An Analytical and Experimental Investigation on the Vibration Isolation Performance of Semi-Active Suspensions

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ABSTRACT

An Analytical and Experimental Investigation on the Vibration Isolation Performance of Semi-Active Suspensions

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In this study, an analytical and experimental investigation on the vibration isolation performance of semi-active (SA) suspensions is carried out. Four types of semi-active suspension configurations are investigated using a 2 d-o-f quarter car bounce model. A discussion is presented on the solution methodology for solving SA-systems characterized by differential equations with discontinuity. The occurrence of system lock-up and the necessity to find a zero-crossing to treat for the lock-up are explained. Analytical investigation of three types of SA-systems is carried out. The response behavior of SA-type 1 and SA-type 3 systems are presented using the steady-state time plots. A modified SA-type 2 system (SA-type 2^*) is proposed and studied for its improved performance. Vibration isolation performance of SA-type 1, SA-type 2^*, and SA-type 3 systems are examined using transmissibility plots of sprung mass bounce acceleration response, suspension rattle space, and unsprung mass relative displacement. The performances of all SA-systems are compared with each other and with the passive suspension system. The isolation performance of all SA-systems are better at the sprung mass natural frequency and inferior at the unsprung mass natural frequency compared to the passive system. In

general, the SA-type 2* system provides a compromise performance compared to SA-type 1, and SA-type 3 systems.

A discussion on the use of a zero-finder in the solution methodology for solving the discontinuous system equations of SA-suspensions is also presented. The performance of an SA-type 4 system is investigated both analytically and experimentally. The experimental analysis of SA-type 4 system is carried out using a laboratory testing of a 2 d-o-f bounce model experimental set-up. A commercial automobile shock-absorber is used in conceiving the SA-type 4 system. The proposed SA-type 4 system uses the firm and soft modes of the adjustable shock absorber for "on" and "off" stages representing high and low damping values. Preliminary tests are carried out to evaluate and compare the damping characteristics of the shock absorber in the firm, normal, and soft modes with the manufacturer supplied data.

Analytical study is carried out to investigate the system responses for variations in damping values for SA-type 4, firm, normal, and soft type suspensions. Laboratory testing is conducted at discrete frequencies for SA-type 4, firm, normal, and soft type suspensions and their vibration isolation characteristics are presented and compared. Using an analytical model with suitable modifications for experimental conditions, computer simulation results for SA-type 4, firm, normal, and soft type suspensions are obtained and validated against the experimental results. The SA-type 4 suspension offers improved vibration isolation properties and a compromising suspension and tire deflection responses over the firm, normal, and soft type of suspensions.

Finally, vibration isolation performance of a typical road vehicle is analytically examined using a one quarter-car model with SA-type 4 suspension system. The results are compared with the performance of firm, normal, and soft type of suspensions.

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Montréal, Canada S.Rengarajan

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NOMENCLATURE

a	Amplitude of harmonic input displacement
c, c_0, c_1, c_2, c_3	Value of Damping coefficient
c_c	Critical damping coefficient
Eqn	Equation
F	Force generator for an active suspension system
$F(.,.)$	Function describing a set of differential equations
F_d	Semi-active Damping force
F_d^{*}	Active damping force
F_d^{**}	Damper force at the instant of lock-up
K_f	Spring force
M_1	Sprung mass
M_2	Unsprung mass
Phiin	Value of condition function before iteration in the integration routine
Phiout	Value of condition function after iteration in the integration routine
R_f	Sum of the inertial and spring force
Sgn(.,.)	sign of (.,.)
t	Time (sec), time step (sec)
t^{*}	Instant of time at which the zero crossing of the condition funtion occurs
T	Transmissibility
x_1	Sprung mass absolute displacement
$(x_1 - x_2)^{**}$	Value of $(x_1 - x_2)$ at the instant of lock-up

$\dot{x}_1$	Sprung mass absolute velocity
$\ddot{x}_1$	Sprung mass absolute acceleration
x_2	Unsprung mass absolute displacement
$\dot{x}_2$	Unsprung mass absolute velocity
$\ddot{x}_2$	Unsprung mass absolute acceleration
$\left\lvert \frac{\ddot{X}_1}{\dot{X}_3} \right\rvert$	Transmissibiliy of Absolute Value of Maximum Sprung Mass Bounce Accelertion Response over Absolute Value of Maximum Input Velocity $(m/s^2)/(m/s)$
$\left\lvert \frac{X_1 - X_2}{\dot{X}_3} \right\rvert$	Transmissibiliy of Absolute Value of Maximum Suspension Deflection Response over Absolute Value of Maximum Input Velocity (m)/(m/s)
$\left\lvert \frac{X_2 - X_3}{\dot{X}_3} \right\rvert$	Transmissibiliy of Absolute Value of Maximum Tire Deflection Response over Absolute Value of Maximum Input Velocity (m)/(m/s)
x_3	Input displacement
X_1	Mid-stroke position
v	Velocity across shock absorber
v_e	Extension velocity across the shock absorber
v_{e1}, v_{e2}	Extension velocity across the shock absorber when the orifice flow starts
v_c	Compression velocity across the shock absorber
v_{c1}, v_{c2}	Compression velocity across the shock absorber when the orifice flow starts
z	Relative displacement between sprung mass and base

$\dot{z}$	Relative velocity between sprung mass and base
z_1	Relative displacement between sprung and unsprung mass
$\dot{z}_1$	Relative velocity between sprung and unsprung mass
α	Value of Tuning factor or Gain in SA-type 3 system
ω	Circular natural frequency
ω_1	Sprung mass natural frequency
ω_2	Unsprung mass natural frequency
ω_n	Natural Frequency
ϕ	Value of Condition Function
$\phi(x_i, y_i)$	Value of function $\phi(x,y)$ at $(i)^{th}$ instant of time
$\phi(x_{i+1}, y_{i+1})$	Value of function $\phi(x,y)$ at $(i+1)^{th}$ instant of time
$\phi(x_*, y_*)$	Value of function at the instant zero-crossing takes place
$\lvert\phi_i\rvert$, $\lvert\phi_{i+1}\rvert$	Absolute value of condition function at the t_i^{th} and t_{i+1}^{th} instant of time, respectively
τ	Period of harmonic input excitation
$\zeta = \frac{c_1}{c_c}$	Value of damping factor at c_1
$\zeta_0 = \frac{c_0}{c_c}$	Value of damping factor at c_0
ζ_e	Value of damping factor for extension stroke
ζ_c	Value of damping factor for compression stroke
ζ_{on}	Value of damping factor for the "on" stroke
ζ_{off}	Value of damping factor for the "off" stroke

$\zeta_{on_{extension\_1}}$	Value of damping factor when $\dot{z}_1 < v_{e1}$
$\zeta_{on_{compression\_1}}$	Value of damping factor when $\dot{z}_1 < v_{c1}$
$\zeta_{on_{extension\_2}}$	Value of damping factor when $\dot{z}_1 > v_{e1}$
$\zeta_{on_{compression\_2}}$	Value of damping factor when $\dot{z}_1 > v_{c1}$
$\zeta_{off_{extension\_1}}$	Value of damping factor when $\dot{z}_1 < v_{e2}$
$\zeta_{off_{compression\_1}}$	Value of damping factor when $\dot{z}_1 < v_{c2}$
$\zeta_{off_{extension\_2}}$	Value of damping factor when $\dot{z}_1 > v_{e2}$
$\zeta_{off_{compression\_2}}$	Value of damping factor when $\dot{z}_1 > v_{c2}$

CHAPTER 1

INTRODUCTION

1.1 General Introduction

A commercial ground vehicle in general, should provide comfortable ride to passengers while the stability of the vehicle is maintained. The terrain input motion which is transmitted to a vehicle through its suspension should be isolated to ensure a comfortable ride. Suspension systems on any ground vehicle provide certain strategic features which are of practical importance to both the rider and the vehicle. The major function of a suspension system in ground transport vehicles is to isolate both passenger and freight from disturbances due to terrain irregularities and external forces. This must be achieved with a minimum loss of road to wheel contact in order to maintain good vehicle stability. Hence, the practical design of a suspension system is a compromise between road holding ability and ride comfort.

Suspension design concepts and their advancements are reported in this chapter. Discussions on vehicle suspension state-of-the art surveys and several aspects of suspension design are presented and relevant literatures are reviewed. The various school of thoughts and arguments with regard to the general classifications of suspensions such as Passive, Semi-active (SA), and Active suspensions are also presented.

1.2 Review of the Past Work

1.2.1. The State-of-the-Art Survey on Vehicle Suspensions

There are several state-of-the-art surveys on vehicle suspensions [1-5]* which discuss work done in the past. These include literature

* Numbers inside a square bracket represent references

review of road, off-road, and rail vehicles. Generally in the case of road vehicles, low frequencies are of prime importance, with the chassis structural flexibility and tire carcass elongation properties, having secondary influence in these frequencies. However, some discussions are also reported on high frequency vibration isolation in ground vehicles.

Sharp and Crolla [1] have reported the work done in vehicle suspensions with an emphasis on road vehicles and their vertical motion, and modeling of road inputs. Their work also focused on tires and overall vehicle systems. While considering the cases of passive, SA, active, and slow-active suspensions, various methods of control strategies and comparative analyses on performances were discussed. Bernard, Vanderplog, and Shannon [2] have reviewed active suspensions and their development in road vehicle dynamics.

Goodall and Kortum [3] presented a detailed state-of-the-art survey on active control strategies that are applicable to both road and guided rail vehicles. The survey is mostly focused on guided rail vehicles and their lateral dynamic problems. However, some discussions on steering controls and damper orifice controls were also presented. Hedrick [4] had reported on railway vehicle active suspensions. Law and Cooperider [5] reviewed conventional railway vehicles mostly concentrating on the lateral dynamics, stability and kinematics of the wheel/rail contact and the guidance tracks. The different aspects of suspension design which were dealt with, in the surveys can be listed below:

. Ride quality improvement, road holding or both.

. Analytical concepts related to vehicle suspension modeling.

- Controller design.
- Pneumatic and hydraulic components in force generation.
- Analytical techniques for solving system equations.
- Experimental set-up and implementations.
- Road profile characteristics.
- Tracking and stability.
- Performance assessment.
- Future scope and new trends.

A review on some of the literatures related to these aspects of vehicle suspensions is presented in the next section.

1.2.2 Review of Literature

Ride quality was investigated extensively, and analytical concepts relating to suspension design were examined [6-8]. However, in achieving active vibration isolation, active control techniques have been growing with interest. Researchers have been using various types of control strategies to achieve active vibration isolation while not compromising handling and stability characteristics during cornering and braking. Analytical investigations using computer simulations to obtain vehicle system response were validated through experimental investigations [9-11].

The performance evaluation of a suspension system is often expressed in terms of transmissibility ratios of the desired system response such as sprung mass acceleration level, rattle space requirement, and tire deflection characteristics. There were also other performance indices [12,13] reported, as a combination of these common performance indicators such as, passenger discomfort in terms of weighted rms acceleration levels, and dynamic tire load variation for a given suspension working space.

Passive Suspension

The conventional passive system is shown schematically in Fig.1. It mainly consists of a spring and a dissipative damper with a possible addition of a self-leveling mechanism. A good design of a passive system requires a compromise between the different performance characteristics of the vehicle system. The conventional passive system have been studied widely [14-19]. Thompson [14], and Ruzika [15] discussed the vibration isolation performance and limitations due to the fixed parameters of the system. The compromising performance effects of the system were reported by A.G. Thompson [16], and Ryba [17].

Passive Suspensions with Self-Leveling Mechanism

For a conventional passive system, the spring rates and damping values are constants. Hence the performance of passive systems cannot be improved beyond a certain level. However with certain changes to the configuration of passive suspensions, such as a self-leveling feature, passive systems could be developed to achieve some advantages which are generally not available from simple passive suspensions. Self-leveling is required to compensate for the variations due to the static load when in ideal case, the suspension system, initially loaded would respond

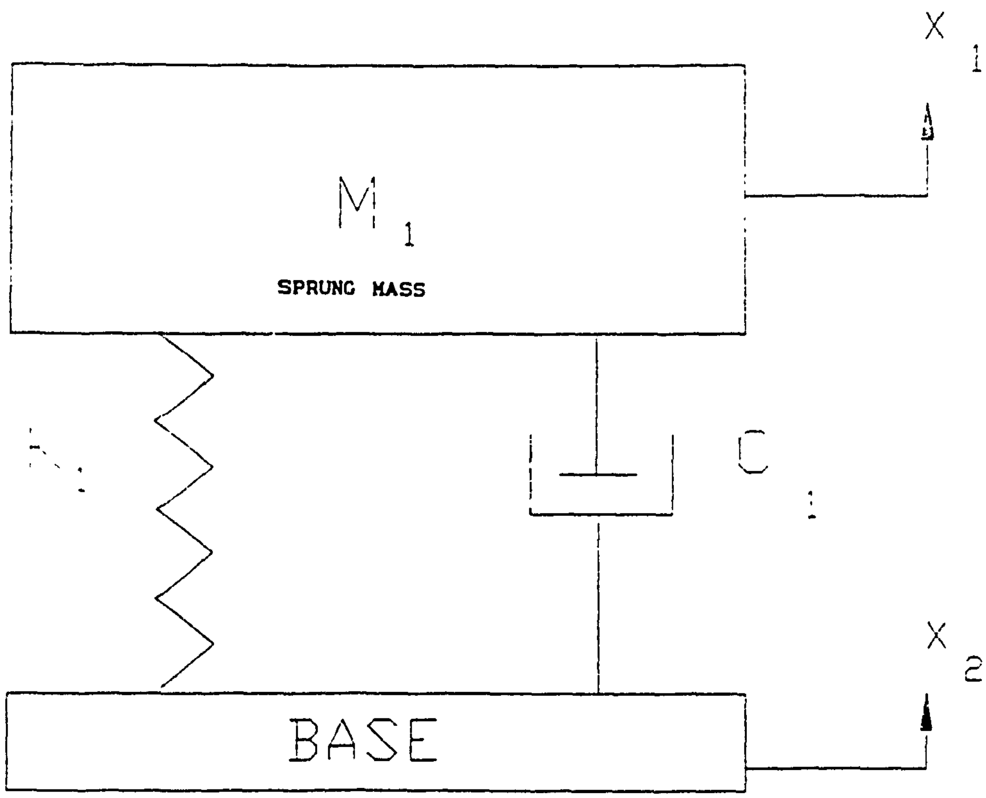


Fig. 1.1 A single degree-of-freedom model of a passive suspension

with some deviations in its original characteristics. There exists a time lag in response of such systems due to the initial static load phenomenon. A very small amount of external energy (a hydraulic strut with oscillating pendulum) is usually available to maintain constant suspension deflections to ensure that the suspension travel is always available.

Commercial suspension systems using air as medium, and capable of self-leveling have been promoted by various industries like Citroën, Ford, Lucas-Girling, and Dunlop [9]. This type of suspension system is developing fast and is often of interest for future research.

Semi-Active (SA) Suspension

Semi-active suspensions are a compromise between passive and active suspensions, as the fully active suspension systems are complex, expensive and hard to maintain. The concept of Semi-Active suspensions was first proposed by Karnopp and Crosby [20] in 1973. The damping orifice is modulated to generate desired forces. It attempts to emulate the sky-hook active damper for certain parts of vibration cycle. That is when the actual damper force and the desired force are of opposite signs the damper assumes its off state, generating no force. The response of an SA-suspension system, reported in [20,21] showed that the response followed closely an active suspension system when a control scheme based on relative and absolute velocities is employed. The hardware required for this SA-suspension system is accelerometers, relative velocity and displacement transducers (operates at low signal power level) and low level power supplies for modulating the fluid flow through the orifice of a hydraulic cylinder with a pair of poppet valves to control damping force. The force required to open the valve is set by a torque motor or a staged force amplifier requiring small power supply.

SA-suspensions can have either a continuous or an on-off control for the damping orifice. In an SA-suspension system with on-off type of control of damping, orifice complexities are lesser, and there is no need for any external power.

The on-off SA-suspension systems, are passive damping devices that modulates flow through an orifice area, according to the control algorithm. When the control is on, the damping will be corresponding to a fixed position of the orifice. For the off position, the orifice is fully open. This process can be achieved by using a hydraulic damper in which the oil flow path area can be modulated to have the required energy dissipation.

For higher frequency bandwidth, the practical realization of such SA-suspension systems may be difficult, but efforts are under way to implement various such SA-systems. The control strategy demands the switching of forces in SA-systems, similar to active suspension systems. In the active suspension systems, the passive damper is replaced by an active force generator (Fig 1.2) or an ideal actuator capable of producing any control force with any sign (direction) dictated by any control law employing state feedbacks. It is also possible to have an actuator or a force generator work semi-actively in response to the control laws. The force generator replaced by a passive damper

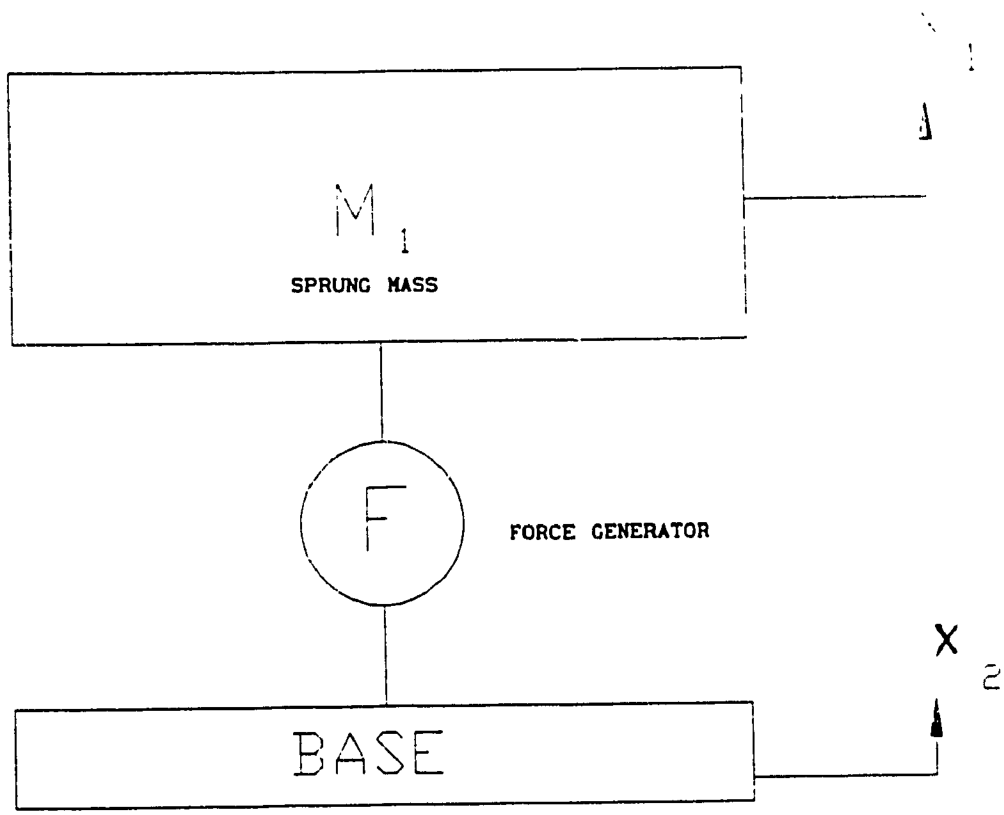


Fig. 1.2 A single degree-of-freedom model of an active suspension

(modulated) in the case of SA-suspensions can be controlled by the same measurement of the state as that of the active system of suspensions. Hence, the feedback laws controlling active systems could be applicable to semi-active system of suspensions.

Krasnicki [11] examined an SA-scheme for the rear suspension of a motorcycle. Extensive studies were carried out on SA-suspensions and reported [22-25]. Margolis [23] presented a study of a 2 d-o-f model SA-suspension system in an effort to provide control of the wheel hop. Alanoly and Sankar [24] analyzed various SA-suspension schemes in a single degree-of-freedom model. While efforts were made to study SA-suspensions that require on-off type of orifice control, Boonchanta [25] presented studies on SA-schemes that operates using continuous type of orifice control.

Karnopp [26] reported the advantage of semi-active suspensions that uses control function based on velocity signals, and generation of damping force through hydraulic and electrical devices. A criteria was also presented indicating whether or not the damping will give improved performance. He also, presented [27], a study based on a hydraulic SA-damper with a double ended piston rod, and mechanically coupled valves. An electro-magnetic SA-system providing improved damping (or dissipation of energy) through electrical brakes, by causing higher resistance, was also reported [27]. Rakheja and Sankar [28] presented a study on the different control schemes involving absolute and relative velocities. Oueslati [29] studied a comparative performance of active and semi-active suspensions based on an in-plane vehicle model.

Active Suspension

Active suspensions are defined as a system in which an actuator either replaces or acts in parallel with the passive components. Energy, usually a significant amount, may be put into or taken out of the system and the operational frequency bandwidth is defined by the frequency response characteristics of the actuator and the control components. In short, the active force generator controls the movement of vehicle mass. This is normally conceived so as to minimize the transmitted acceleration while staying within its travel limits. The principle of such active force generators is to produce a force independent of the input, within the smallest time interval.

The concept of theoretical and experimental active suspension systems have been investigated by several researchers earlier [10,30]. Also several research studies have been reported [31-37] in the use of active suspension concepts. Hac [31] has reported suggestions for use of an adaptable control method based on linear optimal control theory in active vehicle suspensions. The use of active suspension systems in rail vehicle studies were reported in [1,3]. Lotus [38,39] showed the evolution of active suspensions in its racing cars with the use of ground effects. Study on full car model with 7 degrees-of-freedom [7,31,40,41] has been reported with active suspensions. Some of the practical on road car models having adaptive suspensions, like TEMS [42] (Toyota Electronically Modulated Suspensions) use two or three options of suspension (damping) settings.

The active control techniques have been developed extensively, and the wealth of literatures reported on the same are numerous. Some of the commonly used control methods are, however, reported in the study of active suspensions. Burdess and Metcalf [43] used a robust controller while Fruehauf, Kasper, and Luckel [40] used a Ricatti state controller in solving optimization problem. Towards the application of optimal control theory in active vehicle suspension systems, various studies were conducted [6,7,33,35,41,44,45] in the past. A study on verification of the physical realization of optimal control system in active suspensions was also presented [33].

1.3 Scope of the Study

In this thesis, different types of semi-active suspension configurations are investigated using a quarter car model. The vehicle model is based on a 2 d-o-f representing the bounce motion of both sprung and unsprung masses.

The SA-vehicle response in terms of bounce acceleration of sprung mass, and relative displacement of the sprung and unsprung masses are presented as transmissibility plots. Computer simulation results for one type of SA-suspension system are compared with laboratory testing of an experimental set-up.

In Chapter 1, a brief general introduction to the problem is presented. Review of the past work is also presented to highlight the state-of-the-art survey on vehicle suspension, and to discuss literatures on passive, semi-active and active suspensions.

Chapter 2 deals with the system concepts and modeling of various types of SA-suspensions, their control schemes, and conditions for system lock-up. A detailed discussion is presented on the solution procedure for solving the equations of motion of SA-suspension systems. The importance, and the effect of using a zero-finder in the solution methodology for evaluating the responses of SA-suspensions are also presented.

In chapter 3, the study on vibration isolation properties of 3 types of SA-systems are presented and compared with passive suspension system. The response of SA-suspensions are investigated in terms of transmissibility ratio of sprung mass acceleration response, suspension rattle space, and unsprung mass relative displacement. While evaluating the response of the SA-systems, the discontinuous equations are solved numerically in conjunction with a zero-finder. Steady-state responses are presented to provide a detailed insight into the response behavior of SA-suspensions. An effort is made to study the influence due to solving the discontinuous equations of SA-systems without using a zero-finder and observations are made.

One type of SA-system, SA-type 4, is both analytically and experimentally studied using a two d-o-f quarter car bounce model. A Nissan 300ZX automobile's adjustable rear shock absorber is used in the laboratory testing. The damping characteristics of the shock absorber is also evaluated using the test rig. A parametric study is presented to investigate the influence of various damping values for the shock absorber. The Experimental responses of SA-type 4 system are compared with firm, normal and soft type suspensions. The analytical model is suitably modified according to the experimental conditions and the responses for SA-type 4, firm, normal, and soft type suspensions are evaluated and validated against the experimental results. The vibration isolation performance of a typical road vehicle quarter car model (Nissan 300ZX) is then analytically evaluated using SA-type 4, firm, normal, and soft type suspension systems.

Chapter 5 presents the conclusion with highlights on the overview of both analytical and experimental analysis. Discussions are presented on performance abilities of SA-suspensions over passive systems. Finally the recommendations for future study on both analytical and experimental investigations of SA-systems are presented.

CHAPTER 2

VEHICLE MODELING AND SOLUTION METHODOLOGY

2.1 Introduction

The objective of the study is to analyze different types of semi-active (SA) suspension configurations and to provide a detailed insight into the various feasible control strategies. In the following section, details of the various models studied in the past, and the model currently being proposed are presented. The analytical model and the solution methodology are discussed.

2.2. Vehicle Modeling for SA-Suspensions

Several vehicle models have been used to study semi-active and active suspensions. Malek and Hedrick [46] reported a study on active suspension using a full-car model with 7 d-o-f allowing the suspension inter-connection effects of one on the other. The roll and pitch modes were decoupled, while taking into consideration the coupling stiffness and damping rates between various modes. Although the complete response of vehicle suspension systems are studied using a full-car model, often in-plane models are used on vehicle suspensions, limiting the study to pitch and bounce response. Krasnicki [21] tested a motorcycle prototype model with a semi-active rear suspension. Vehicle models using equivalent springs, and damping have been conceived in the past [20,47].

Ananthanarayana and Yoshima [48] investigated the use of active secondary suspensions using a simple 4 d-o-f track/vehicle system model having pitch, c.g.bounce, and bounce of leading and trailing truck. Chalasani [6,7] investigated, both a single-wheel station, and a

7 d-o-f full-car model. Boonchanta [25] performed a study based on an in-plane or bi-cycle model, and compared the performance of passive and semi-active suspensions.

Horton and Crolla [8] presented a model, using a hydro-mechanical system fitted to each of the single-wheel station, for an off-road vehicle suspension. A pendulum mass attached to the sprung mass was used to control the orifice for the required hydraulic flow. The detailed study and conceptual knowledge on semi-active systems are generally introduced using a single-wheel station or an 1 d-o-f quarter car vehicle model. Karnopp [49] proposed a class of active suspensions using 1 d-o-f quarter car models of different configurations.

Céch [50] reported an interesting in-series 2 d-o-f quarter car, and 4 d-o-f in-plane models. The study showed improved performance when analyzed for roll and bounce modes. Oueslati [29] used a 1 D-O-F, and a 4 d-o-f in-plane model to compare three different types of SA-suspensions. He also compared their performance against fully active and passive suspensions. The detailed arguments about how it is reasonable to use a simple model rather than the multi degrees-of-freedom system were presented [18]. Healey, Nathan, and Smith [19] emphasized that if the events at any particular position or corner of a vehicle is dominating due to its suspension in that location, then a simple model is preferable, and considered practical and significant.

Although a number of different models with single and multi-d-o-f have been investigated, there is generally a lack of experimental results to validate the models or to compare their performances. Hence, in this investigation a 2 d-o-f model with different types of

SA-suspensions are investigated with the laboratory testing and validation of one of the types of SA-suspension system.

2.3. Condition Function

In SA-suspension systems, a control logic is used to switch the damper force between two different values (high/low). Different control logics have been developed and used by several researchers based on different reasoning. In this section, a brief discussion on different control logics is presented.

The control logic, also referred to as the condition function, is derived from system sprung and unsprung mass, absolute and relative response variables such as velocity, displacement, acceleration, and the pressure differences across the damper chambers, etc., to keep the system damping forces ON or OFF. In a semi-active scheme, since no power is supplied, the force produced is such that the power associated with the damping force is always dissipative.

A review of the semi-active scheme of controls and their basis with an insight for their use are discussed in the following text.

With reference to a single d-o-f system, Alanoly [51] studied the effect of 6 types of control schemes: three were based on continuous control and three used on-off control. An experimental testing on one type of control was also carried out.

In his work, the damper force and the control function used are as follows:

a)Type 1 [51]:

$$\left.\begin{array}{lll} F_d = C\dot{X} \;, & \dot{X}(\dot{Z}) > 0 \\ F_d = 0 \;, & \dot{X}(\dot{Z}) < 0 \end{array}\right\} \qquad (2.1)$$

where C = Damping coefficient (or $2\,\zeta\,\omega_1 m_1$)

F_d = Damping force

Z = Relative displacement

$\dot{Z}$ = Relative velocity

X = Absolute displacement

$\dot{X}$ = Absolute velocity

The logic for this control function is based on the principle that, when the conventional damping force, ($C\dot{Z}$), and sky-hook force, ($C\dot{X}$), oppose each other, the damping force for the semi-active suspension is turned to the "off" stage, and when they are in the same direction, the damper force is turned to the "on" stage with a value of $F_d = C\dot{X}$

b)Type 2 [51]:

$$\left.\begin{array}{lll} F_d = C\dot{Z} \;, & \dot{X}(\dot{Z}) > 0 \\ F_d = 0 \;, & \dot{X}(\dot{Z}) < 0 \end{array}\right\} \qquad (2.2)$$

The control logic is the same as in Type 1 SA-suspension, but the damper force is a function of relative velocity.

c)Type 3 [51]:

$$\left.\begin{array}{lll} F_d = -\alpha\,\omega_n^2\,Z\,, & Z(\dot{Z}) & < 0 \\ F_d = 0 \,, & Z(\dot{Z}) & > 0 \end{array}\right\} \qquad (2.3)$$

Where α is the value of tuning factor

When the spring force and the conventional damping force oppose each other, the damping force for the semi-active scheme is kept "on" with a magnitude proportional to the spring force but opposite in sign. If the spring force and the conventional damping force are in the same direction, then the damper force is set to "off" with zero force.

d)Type 4 [51]:

$$\left.\begin{array}{lll} F_d = C\,\dot{Z} \,, & Z(\dot{Z}) & < 0 \\ F_d = 0 \,, & Z(\dot{Z}) & > 0 \end{array}\right\} \qquad (2.4)$$

The control logic is the same as in Type 3 SA-suspension, but the damper force is a function of relative velocity.

e)Type 5 [51]:

$$\left.\begin{array}{ll} F_d = F_0 \,, & \dot{Z} > 0 \\ F_d = -F_0 \,, & \dot{Z} < 0 \end{array}\right\} \qquad (2.5)$$

The condition function is based on the relative velocity of the system and the damper force is a constant value opposing the motion. Though the damper force can be produced passively, as mentioned in [51] this has been included in SA-systems for the sake of completeness.

f)Type 6 [51]:

$$\left.\begin{array}{llll} F_d = & C_1 \dot{z} & , & |\dot{z}| \le V_s \\ F_d = & C_2 \dot{z} & , & |\dot{z}| > V_s \end{array}\right\} \quad (2.6)$$

V_s, is a critical velocity depending on the natural frequency of the system, and C_1 and C_2 are the damping coefficients. The condition function is based on the relative velocity of the system and the damper force is a function of relative velocity.

Rakheja and Sankar [28,52] have explained the control schemes employing absolute and relative velocity, and relative velocity with relative displacement, and relative acceleration respectively. An on-off concept with 2 different control schemes were also investigated for vibration and shock isolations. Margolis [23] examined the possibilities of different control schemes on a 2 d-o-f model for wheel hop. In this model, inertial control of both the masses represents the best isolation to resonance control.

2.4 Types of Semi-active Suspensions Proposed in this Thesis

In this investigation on SA-suspension systems, a two d-o-f quarter car model in bounce mode, as shown schematically in Fig.2.1 is considered. Four types of condition functions are considered for the SA-suspension schemes. Type 1 and Type 3 are based on continuous control (meaning continuous control of damping orifice with a servo control or a continuously controlled actuator as a force generator). The other two condition functions, Type 2 and Type 4 require only an on-off control of the orifice. As explained earlier, the modulation of the damping force takes place through the control of the flow area in the orifices.

In order to discuss the four types of SA-suspension schemes, the general equations of motion of the SA-system shown in Fig.2.1 are derived:

$$\ddot{X}_1 + \omega_1^2(X_1 - X_2) + F_d / m_1 = 0 \tag{2.7}$$

$$\ddot{X}_2 + \omega_2^2(X_2 - X_3) - F_d / m_2 + (K_1 / m_2)(X_2 - X_1) = 0 \tag{2.8}$$

where $\omega_1^2 = K_1 / m_1$, $\omega_2^2 = K_2 / m_2$, and F_d is the damping force.

The magnitude of the damping force, F_d and the condition function corresponding to each type is discussed as follows and illustrated in Table 2.1.

<u>Type 1</u>:

Type 1 SA-system is based on the sky-hook control of sprung mass as introduced by Crosby and Karnopp [20,21]

This type of semi-active control provides good resonance control of

$$F_d = \left\{ \begin{array}{ll} 2\,\zeta_1 \omega_1 \dot{X}_1 m_1 , & \dot{X}_1 (\dot{X}_1 - \dot{X}_2) > 0 \\ 0 \quad , & \dot{X}_1 (\dot{X}_1 - \dot{X}_2) < 0 \end{array} \right\} \tag{2.9}$$

the sprung mass for a single d-o-f model [51]. Further, this control was also examined for control of wheel chatter [53]. This scheme requires a continuous control of damper orifice. This could be achieved by using a servo-valve with high bandwidth and measurement of absolute and relative velocities.

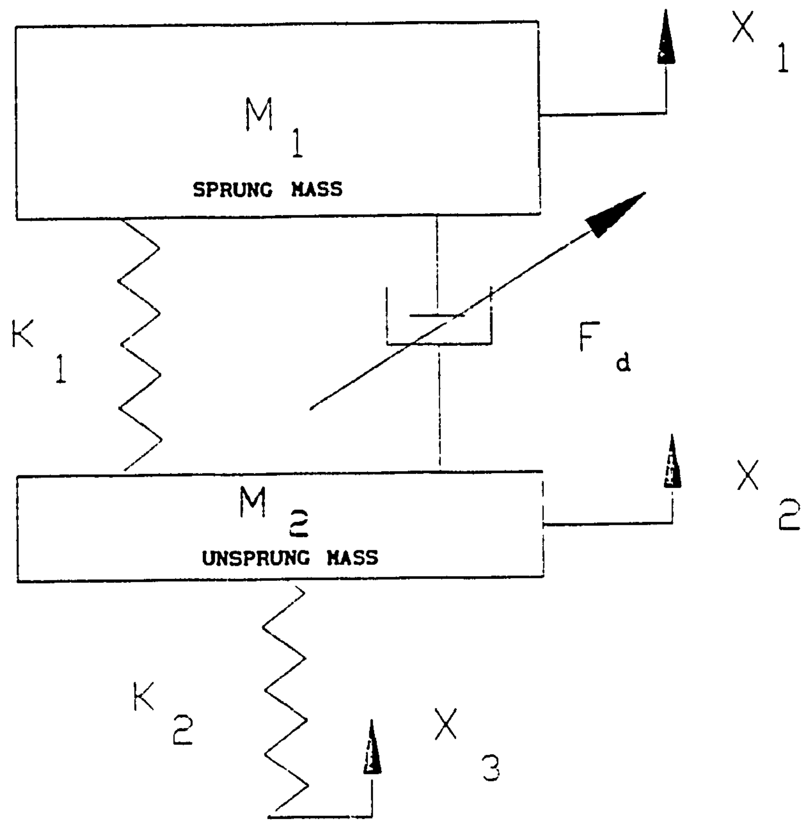


Fig. 2.1 A two degree-of-freedom model of a semi-active suspension

TABLE 2.1. Types of semi-active (SA) suspension system

Type (I)	Condition Function $\Phi(I)$	Damping Force, F_d for $\Phi(I) > 0$	Damping Force, F_d for $\Phi(I) < 0$	Type of Orifice Control
1	$\dot{X}_1(\dot{X}_1 - \dot{X}_2)$	$2\zeta_1\omega_1\dot{X}_1 m_1$	0	Continuous
2	$\dot{X}_1(\dot{X}_1 - \dot{X}_2)$	$2\zeta_1\omega_1(\dot{X}_1 - \dot{X}_2)m_1$	0	On-off
3	$-(X_1 - X_2)(\dot{X}_1 - \dot{X}_2)$	$-\alpha\omega_1^2(X_1 - X_2)m_1$	0	Continuous
4	$-(X_1 - X_2)(\dot{X}_1 - \dot{X}_2)$	$2\zeta_{on}\omega_1(\dot{X}_1 - \dot{X}_2)m_1$	$2\zeta_{off}\omega_1(\dot{X}_1 - \dot{X}_2)m_1$	On-off

Type.2:

This scheme was originally proposed by Roley [54] and the control logic has been tested by others [51] in a single d-o-f model. The type 2 SA-system is a simpler version of Type 1 SA-system with an on-off orifice control.

$$F_d = \begin{Bmatrix} 2\,\zeta\,\omega_1(\dot{X}_1 - \dot{X}_2)m_1, & \dot{X}_1(\dot{X}_1 - \dot{X}_2) > 0 \\ 0 \quad , & \dot{X}_1(\dot{X}_1 - \dot{X}_2) < 0 \end{Bmatrix} \tag{2.10}$$

The type 2 SA-system is basically similar to type 1 SA-system in terms of the control logic. But the expression for force is different in this case with the magnitude proportional to the relative velocity across the damper.

Type.3:

Alanoly [51] studied this condition function in a single d-o-f model.

$$F_d = \begin{Bmatrix} -\alpha\,\omega_1^2(X_1 - X_2)m_1 \;, & (X_1 - X_2)\;(\dot{X}_1 - \dot{X}_2) < 0 \\ \\ 0 \quad , & (X_1 - X_2)\;(\dot{X}_1 - \dot{X}_2) > 0 \end{Bmatrix} \tag{2.11}$$

where α is the tuning factor.

During certain parts of the vibration cycle, the damping force in a passive system tends to increase the acceleration of the sprung mass when the spring force is in the same direction as the damper force. Hence, when the spring force and the damping force are in opposite directions, and have equal magnitude, then the net force acting on the sprung mass will be zero. This concept is the basis for this scheme. The control function in this scheme is based on the relative velocity and the relative displacement of the sprung mass which makes this type more attractive for implementation in moving vehicle suspensions. This scheme also requires a continuous control of damper orifice.

Type 4:

Earlier studies were carried out on this condition function [28,55] with quadratic damping. A study was also conducted in a single d-o-f model SA-suspension for the same control logic but with the viscous damping instead of quadratic damping [24,51]. In this scheme,

$$F_d = \begin{cases} 2\,\zeta_{on}\,\omega_1\,(\dot{X}_1 - \dot{X}_2)\,m_1\,, & (X_1 - X_2)\quad(\dot{X}_1 - \dot{X}_2) < 0 \\ 2\,\zeta_{off}\,\omega_1\,(\dot{X}_1 - \dot{X}_2)\,m_1, & (X_1 - X_2)\quad(\dot{X}_1 - \dot{X}_2) > 0 \end{cases} \qquad (2.12)$$

In this control function, the damper force is generated by a viscous damper having two different damping values namely high, and low (firm/soft). This scheme could be implemented using an on-off orifice control.

2.5.Solution Methodology

Semi-active systems are inherently non-linear due to the discontinuous system equations. It has been customary to compare the performance of suspension systems using frequency response analysis. In the case of SA-systems, a comprehensive frequency response transfer function does not exist. Many a times it has been found to be convenient to solve for the system response in time domain for different input frequencies using a sinusoidal base excitation. In addition to the conventional integration routines like Runge-Kutta fourth order method, other methods [8,56,57] were also used for obtaining the system response. Adams-Merson method of integration technique [8], and an iterative technique based on Quasi-Newton method [56] are some of the methods that were used in solving for the time histories.

2.5.1.Solution of Discontinuous Ordinary Differential Equations (ODE)

The governing equations of a physical system that are discontinuous,in general, require special treatment during the solution process. In the case of a semi-active suspension system, the governing equations are discontinuous due to the switching of the condition function. A changing sign from a positive to a negative value of the condition function, requires the magnitude of the damping force to cross a zero value. Thus, there is a definite discontinuity in the system equations.

In order to treat this condition of discontinuity, the differential equations are solved until the condition function switches from one mode to the other. At that instant, the differential equations are modified according to the condition function and the solution proceeds with a new set of initial conditions. The initial values are nothing but the final values of the dependent variables before switching.

Since the accuracy of the solution of the discontinous differential equation depends on how exactly the initial conditions are approximated at the time of switching, there is a need to find precisely the zero crossing of the condition function. When the condition function is zero, the system may have lock-up as described in section 2.3.3. Hence, there is also a need to find precise zero-crossing of the condition function in order to treat for the lock-up condition. A discussion is presented on lock-up in the end of this section. Thus, the purpose of finding out the zero value of the condition function is summarized as:

1. To estimate precisely the new set of initial values for continuing the solution process.

2. To treat precisely the lock-up condition at the zero-crossing of the condition function.

Methods for Solving the Discontinuous ODE

Simon Ola Fatunla [58] discussed the various available methods for solving initial value problems with discontinuity. Gear [58] provided a procedure to restart integration beyond the point of discontinuity:

a) pass the point of discontinuity,

b) detect and locate the presence of discontinuity, and

c) restart the integration process.

For a condition function, ϕ (X,Y), when there is a change in sign i.e.,

$$\phi\ (X_i,\ Y_i)\ > 0 \text{ and } \phi\ (X_{i+1},\ Y_{i+1}) < 0,$$

or

$$\phi\ (X_i,\ Y_i)\ < 0 \text{ and } \phi\ (X_{i+1},\ Y_{i+1}) > 0,$$

there exists a zero-crossing at $t = t^*$ which lies between (t_i) and (t_{i+1}) and the condition function satisfies $\phi\ (X,Y)_{t=t^*} = \phi^* \cong 0$ This implies the existence of a root or zero crossing which causes the system of the original set of equations to have discontinuity at this instant.

