitudes during braking are shown in Figs. 10, 11 and 12 against the values of braking

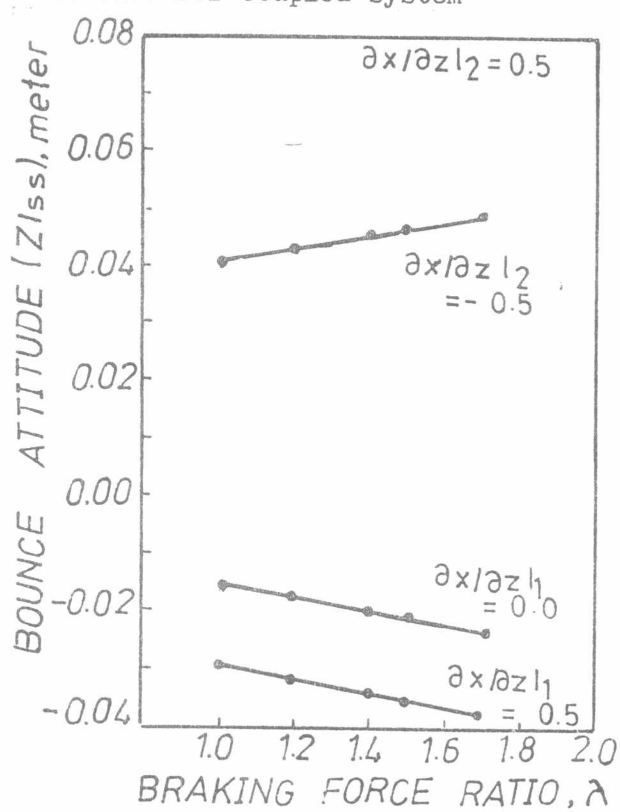

Fig. 12 The steady-state bounce attitude for coupled system

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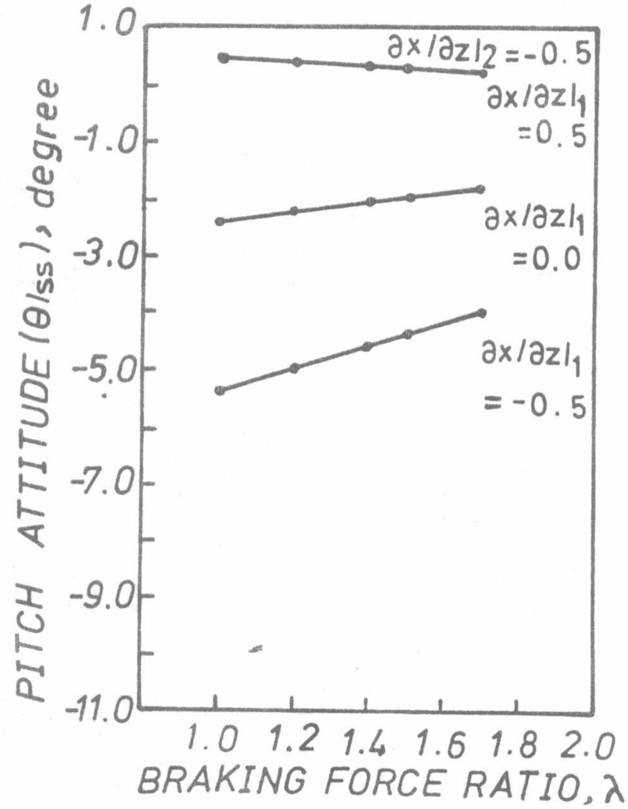