Solution of ODE with discontinuities can be obtained using methods that require derivative of the condition function to be continuous [52,53,55,56]. There are, however, direct methods that does not require time derivative of the condition function and will always converge to the solution [51]. These direct methods for finding out the zero crossing of the condition function are based on finding the roots of the function using methods such as the secant method, false position and the bisection methods. Alanoly [51] used the bisection method as the zero finder. These methods require more number of iterations to find a zero depending on the level of accuracy of the solution desired. In this study, a single step secant method is used to find the zero crossing of the condition function.

A check is made at every time step in the process of simulation to find out whether a zero crossing occurs. When it is verified that the zero crossing occurs, then the time, t^* that approximates the time at the zero crossing is calculated from Fig. 2.2 as:

$$t^* = t_i + \frac{|\phi_i|}{|\phi_i| + |\phi_{i+1}|} . \partial t \qquad (2.14)$$

where

∂t = Step size

ϕ_i = Value of the condition function at the instant of time t_i

ϕ_{i+1} = Value of the condition function at the instant of time t_{i+1}

t^* = The instant of time at which the zero crossing takes place

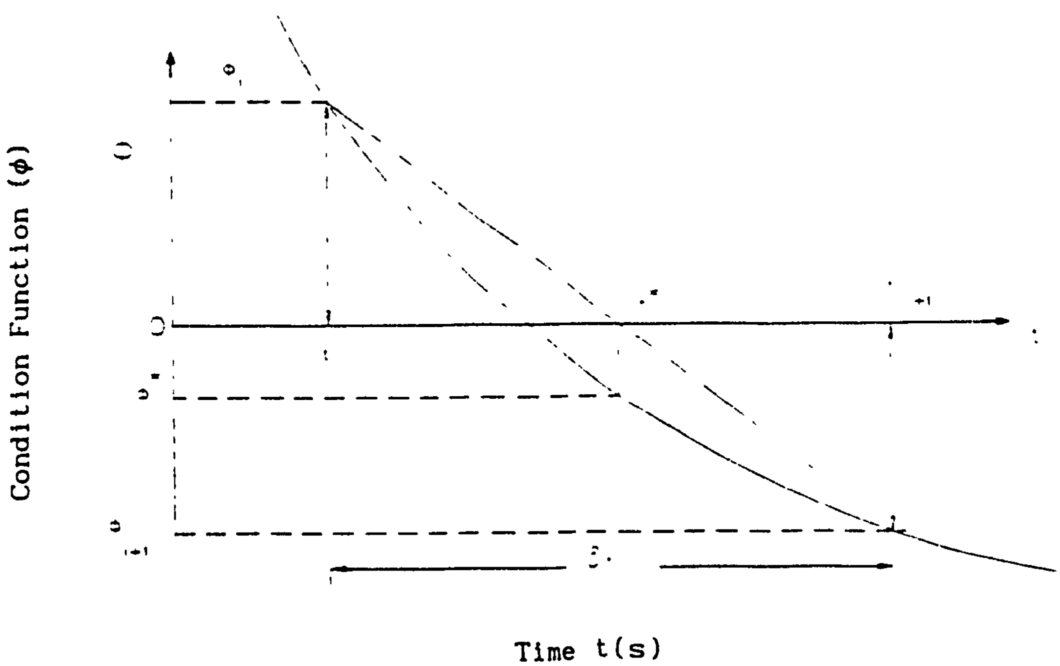


Fig. 2.2 A schematic of the zero-finder method

Since a single step secant method is used, the value of ϕ^*, which should ideally be equal to zero at the zero-crossing of the condition function, may not be zero. The value of ϕ^* will reflect the error involved in this process.

Lock-up Condition

The damping force switches between the two values according to the condition function. When the condition function crosses a zero value, the system has to be checked for the lock-up. The lock-up condition occurs at the exact instant of zero value of the condition function. When the condition function reaches a zero value due to a zero relative velocity between the two masses, the relative motion between the two masses ceases and the system is characterized by a lock-up behavior. The treatment of a lock-up behavior is dealt with in this section with a derivation of an expression for the [illegible] required to break-up the lock-up condition.

For each condition function, there are two possibilities when a zero crossing can occur. The following describes an illustrative example of how the lock-up condition is treated. For this purpose, the Type 1 Semi-Active (SA) suspension is considered. In this case, the conditions for a zero crossings are:

$$\left.\begin{array}{c} \dot{X}_1 = 0 \\ \text{or} \\ (\dot{X}_1 - \dot{X}_2) = 0 \end{array}\right\} \qquad (2.15)$$

When absolute velocity ($\dot{X}_1 = 0$) is equal to zero, the damper force, F_d

is equal to zero. When the system relative velocity is equal to zero, $(\dot{X}_1 - \dot{X}_2 = 0)$, the damper force will be equal to the lock-up force at that instant. The force required to break-up the lock-up condition can be obtained as follows: When the lock-up occurs, the relative velocity between the two masses is equal to zero (or $\dot{X}_1 = \dot{X}_2$). Using this condition, the expression for the force required to break-up the lock-up can be obtained from Equations 2.8 and 2.9, as:

$$m_1 \ddot{X}_2 + K_1 (X_1 - X_2)^{**} \geq F_d^{**} \tag{2.16}$$

where $$\ddot{X}_2 = \frac{K_2 (X_3 - X_2)}{m_1 + m_2}, \text{ and}$$

$(X_1 - X_2)^{**}$ is the value of $(X_1 - X_2)$ at the instant of lock-up, and F_d^{**} is the damper force at the instant of lock-up.

2.6 Summary

In this chapter, the main types of semi-active vehicle suspension models studied in the past are outlined and the model for the present study is proposed. A discussion is presented on the solution methodology for solving the discontinuous system equations. In conjunction, the occurrence of lock-up, and the necessity to find a zero crossing to treat for the lock-up are explained.

CHAPTER 3

RIDE PERFORMANCE ANALYSIS OF A QUARTER CAR BOUNCE MODEL WITH SEMI-ACTIVE SUSPENSIONS

3.1 Introduction

The equations of motion for a quarter car bounce model with semi-active suspension configurations are presented in the previous chapter. The computer simulation results and their comparison with passive suspension systems are presented in this chapter. The proposed study is to examine various semi-active suspension systems, and to evaluate and compare their vibration isolation properties with a passive suspension. A detailed discussion on the feasible semi-active suspension schemes that can be implemented in a laboratory set-up for an experimental investigation is also presented. It is intended that a comprehensive study be provided on the basis of analytical and experimental investigations.

3.2 Performance Characterization

The performance characteristics of vehicle suspensions can be illustrated in terms of vehicle response behavior such as ride quality, road holding ability, cornering properties, and vehicle stability [6,7,13,59,60]. For this study on a quarter car bounce model, the vehicle performance is identified in terms of the three main characteristics [6,7]:

1.Maximum bounce acceleration response of the sprung mass.

2.Maximum suspension deflection response.

3.Maximum tire deflection response.

These three characteristics directly measure the vibration isolation property of a vehicle, namely the ride quality, the suspension rattle space requirement, and the road holding ability. The basic set of system parameters used for calculating the system response are given in Table 3.1 [6,7]:

Table 3.1 Basic System Parameters [6,7]

M_1	240 kg
M_2	36 kg
K_1	16000 N/m
K_2	160000 N/m

3.3 Passive Suspension

Semi-active suspensions have been considered because of their superior ride performance compared to passive suspensions. In order to compare quantitatively the performance behavior of these two types of systems, a detailed response analysis of the passive suspension system is needed. For this purpose, in this section, the ride performance analysis of a quarter car bounce model with a passive suspension system is presented.

Simplified models of vehicles have been used in the past for analyzing the behavior of passive and semi-active systems. Chalasani [6] has carried out a detailed study of a two degree-of-freedom quarter car model and extended the analysis to a full car model [7] for both passive and active suspensions. The schematic of the quarter car model used for passive suspension is presented in Fig 3.1. The sprung mass corresponding to one corner of the vehicle is represented as M_1 and the wheel at one (same) corner of the vehicle as M_2. The suspension is modelled with a linear spring of stiffness K_1 and a linear damper with a damping rate C_1. The tire is represented by a linear spring of stiffness K_2. Since the damping in the tire is very small, it is neglected in this analysis. It is assumed that the tire behaves as a point contact follower, which is in contact with the road at all times.

The ride performance of a quarter car bounce model with the passive suspension system is shown in the form of frequency response plots in Fig. 3.2. The terrain input is considered to be a sinusoidal displacement with constant amplitude.

The influence of damping on the vertical acceleration response of the passive suspension is shown in Fig. 3.2a for three values of damping:high (ζ = 1.0), medium (ζ = 0.25), and low (ζ = 0.050). Since the damping value is very large for the case of passive suspension with high damping, no resonance is observed in the region of sprung and unsprung mass natural frequencies. However, as the damping value is reduced from high to low, the acceleration response is adversely affected with a pronounced peak for the case of low damping.

The maximum suspension deflection response is illustrated by the frequency response plot shown in Fig. 3.2b. ..eduction in damping values show an increase in maximum suspension deflection response at frequencies close to the natural frequencies.

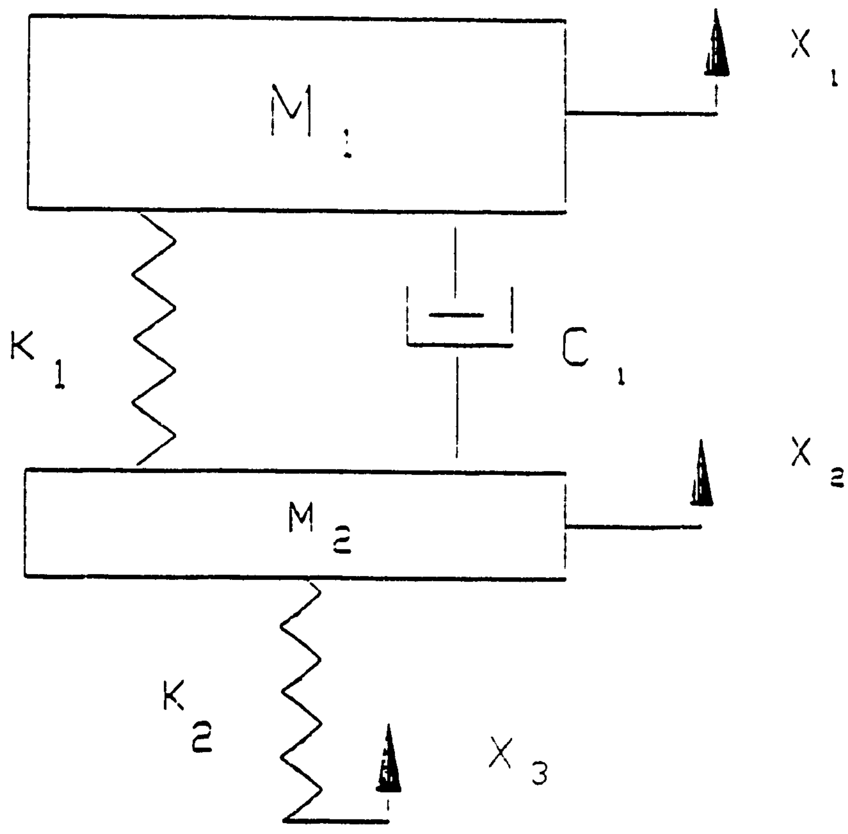
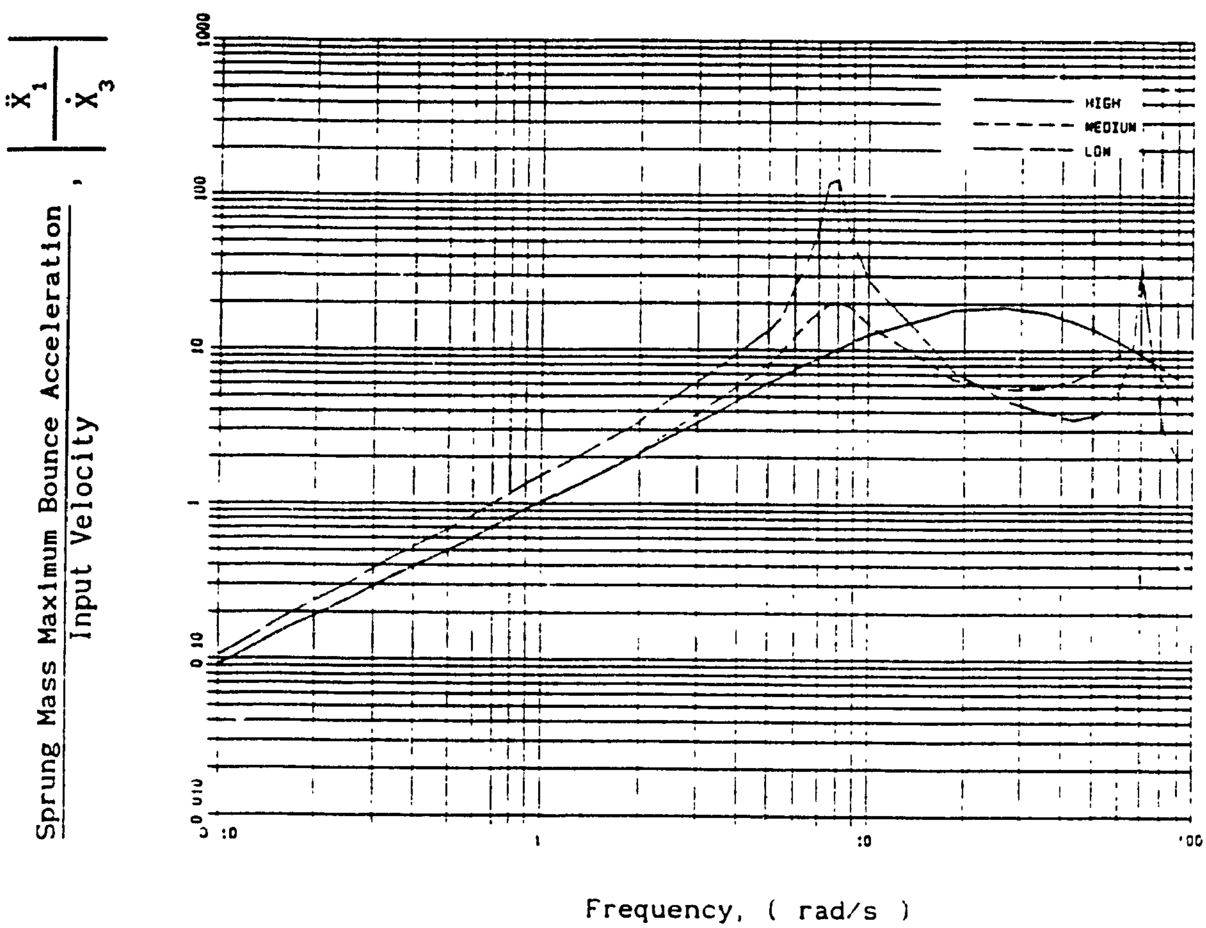


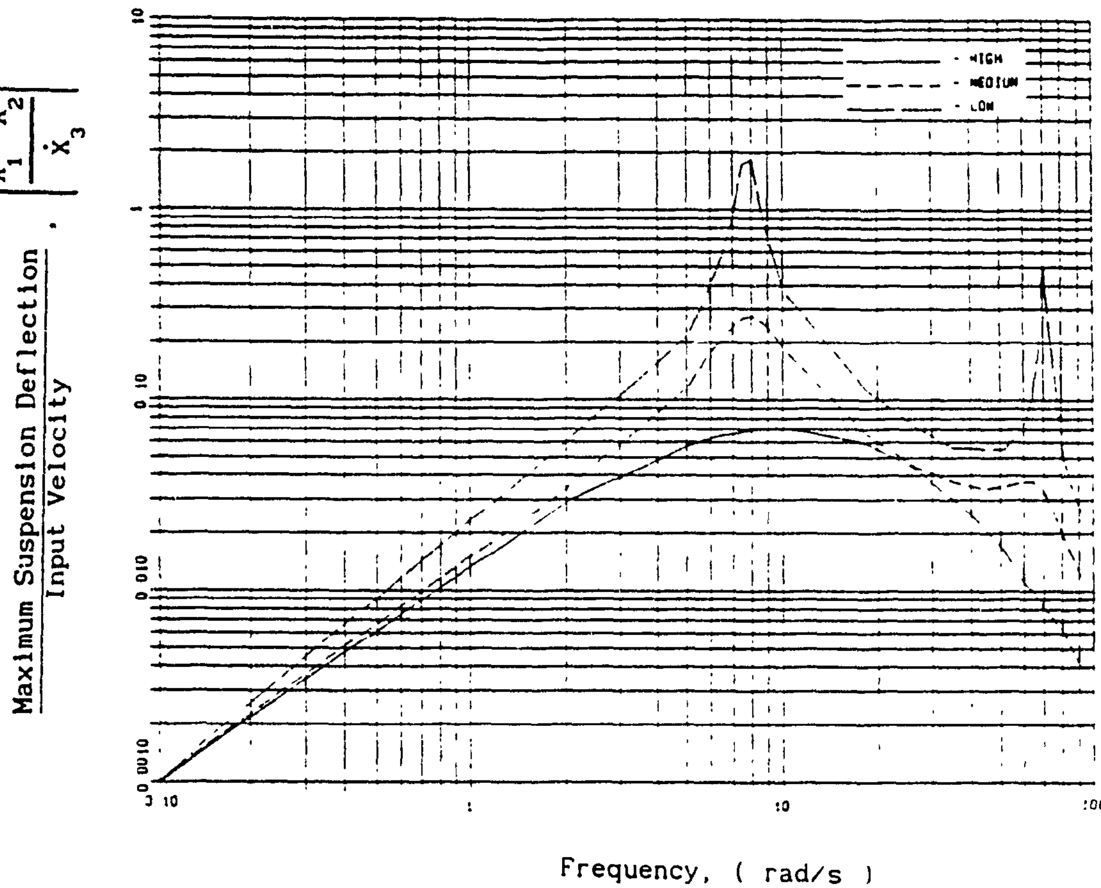
Fig. 3.1 A two degree-of-freedom model of a passive suspension



High : ζ = 1.000 ; Low : ζ = 0.050 ;

Medium : ζ = 0.25 ;

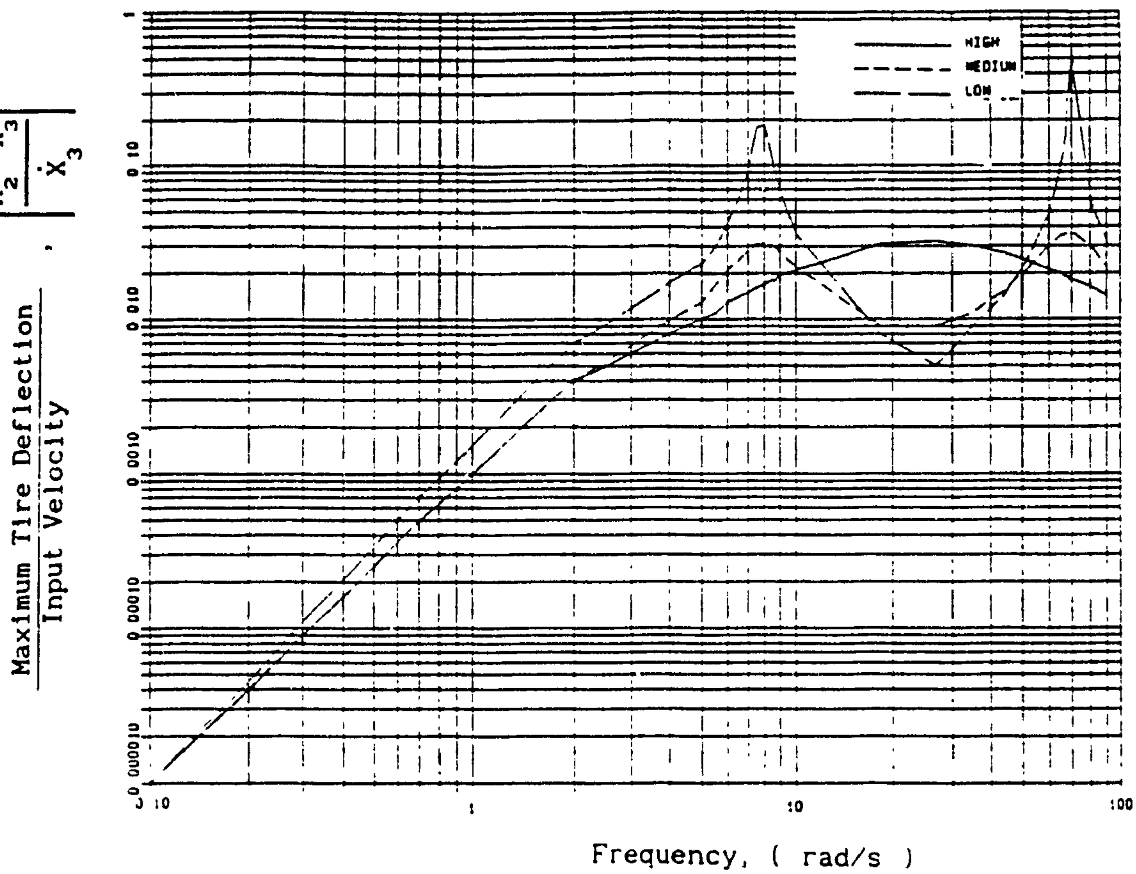
Fig. 3.2a Influence of damping on the maximum bounce acceleration response for a passive suspension



High : ζ = 1.000 ; Low : ζ = 0.050 ;

Medium : ζ = 0.25 ;

Fig. 3.2b Influence of damping on the suspension deflection response for a passive suspension



High : ζ = 1.000 ; Low : ζ = 0.050 ;

Medium : ζ = 0.25 ;

Fig. 3.2c Influence of damping on the tire deflection response for a passive suspension

The influence of variations in damping values, on maximum tire deflection response is presented in Fig. 3.2c. Reduction in the damping value from high to medium, results in increased peaks at the natural frequencies. However for the range of frequencies between the two natural frequencies, the maximum tire deflection response is considerably better than the case with the high damping. Further reduction in the damping value to low damping, the maximum tire deflection response at the natural frequencies become inferior. The response, however, in the region between the natural frequencies, is improved.

3.4 Semi-Active Suspension Systems

The ride performance of a quarter car bounce model with semi-active suspension schemes is illustrated in this section. The terrain input is again considered to be sinusoidal displacement with constant amplitude. The responses presented in this analysis are obtained using a zero-finder in the solution methodology as explained in the chapter 2. Results are also presented at the end of this chapter for response analysis obtained without a zero-finder.

3.4.1 SA-type 1 system

Figures 3.3 , 3.4 and 3.5 show the steady-state time response of the system for frequency ratios ω/ω_1= 0.5, 1.0 and 5.0. In Fig. 3.3 at frequency ω/ω_1= 0.5, though the system undergoes the lock-up mode, one cannot identify explicitly this event from the time history plots. However, it can be seen that the sprung mass has a sharp spike in

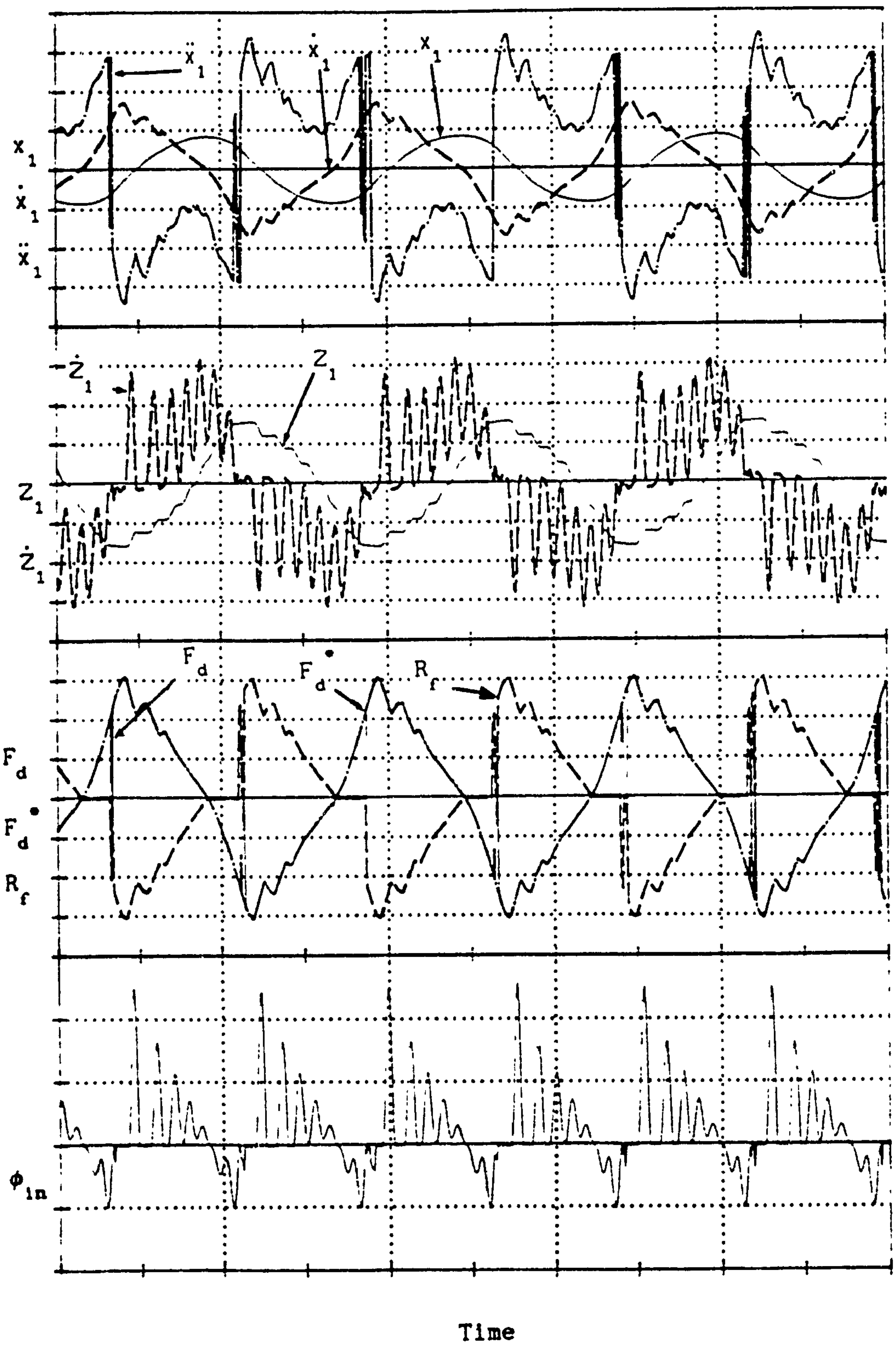


Fig. 3.3 Steady-state response of an SA-type 1 system for $\omega/\omega_1=0.5$

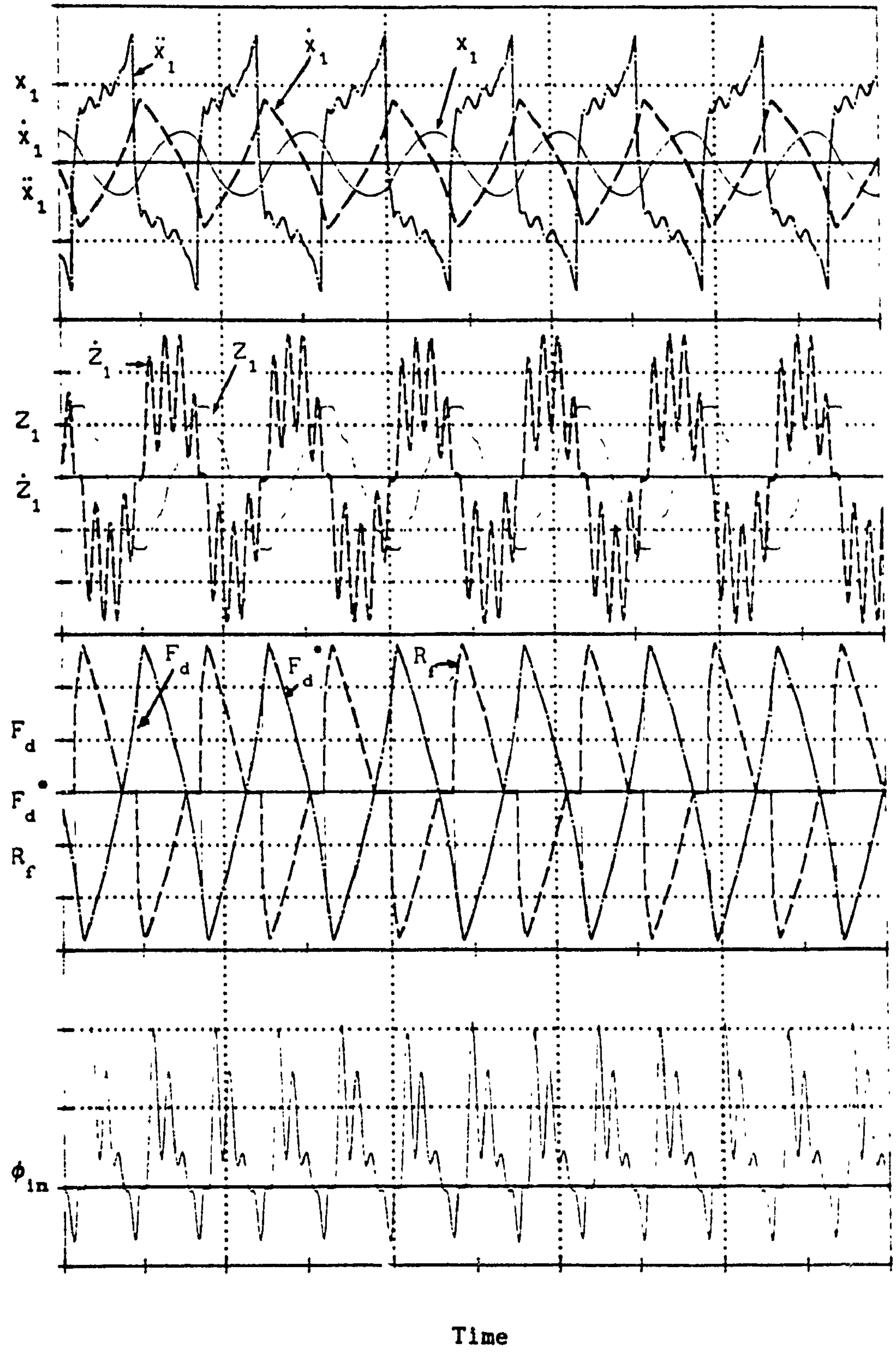


Fig. 3.4 Steady-state response of an SA-type 1 system for $\omega/\omega_1=1.0$

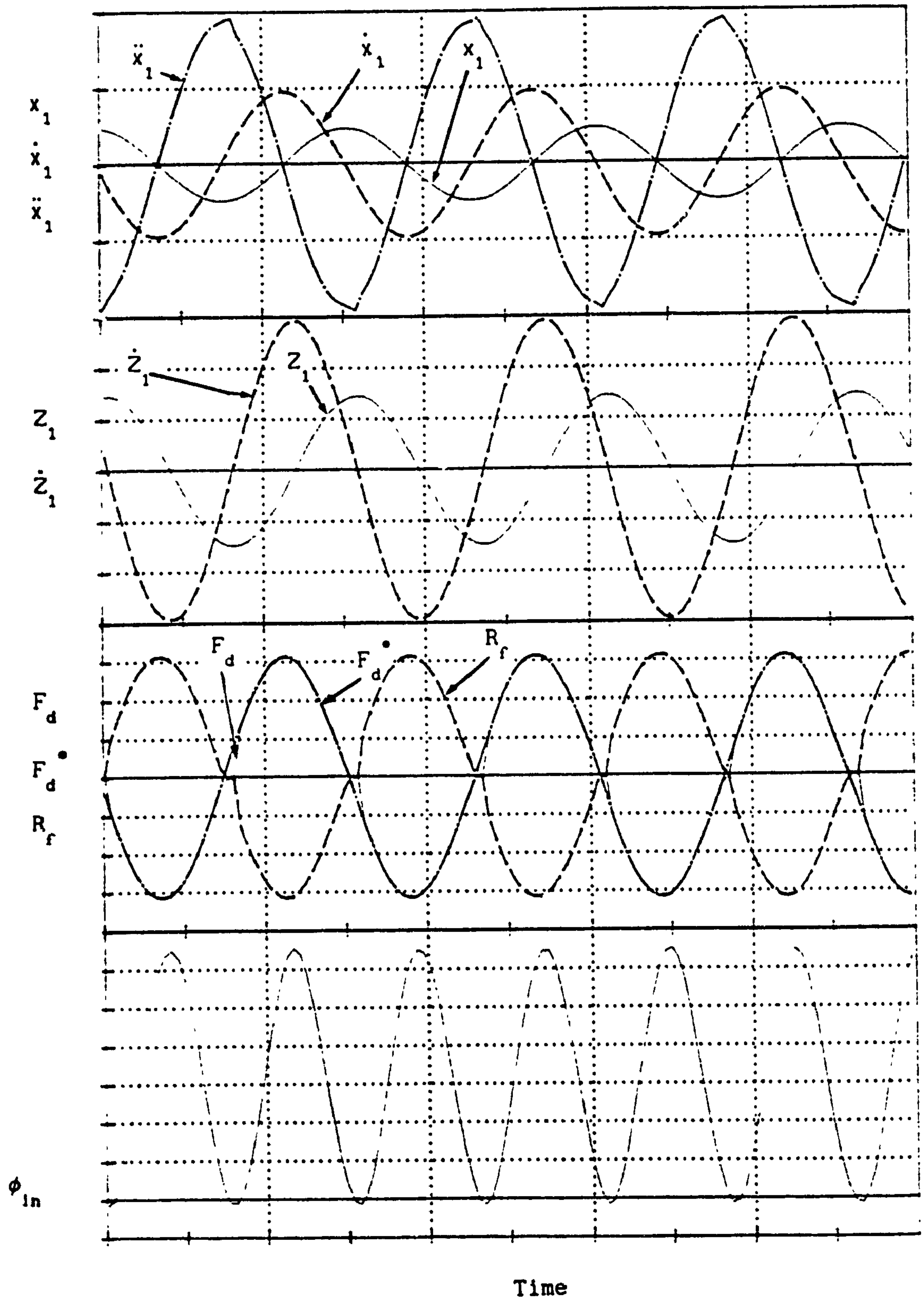


Fig. 3.5 Steady-state response of an SA-type 1 system for ω/ω_1=5.0

acceleration response when it breaks-away from the lock-up condition. During lock-up, the relative velocity response tends to switch back to its original value before lock-up. Due to this behavior, a fluctuation in the condition function is observed just at that instant which introduces a large variation in the damping force and hence a sharp spike in the acceleration response of the system. As outlined in the previous chapter, the discontinuous equations of semi-active (SA) systems are solved using a suitable zero-finder. The use of a zero-finder in the solution facilitate the treatment of lock-up condition. A discussion on the influence of a zero-finder in the solution is presented at the end of this chapter. In addition to the use of a zero-finder, the system is also treated for the lock-up condition.

In Fig. 3.4, the system is excited at its natural frequency and there are inherent fluctuations in the system responses which may be attributed to the lock-up behavior of the system at this frequency also. At higher frequencies as shown in Fig. 3.5, the system does not lock-up and the condition function is smooth and remains almost in the positive region. This nature of the condition function requires the damping force to be "on" for the majority of the time in one period of oscillation. It is also observed that the value of the condition function remains positive for larger period of time in one cycle for all the frequencies. For higher frequencies, the condition function remains positive for almost the entire cycle.

In all the three figures, the semi-active damping force, F_d, and the sky-hook force, ($F_d^* = c_1 \dot{x}_1$), follow exactly each other except for the off stage and at lock-up. The resultant force R_f of the system, namely the sum of the inertial and spring forces are truly opposed by the damping force during the period before and after the change of sign. Thus, it is noted that the responses presented in the steady-state analysis have a trend which is consistent with the system behavior.

For the SA-type 1 suspension system, the steady-state maximum bounce acceleration response over the maximum input velocity is presented in Fig. 3.6 and is compared with the passive suspension system. The SA-type 1 suspension system, which is based on the sky-hook force as the control force, provides similar vibration isolation performance in terms of the maximum acceleration response, throughout the lower frequency range of interest. At the sprung mass natural frequency, the SA-system provides better resonance control. Whereas at the unsprung mass natural frequency, the sprung mass maximum acceleration response reaches a peak value that is comparable to the passive suspension system.

Fig. 3.6 also shows that, by using the sky-hook force as the damping force, the SA-type 1 system can achieve good isolation of the sprung mass between the two natural frequencies of the system.

In Fig. 3.7, the maximum tire deflection response of the SA-type 1 suspension system is compared to the passive suspension system. The response for the SA-type 1 suspension system is inferior than the passive suspension system, for frequencies lower than the sprung mass natural frequency. The response near the sprung mass natural frequency is, however, comparatively better for the SA-type 1 suspension system.

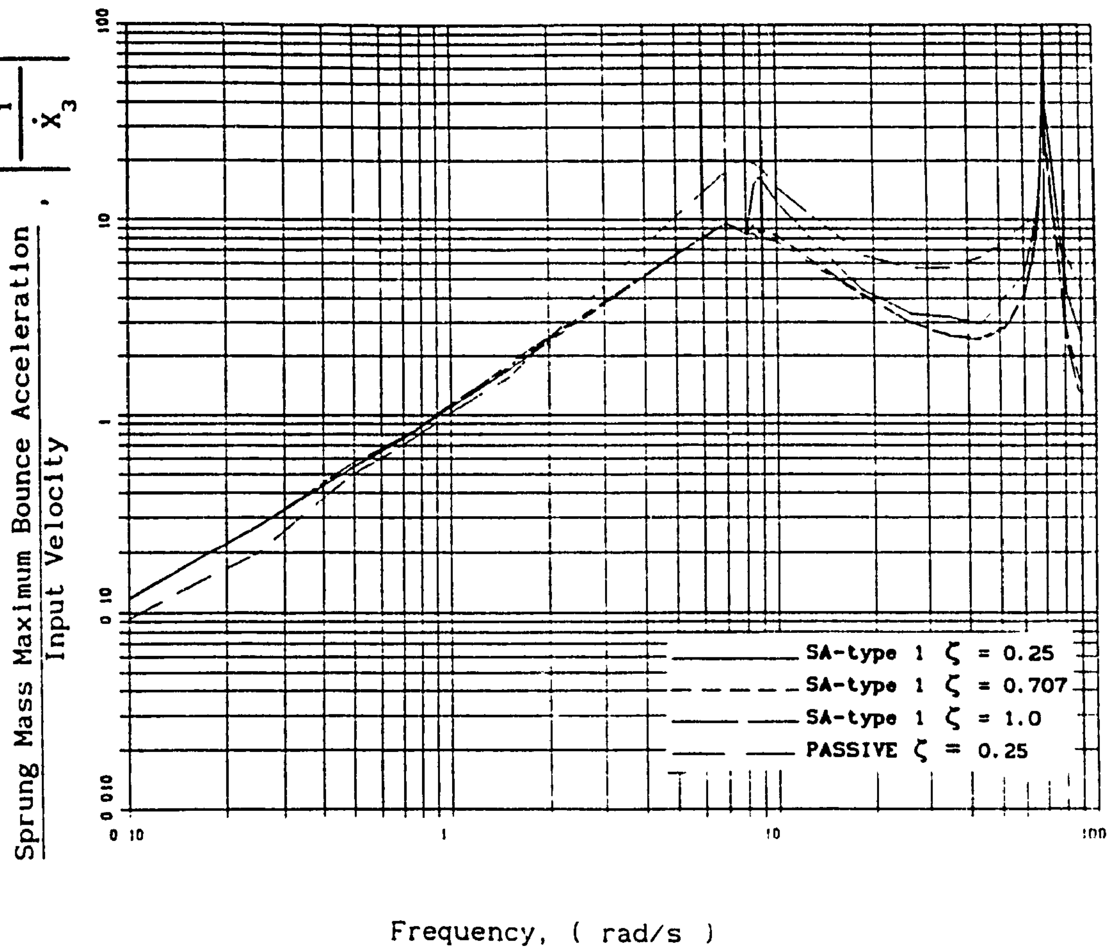


Fig. 3.6 Comparison of the sprung mass maximum bounce acceleration for passive and SA-type 1 suspensions

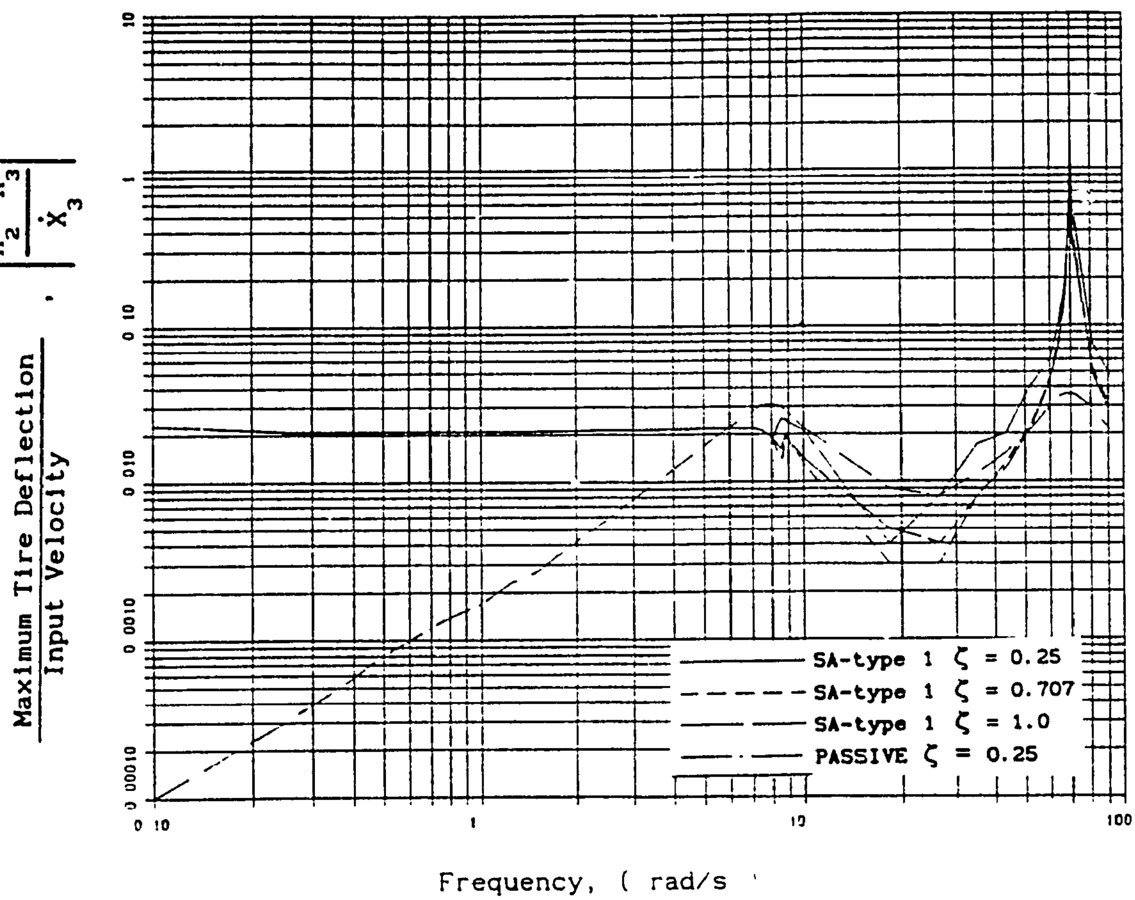


Fig. 3.7 Comparison of maximum tire deflection for passive and SA-type 1 suspensions

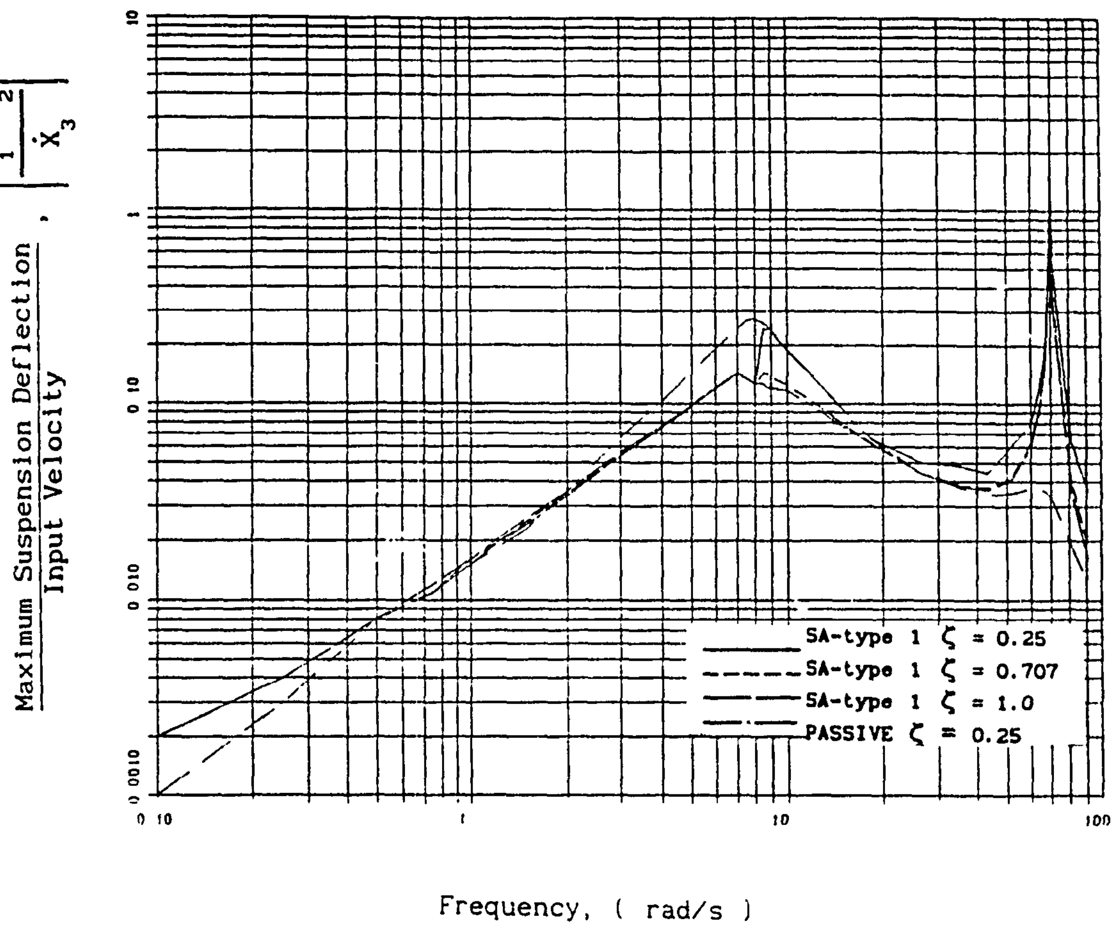


Fig. 3.8 Comparison of maximum suspension deflection of passive and SA-type 1 suspensions

The response is inferior at and beyond unsprung natural frequency. It should be noted that higher the tire deflection, the tire wear and pavement load will be higher.