Fig. 13 The steady-state pitch attitude for coupled system

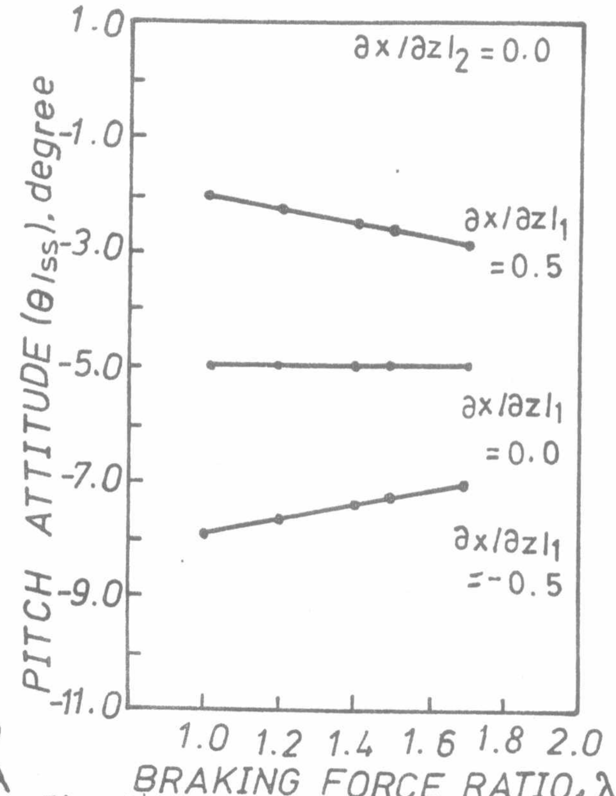

Fig. 14 The steady-state pitch attitude for coupled system

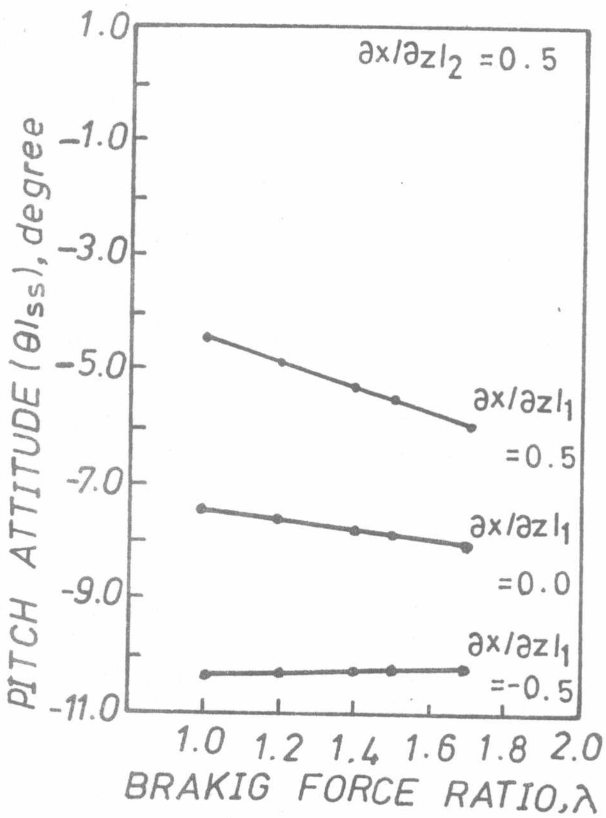

Fig. 15 The steady-state pitch attitude for coupled system

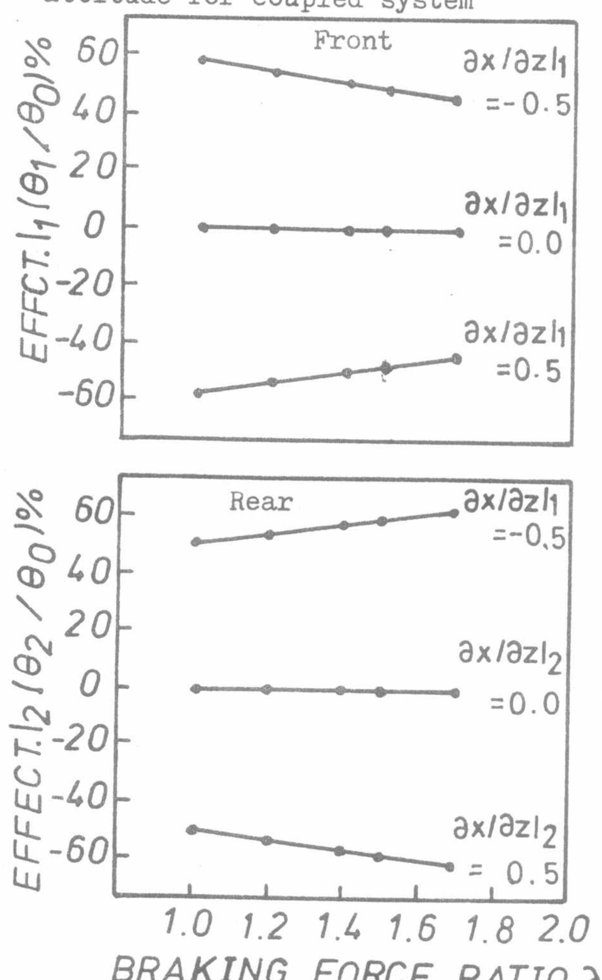

Fig. 16 Effectiveness of pitch attitude for coupled system

force ratio(λ) stated in Table 2. Fig. 10 shows this attitude when the front suspension derivative($\partial X/\partial Z|_1$)= -0.5, while the rear suspension derivative($\partial X/\partial Z|_2$) varies from -0.5,0.0,0.5. The results in this figure are similar to that obtained for the conventional suspension system (Fig. 3), but the reduction in bounce attitude here is bigger. The same comments can be observed in Figs. 11 and 12. Only when the case of $\partial X/\partial Z|_1$ and $\partial X/\partial Z|_2$ are equal to zero the attitude remains constant at all the values of braking force ratio(λ).

The total vehicle pitch attitudes during braking are shown in Figs. 13,14 and 15 against the values of braking force ratio(λ) stated in Table 2. Fig. 13 shows this attitude when the front suspension derivative ($\partial X/\partial Z|_1$) = -0.5, while the rear suspension derivative($\partial X/\partial Z|_2$) varies from -0.5, 0.0,0.5. In the cases calculated here the reduction of total pitch attitudes due to the increase of front and rear suspension derivatives are nearly the same as in total bounce attitudes. Only the case of $\partial X/\partial Z|_1$ and $\partial X/\partial Z|_2$ are equal to zero the attitude remains constant.