The suspension deflection response which is an important parameter in terms of good cornering property and available rattle space for the suspension of the vehicle, is presented in Fig. 3.8. The maximum suspension deflection response for the SA-type 1 suspension system is lower than the passive suspension system at sprung mass natural frequency. This response is, however, inferior to the passive suspension system around and beyond the unsprung mass natural frequency.

The suspension deflection response of the SA-type 1 system can be described using the nature of the condition function. The SA-type 1 condition function is very effective in the neighborhood of the sprung mass natural frequency due to the fact that the semi-active force in this control logic is derived to isolate the sprung mass effectively. The response is inferior around the unsprung mass natural frequency, due to the reason that the employed control logic does not isolate the vibration of the unsprung mass as effectively as the sprung mass. For frequencies beyond the second natural frequency, the suspension deflection is, however, controlled adequately and is similar to the passive system.

In conclusion, the response of the SA-type 1 suspension system, in terms of the vibration isolation of the sprung mass is far superior to the passive system. This is achieved at the expense of inferior performance of suspension and tire deflections around and beyond unsprung mass natural frequency.

3.4.2 SA-type 2 system

The SA-type 2 suspension system in principle controls the vibration isolation of the sprung mass in terms of the relative motion across the suspension. This scheme was studied by various researchers [51,53,54]. It has been reported [51] that this system experienced instability at low frequencies when analyzed for a single d-o-f model. It was also reported that solution exists only at certain discrete frequencies and there were numerical instability in the solution process, referred to as "chatter". This is caused due to the instantaneous on/off switching. In a real physical system because, the on/off valve switching will not be instantaneous and the switching takes place in a finite duration, the chatter is not a physical phenomenon.

The Two D-O-F SA-type 2 suspension model was solved for different discrete frequencies in the range of 0.1 to 150 rad/s. The system was found to be unstable at certain frequencies as was reported [51] for the case of a single d-o-f model. A detailed study, on the response behavior of the 2 d-o-f model shows that for the SA-type 2 control logic, the damping force may be aiding the motion of the sprung mass, leading to unstable situation. Further, the absence of the damping force (zero value during the off stage) may result in an undesirable effect on the system's stability.

Thus, it may be concluded that the control force of the SA-type 2 system results in increasing the instantaneous acceleration response for certain parts of the vibration cycle. To overcome the drawbacks of the SA-type 2 system, a modified version (SA-type 2^*) which consists of a small amount of passive damping (ζ_0) operating in parallel with the SA-type 2 system is introduced. The schematic of the SA-type 2^* system is presented in Fig. 3.9. The expression for the damping force in SA-type 2^* system can be written as:

$$F_d = \begin{Bmatrix} 2\ \zeta\, \omega_1 (\ \dot{X}_1 - \dot{X}_2) m_1 + 2\ \zeta_0 \omega_1 (\ \dot{X}_1 - \dot{X}_2) m_1 & , & \dot{X}_1 (\ \dot{X}_1 - \dot{X}_2) > 0 \\ 2\ \zeta_0 \omega_1 (\ \dot{X}_1 - \dot{X}_2) m_1 & , & \dot{X}_1 (\ \dot{X}_1 - \dot{X}_2) < 0 \end{Bmatrix} \quad (2.10)$$

As explained in the SA-type 1 system, the SA-type 2^* system also would result in the amplification of the sprung mass motion around the unsprung mass natural frequency. This is observed from the sprung mass acceleration response presented in Figures 3.10a, 3.10b, and 3.10c, for variations in ζ (0.25, 0.707, 1.0) and ζ_0 (0.08, 0.16, 0.24).

The availability of additional passive damping may improve the ride performance at and frequencies lower than the sprung mass natural frequency, and may degrade the ride performance at higher frequencies during the "on" stage. At higher frequencies the SA-type 2^* system may be aiding the motion of the sprung mass during some parts of the vibration cycle, due to the availability of the higher passive damping. Hence as a compromise the value of the additional passive damping of $\zeta_0 = 0.16$ is considered better. The SA-type 2^* system with $\zeta = 0.25$ and $\zeta_0 = 0.16$ shows an optimal response for reducing the peak acceleration at both the sprung and unsprung mass natural frequencies.

The sprung mass acceleration response is comparatively better than

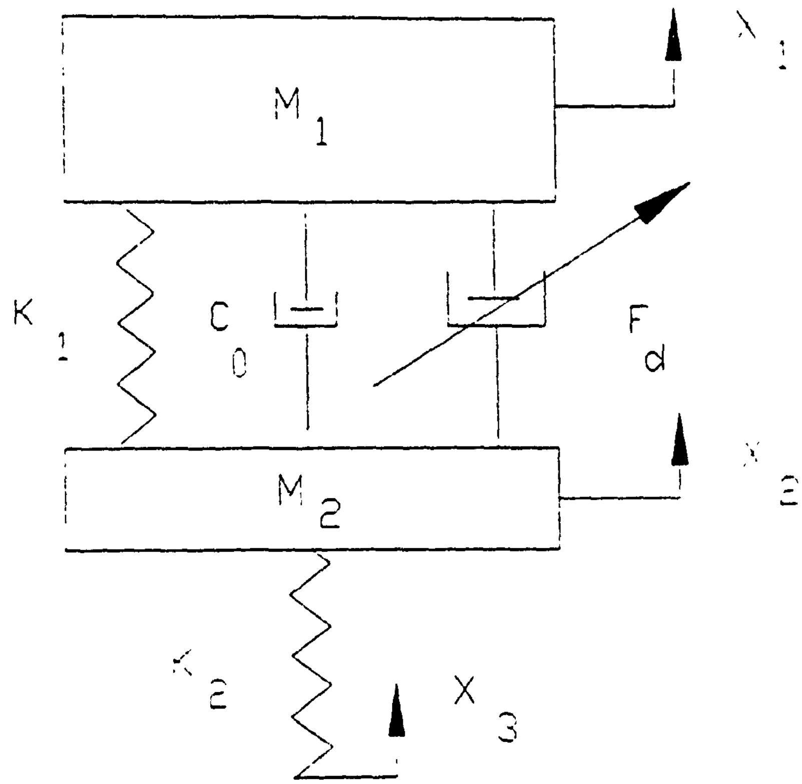


Fig. 3.9 Schematic of an SA-type 2* model suspension

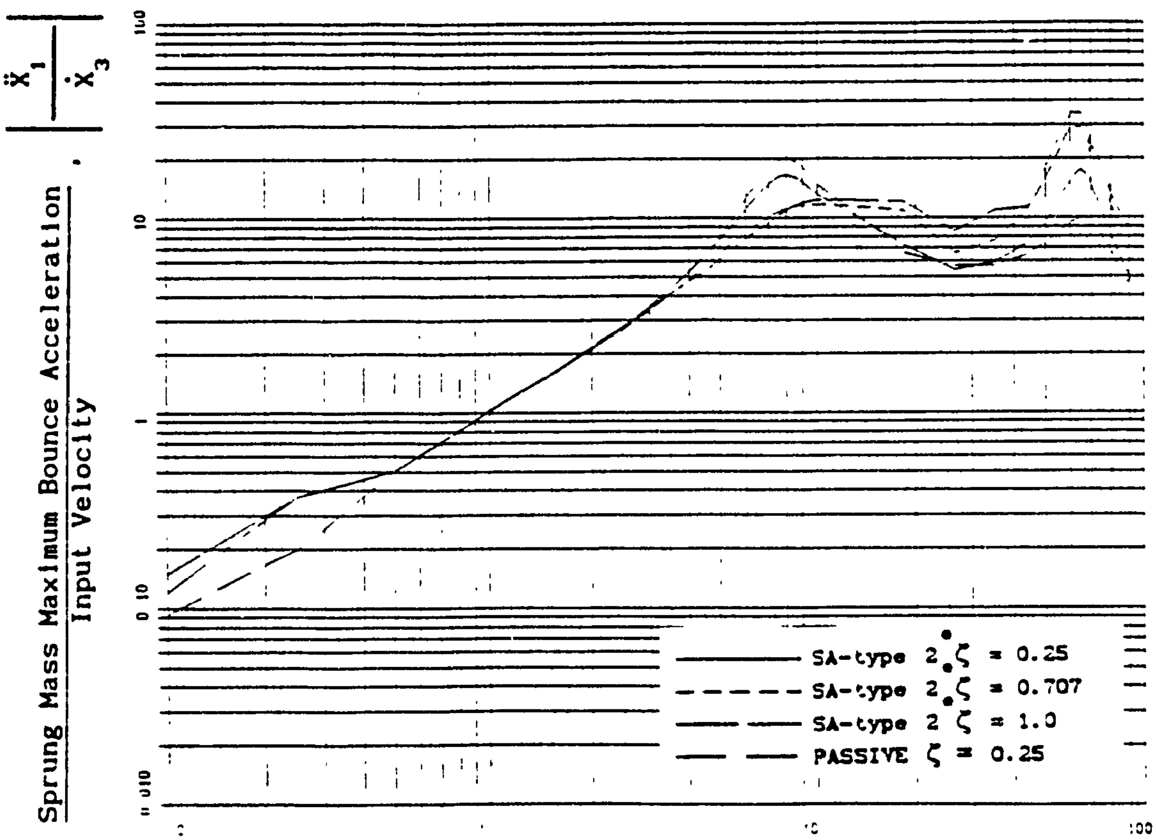


Fig. 3.10a Comparison of the sprung mass maximum bounce acceleration for passive and SA-type $2^{\bullet}$ (with $\zeta_0 = 0.08$) suspensions

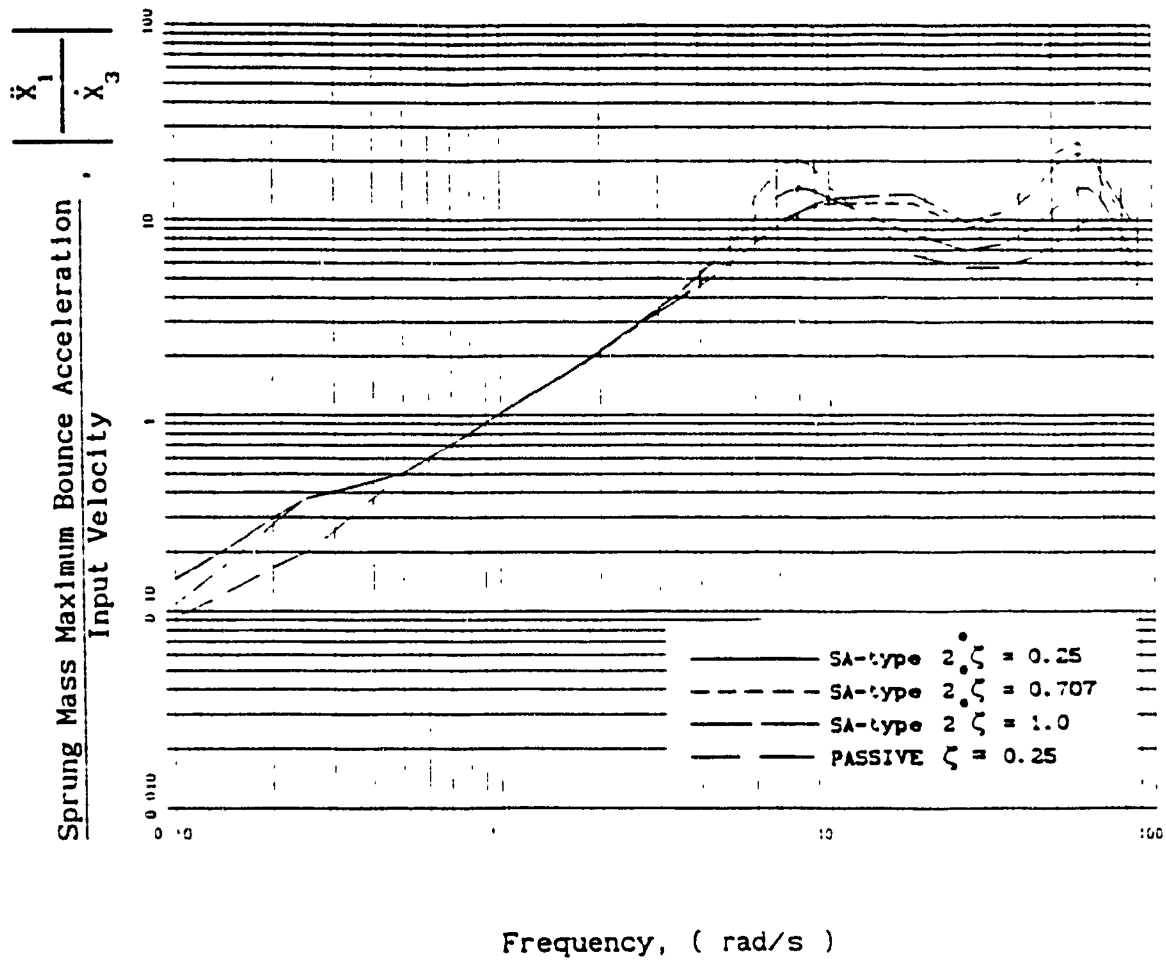


Fig. 3.10b Comparison of the sprung mass maximum bounce acceleration for passive and SA-type $2^{\bullet}$ (with ζ_0 = 0.16) suspensions

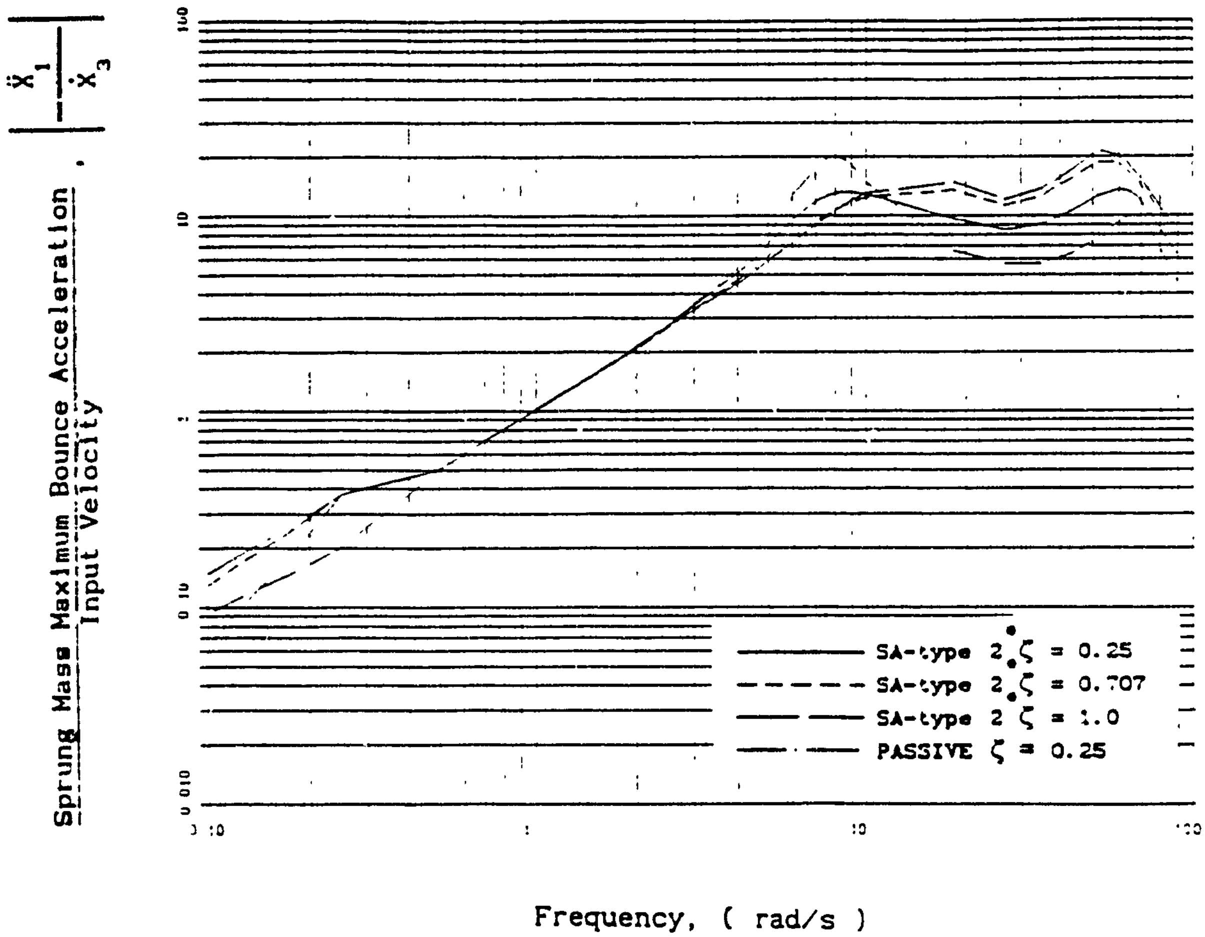


Fig. 3.10c Comparison of the sprung mass maximum bounce acceleration for passive and SA-type 2• (with ζ_0 = 0.24) suspensions

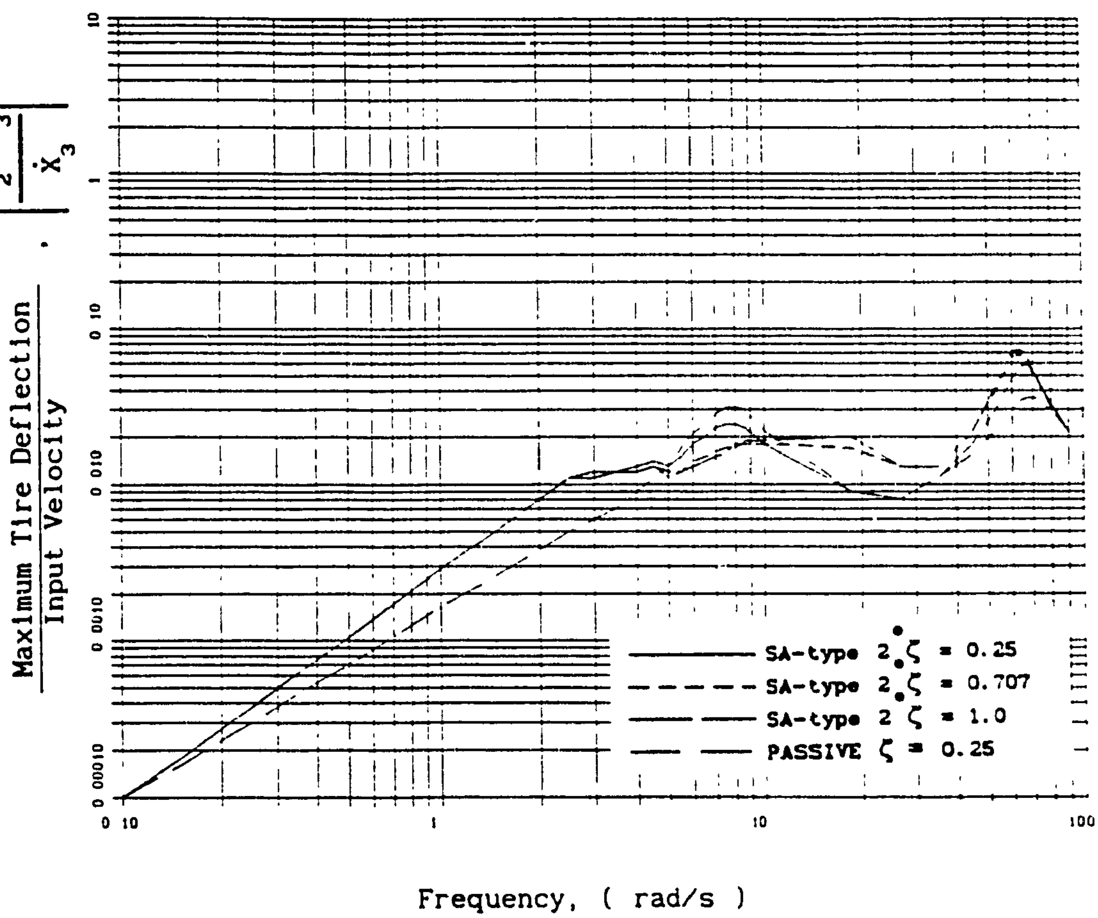


Fig. 3.11a Comparison of maximum tire deflection for passive and SA-type 2* (with $\zeta_0 = 0.08$) suspensions

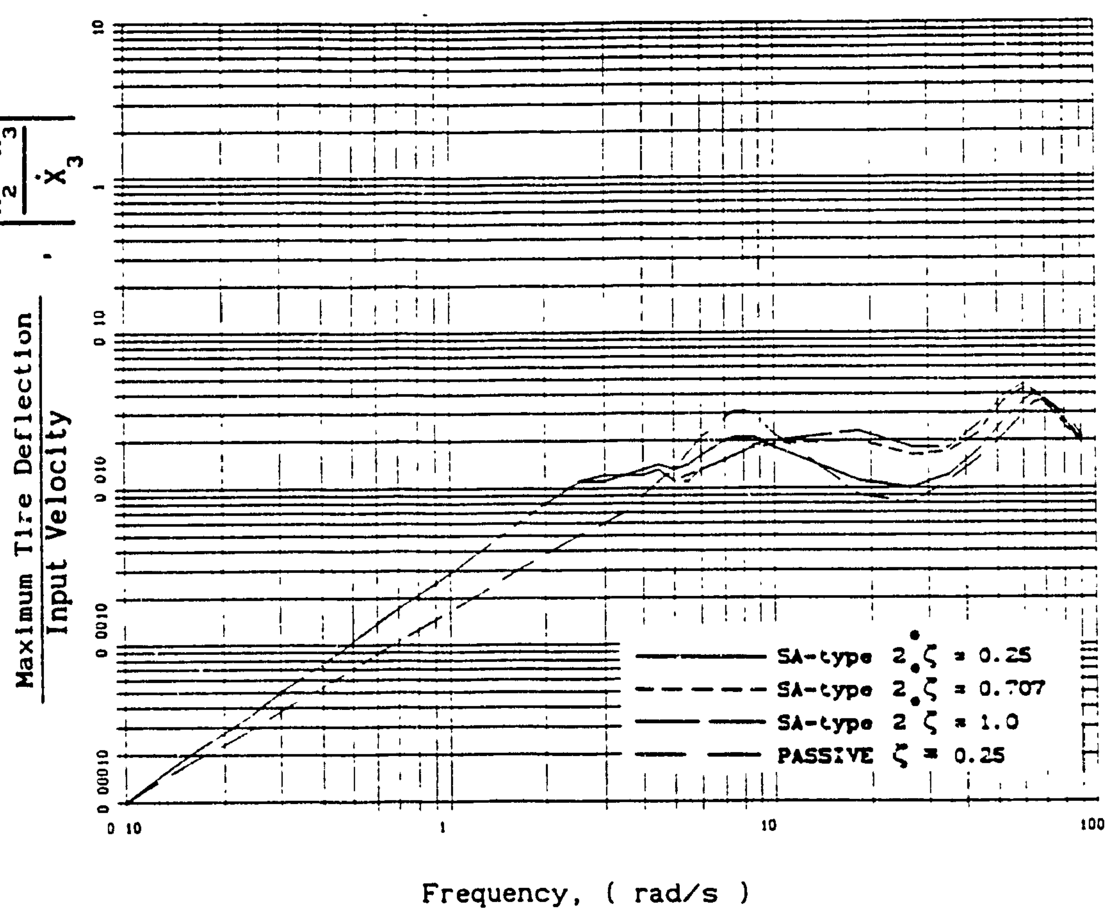


Fig. 3.11b Comparison of maximum tire deflection for passive and SA-type $2^{\bullet}$ (with ζ_0 = 0.16) suspensions

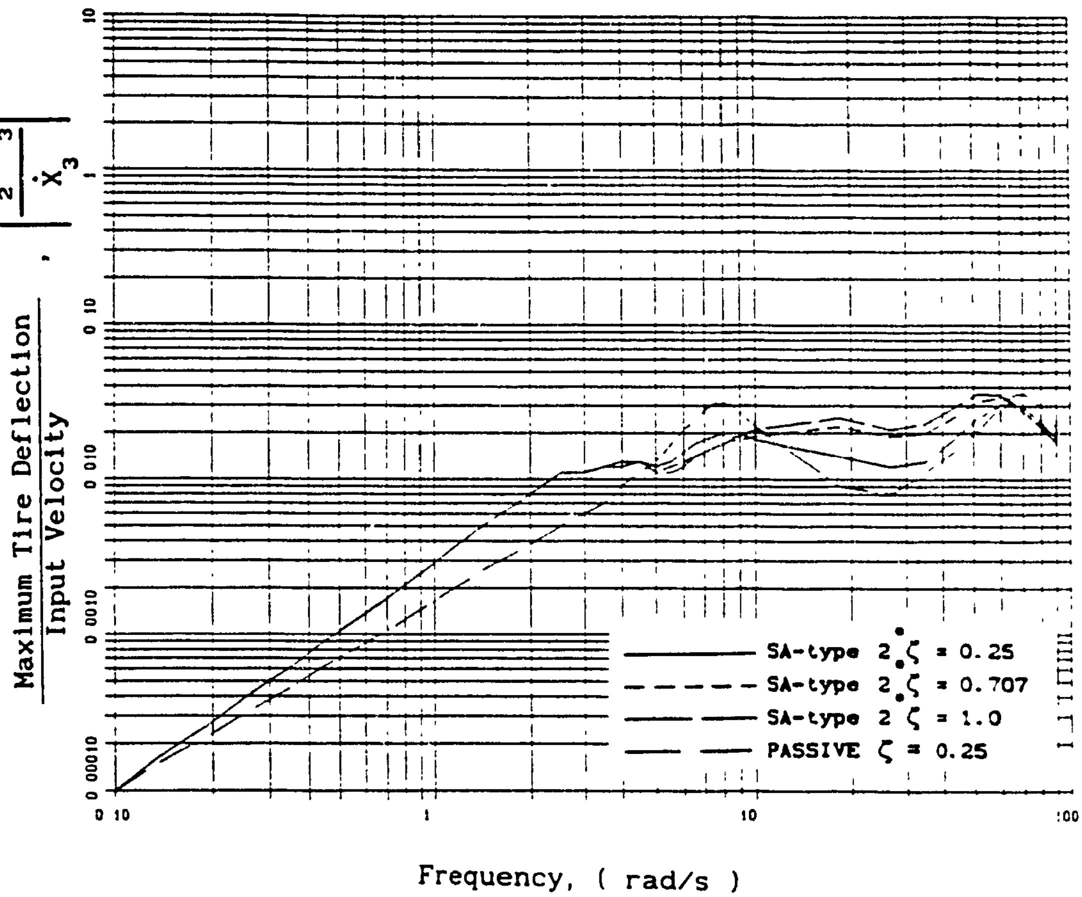


Fig. 3.11c Comparison of maximum tire deflection for passive and SA-type 2* (with $\zeta_0 = 0.24$) suspensions

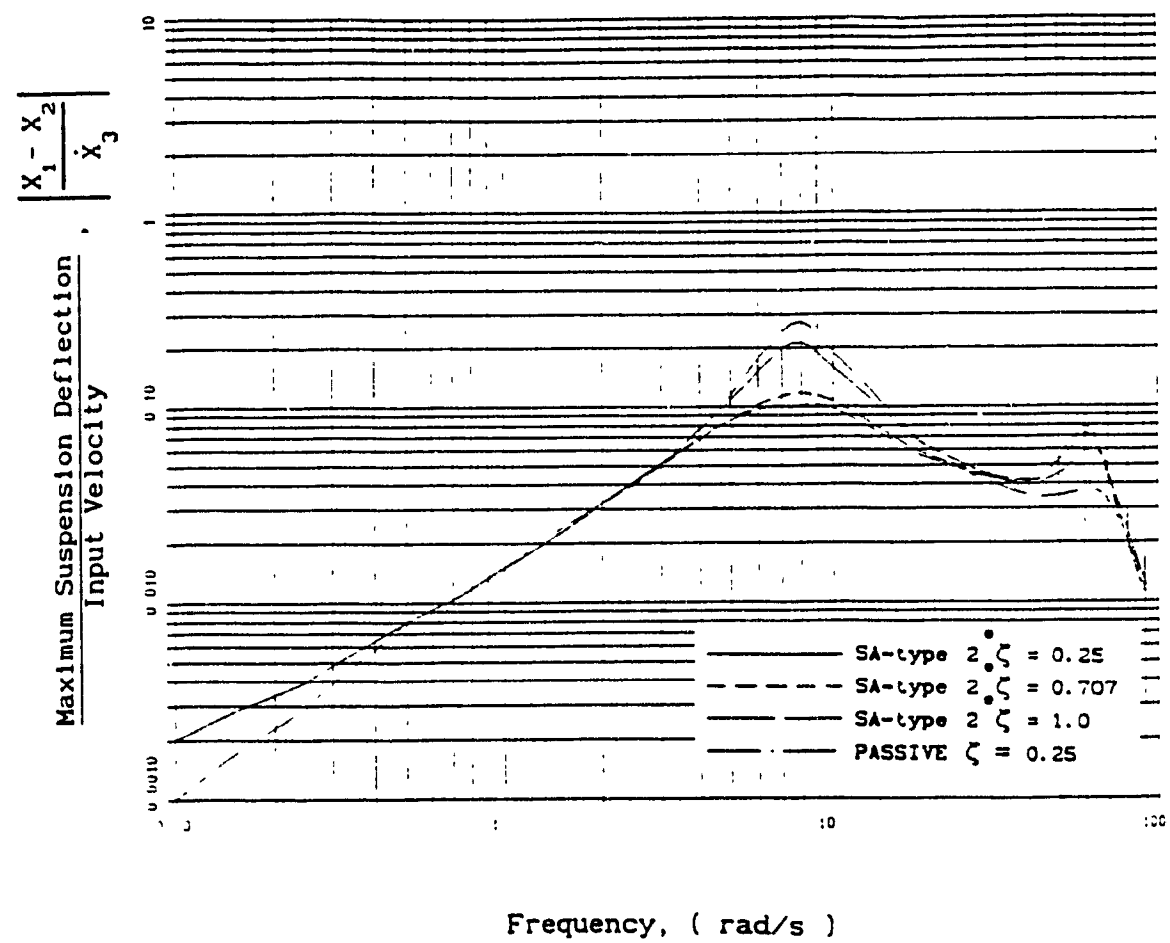


Fig. 3.12a Comparison of maximum suspension deflection of passive and SA-type 2* (with $\zeta_0 = 0.08$) suspensions

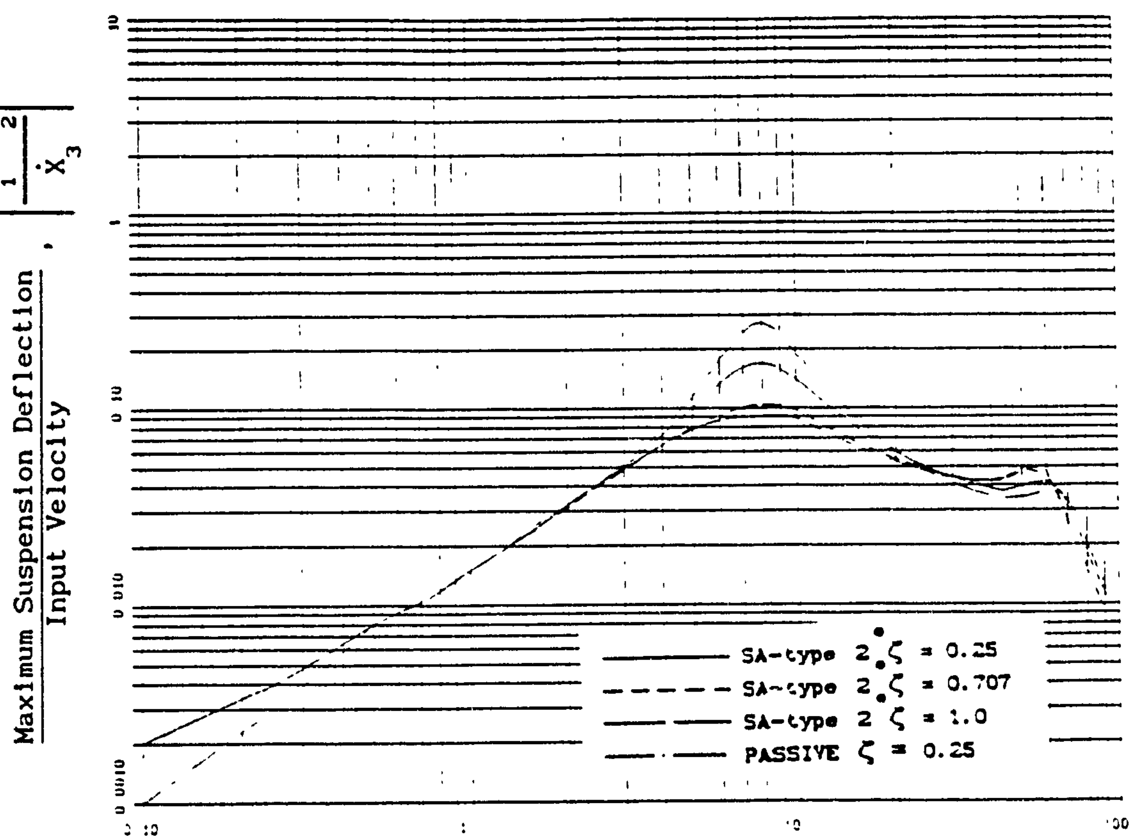


Fig. 3.12b Comparison of maximum suspension deflection of passive and SA-type 2* (with $\zeta_0 = 0.16$) suspensions

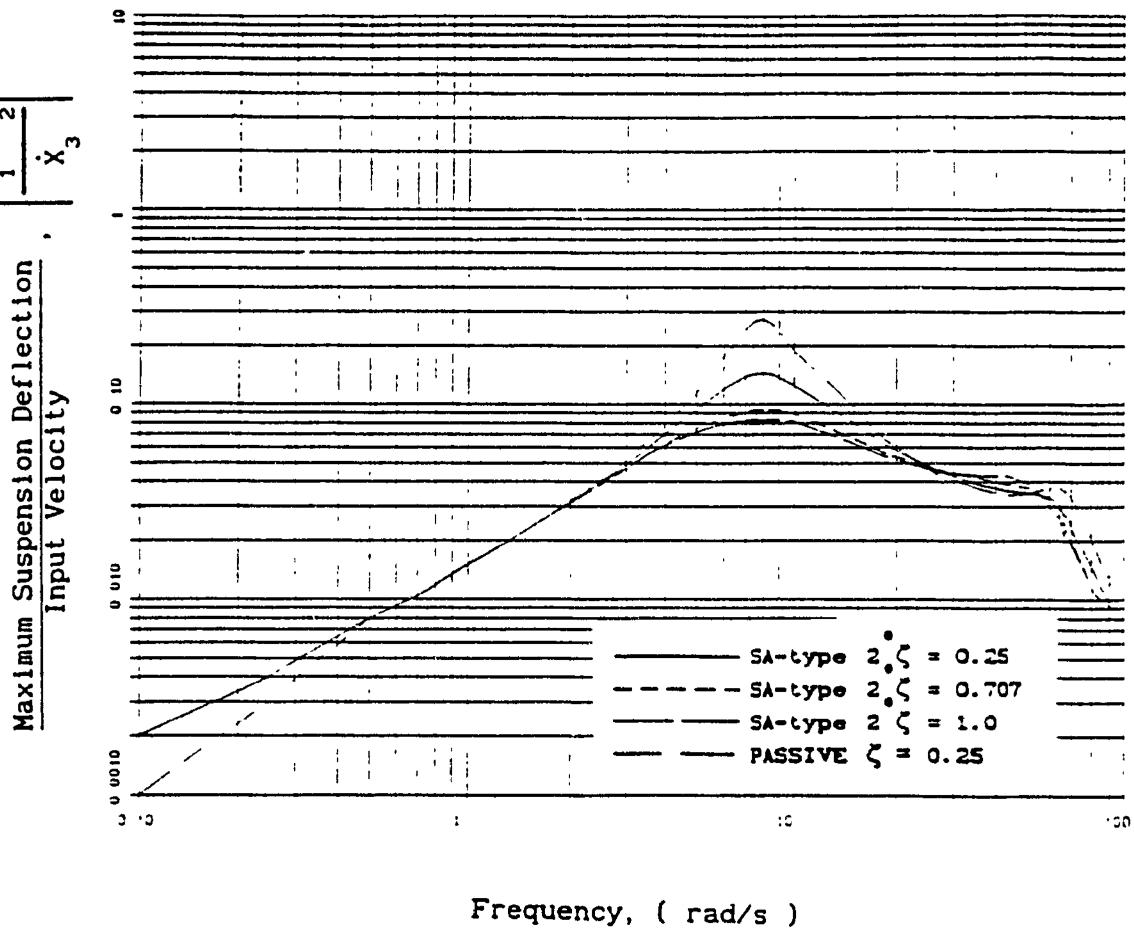


Fig. 3.12c Comparison of maximum suspension deflection of passive and SA-type 2* (with $\zeta_0 = 0.24$) suspensions

the passive system, around the sprung mass natural frequency, however, it is inferior for frequencies in the neighborhood of the unsprung mass natural frequency.

The tire deflection response presented in Figures 3.11a, 3.11b, and 3.11c shows that the response of SA-type 2^{*} system behaves very similar to the passive system especially for $\zeta = 0.25$ and the response in general is better than the passive system at the sprung mass natural frequency. The response is, however, inferior at the unsprung mass natural frequency.

The suspension deflection response presented in Figures 3.12a, 3.12b, and 3.12c shows that the SA-type 2^{*} has inferior performance compared to passive suspension for very low frequencies lower than the sprung mass natural frequency and around the unsprung mass natural frequency. However, the response of SA-type 2^{*} is better than the passive system at the sprung mass natural frequency.

Again for the SA-type 2^{*} system, $\zeta = 0.25$ and $\zeta_0 = 0.16$ provides a good compromise in the suspension deflection response at the two natural frequencies. However, the response of the SA-type 2^{*} system with the value of $\zeta = 0.707$ and $\zeta_0 = 0.16$ shows the best suspension deflection response (Fig. 3.12b).

It is observed from the overall study of the SA-type 2^{*} system responses, that this system with $\zeta = 0.25$ and $\zeta_0 = 0.16$ provides a good compromise between acceleration and suspension deflection responses at both the natural frequencies. The tire deflection is also better than the passive system response.

3.4.3 SA-type 3 system

The steady-state time responses of the SA-type 3 system with the value of unity for the tuning factor are presented, for different frequency ratios, in Figures 3.13, 3.14, and 3.15. In Figure 3.13, though the system undergoes lock-up mode at frequency $\omega/\omega_1 = 0.5$, one cannot identify this event explicitly from the time history plots. However, as in the case of SA-type 1 system, it can be seen that the sprung mass has similar sharp spike in the acceleration response when it breaks away from the lock-up. During lock-up, the relative velocity response tends to switch back to its original value before lock-up. Due to this behavior, a fluctuation in the condition function can be observed just at that instant which introduces a large variation in the damping force and hence a sharp spike in the acceleration response of the system.

The semi-active damping force (F_d) and the resultant forces (R_f)are truly opposing each other. The active damping force (F_d^*) and the semi-active damping force (F_d) are identical excepting at lock-up, and for the off stage of the condition function. In one period of oscillation, the condition function switches periodically between positive and negative values for all the frequencies. Because of this behavior, this system unlike the SA-type 1 system, does not require instantaneous switching at higher frequencies. It is also noted that when the damping force is zero, the acceleration response increases sharply. However, the acceleration response remains zero during some parts of the vibration cycle when the damping force is on. This is in

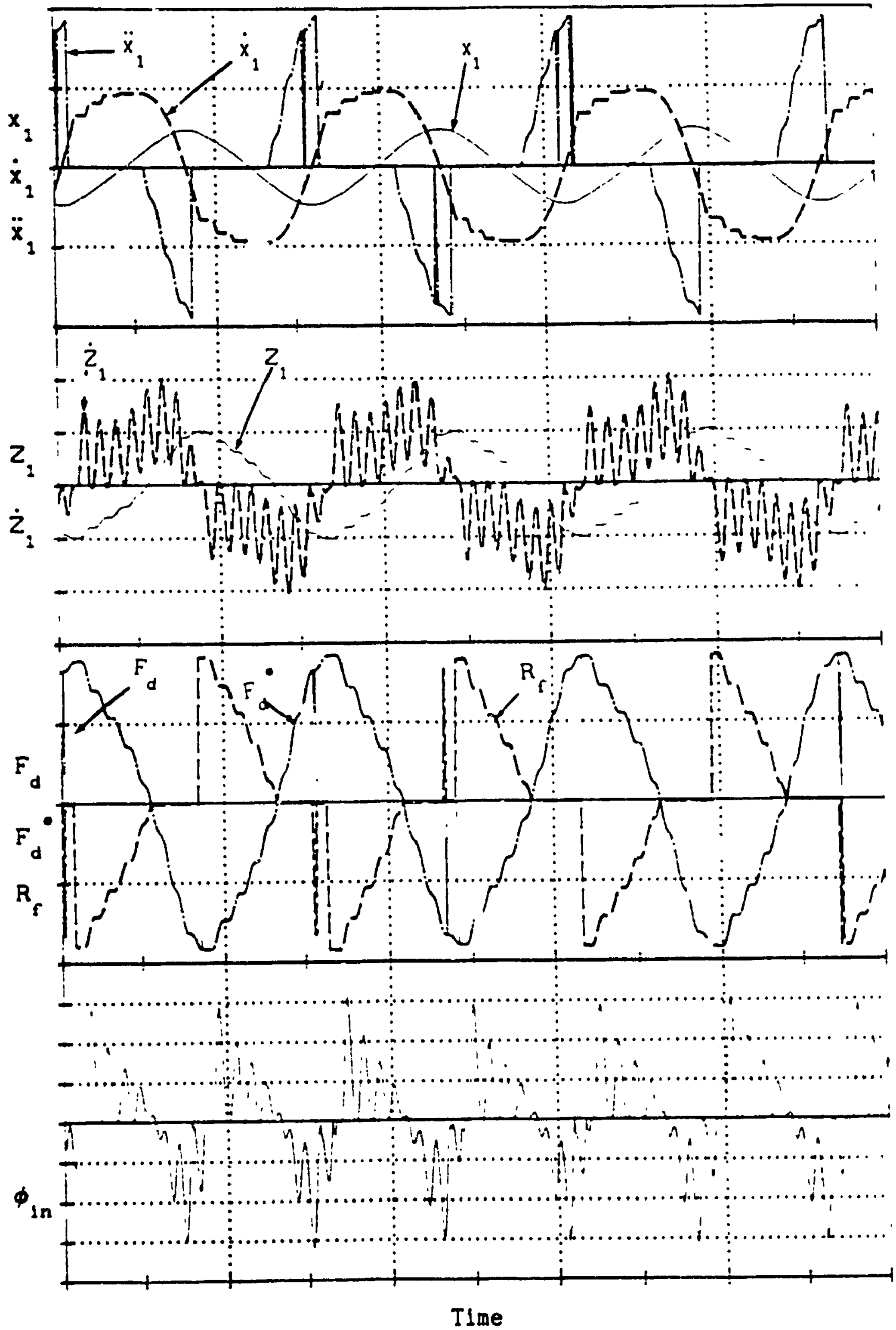


Fig. 3.13 Steady-state response of an SA-type 3 system for ω/ω_i=0.5

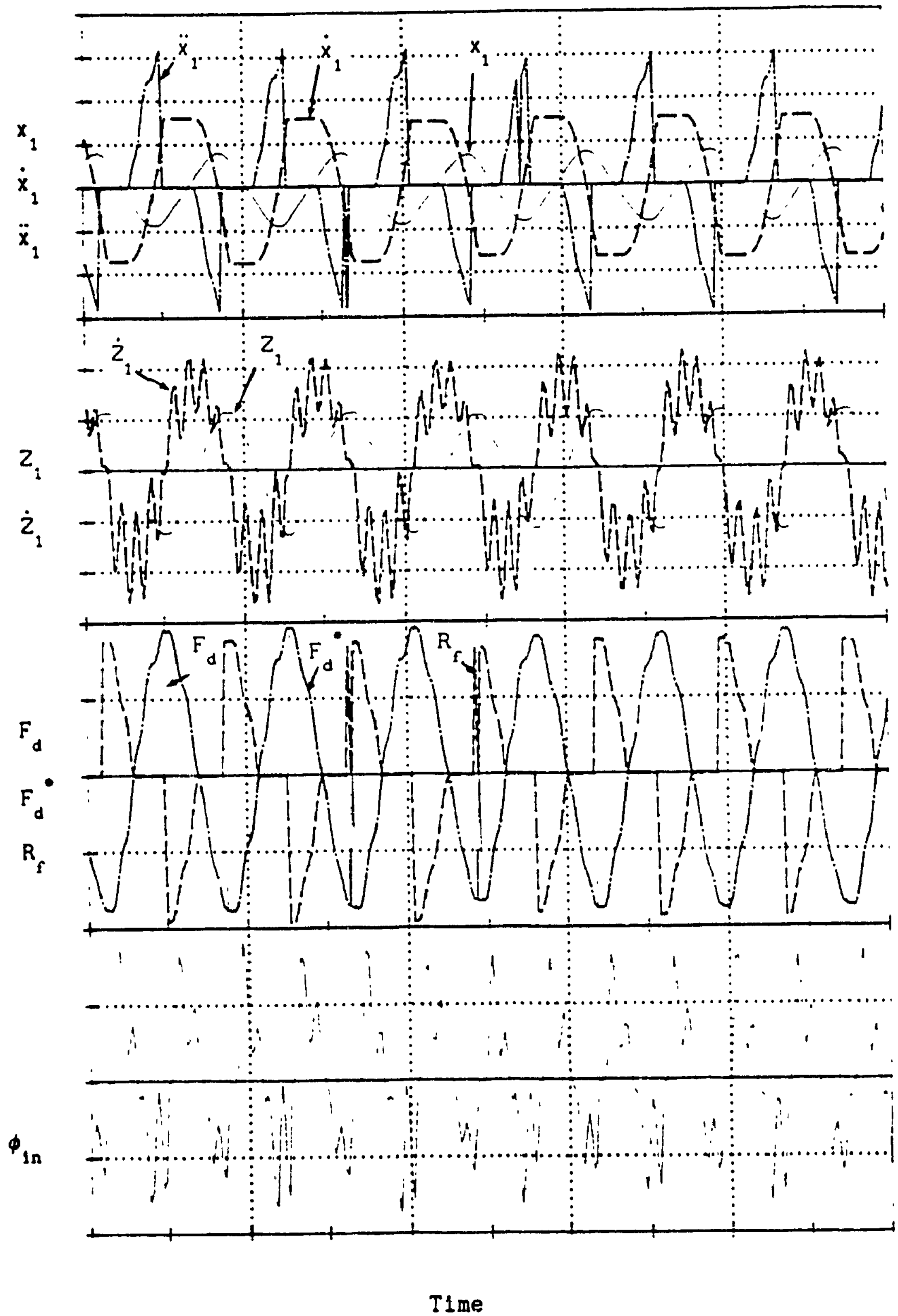


Fig. 3.14 Steady-state Response of an SA-type 3 system for $\omega/\omega_1=1.0$

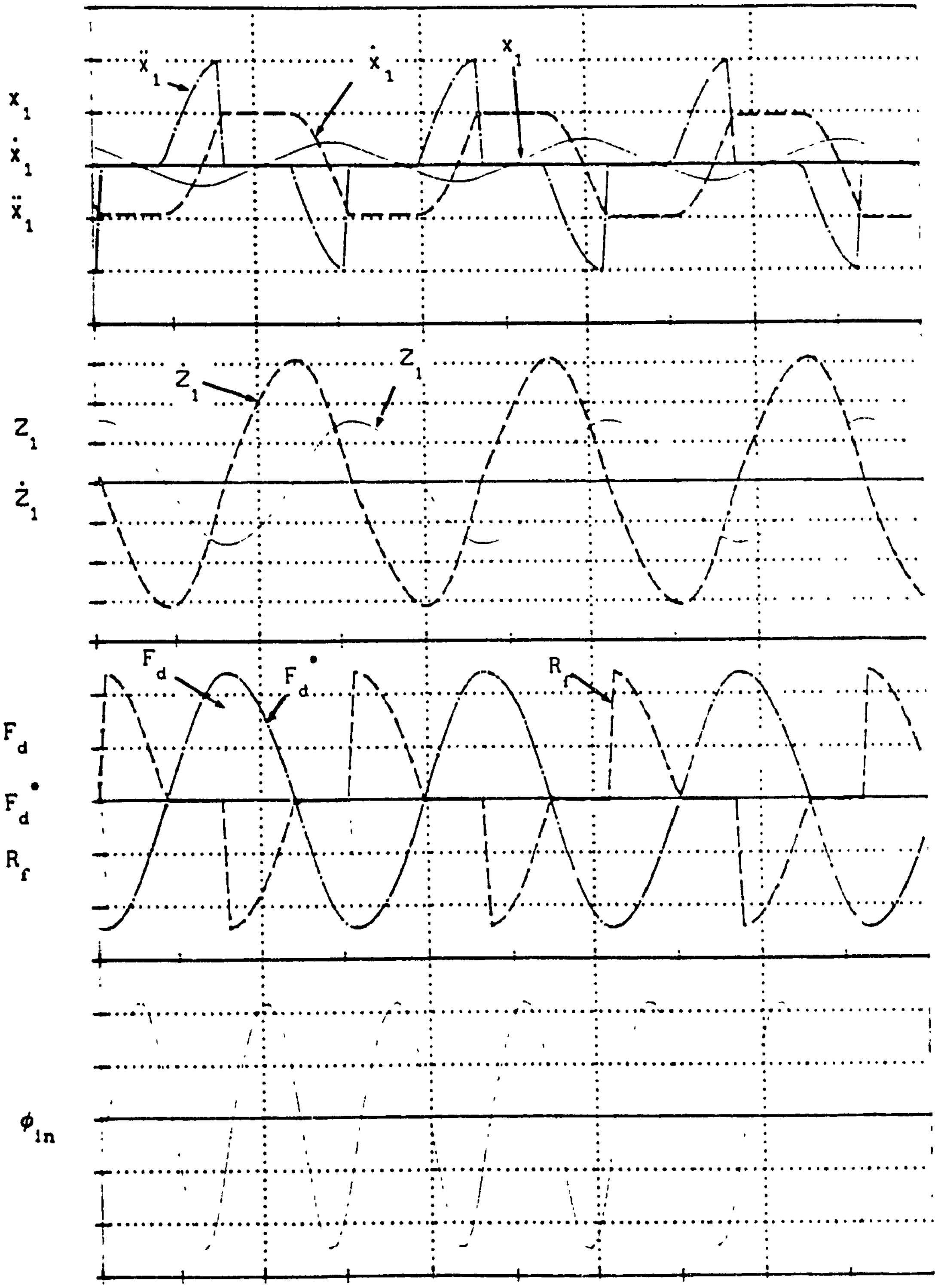


Fig. 3.15 Steady-state Response of an SA-type 3 system for $\omega/\omega_1=5.0$

conformity with the nature of the control logic.

The SA-type 3 system will be unstable when the value for the tuning factor, α, is greater than unity. This is due to the fact that when $\alpha > 1$, the semi-active damper will introduce negative damping on the system and will cause instability.

The Sprung mass maximum acceleration response of the SA-type 3 system, for different values of tuning factors less than unity, is presented in Fig. 3.16 and compared to the response of the passive suspension. Since the SA-type 3 system is derived on the basis that the value of the instantaneous sprung mass acceleration response be reduced, it provides better acceleration attenuation of the sprung mass at its natural frequency. The response at the unsprung mass natural frequency, however, is slightly inferior to the passive system. However, the acceleration response between the two natural frequencies and beyond the unsprung mass natural frequency is superior. This behavior of the system is attributed to the fact that, the damping force only nullifies the amount of spring force acting on the sprung mass during the on stage.

At frequencies lower than the sprung mass natural frequency, the response of the SA-system is slightly inferior when the tuning factor is unity. This is due to the fact that in this region the system undergoes a lock-up and hence the performance is little degraded. But for tuning factors less than unity, the acceleration response near the sprung mass natural frequency increases slightly which means that the force acting on the sprung mass is best countered only when damper force is equal in magnitude to that of the spring force. The response of the SA-type 3 system is inferior to the passive system at the unsprung mass natural frequency.

In Fig. 3.17, the tire deflection response is compared with the passive system. The maximum tire deflection response of the SA-type 3 system is generally inferior to the passive system. The maximum suspension deflection response of the SA-type 3 system is presented in Fig. 3.18. The response is inferior to the passive system for the entire range of frequencies, except for the frequencies between the two natural frequencies of the system. In this frequency range, the SA-type 3 and passive systems have similar responses.

On the over all review of the response of the SA-type 3 suspension system, it is observed that the vibration isolation in terms of the acceleration response at the sprung mass is superior to the passive suspension. Although the system performance is poor compared to the passive system for a narrow bandwidth of frequencies around the unsprung mass natural frequency, the performance is better between the two main natural frequencies of the system. The variations in the value of the tuning factor to more than unity would cause instability and hence the selection of a value less than unity is important in this control logic. The value of alpha more than unity would create conditions of negative damping. Alpha being very small in value would result in the magnitude of available damping force less than the spring force, this may not provide effective and intended ride performance. Hence, it would be appropriate to have the value of alpha between 0.5 and unity.

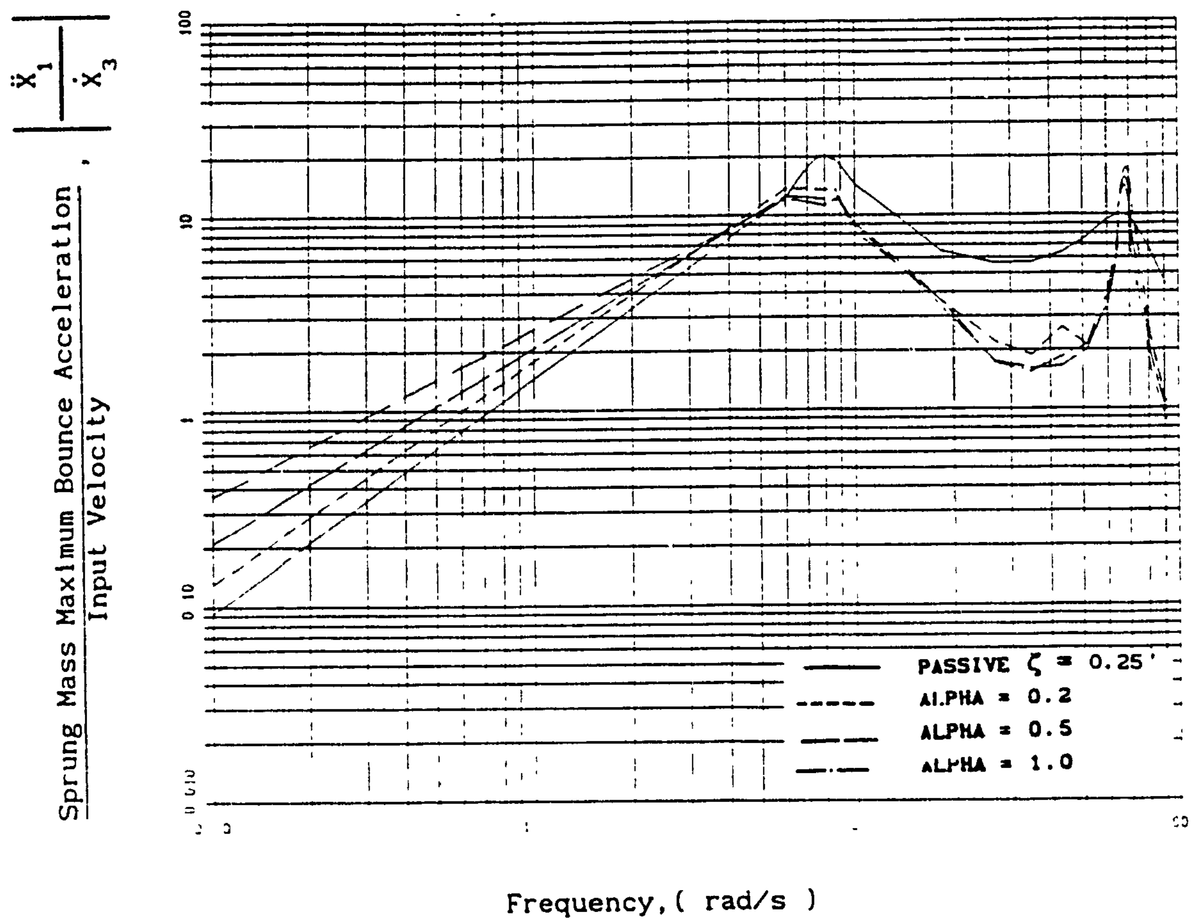


Fig.3.16 Comparison of the sprung mass maximum bounce acceleration for passive and SA-type 3 suspensions

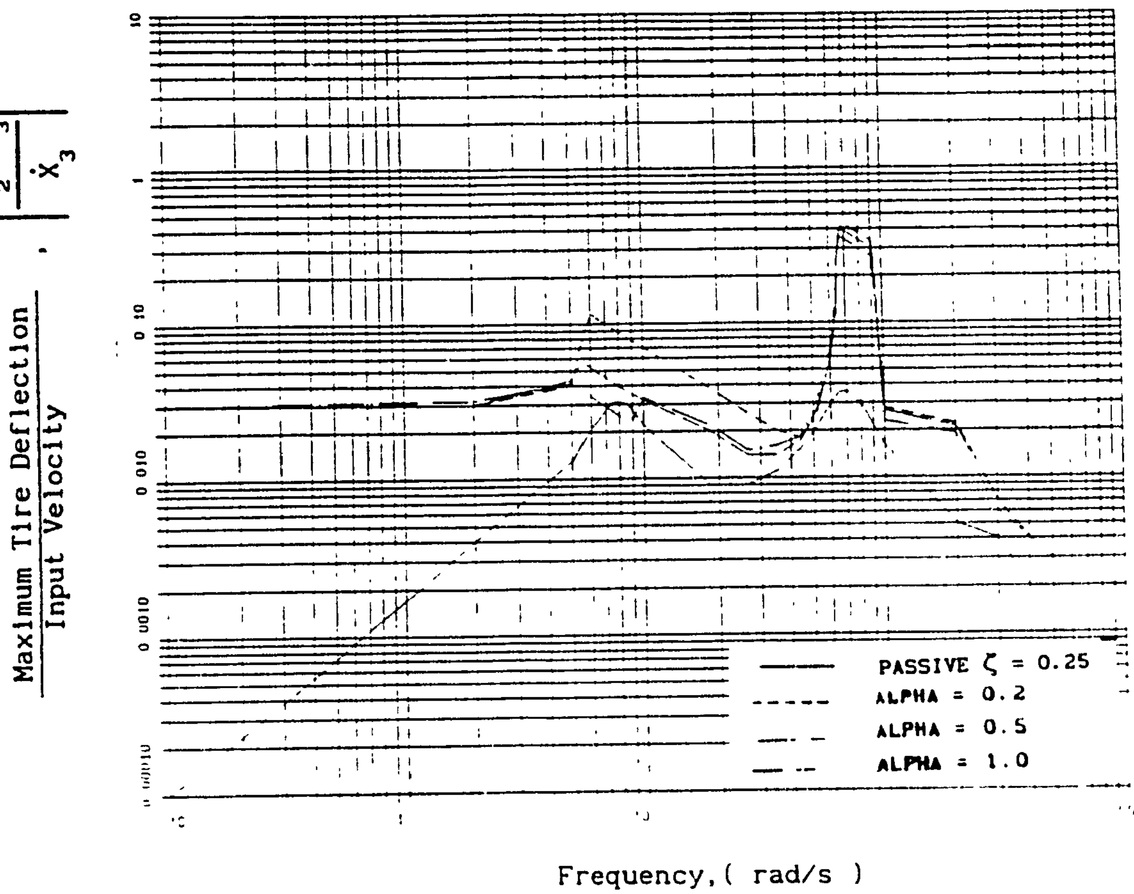


Fig.3.17 Comparison of maximum tire deflection for passive and SA-type 3 suspensions

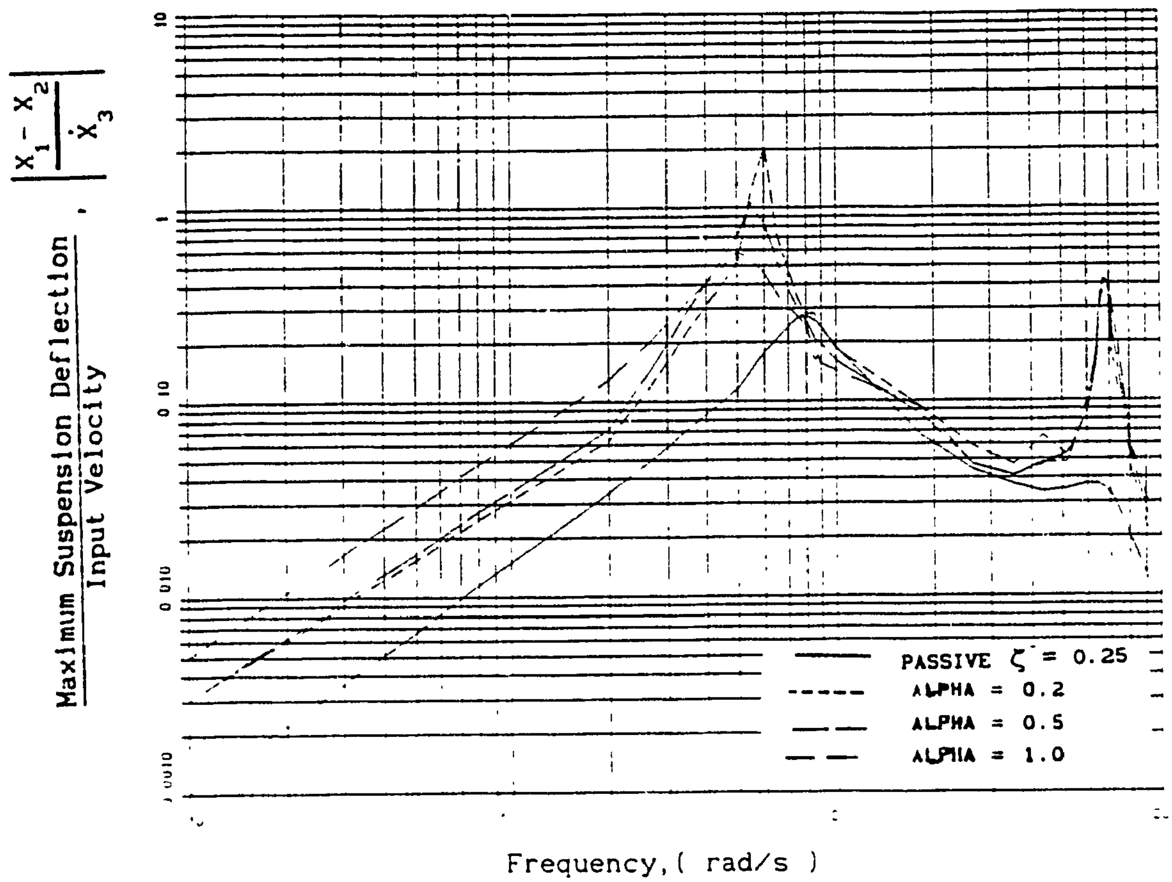


Fig.3.18 **Comparison of maximum suspension deflection of passive and SA-type 3 suspensions**

3.4.4 SA-type 4 system

This scheme is implemented in the experimental study with different sets of values for the damping coefficient in compression and expansion strokes of the shock absorber. The analytical results are presented along with the experimental results in Chapter 4 for variation in damping values. The equation characterizing the damping force for this configuration is given as:

$$F_d = \begin{cases} 2\,\zeta_{on}\,\omega_1\,(\dot{X}_1 - \dot{X}_2)\,m_1, & (X_1 - X_2)\,(\dot{X}_1 - \dot{X}_2) < 0 \\ 2\,\zeta_{off}\,\omega_1\,(\dot{X}_1 - \dot{X}_2)\,m_1, & (X_1 - X_2)\,(\dot{X}_1 - \dot{X}_2) > 0 \end{cases} \qquad (2.12)$$

$$\zeta_{on} = \begin{matrix} \zeta_{on_{extension\_1}} & , & \dot{z}_1 < v_{e1} \\ \zeta_{on_{compression\_1}} & , & \dot{z}_1 < v_{c1} \end{matrix}$$

$$\zeta_{on} = \begin{matrix} \zeta_{on_{extension\_2}} & , & \dot{z}_1 > v_{e1} \\ \zeta_{on_{compression\_2}} & , & \dot{z}_1 > v_{c1} \end{matrix}$$

and

$$\zeta_{off} = \begin{matrix} \zeta_{off_{extension\_1}} & , & \dot{z}_1 < v_{e2} \\ \zeta_{off_{compression\_1}} & , & \dot{z}_1 < v_{c2} \end{matrix}$$

$$\zeta_{off} = \begin{matrix} \zeta_{off_{extension\_2}} & , & \dot{z}_1 > v_{e2} \\ \zeta_{off_{compression\_2}} & , & \dot{z}_1 > v_{c2} \end{matrix}$$

where, ζ_{on}, and ζ_{off} are the damping values for the "on" and "off" stages of the SA-scheme. The numerical values of damping factors are based on the extension and compression strokes of the shock absorber, and the break velocity. A detailed study of this system with various damping characteristics of the shock absorber, is presented in Chapter 4.

3.5 Study on the Influence of the Zero-Finder in the Solution Methodology

A study on the solution methodology for the response of the SA-system with and without the use of the zero-finder is presented. The necessity of a zero-finder is discussed in the previous chapter. The SA-type 1 and SA-type 3 systems are solved with, and without using the zero-finder. It is learnt form the following results that the force when takes on one of the two values at the end of the time step after the condition function crosses a zero value, the system response is considerably affected by the zero-finder at higher frequencies, especially around the unsprung mass natural frequency.

In Fig. 3.19, the maximum acceleration response for the SA-type 1 suspension system with the zero-finder is presented and compared with the response obtained without the zero-finder. It is evident that the response obtained with out a zero-finder is influenced throughout the frequency range of interest with higher response compared to the result with the zero-finder. The influence is dominant around the system natural frequencies.

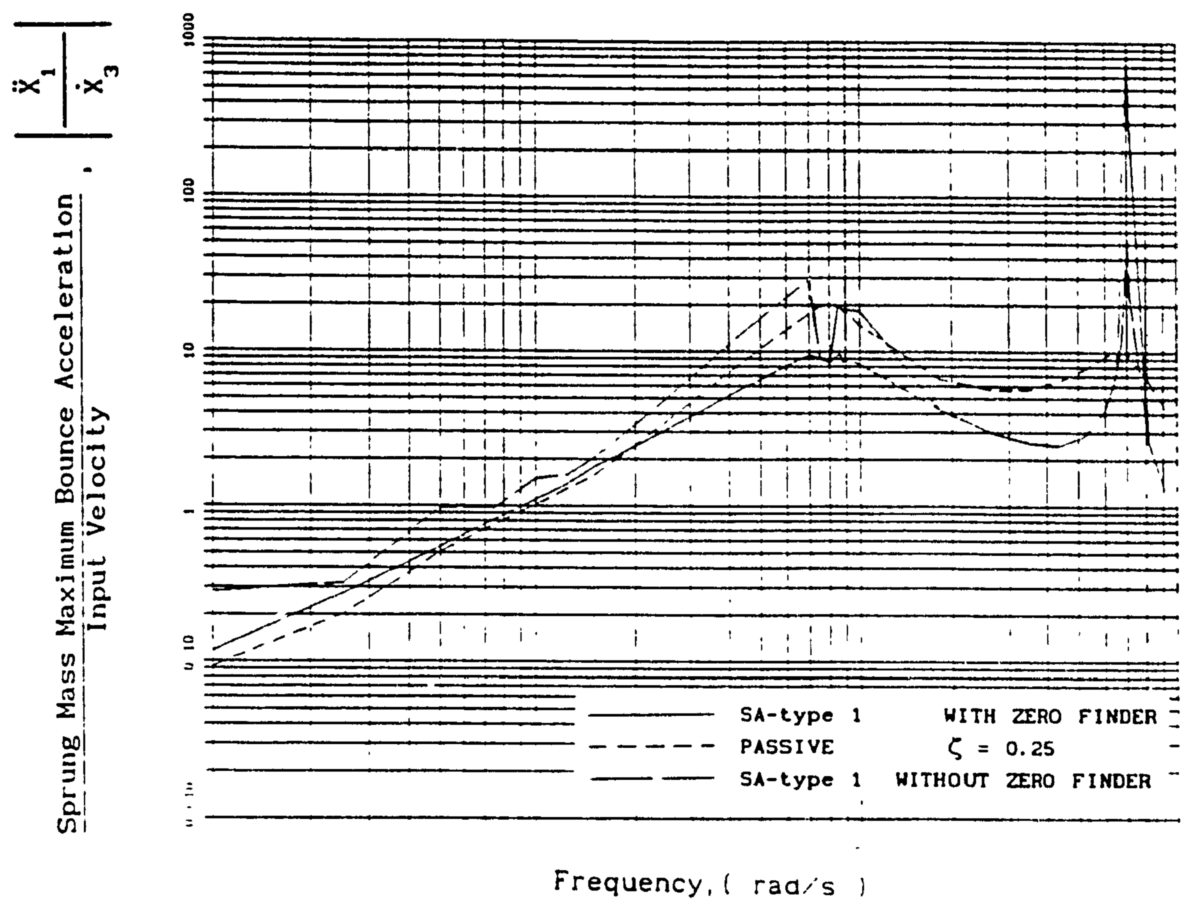


Fig.3.19 Influence of a zero-finder on the sprung mass maximum bounce acceleration response in an SA-type 1 suspension

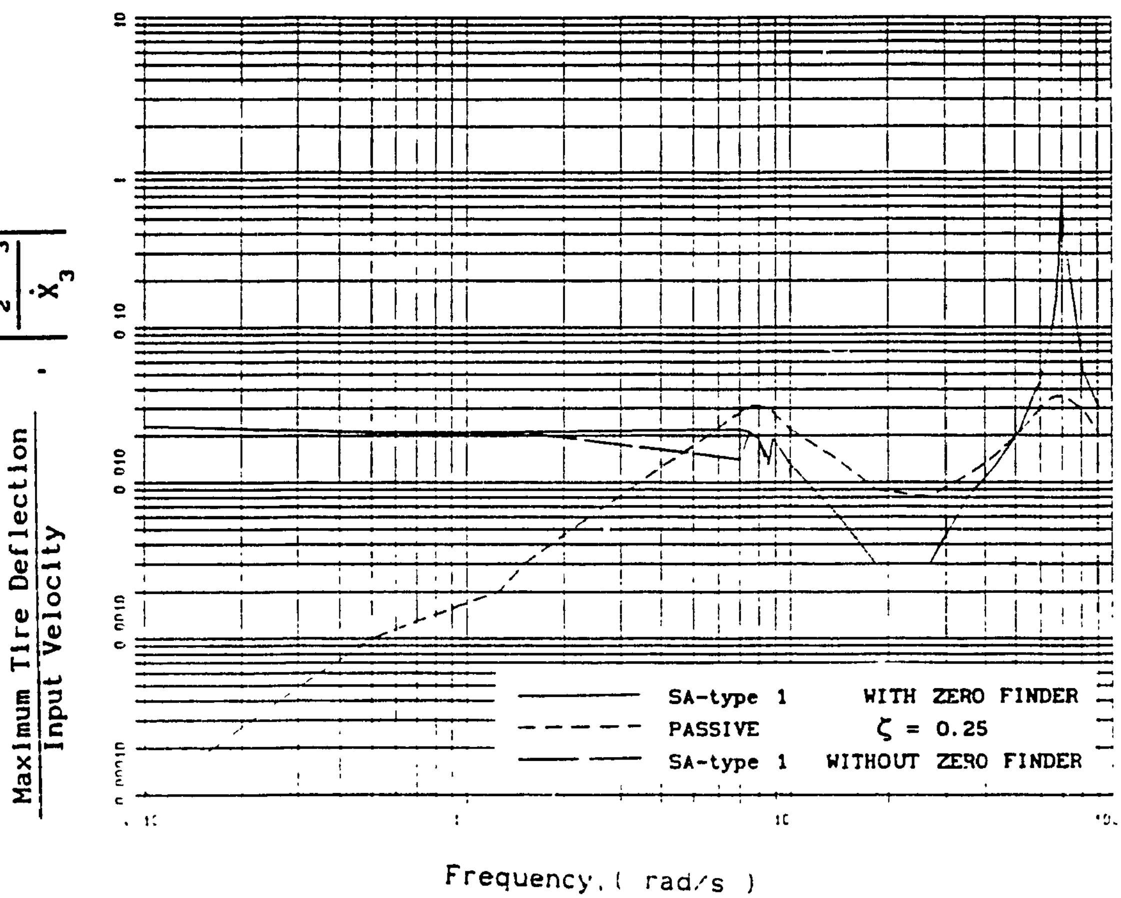


Fig.3.20 Influence of a zero-finder on the maximum tire deflection response in an SA-type 1 suspension

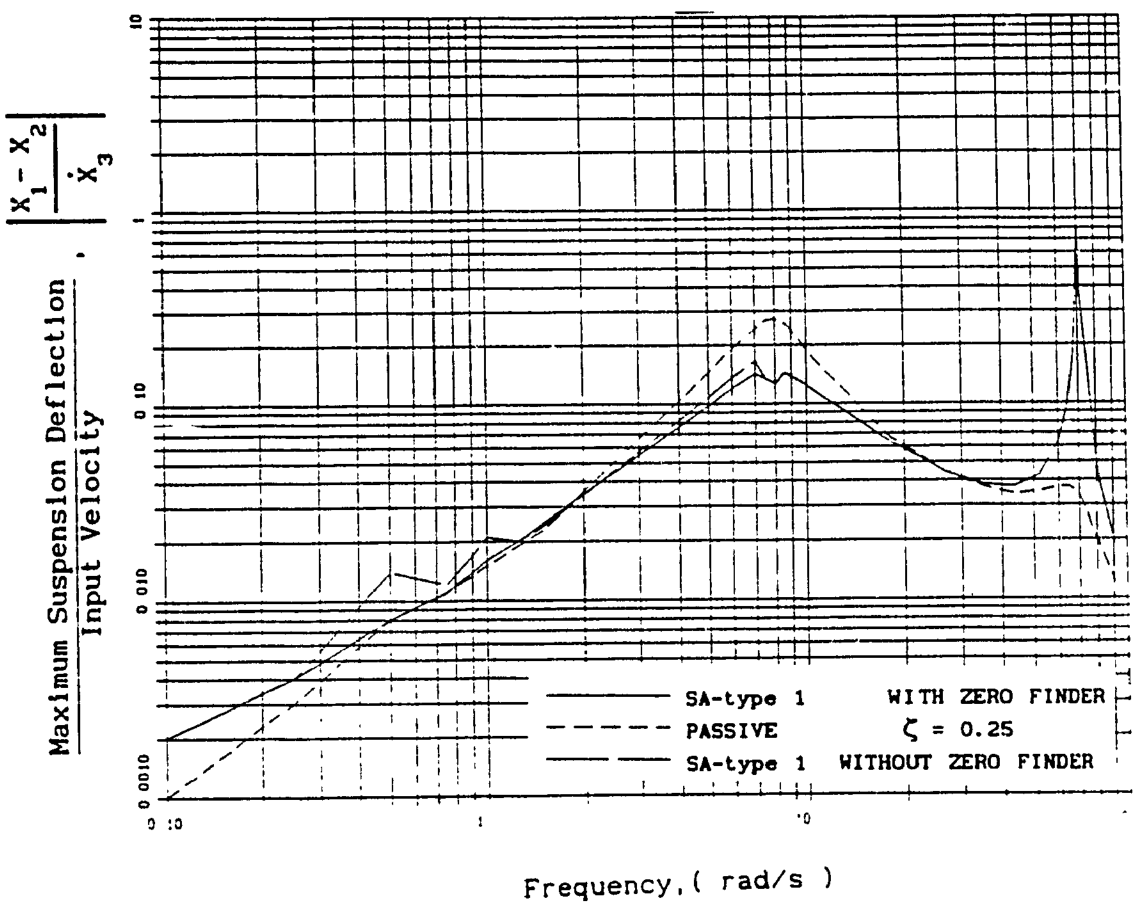


Fig.3.21 Influence of a zero-finder on the maximum suspension deflection response of an SA-type 1 suspension

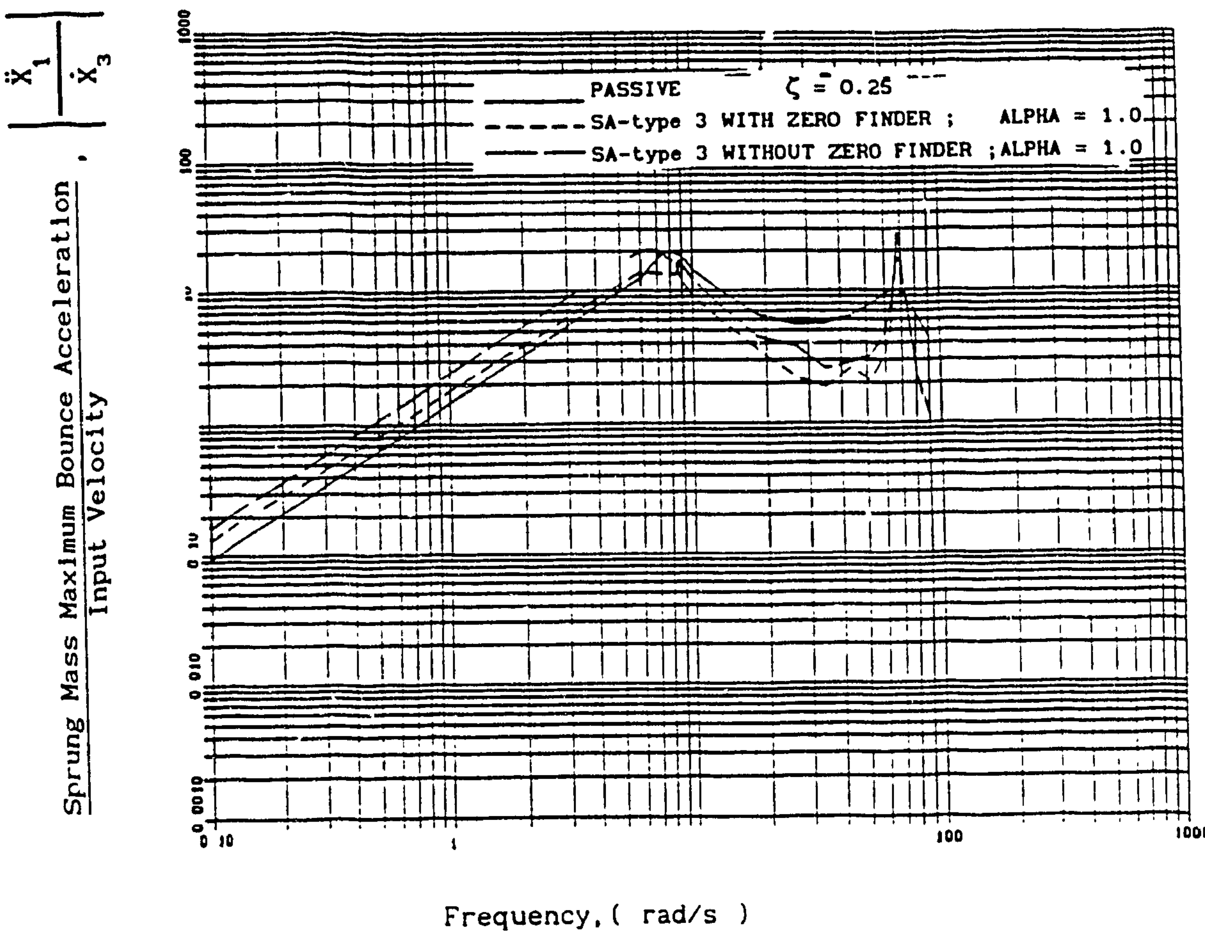


Fig.3.22 Influence of a zero-finder on the sprung mass maximum bounce Acceleration response in an SA-type 3 suspension