Figure 16 shows the relative effectiveness of the front and rear suspensions. The same comments can be stated here, despite of the effectiveness here is much more than that for the case of conventional suspension system. It is seen that the rear suspension makes a very significant contribution to the anti-dive properties of coupled suspension system.

CONCLUTIONS

1. A logical analysis of the anti-dive characteristics of suspension designs has been demonstrated.
2. The method can be employed to study vehicle attitudes during both braking and acceleration and is completely general.
3. It has been demonstrated that the rear suspension system is important in controlling the ride performance of vehicles occupied by either conventional or coupled suspension system.
4. It has been found that the coupled suspension system is more effective in controlling the pitch attitude than that of the conventional suspension system.
5. It must be noted that the effect of non-linearity in the suspension system parameters have not been included in the analysis, that will be done in the near future.

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NOMENCLATURE

a	Distance from c.g(centre of gravity) to front suspension
b	Distance from c.g(centre of gravity) to rear suspension
C_1	Front damping characteristic
C_2	Rear damping characteristic
F_x	Force in the x-direction
F_z	Force in the z-direction
h	Height of c.g. above ground plane
I	Moment of inertia in pitch
K_1	Effective front spring stiffness
K_2	Effective rear spring stiffness
${}_1K_2$	Effective coupling spring stiffness
${}_2K_1$	Effective coupling spring stiffness
L	Wheel base
M	Mass of vehicle
Z	Vertical displacement