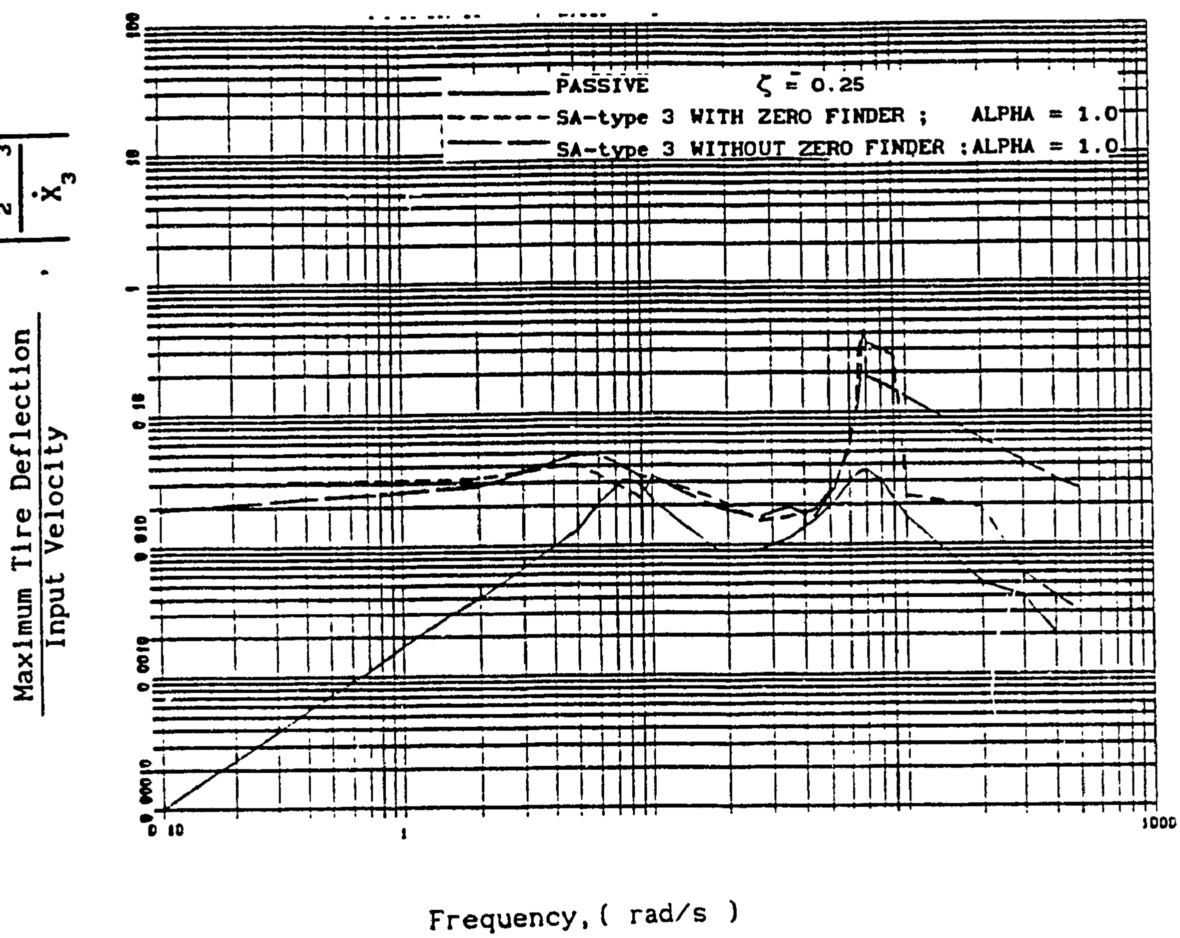


Fig.3.23 Influence of a zero-finder on the maximum tire deflection response of an SA-type 3 suspension

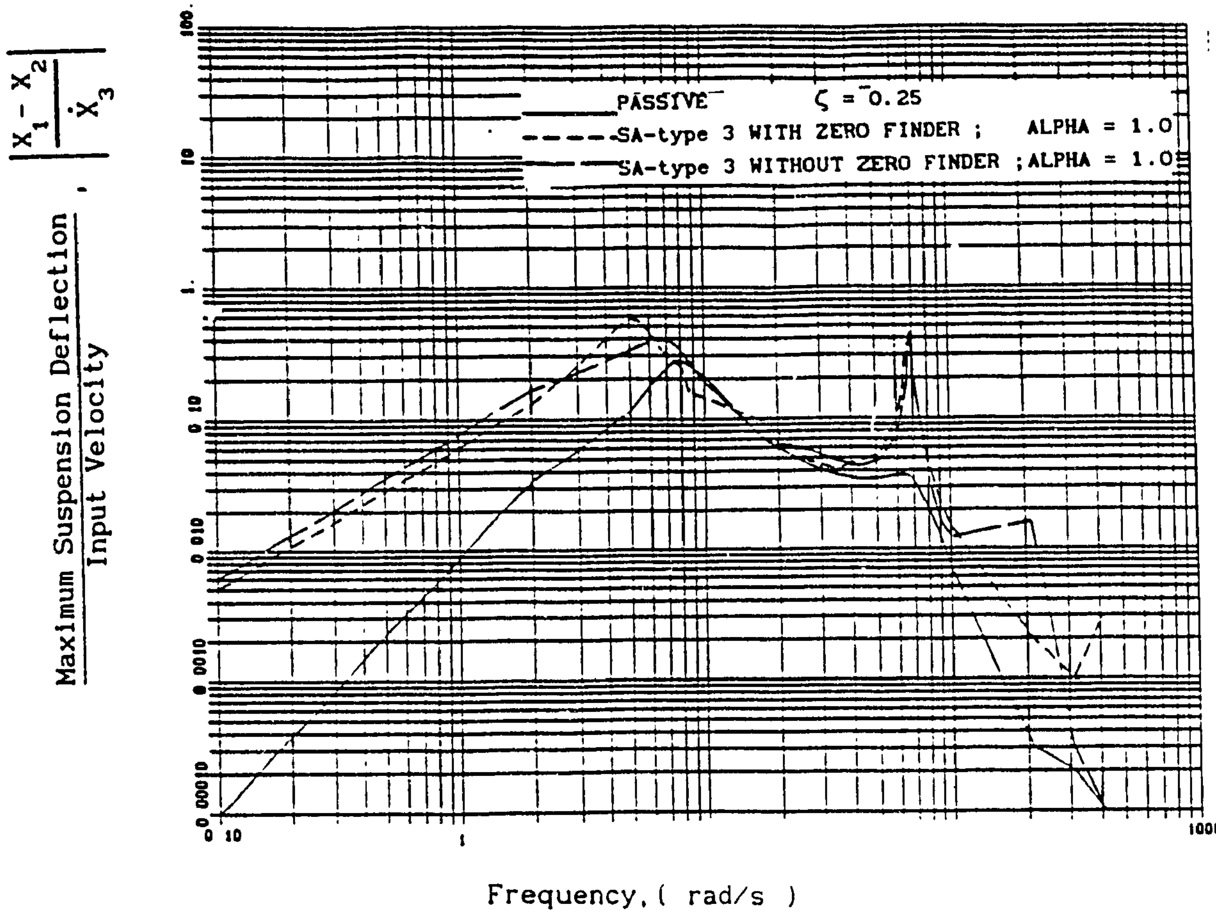


Fig.3.24 Influence of a zero-finder on the maximum suspension deflection response of an SA-type 3 suspension

Figs. 3.20 and 3.21 show the influence of the zero-finder on the tire and suspension deflections for the SA-type 1 system. It can be seen for these two responses, that the zero-finder has an influence in the low frequency regions up to the sprung mass natural frequency.

In the case of the SA-type 3 system, the responses with and without the zero-finder are illustrated in Figs. 3.22, 3.23, and 3.24. It is clear from these plots that the influence of the zero-finder is significant around the system natural frequencies.

Thus it is observed that the solution methodology when used without the zero-finder tends to deviate from the solution obtained with the zero-finder at regions near the natural frequencies. This is due to the fact that the magnitude of forces are higher in these regions and hence while switching is done it is preferable to find out the true initial condition at the point of discontinuity to continue the solution process and hence the response of the system.

3.6 Comparison of Responses

The comparison of sprung mass maximum acceleration responses between the three SA-systems are presented in Fig. 3.25. The response of SA-type 1 system is the best at the sprung mass natural frequency. The response of SA-type 3 system, however, shows an excellent behavior between the two natural frequencies. In the case of SA-type 2* system, the response is better than the passive system, but is marginally inferior in the frequency range between the two natural frequencies. On an overall basis, the response of

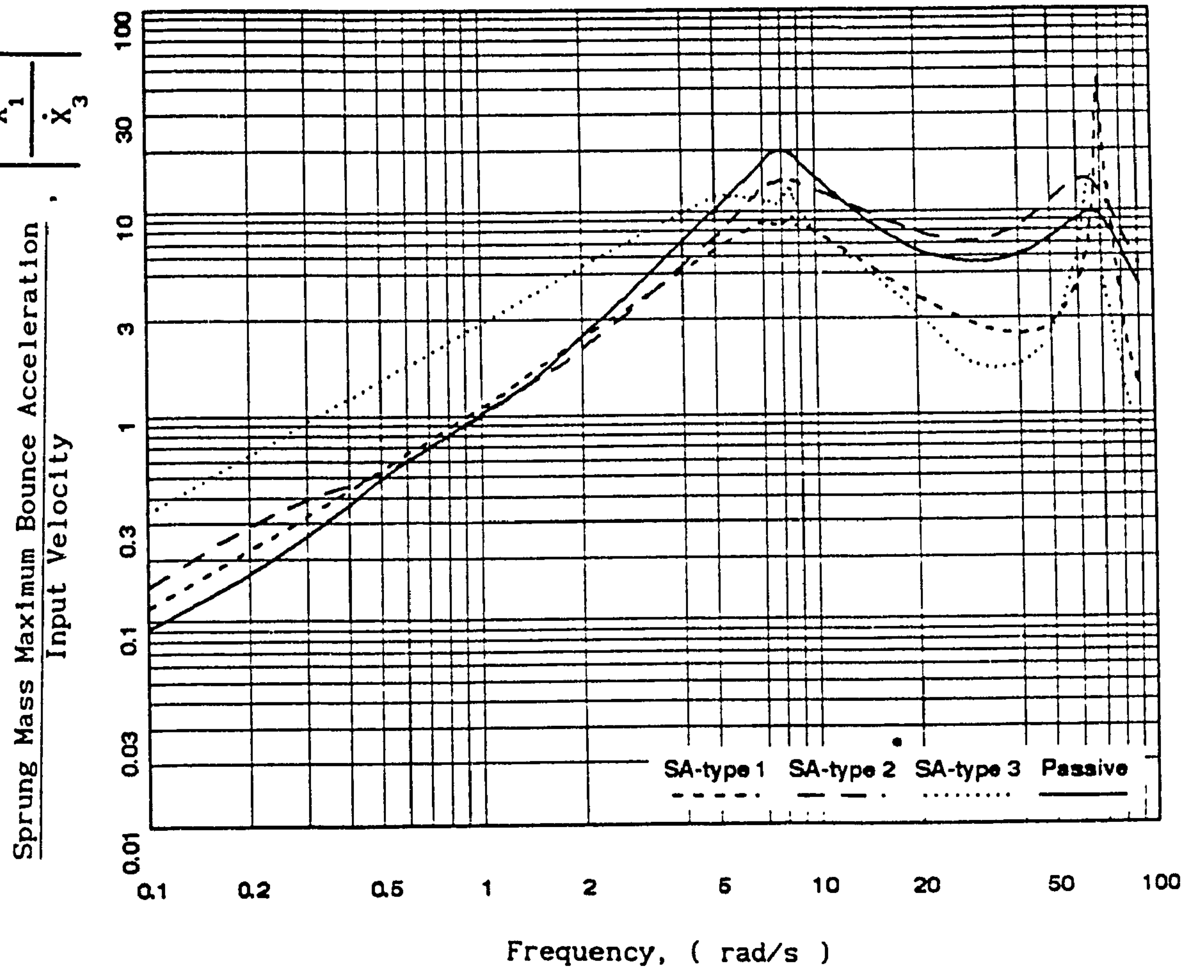


Fig. 3.25 Comparison of the sprung mass maximum bounce acceleration for passive (ζ = 0.25), SA-type 1 (ζ = 0.707), SA-type 2* (ζ = 0.707 and ζ_0 = 0.16) and SA-type 3 (ζ = 0.707 and α = 1.0) suspensions

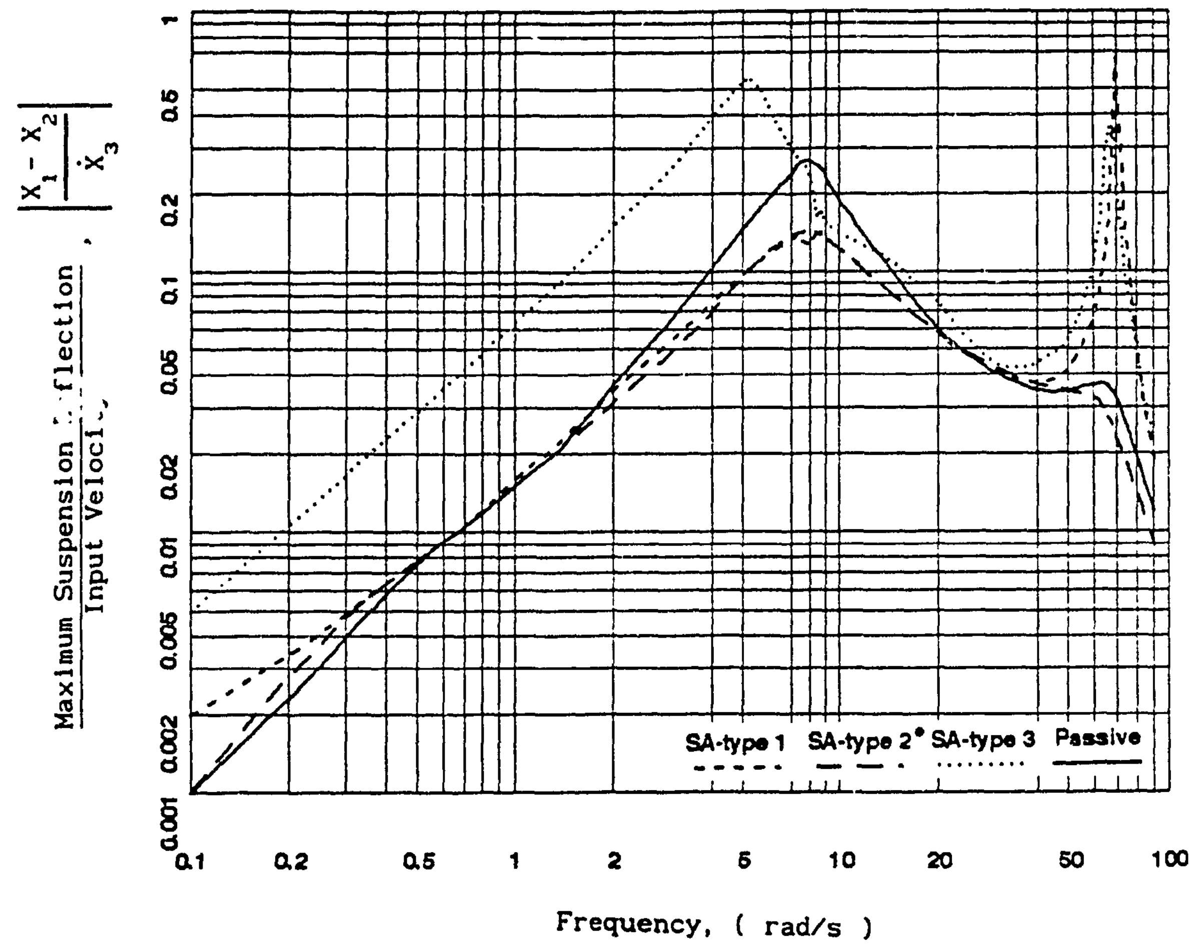


Fig. 3.26 Comparison of the maximum suspension deflection for passive (ζ = 0.25), SA-type 1 (ζ = 0.707), SA-type 2* (ζ = 0.707 and ζ_0 = 0.16) and SA-type 3 (ζ = 0.707 and α = 1.0) suspensions

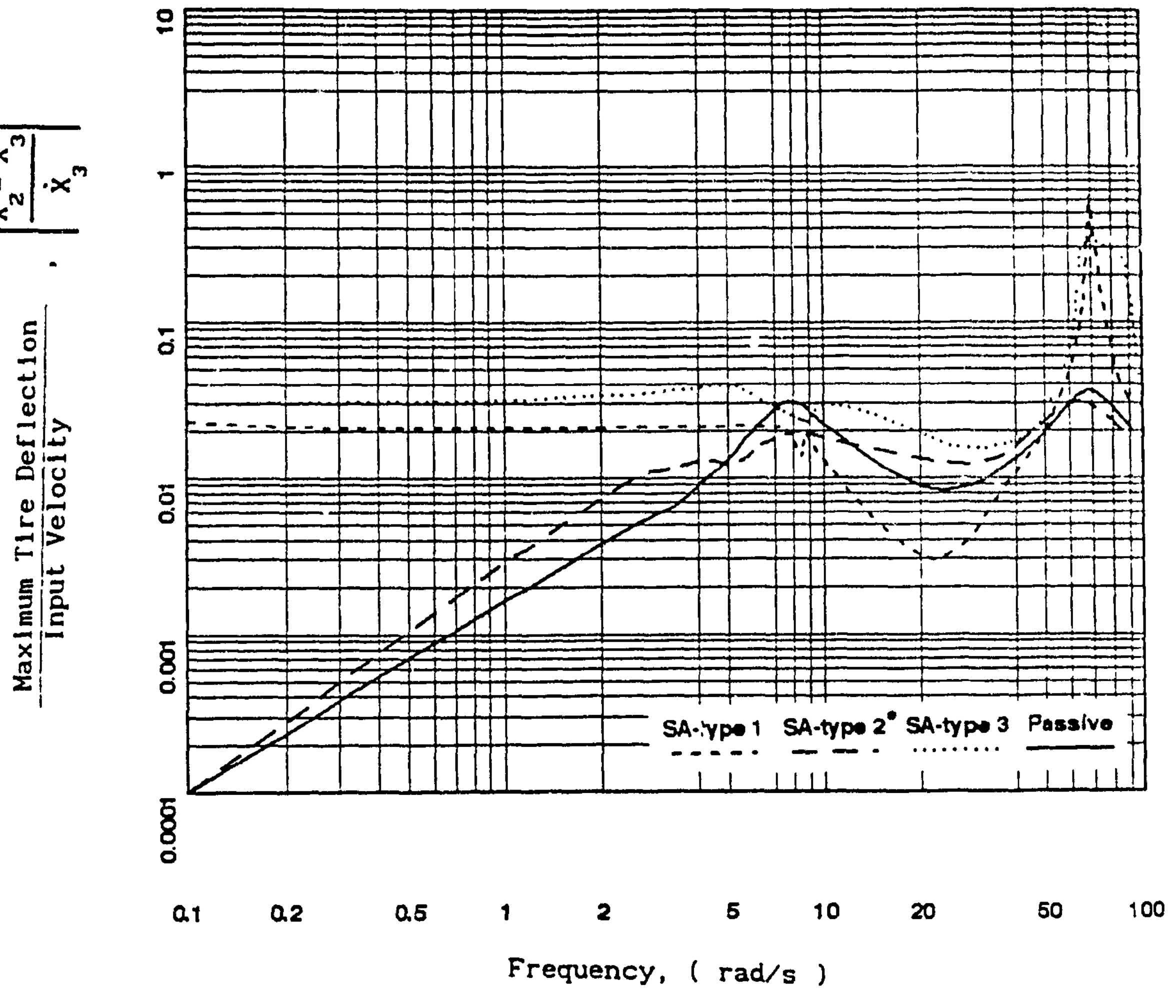


Fig. 3.27 Comparison of the maximum tire deflection for passive (ζ = 0.25), SA-type 1 (ζ = 0.707), SA-type 2* (ζ = 0.707 and ζ_0 = 0.16) and SA-type 3 (ζ = 0.707 and α = 1.0) suspensions

SA-type 2^{*} system is a good compromise at both the natural frequencies, and hence it can be considered to provide an optimal ride performance among the three SA-systems.

The suspension deflection responses are presented in Fig. 3.26, which shows that the response of SA-type 2^{*} system is better than the other two SA-systems and the passive system through out the frequency range with an exception at very low frequencies.

In Fig. 3.27, the tire deflection responses of the SA-systems are compared. The SA-type 1 and SA-type 3 systems provide inferior tire deflection responses at lower frequencies, and unsprung mass natural frequency. The SA-type 2^{*} system, again provides a compromise solution. In general, the SA-systems encounter more suspension deflection than passive systems ana hence require more rattle space requirement.

3.7 Summary

The Two D-O-F quarter car bounce model with semi-active suspension systems are examined for vibration isolation properties. The performance of the SA-systems are evaluated and compared with passive suspension. The discontinuous equations of semi-active suspension systems are solved in conjunction with a zero-finder. A detailed insight is made to study the response behavior and configuration of the SA-systems through steady-state time plots. The steady-state time responses are presented with detailed discussions on the responses and nature of the control logic of the systems.

The SA-type 1 and SA-type 3 systems undergo a lock-up mode for certain parts of the vibration cycle due to the relative velocity response crossing a zero value at lower frequencies. Though this cannot be identified explicitly in the time plot, it can be seen that there is a sharp spike in the acceleration response when the system breaks away from a lock-up mode. The relative velocity, condition function, and hence the damping force responses have fluctuations due to the tendency of the system to switch back to its original course of motion at lock-up condition. The treatment of lock-up is facilitated by finding out the zero of the condition function using a zero-finder.

From the high frequency time histories, it is observed that for the SA-type 1 system, the value of the sky-hook condition function remains positive for most of the time in one period of oscillation, requiring the semi-active damping force (F_d) to be "on" for a long period of time. Consequently, the duration of time for the negative value of the condition function is almost zero in the same one period. This implies that the damping force (F_d) is to be switched instantaneously back to "on" position. Though this switching behavior of the condition function and the damping force is identified at high frequencies with an instantaneous breaking of the lock-up mode, difficulties may be encountered in practice while implementing this scheme.

For SA-type 3 system, in one cycle, the condition function switches periodically between positive and negative values for all the frequencies. Because of this behavior, this SA-system unlike the SA-type 1 system does not require instantaneous switching at higher frequencies. The acceleration response of the SA-type 3 system remains zero for certain parts of the vibration cycle when the damping force is "on". This is in conformity with the objective of the control logic.

The ride performance in terms of acceleration response for all the SA-systems is better than the passive system at the sprung mass natural frequency and inferior at the unsprung mass natural frequency. The response of SA-type 1 system is superior at the sprung mass natural frequency, however, the response of SA-type 3 system is better in the frequency range between the two natural frequencies. The SA-type 2 suspension system is unstable at certain frequency ranges and hence a modified system, SA-type 2^*, having a small value of passive damping in parallel to the SA-type 2 damping is studied. The response of SA-type 2^* system is better than the passive system though it is marginally inferior between the two natural frequencies. On the overall basis, the response of SA-type 2^* system is a good compromise at both the natural frequencies. Hence SA-type 2^* can be considered to provide an optimal ride performance among the three SA-systems.

The suspension deflection responses show that the response of SA-type 2^* system is better than the other two SA-systems and the passive system through out, with an exception at very low frequencies. The response of SA-type 3 system is inferior at lower frequencies.

The tire deflection responses of the SA-type 1 and SA-type 3 systems are inferior at the lower frequencies, and unsprung mass natural frequency. The SA-type 2^* system provides a compromise in general though the response is marginally inferior between the two natural frequencies. Since the tire deflection is inferior for SA-systems at lower frequencies, heavier pavement load is expected, resulting in more wear and tear of the tire.

An effort is made to study the effect of a zero-finder in the solution methodology. Though, the study on the SA-systems without using a zero-finder in the solution methodology shows significant variations on the system responses for all the frequencies, the influence is dominant at the natural frequencies. This is due to the fact that the magnitude of the forces are high at these frequencies and hence solving for the discontinuous equations of SA-systems without proper initial conditions at the instant of switching could cause a dominant and obvious deviation in the responses.

CHAPTER 4

ANALYTICAL AND EXPERIMENTAL INVESTIGATIONS ON THE SA-TYPE 4 SUSPENSION SYSTEM

4.1. Introduction

There are numerous analytical studies [21-29,61] reported in the past on the ride performance of various SA-suspension systems. However, there are only very few reported results on experimental investigations [51,62-69]. Many of the experimental analyses reported earlier were using sky-hook control logic for SA-suspension [63,67,68]. The concept of SA-suspension using the SA-type 3 system was studied in a single D-O-F model using a combination of spring loaded poppet and check valve operated mechanism, and a dual chamber damper [69]. In this investigation, a laboratory testing of the SA-type 4 suspension system was conducted, and the results are presented and compared with analytical responses. A detailed parametric study is also presented to examine the influence of damping. This investigation is thus intended to provide a comprehensive study based on both analytical and experimental means.

The present study is carried out using a two d-o-f quarter car bounce model with an adjustable shock absorber and the SA-type 4 semi-active suspension control logic. The adjustable shock absorber is a commercially available unit used on the rear suspension of Nissan 300ZX (1987) [70,71] automobile. The shock absorber has three types of damper orifice settings, namely firm, normal and soft which correspond to high, medium and low damping values, respectively.

In the proposed laboratory testing, since the objective is to investigate the the SA-type 4 suspension scheme, the control logic must be proportional to the relative displacement and velocity of the sprung mass, and the magnitude of the damping force must switch between two sets (high and low) of damping values.

The damping characteristics of the soft mode in the adjustable shock absorber can be used to represent the "off" mode with small damping value. The damping characteristics of the firm mode will represent the "on" mode in the control logic with high damping value. The control logic used for modulating the orifice of the shock absorber will be based on the relative displacement and velocity of the sprung mass. The SA-type 4 control logic switches the damping force between the firm and soft modes by adjusting the orifices in the shock absorber using a microprocessor controller.

4.2. Description of the Experimental Set-up

The photograph of the experimental set-up is presented in Fig. 4.1. The experimental set-up represents a two d-o-f quarter car model with the sprung and unsprung masses of the vehicle connected rigidly by a coil spring and an adjustable shock absorber mounted in parallel. The unsprung mass is supported by a coil spring at its bottom representing the compliance characteristics of the tire, and is connected at one end to an electro-hydraulic shaker. The masses are aligned in line and constrained to move only in vertical direction by a set of four linear guide bearings. The linear bearings are rigidly attached to the inertial frames both at the top and bottom.

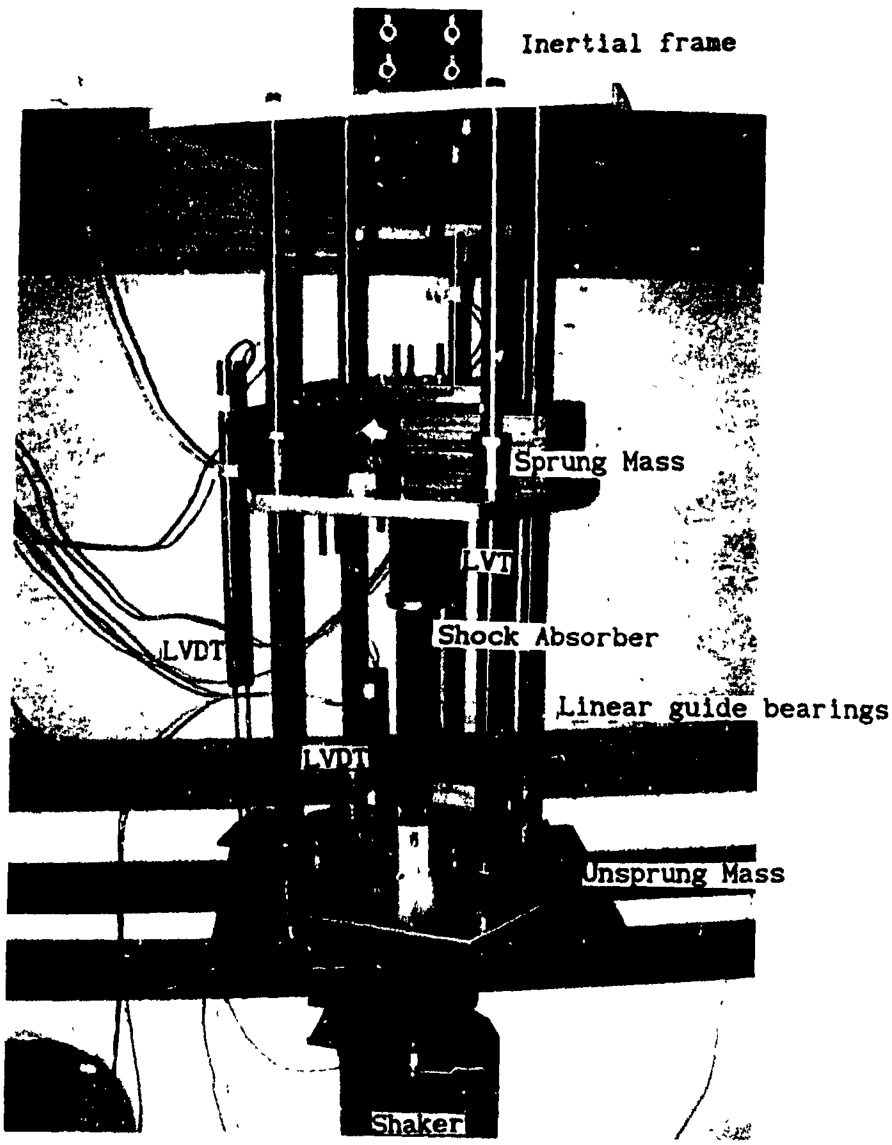


Fig. 4.1 Experimental set-up used for testing the SA-type 4 suspension model

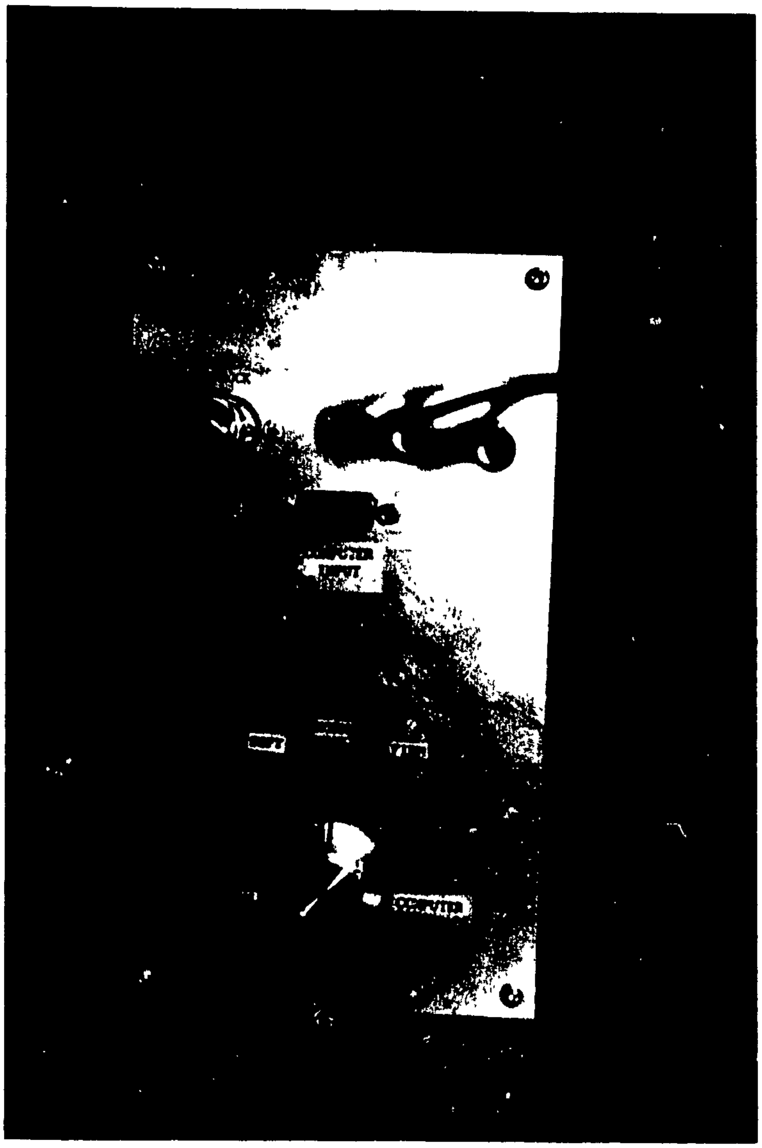


Fig. 4.2 Picture of the electronic control box

The use of a commercial shock absorber in the experimental testing provides a comprehensive understanding of both the adaptability and effectiveness of such an adjustable shock absorber in the semi-active (SA) control policy. Also it can be used for comparing the performance of the SA-suspension against the passive modes with the damper setting in the firm, normal and soft configurations. The shock absorber has a built-in torque motor to turn the shutter between the different orifice positions. Hence no external device or fabrication is needed to convert the commercially available adjustable shock absorber for this testing.

The SA-type 4 control logic is employed in this experimental analysis using an electronic control box for switching between the "on" and "off" modes. The picture of the control box is presented in Fig. 4.2. The control box has two modes, namely the direct computer control for SA-switching, and manual control. During the manual operation it has a switch with three positions for firm, normal and soft type suspensions. The torque motor in the adjustable shock absorber turns the shutter between each orifice position in 180ms. Hence, using the control box, the switching between each of the type of suspension settings is done in 180ms.

The parameter values for the masses and springs used in the experimental set-up are selected to be one quarter of the value of a Nissan 300ZX automobile. Hence the undamped natural frequencies of the sprung and unsprung masses of the scaled down experimental model will have values identical to the full vehicle. Since it is not possible to scale down the damping value to 25% of its original value without design changes to the orifices in the shock absorber, the original shock absorber is used in the experimental set-up without any modification. This introduces a high damping value and the experimental results have over damping characteristics in the laboratory model. The parameters used in the experimental model are presented in Table 4.1.

In the experimental set-up, the ride performance of the suspension system is evaluated by measuring the sprung mass acceleration response using an accelerometer. The suspension deflection response is measured by an LVDT (Linear Variable Differential Transformer) mounted between the sprung and unsprung masses. The tire deflection response is also measured by an LVDT installed between the unsprung mass and the moving platform of the shaker. Since the control function requires the relative velocity between the two masses, an LVT (Linear Velocity Tranducer) is used.

The instrumentation used for measuring the required responses are also shown in Fig. 4.1. The details of the instruments used are listed as follows:

1. The Bruel & Kjaer 4370 piezoelectric, delta shear type accelerometer and the charge amplifier type 2651 to measure acceleration of the sprung mass.
2. Two LVDT's with displacement rating of 10" and 3" : Schavitz model nos. 3000 DC-E and 10000 DC-E
3. The LVT Schavitz model no. 7L VT-2 10.0

The experimental suspension model is excited by an electro-hydraulic shaker. The shaker has a stroke capacity of ± 3 inches

TABLE 4.1. Parameter Values of Experimental Set-up

M_1	106	kg
M_2	13.5	kg
K_1	6842	N/m
K_2	28643	N/m
$f_1(\omega_1)$	1.28 (8.03)	Hz (rad/s)
$f_2(\omega_2)$	8.16 (51.27)	Hz (rad/s)
C	*	

* The damping values are presented in the Appendix 1

in the 0-200 Hz frequency range with a dynamic load capacity of approximately 22268.7 N (5000 lb). Harmonic input excitations with peak-to-peak amplitudes of 12.7 mm (0.5 inch), 19.1 mm (0.75 inch), and 25.4 mm (1.0 inch), at frequencies of 0.1 Hz to 2.0 Hz for the SA-type 4 suspension system and 0.1 Hz to 10.0 Hz for the firm, normal and soft type suspension systems are selected. A data acquisition system, RC electronics computer scope EGAA (Enhanced Graphic Acquisition and Analysis) version 3 is used for acquiring data simultaneously through 8 channels. Since, the data acquisition system is capable of acquiring data at 1 MHz when only one channel is used, the minimum sampling time available for the present study with 8 channels is 8 μs. The rate of data acquisition is set manually for each frequency of excitation. The results are presented for experiments conducted with 12.7 mm (0.5 inch) peak-to-peak input excitation.

4.2.1 Estimation of Suspension System Parameters

The coil springs used in the suspension set-up are tested on an universal testing machine to estimate their stiffness values. The load vs deflection characteristics are obtained and the spring rates are estimated as given in Table 4.1. The damping values are estimated by conducting a detailed study on the damping characteristics of the adjustable shock absorber. In general, the design and modeling of a shock absorber in a suspension system [42,70 , 72] is based on different needs a venicle has to encounter. The adjustable shock absorber used is designed by the Nissan Motor Company [70,71] to provide better high speed performance, critical turning performance, and improved steering. As mentioned earlier, the adjustable shock absorber shown in Fig. 4.3a, has a motor which operates a shutter to vary the orifice opening to three different settings to provide firm, soft, and normal ride performance. In the commercial vehicle, these settings are operated by the driver through a dashboard mounted switch inside the vehicle according to the preference on ride comfort. However, it should be noted that for the SA-type 4 suspension system, switching between the firm and soft modes is accomplished automatically according to the control function in the experiment set- up.

Preliminary tests are conducted on the shock absorber to identify the factors that could cause deviations from a desired damping behavior. The potential source of performance degradation could occur due to compliant characteristics of the bushing at the mounting eyelid of the shock absorber. A test rig to estimate its compliance is set-up. The set-up is constrained to move the shock absorber at the top flange with the load applied at the bottom. A load cell of capacity 10000 N is attached to the bottom of the shock absorber. The force applied is directly measured using the data acquisition system installed on-line through a PC based software, DADS-16. A monitoring scope is also used to observe the input excitation and force applied.

The observations are made in terms of the force vs deflection response which reveal that the bushing has a very high stiffness and the amplitude of deflection is negligible. Hence, it is assumed in this study that the bushing will have negligible influence on the performance of the shock absorber in terms of the damping characteristics and that

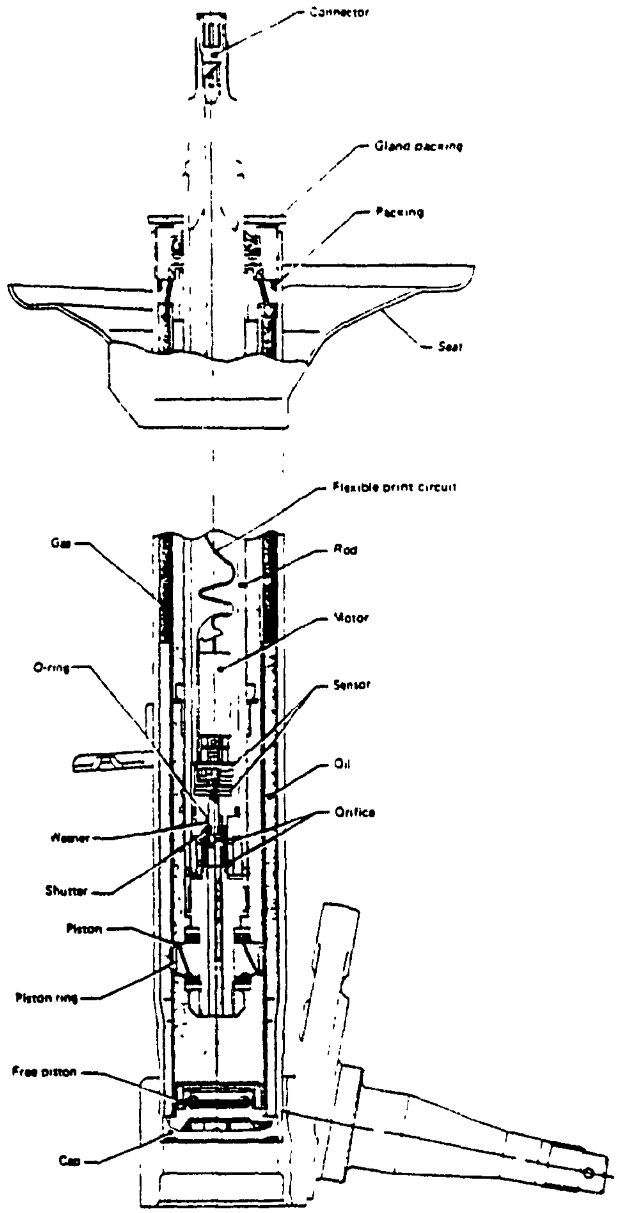


Fig. 4.3a The schematic of the adjustable rear shock absorber of Nissan 300ZX [70]

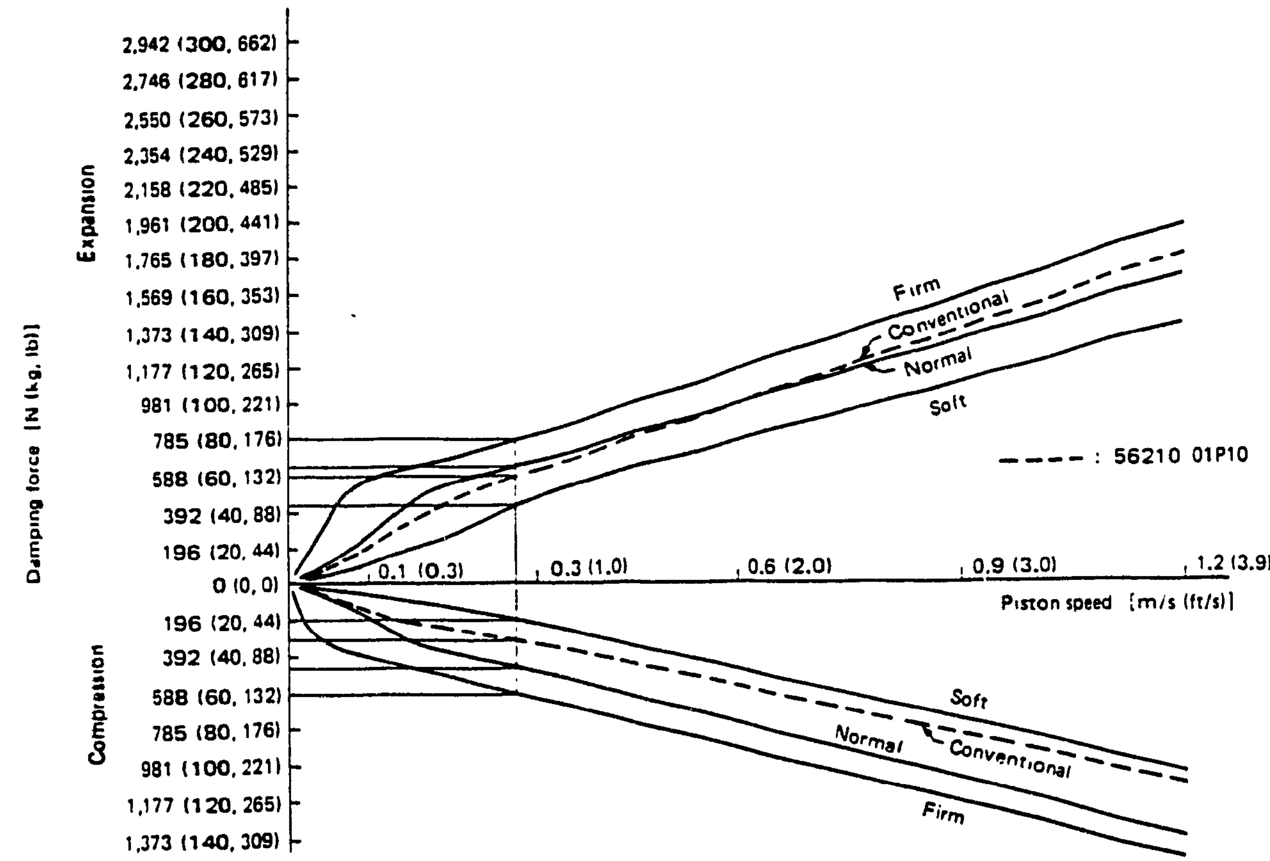


Fig. 4.3b Damping force vs velocity of Nissan 300ZX model rear shock absorber for the three modes of damping settings [70,71]

its influence on the overall system response is negligible.

The Nissan Motor Company has supplied [70,71] the damping characteristics of the shock absorber for all the three modes (firm, normal and soft) of the operational settings as shown in Fig. 4.3b. A detailed experimental testing of the shock absorber is carried out to verify the damping characteristics specified by the manufacturer [70,71]. Results of this testing are presented in Appendix 1.

The maximum damping force is measured for discrete frequencies and hence the damping behavior of the shock absorber is approximated over the given range. The shock absorber characteristics established from the testing are in general agreement with the manufacturer's specification in the low velocity region. However, there are variations in the high velocity region which may partly be attributed to the factors such as the influence of seal friction, change in viscosity of fluid at high speed operating conditions. Also, the estimate of the damping force from the tests are based on the measurement of peak force.

The damping characteristics estimated from this experimental study are used in the analytical study to validate the experimental results.

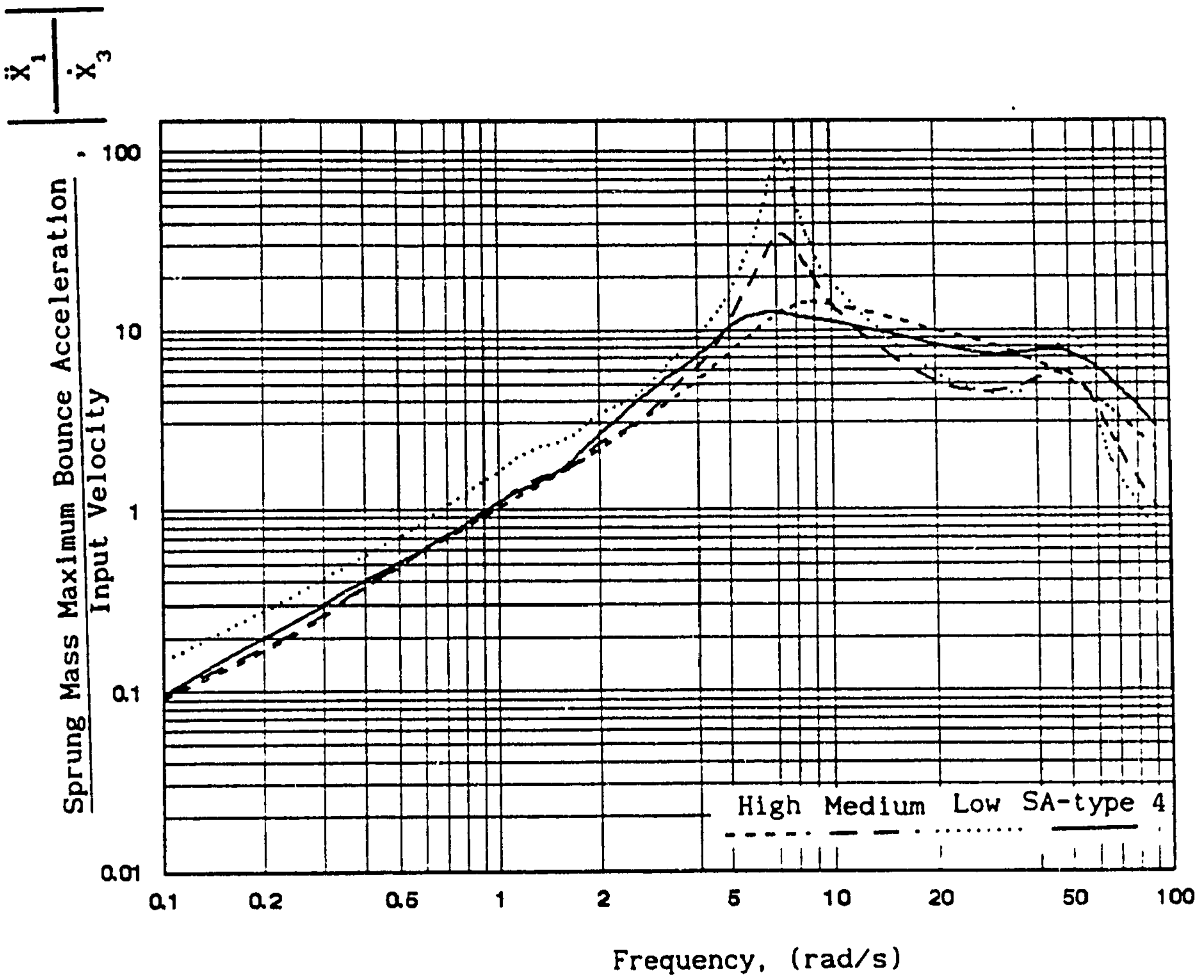
4.3. Analytical Study Using the SA-type 4 Suspension

The SA-type 4 suspension system is examined analytically for vibration isolation performance. Since this SA-system switches between the firm and soft modes of damping characteristics of the shock absorber, the performance of the suspension system, using these modes are also individually evaluated. The analytical study is carried out with a constant amplitude (0.5 inch peak-to-peak) sinusoidal base excitation for frequencies 0.1 to 10 Hz.

In Fig 4.4, the maximum bounce acceleration response of SA-type 4 suspension system with damping values of $\zeta_{on} = 0.707$ and $\zeta_{off} = 0.05$, representing firm and soft modes is presented. It should be noted that the damping values for firm and soft modes are constants for both compression and extension stroke of the shock absorber. The figure also presents results on the performance of passive suspensions with various damping values.

The response of SA-type 4 system at the sprung mass natural frequency is superior compared to the passive system with medium ($\zeta = 0.25$) and low ($\zeta = 0.05$) damping values and marginally better than the passive system with high damping ($\zeta = 1.00$). However, the SA-type 4 system's response is inferior to the passive system with medium and low damping for the frequency range between the two natural frequencies. At the unsprung mass natural frequency, the response of SA-system is marginally superior than the system with low damping , and inferior than the system with medium and high damping. In general, the response of the SA-type 4 system is similar to the passive system with high damping.

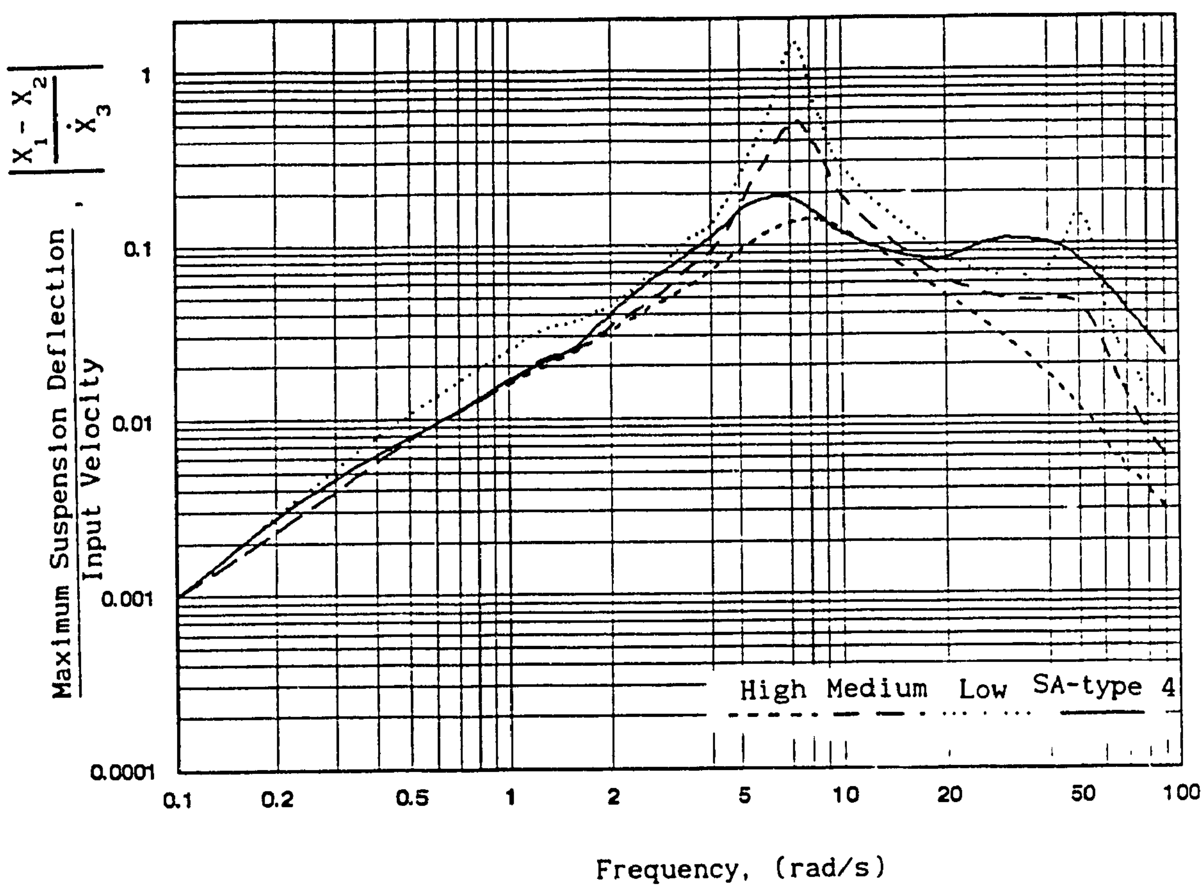
The suspension deflection response of the SA-type 4 system is presented in Fig 4.5 and compared with passive systems with high, medium and low damping. The response of SA-type 4 system is better than both



Low : ζ = 0.05 ; Medium : ζ = 0.25 ;

High : ζ = 1.00 ;

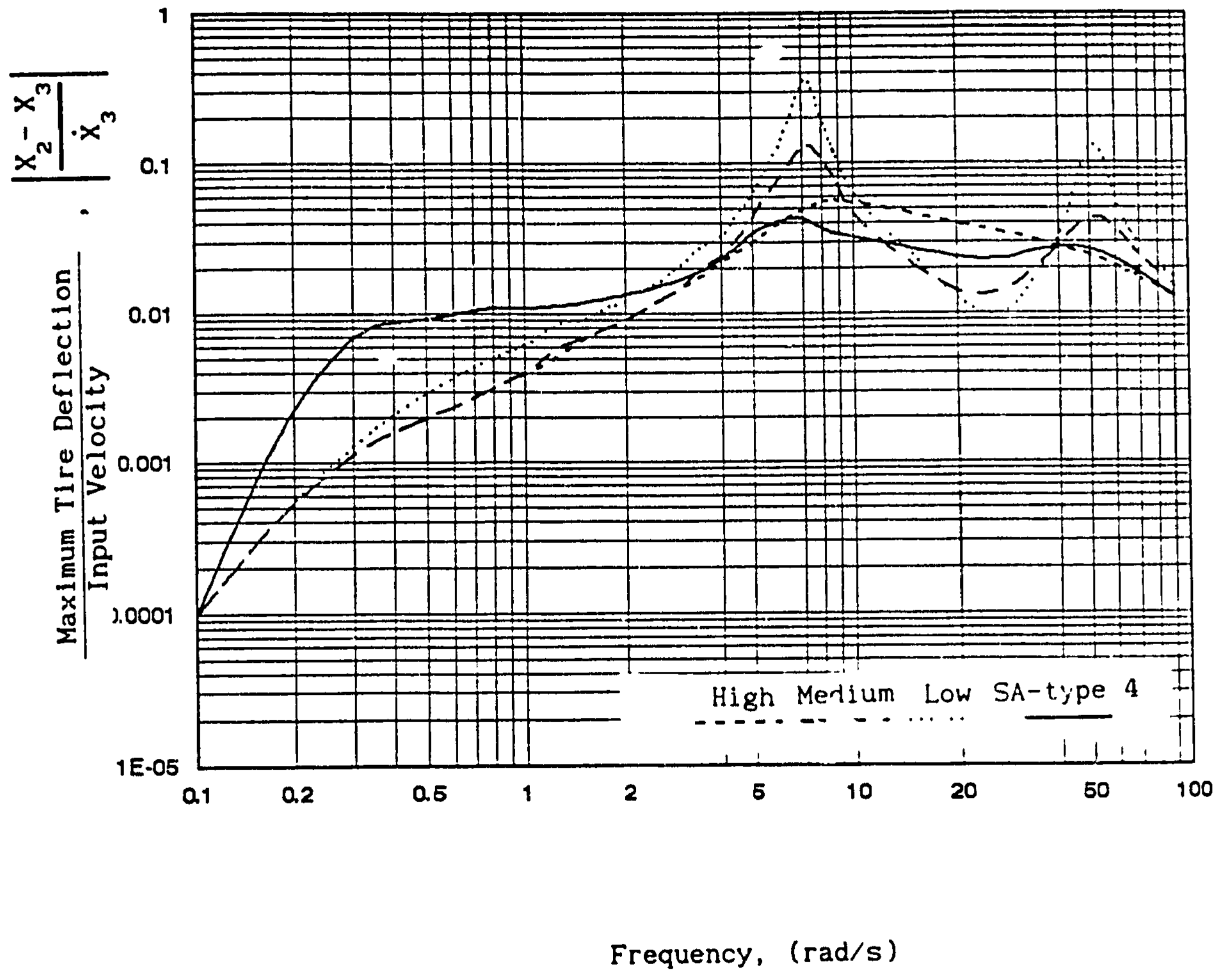
Fig. 4.4 Comparison of maximum bounce acceleration response of passive and SA-type 4 (ζ_{on} = 0.707, ζ_{off} = 0.05) suspensions



Low : ζ = 0.05 ; Medium : ζ = 0.25 ;

High : ζ = 1.00 ;

Fig. 4.5 Comparison of maximum suspension deflection response of passive and SA-type 4 (ζ_{on} = 0.707, ζ_{off} = 0.05) suspensions



Low : ζ = 0.05 ; Medium : ζ = 0.25 ;

High : ζ = 1.00 ;

Fig. 4.6 Comparison of maximum tire deflection response of passive and SA-type 4 (ζ_{on} = 0.707, ζ_{off} = 0.05) suspensions the medium and low damped passive systems at the sprung mass natural frequency and inferior than the high and medium damped passive systems at the unsprung mass natural frequency.

The tire deflection response of SA-type 4 system is presented in Fig. 4.6 and compared with passive systems with high, medium and low damping. The response of SA-type 4 system is inferior at low frequencies below the sprung mass natural frequency. However, its response is better at both the natural frequencies.

In Fig 4.7, the sprung mass bounce acceleration response is presented. Since the damping values for all the three passive modes are high, the suspension system with the value of the sprung mass considered will be over damped. Hence, the response shown in Fig. 4.7 does not have two distinct resonant peaks corresponding to the sprung mass and the unsprung mass resonant frequencies. It can be noted from the figure, that the response has only one peak at a frequency between the two resonant frequencies. Comparing the responses it can be concluded that the firm type has inferior performance when compared to the other types and the normal type is in turn inferior compared to the soft type of suspension. The performance of SA-type 4 system is similar to the other three passive types at lower frequencies and is better around the sprung mass natural frequency when compared to firm and normal types of suspensions. However, the responses of all the systems are similar at higher frequencies.

The suspension deflection and tire deflection responses for the different systems are presented in Figures 4.8 and 4.9. The suspension

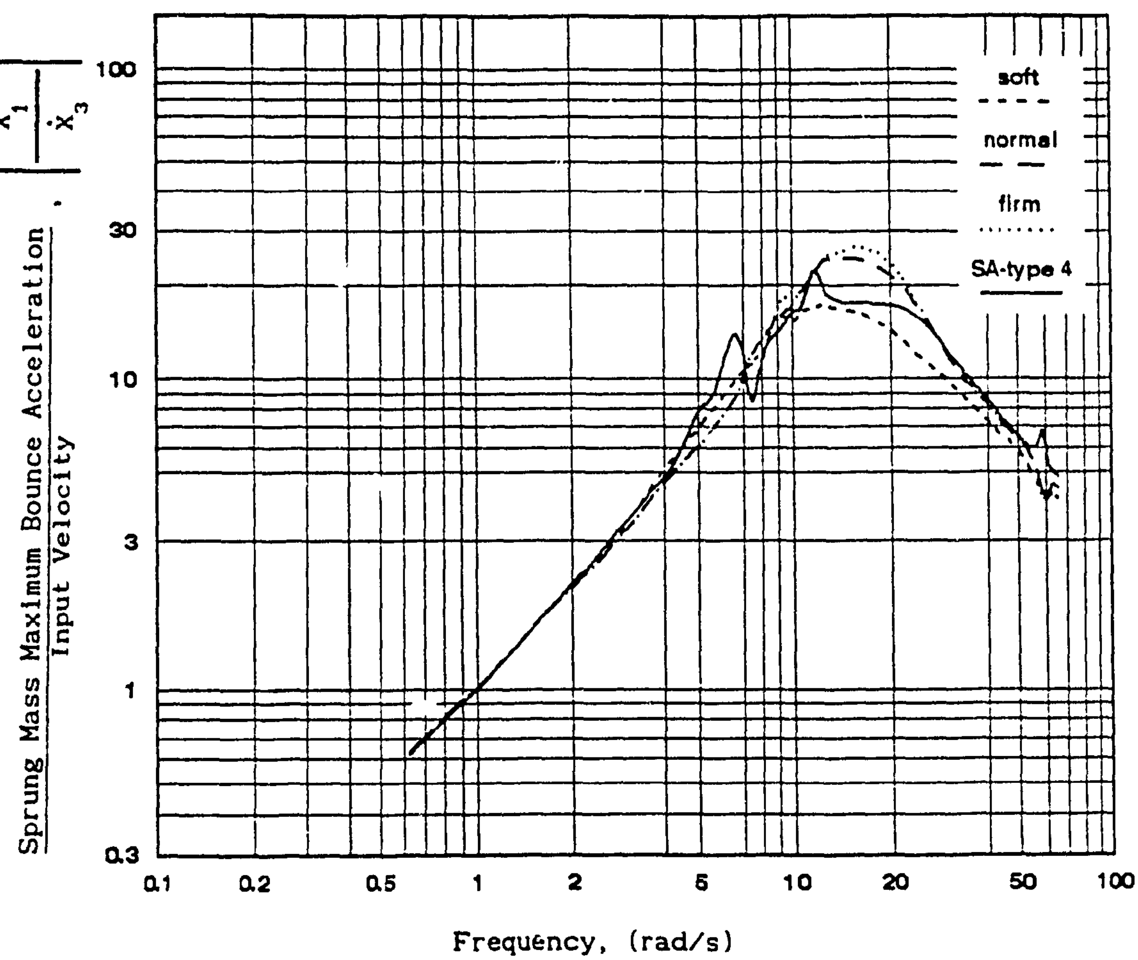


Fig. 4.7 Comparison of sprung mass maximum bounce acceleration for firm, normal, soft, and SA-type 4 suspensions

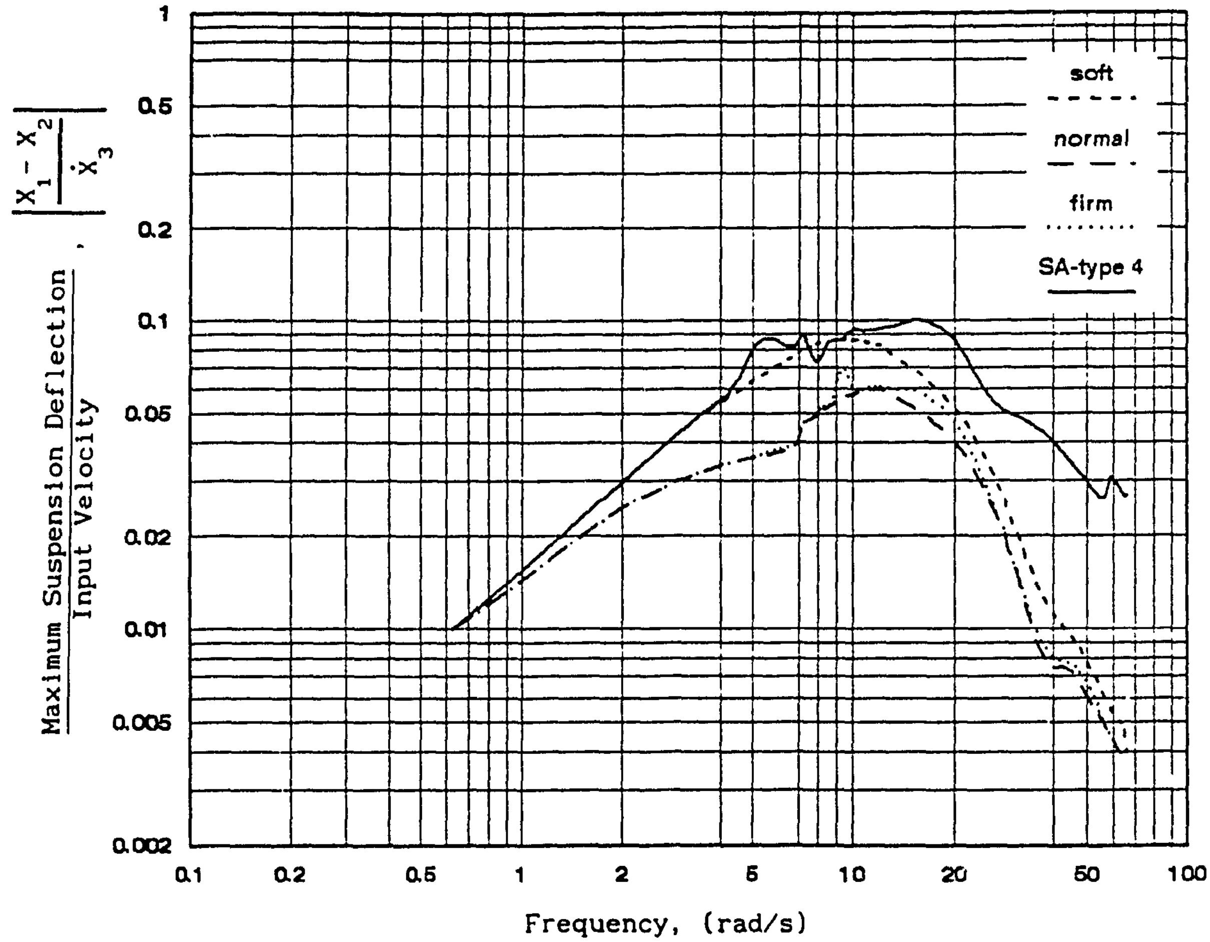


Fig. 4.8 Comparison of maximum suspension deflection response for firm, normal, soft, and SA-type 4 suspensions

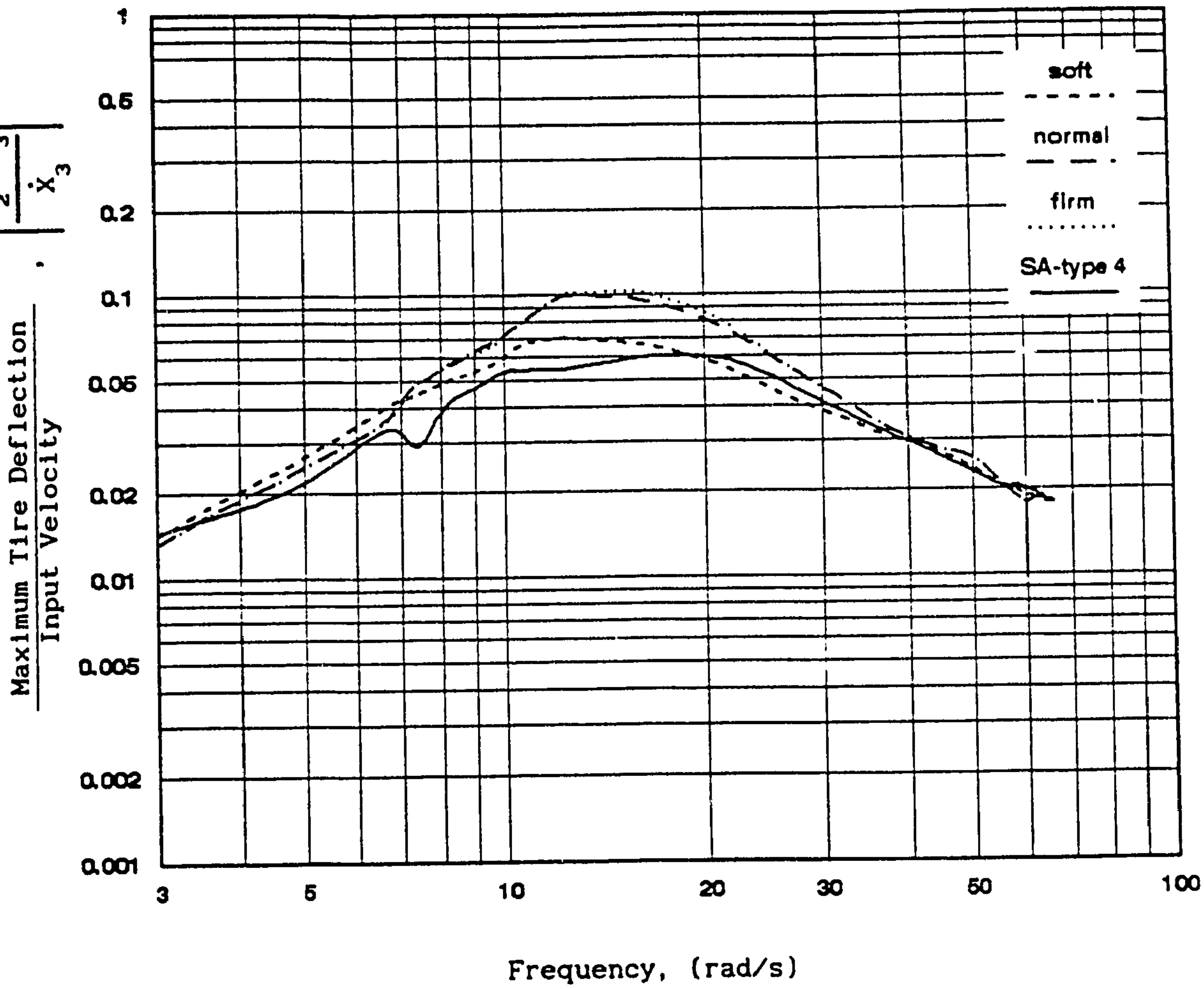


Fig. 4.9 Comparison of maximum tire deflection response for firm, normal, soft, and SA-type 4 suspensions

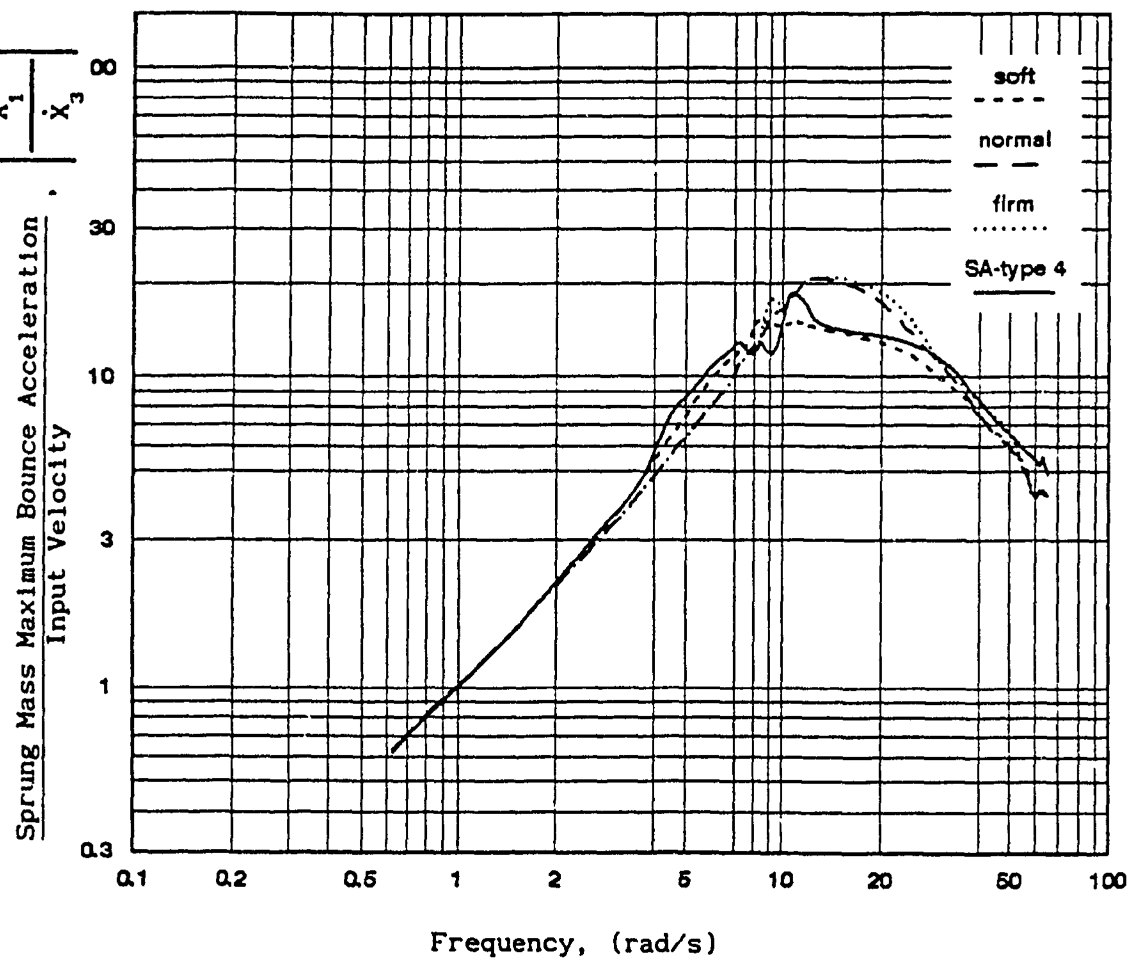


Fig. 4.10 Comparison of sprung mass maximum bounce acceleration for firm,normal, soft, and SA-type 4 suspensions at 75% of the damping capacity

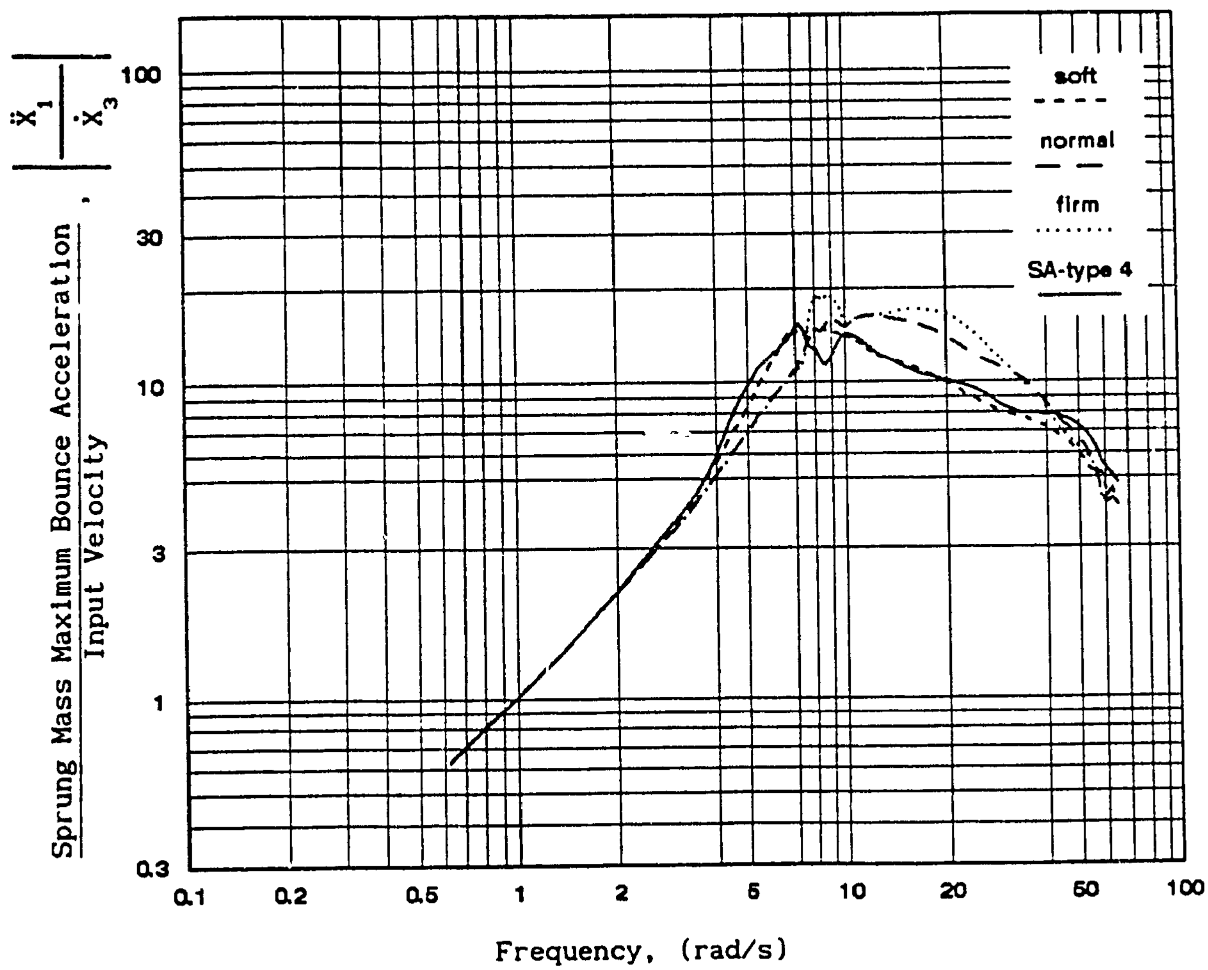


Fig. 4.11 Comparison of sprung mass maximum bounce acceleration for firm,normal, soft, and SA-type 4 suspensions at 50% of the damping capacity

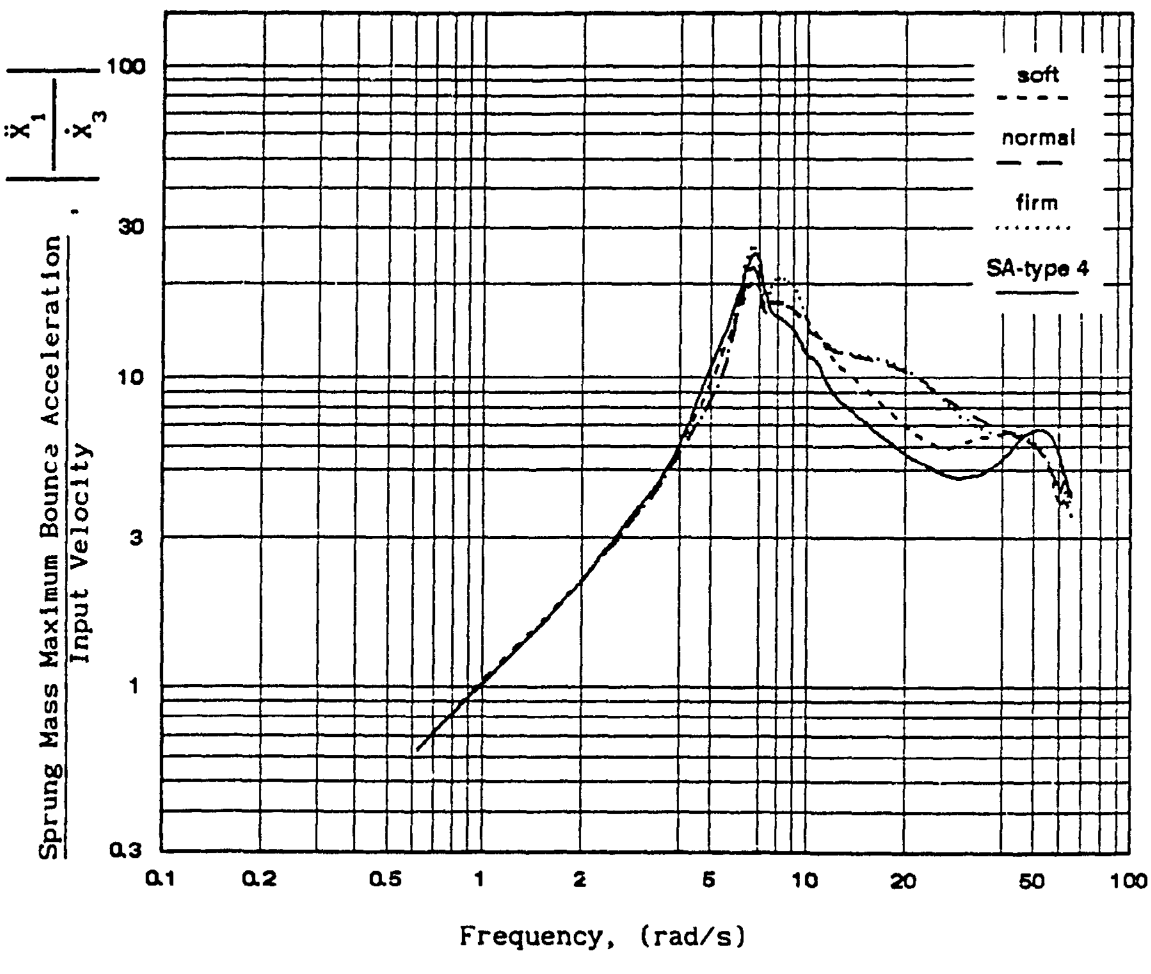


Fig. 4.12 Comparison of sprung mass maximum bounce acceleration for firm, normal, soft, and SA-type 4 suspensions at 25% of the damping capacity

deflection response is higher for the soft mode compared to the normal and firm modes, and the response for the normal mode is in turn more than the firm mode. However, the suspension deflection response of the SA-type 4 suspension is inferior to all passive modes for frequencies higher than 10 rad/s.

The tire deflection responses at low frequnecies for firm, normal and soft type of suspensions are similar. The response, however, is inferior for the SA-type 4 suspension at lower frequencies. The peak tire deflection response for firm and normal modes are higher than the soft and SA-type 4 suspensions and hence will account for higher tire wear and pavement loads.

In order to study the influence of damping in the system, the damping values for all the three modes and hence in the SA-type 4 system are reduced to 75%, 50%, and 25% respectively of the nominal values. Figures. 4.10, 4.11, and 4.12 show the sprung mass bounce acceleration responses of the system with different damping values. The maximum acceleration responses, presented in Fig. 4.10 and 4.11, have only one peak when the damping value of the system is at 75% and 50% of the full capacity respectively. It is observed that for 25% damping value, the acceleration responses for soft and SA-type 4 suspension systems show two resonant peaks corresponding to the system natural frequencies.

In Figures 4.13, 4.14, and 4.15, suspension deflection responses for 75%, 50%, and 25% of the full damping capacity are presented. The two distinct resonant peaks around the natural frequencies are

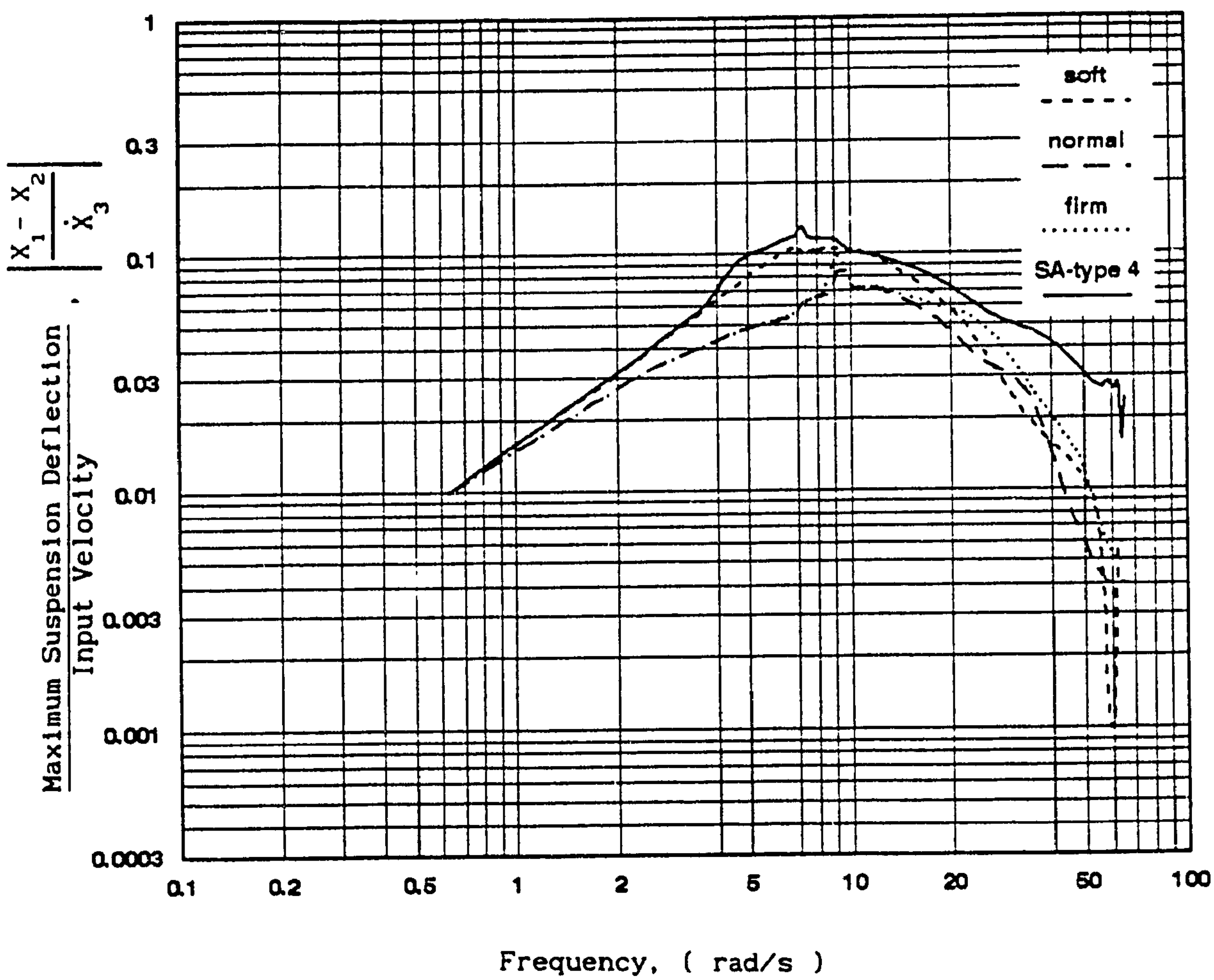


Fig. 4.13 Comparison of maximum suspension deflection response for firm, normal, soft, and SA-type 4 suspensions at 75% of the damping capacity

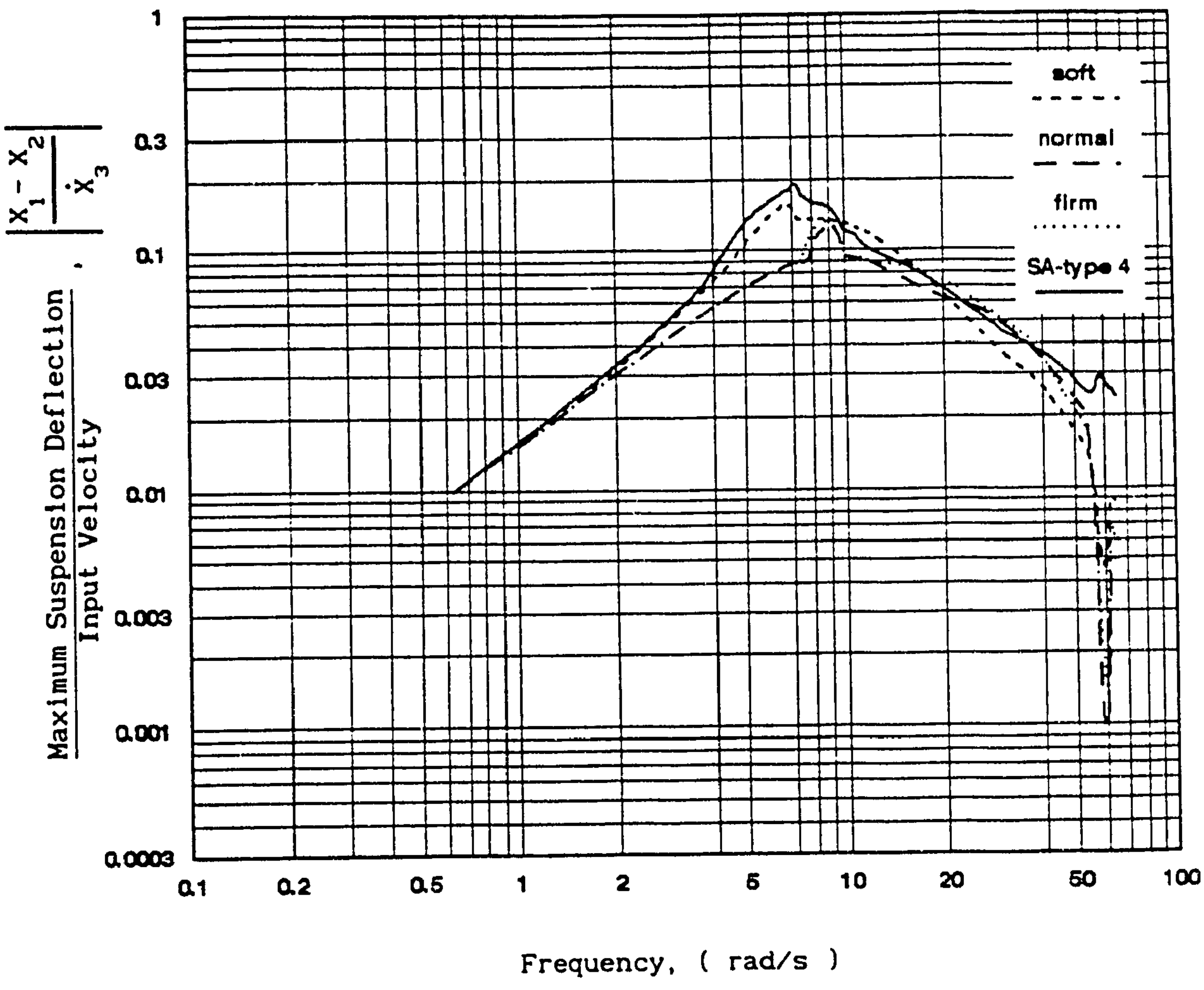


Fig. 4.14 Comparison of maximum suspension deflection response for firm, normal, soft, and SA-type 4 suspensions at 50% of the damping capacity

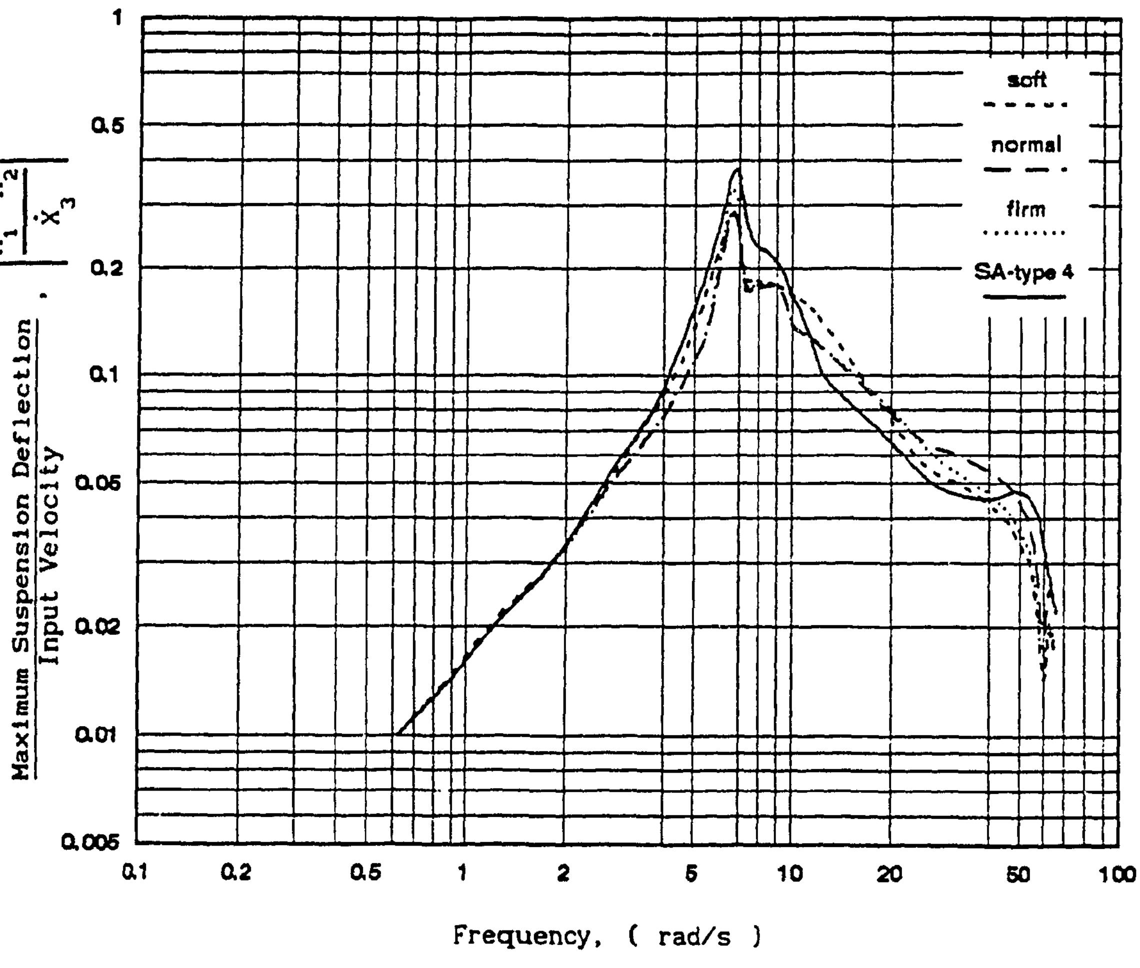


Fig. 4.15 Comparison of maximum suspension deflection response for firm, normal, soft, and SA-type 4 suspensions at 25% of the damping capacity

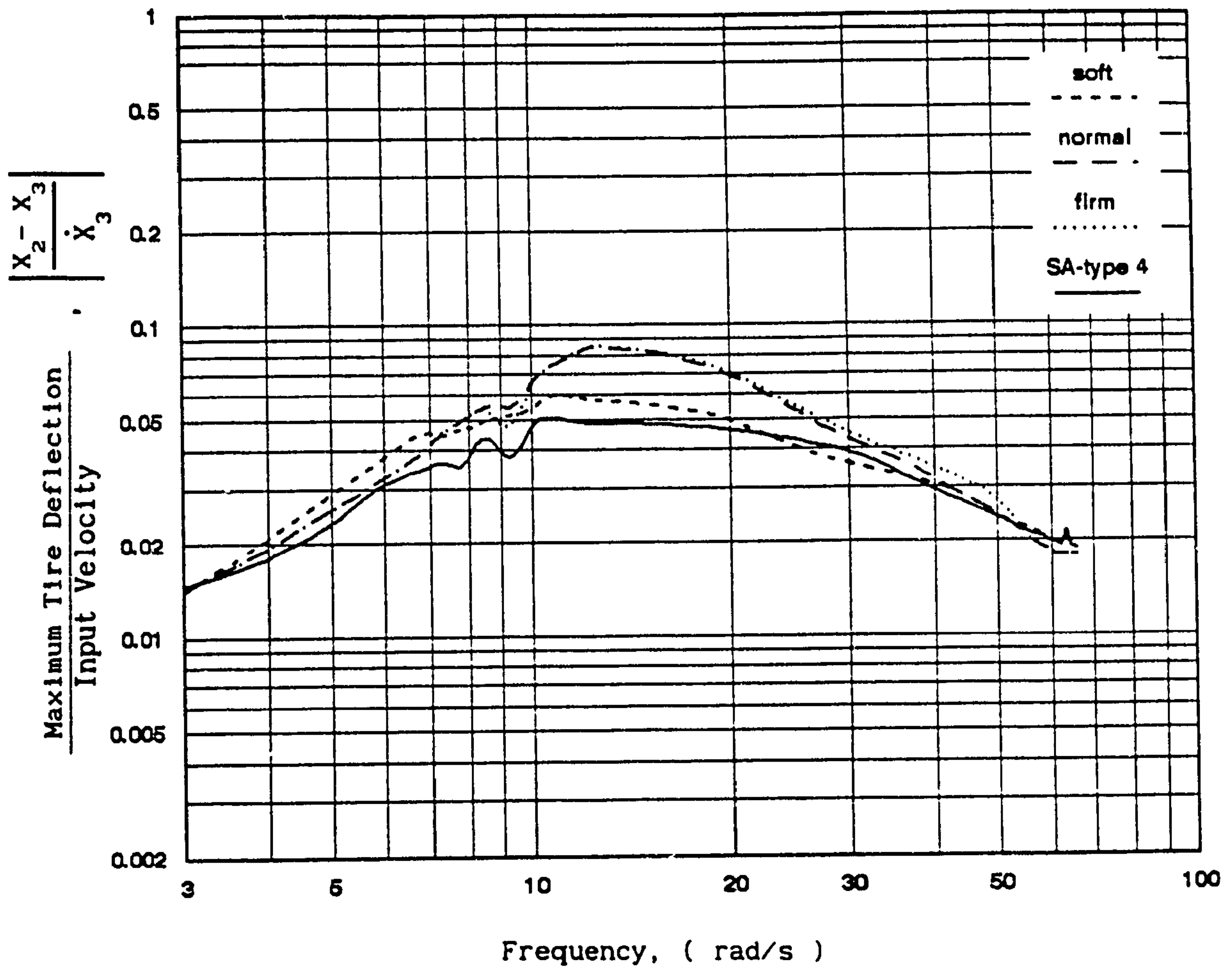


Fig. 4.16 Comparison of maximum tire deflection response for firm, normal, soft, and SA-type 4 suspensions at 75% of the damping capacity

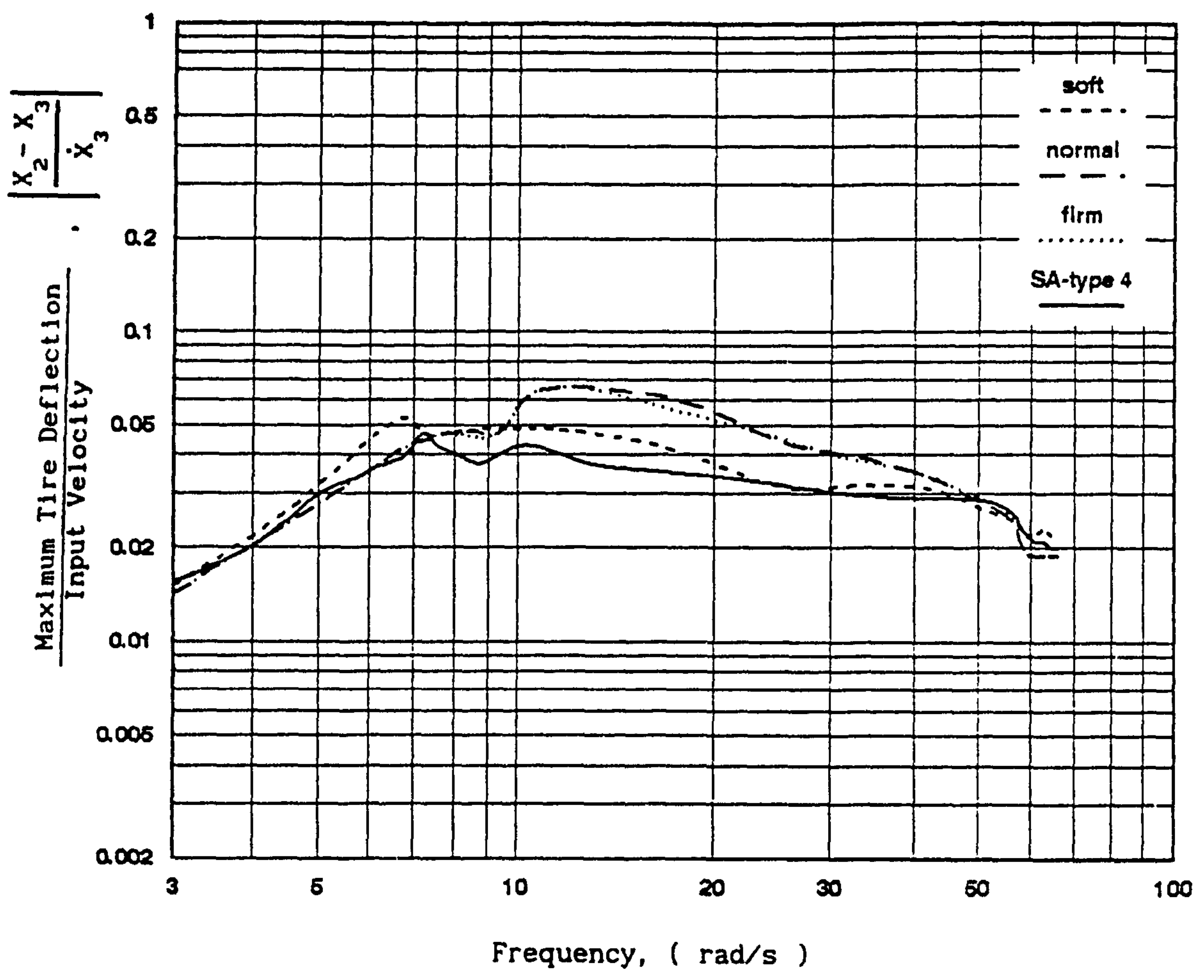


Fig. 4.17 Comparison of maximum tire deflection response for firm, normal, soft, and SA-type 4 suspensions at 50% of the damping capacity

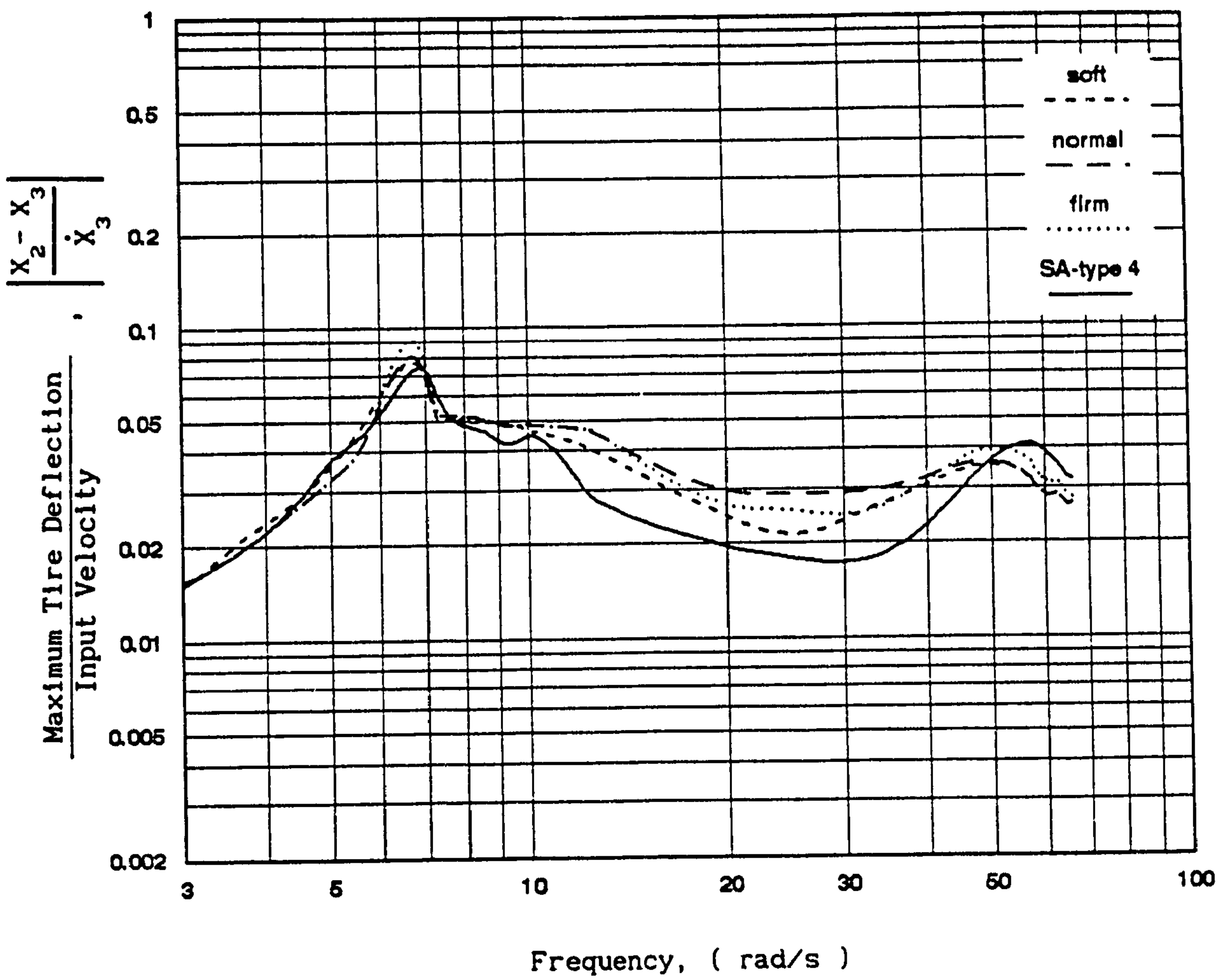


Fig. 4.18 Comparison of maximum tire deflection response for firm, normal, soft, and SA-type 4 suspensions at 25% of the damping capacity

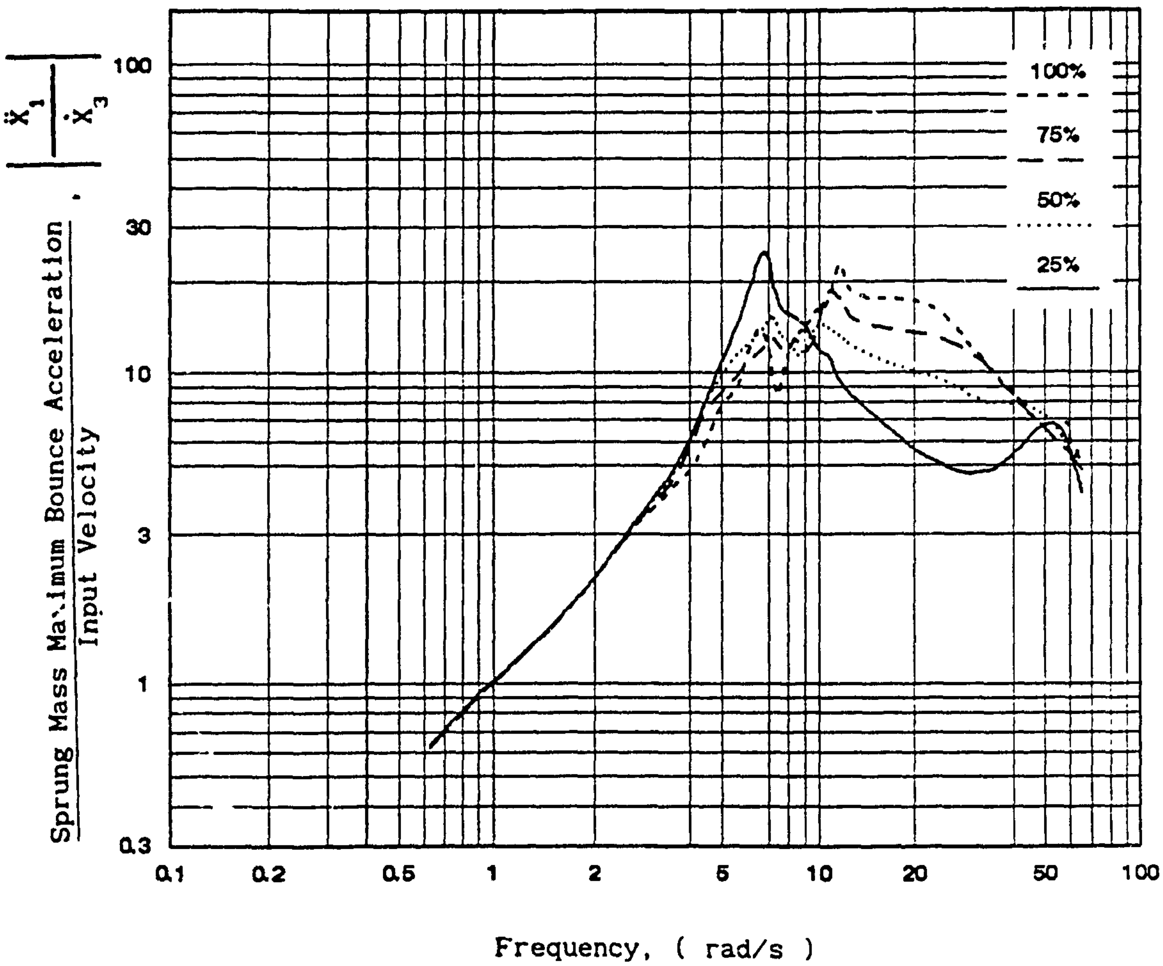


Fig. 4.19 Comparison of maximum bounce acceleration response of SA-type 4 suspension at different damping capacities

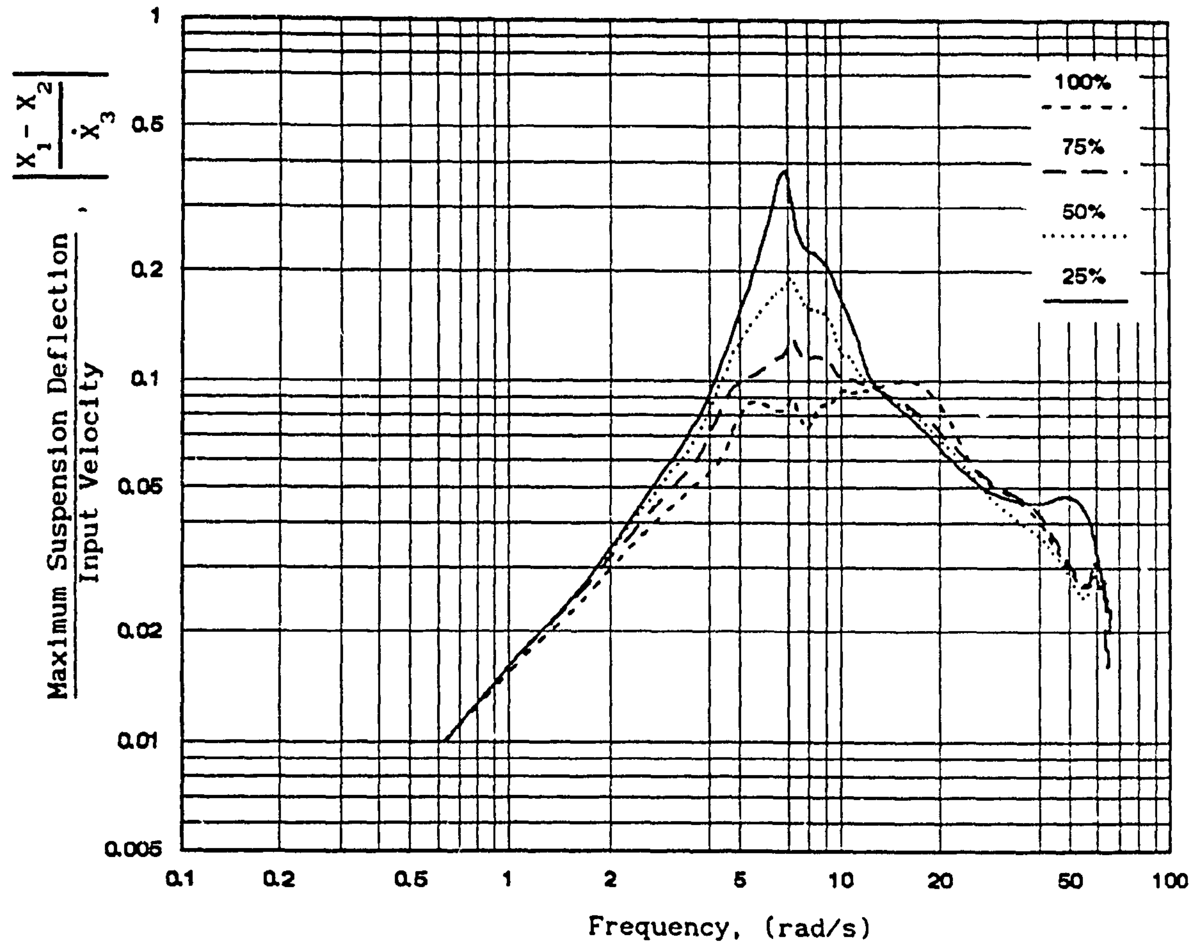


Fig. 4.20 Comparison of maximum suspension deflection response of SA-type 4 suspension at different damping capacities

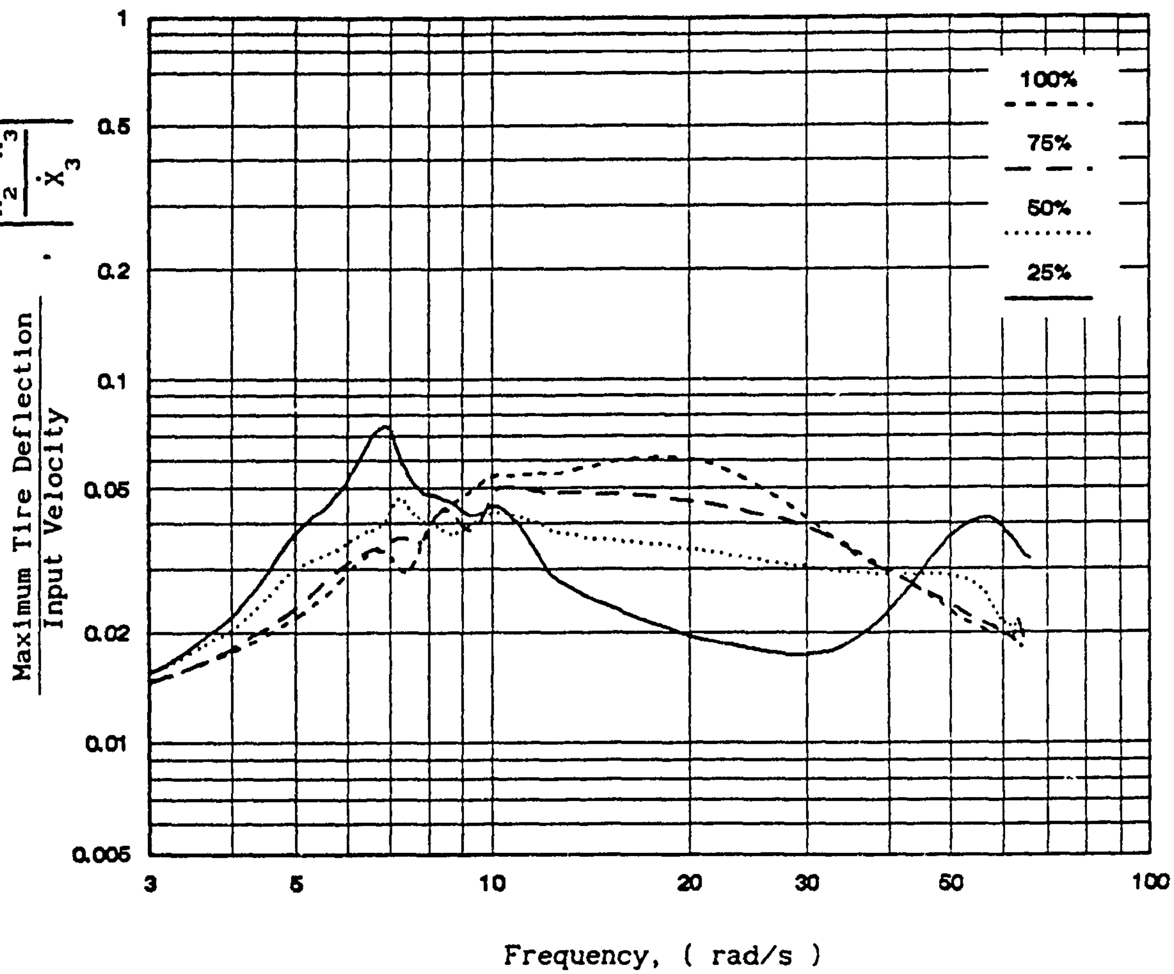


Fig. 4.21 Comparison of maximum tire deflection response of SA-type 4 suspension at different damping capacities

identified only for the SA-type 4 system when the damping is 25% of the full capacity. The tire deflection responses are presented in Figures 4.16, 4.17, and 4.18, for 75%, 50%, and 25% of the full damping capacity. Two distinct peak responses are observed for all the systems only when the damping is at 25% of the full capacity.

To summarise the results, on the performance of SA-type 4 systems due to variations in damping values, Figures 4.19, 4.20, 4.21, are presented. It is observed that sprung mass acceleration, suspension deflection and tire deflection responses for the frequency range provide a good trade-off throughout for the damping value of 50% of the full capacity.

4.4. Experimental Study of SA-type 4 Suspension

In this section, experimental results on SA-type 4 system are presented and compared with analytical results. Since during testing, the responses below the frequency of 3 rad/s were observed to be noisy, the results are presented only for frequencies above 3 rad/s.

The maximum bounce acceleration response for SA-type 4 system obtained from experiment is presented in Fig 4.22 and compared with firm, normal, and soft type passive suspensions. Though the responses are obtained at discrete frequencies they are presented here as continuous plots for the sake of easy comparison. It can be concluded that the system is highly damped. Both peak resonance responses corresponding to the two natural frequencies are replaced by a single peak around the frequency characterized by the oscillation of the two

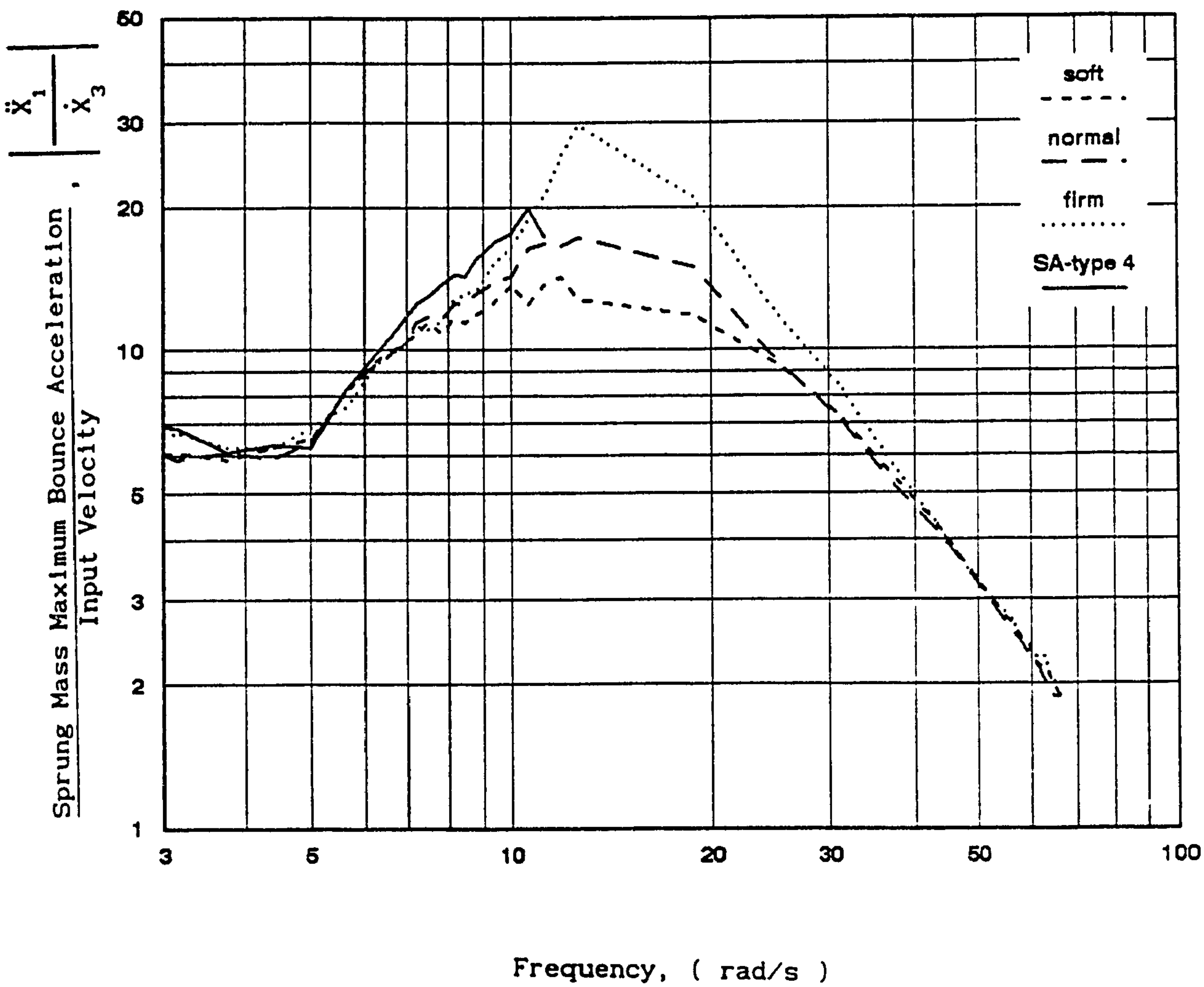


Fig.4.22 Comparison of experimental sprung mass maximum bounce acceleration responses for firm, normal, soft and SA-type 4 suspensions

masses together with respect to the tire. The acceleration response is poor for firm type of suspension and better for soft type suspension. The response of normal type suspension lies in between the firm and soft types of suspensions. The response of SA-type 4 suspension is very similar to soft type suspension at lower frequencies but inferior for frequencies up to 10.7 rad/s (1.7 Hz), and is better at 11.3 rad/s (1.8 Hz). The experimental response of SA-type 4 suspension system could not be obtained beyond this frequency due to the reason outlined below.

The testing could not be proceeded beyond 1.8 Hz frequency due to the limitation of the response time of the motor in the shock absorber to switch between firm and soft modes of damping. The switching is demanded on the basis of the SA-type 4 control logic which is identical to the SA-type 3 control logic whose switching time plots are plotted in chapter 3. However from preliminary testing, it is revealed that during every period of oscillation, the condition function crosses a zero value three times and thus requiring three switching between the firm and soft modes. It is also observed that the time interval between these switchings is 1/3 the period of oscillation. Since, each switching requires 180ms in the adjustable shock absorber, the maximum operating frequency for SA-type 4 suspension is limited to 1.8 Hz. Hence, testing is conducted only up to 1.8 Hz.

Analytical results for the SA-type 4 and firm, normal, and soft type passive suspensions are presented in Figs. 4.7, 4.8, and 4.9. The experimental results presented in Fig. 4.22 may be compared with the analytical results for the purpose of validation. However, before such a comparison is made, one should make sure that the mathematical model in the analytical investigation represents closely the experimental set-up. A detailed examination of the experimental set-up reveals that the dry friction in the linear guide bearings considerably influence the responses measured during testing. The lock-up behavior of the friction in the linear bearing is observed both visually and from the suspension deflection response of the system below a break frequency of 3 rad/s. Because of this physical behavior, the analytical model is approximately modified to incorporate this friction prior to its validation with the experimental results.

During testing, the friction force, F_f, due to the linear guide bearing is estimated as 8.6 N. Mathematically the friction force is represented as,

$$F_f = \left\{ \begin{array}{ll} 8.6 & v > \varepsilon\,, \\ -\,8.6 & v < -\varepsilon, \\ \left(\dfrac{8.6}{\varepsilon} \right) (\dot{X}_1) & -\,\varepsilon \le v \ge \varepsilon \end{array} \right\} \qquad (5.1)$$

where, v = relative velocity and ε = 0.001

In Equation (5.1), the dry friction force in the small region $-\varepsilon$ to $+\varepsilon$ is represented by a linear viscous damper with a very high value for the viscous damping co-efficient in order to overcome any numerical problem during the solution phase.

The introduction of dry friction due to the linear guide bearings will cause both the sprung and unsprung masses to lock-up at low frequencies. Referring to the schematic diagram of the SA-suspension in

Fig 2.2, the lock-up occurs when $\dot{X}_1 = \dot{X}_2$.

At lock-up, $\ddot{X}_1 = \ddot{X}_2$ and the system remains in the lock-up mode until the following condition is satisfied.

$$(m_1 + m_2) \ddot{X}_2 + K_2 (X_2 - X_3) \geq F_f \tag{5.2}$$

where, $\dot{X}_2(0) = 0$; $X_2(0) = 0$; $X_3 = A \sin\omega t$

After the lock-up, the equations of motion characterizing the behavior of the system can be written by modifying the equations 2.7 and 2.8 as :

$$m_1\ddot{X}_1 + \omega_1^2(X_1 - X_2) + F_d + \frac{F_f}{2} (\mathrm{sgn}(\dot{X}_1)) = 0 \tag{5.3}$$

$$m_2\ddot{X}_2 + \omega_2^2(X_2 - X_3) - F_d + (K_1/ m_2)(X_2 - X_1)+ \frac{F_f}{2} (\mathrm{sgn}(\dot{X}_2)) = 0 \tag{5.4}$$

where $\omega_1^2 = K_1 / m_1$,$\omega_2^2 = K_2 / m_2$, F_f is the friction force and F_d is the damping force.

In the case of SA-type 4 suspension, F_d is given by,

$$F_d = \left\{ \begin{array}{ll} 2 \zeta_{on} \omega_1 (\dot{X}_1 - \dot{X}_2) m_1 , & (X_1 - X_2) \quad (\dot{X}_1 - \dot{X}_2) < 0 \\ 2 \zeta_{off} \omega_1 (\dot{X}_1 - \dot{X}_2) m_1 , & (X_1 - X_2) \quad (\dot{X}_1 - \dot{X}_2) > 0 \end{array} \right\}$$

where, ζ_{on}, and ζ_{off} are the damping values for the "on" and "off" stages of the SA-scheme.

The initial conditions in Equations (5.3) and (5.4) are obtained from the solution of Eqn (5.2) at the instant of breaking the lock-up. Also the initial values meet the condition, $X_1 = X_2$; $\dot{X}_1 = \dot{X}_2$;

Using the modified analytical suspension model the acceleration response for SA-type 4, firm, normal and soft type suspensions are evaluated and presented in Fig 4.23. The responses are similar in trend to those from the experimental results. The responses of firm and normal type suspensions are inferior than soft type suspension. The sprung mass acceleration response of SA-type 4 suspension is inferior at the lower frequencies below 3 rad/s and has a characteristic similar to the soft suspension in the frequencies up to the sprung mass natural frequency. The response of SA-type 4 system, however, is better compared to all other types of suspensions for frequencies beyond the natural frequency. Since the damping value in the shock absorber is high, the responses shown in Fig 4.23, does not have two distinct resonant peaks and the response is very similar to previously presented analytical results in Fig 4.7.

In Figures 4.24, 4.25, and 4.26, the experimental maximum bounce acceleration responses for firm,normal and soft type suspension systems are presented and compared with their respective analytical responses. The frequencies of interest are the sprung mass natural frequency at 8.03 rad/s and the natural frequency (15.48 Hz) corresponding to the combined sprung mass - unsprung mass system.

The analytical responses at lower frequencies correlate well with experimental results. In the higher frequencies analytical responses are little higher in value than the experimental responses. This may be due to the reason that in the analytical model, damping value is based on the peak force measured.

The experimental acceleration response for SA-type 4 suspension is

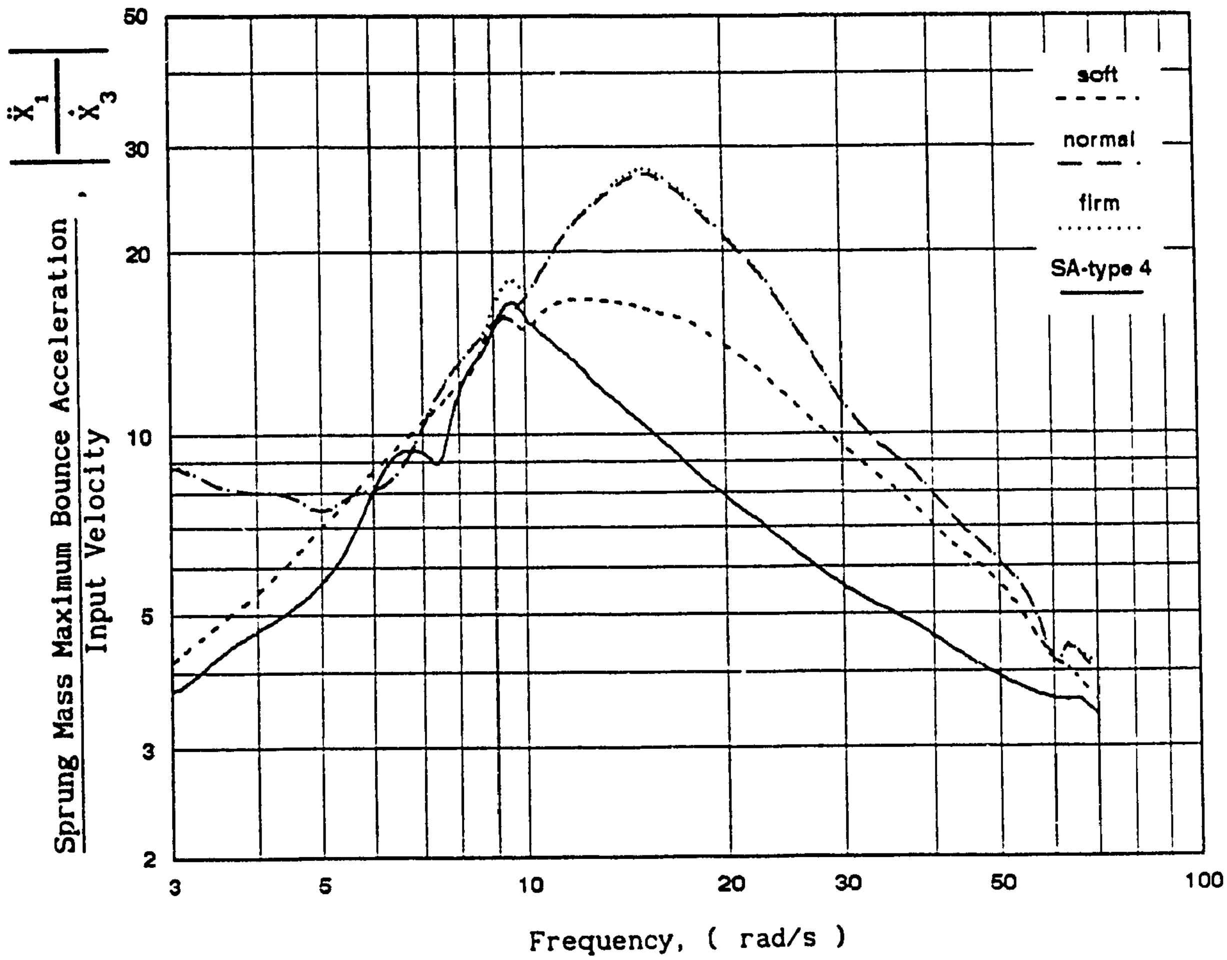


Fig. 4.23 Comparison of sprung mass maximum bounce acceleration response for firm, normal, soft and SA-type 4 suspensions using modified analytical model

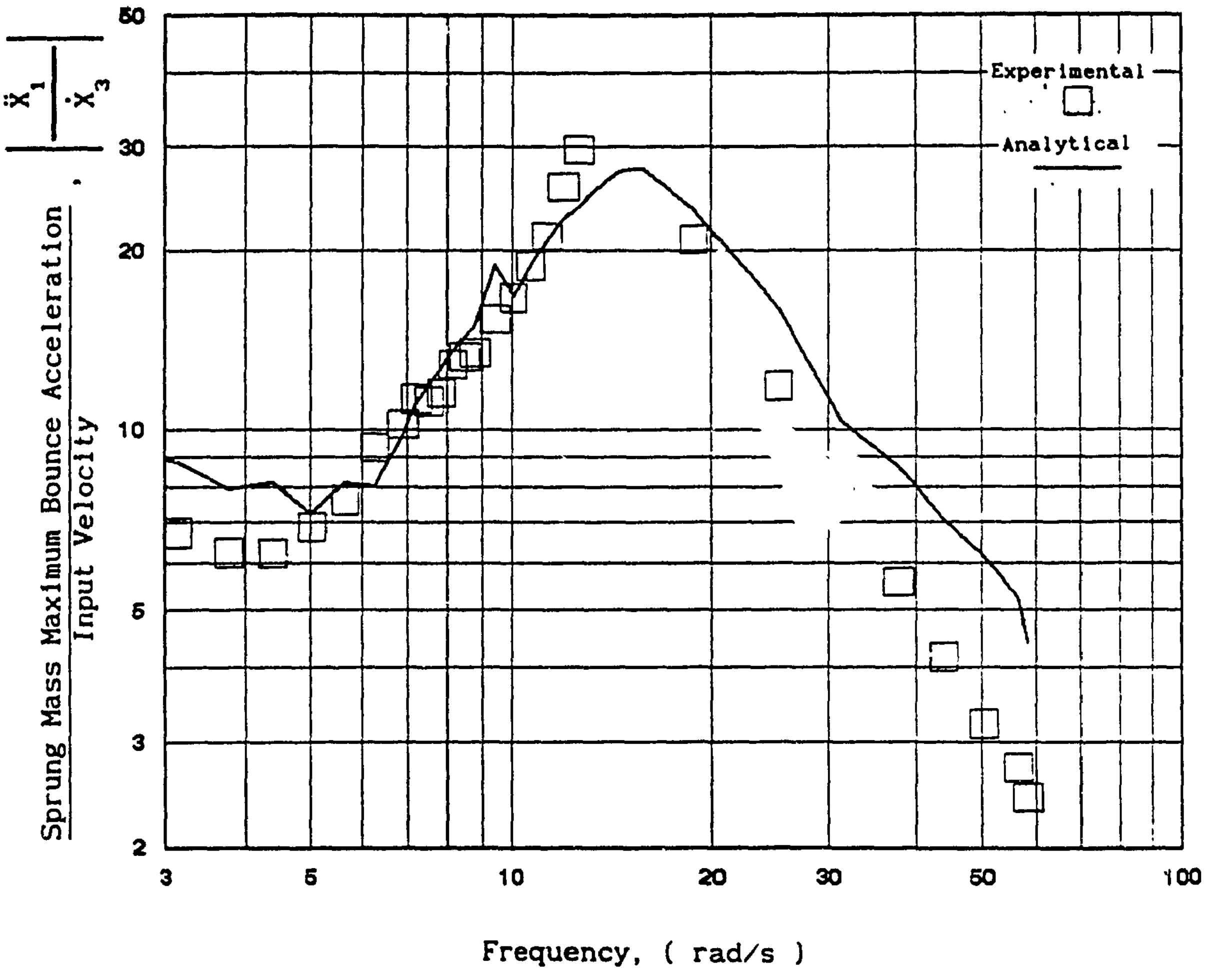


Fig. 4.24 Comparison of sprung mass maximum bounce acceleration response for firm type suspension

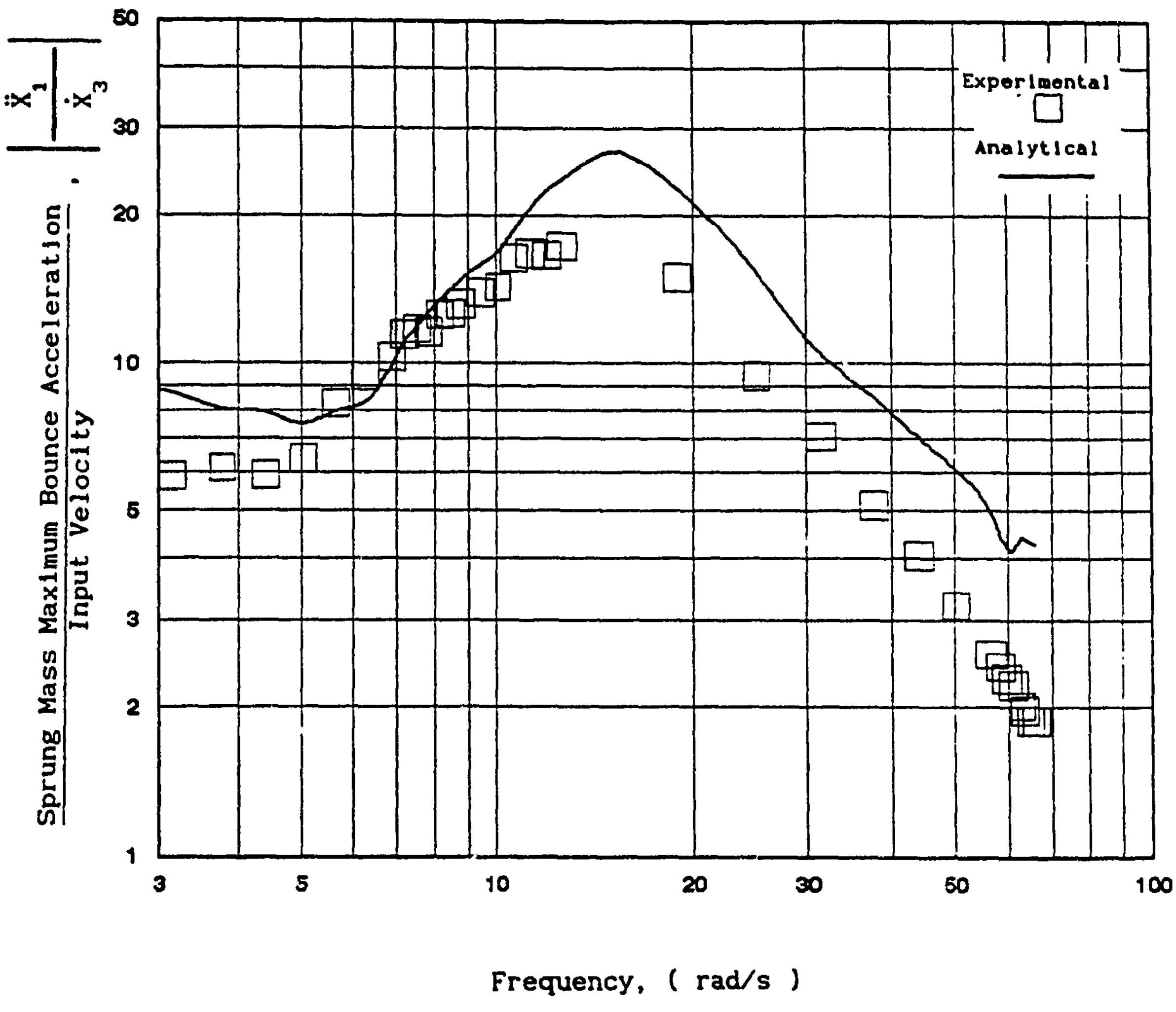


Fig. 4.25 Comparison of sprung mass maximum bounce acceleration response for normal type suspension

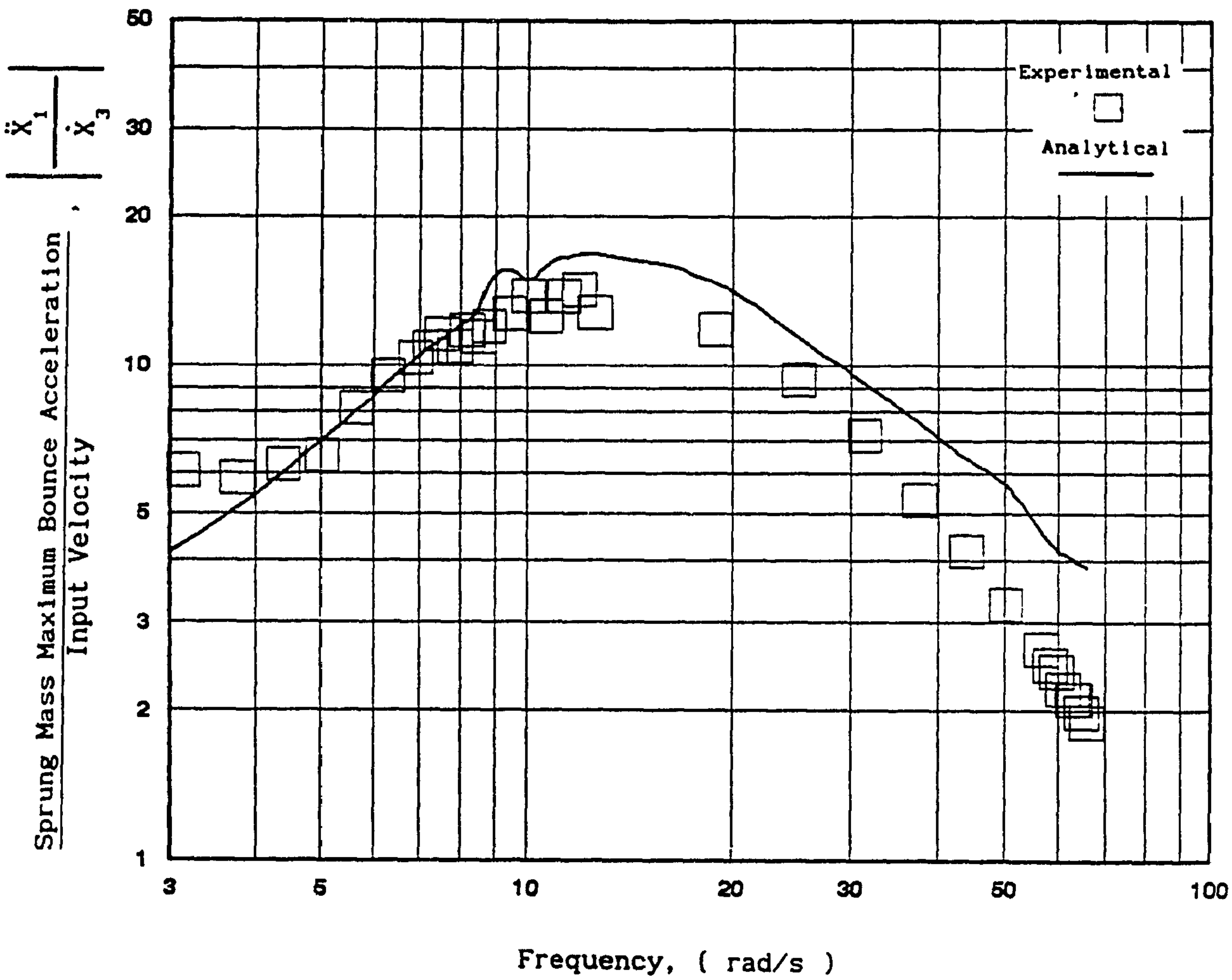


Fig. 4.26 Comparison of sprung mass maximum bounce acceleration response for soft type suspension

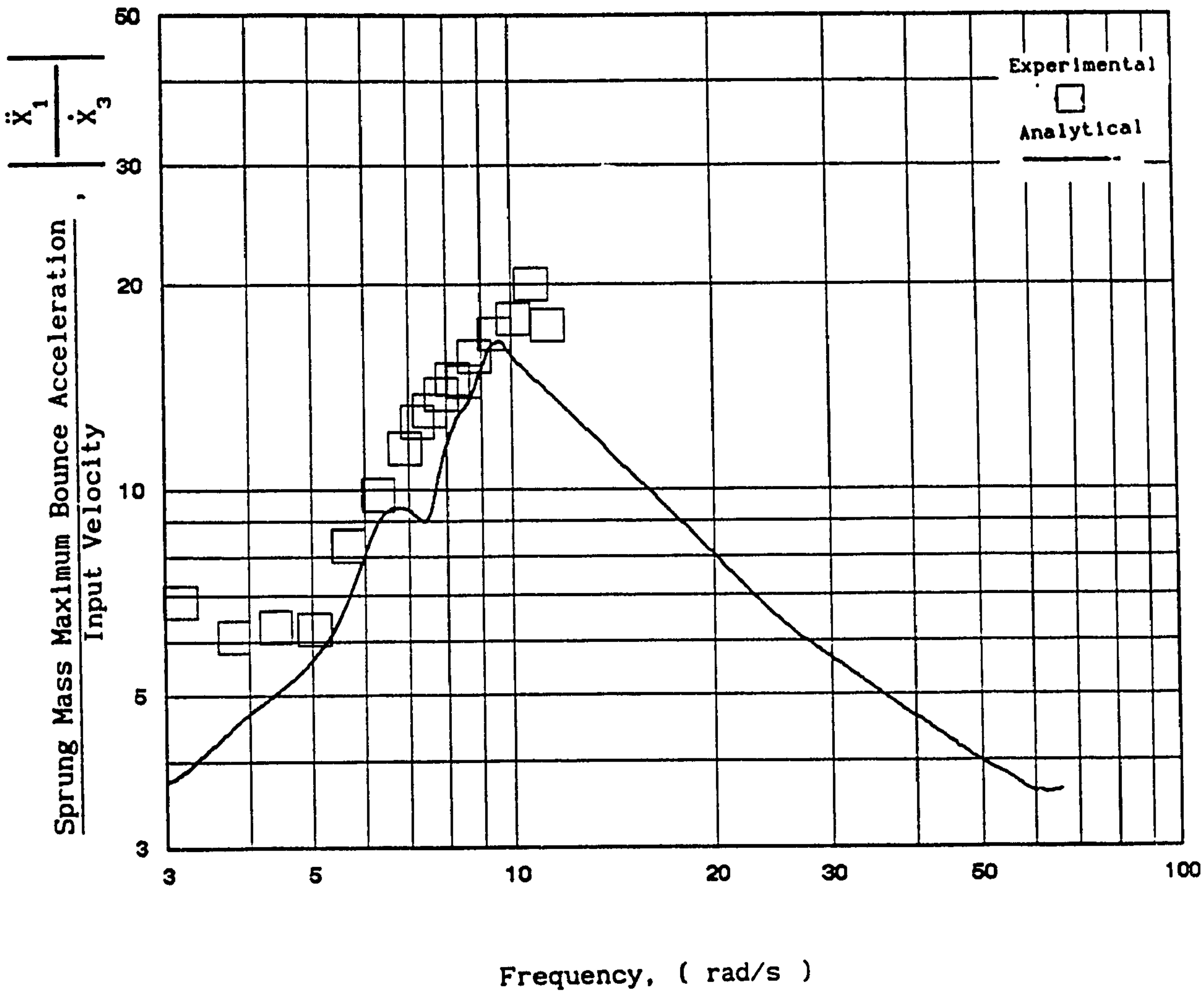


Fig. 4.27 Comparison of sprung mass maximum bounce acceleration response for SA-type 4 suspension

presented in Fig. 4.27, and compared with its analytical result. The analytical result shows reasonable correlation with the experimental results for low frequencies up to 1.8 Hz.

The maximum suspension deflection responses obtained from experimental study are presented in Fig. 4.28, for SA-type 4 and compared with firm, normal, and soft type suspensions. It can be seen that the suspension deflection (rattle space requirement) is more in case of the soft compared to the firm and normal types of suspensions. The suspension deflection response for SA-type 4 suspension is small in the low frequency range up to 5 rad/s and after this frequency the deflection response is similar to that of the soft type suspension. In Fig. 4.29 the suspension deflection responses using the modified analytical model are presented for SA-type 4 system and compared with firm, normal, and soft type suspensions. The SA-type 4 system provides better suspension deflection between frequency of 8-30 rad/s. The responses for all the other types of suspensions in general have similar trend.

In Figs 4.30, 4.31, 4.32, and 4.33 the experimental suspension deflection responses for firm, normal, soft, and SA-type 4 suspensions are compared with their analytical responses respectively. The analytical responses in general have similar behavior as observed in experimental testing and have good correlation at low frequencies.

In Fig. 4.34, the experimental maximum tire deflection response is presented for SA-type 4 and compared with firm, normal, and soft type of suspensions. The tire deflection responses for firm, normal, and soft

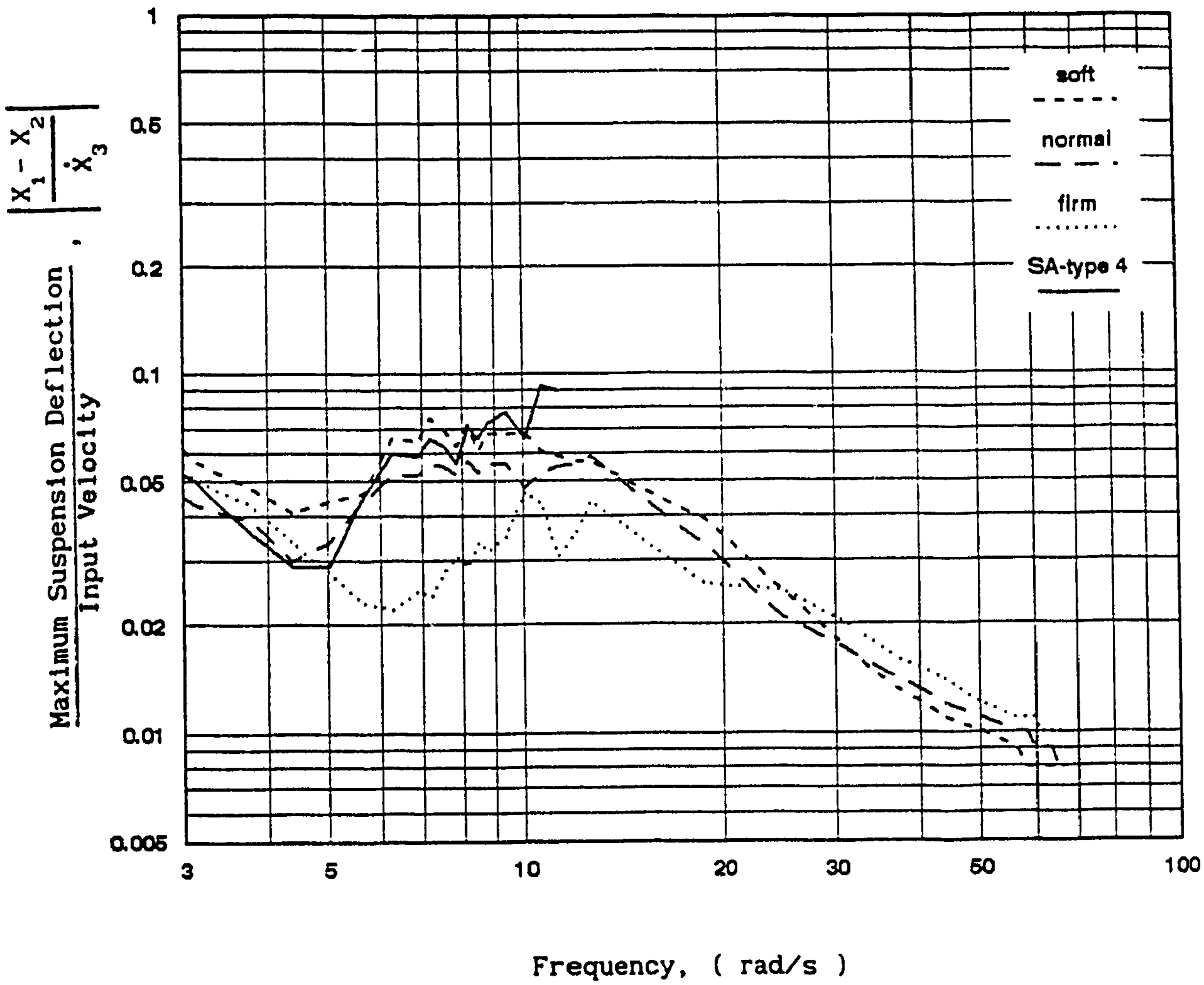


Fig. 4.28 Comparison of experimental maximum suspension deflection responses for firm, normal, soft, and SA-type 4 suspensions

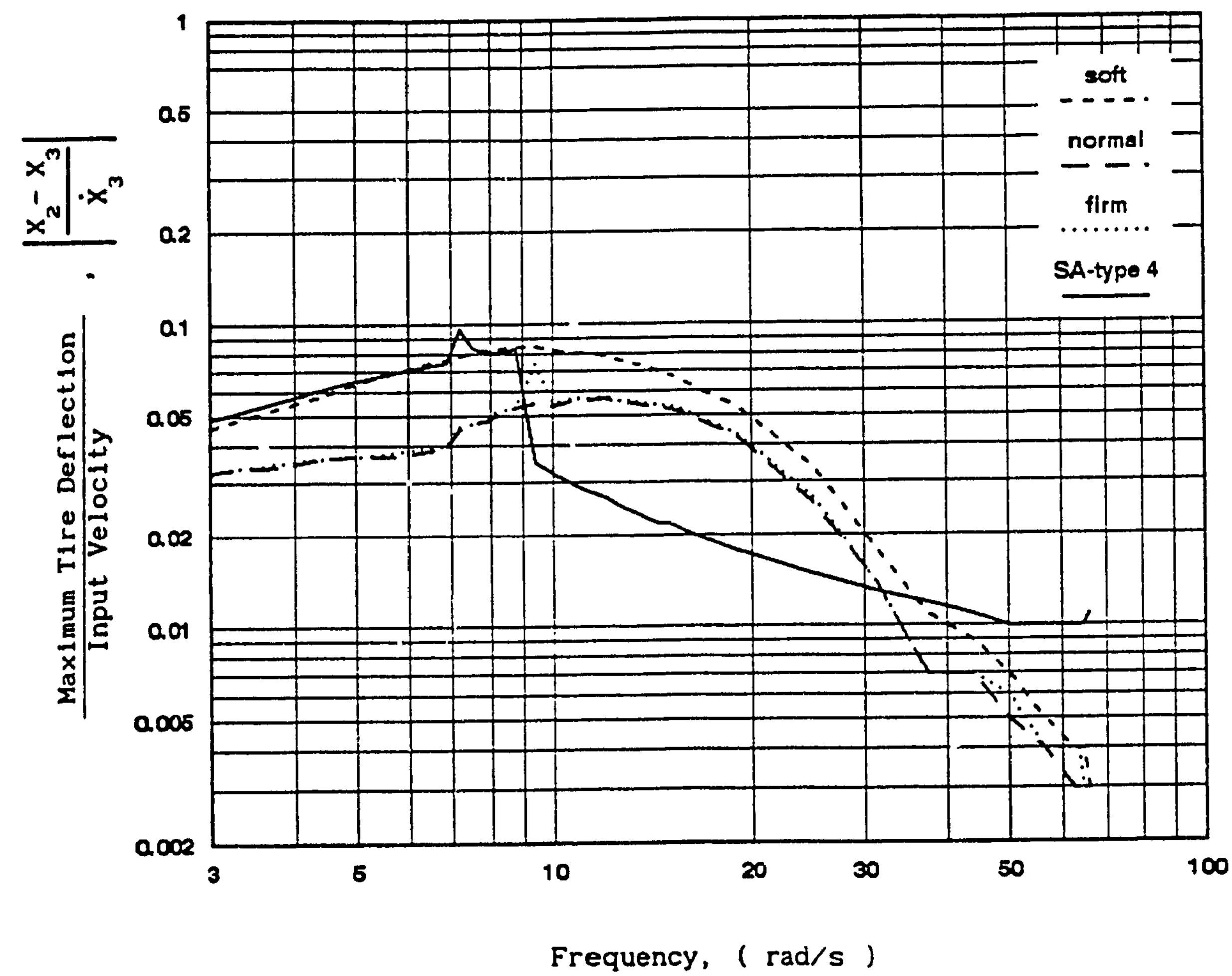


Fig. 4.29 Comparison of maximum suspension deflection response for firm, normal, soft, and SA-type 4 suspensions using modified analytical model

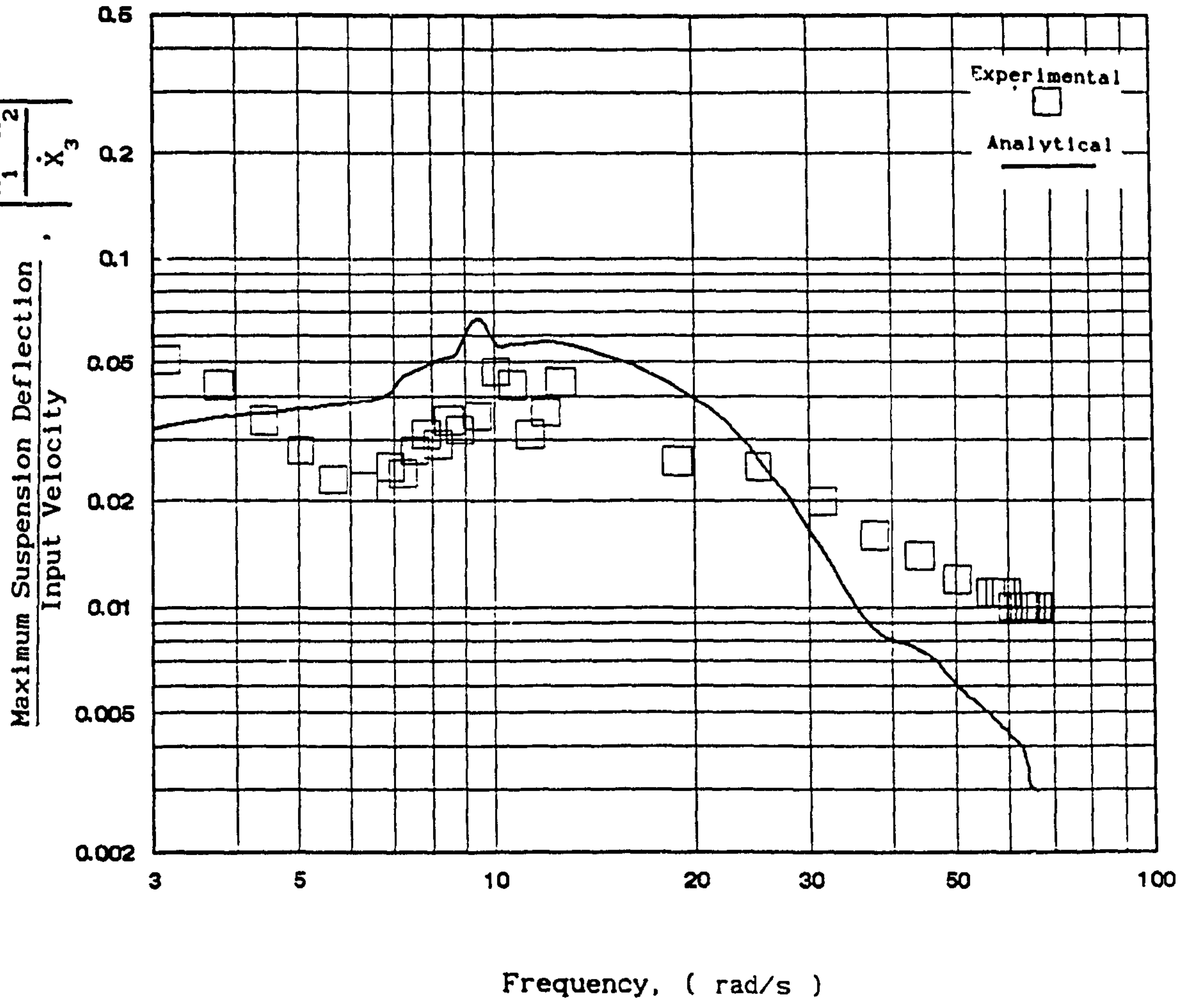


Fig. 4.30 Comparison of maximum suspension deflection responses for firm type suspension

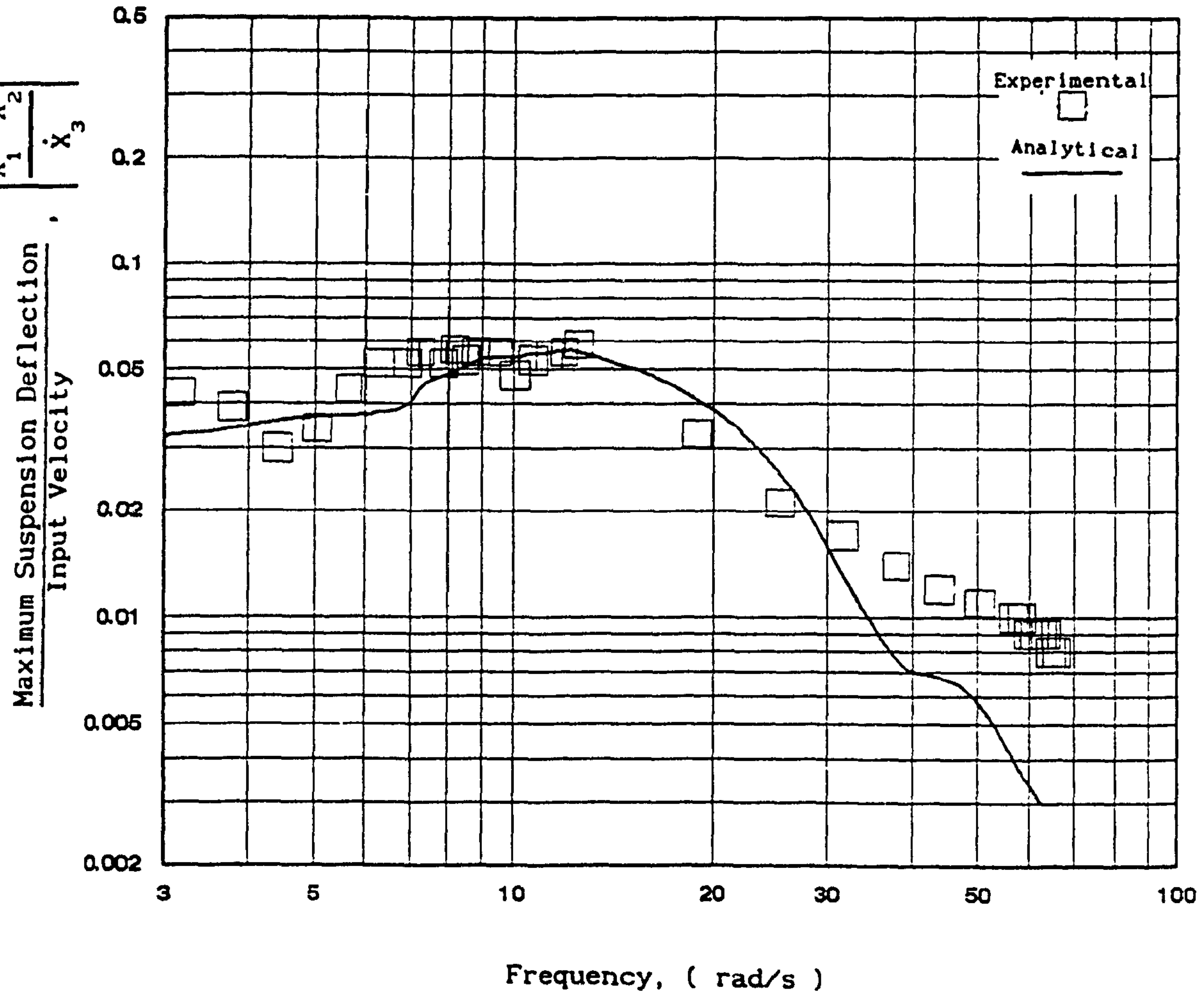


Fig. 4.31 Comparison of maximum suspension deflection response for normal type suspension

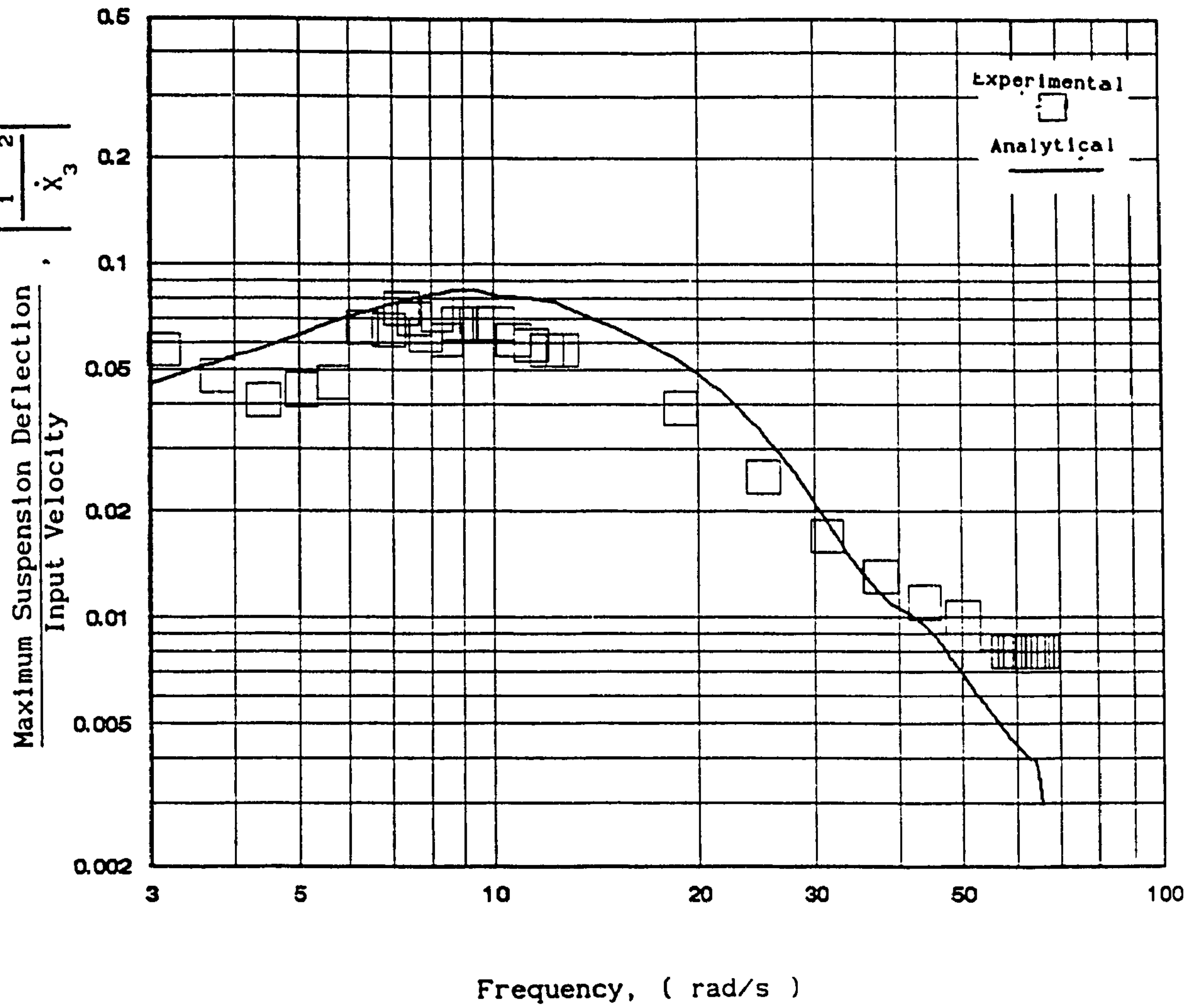


Fig. 4.32 Comparison of maximum suspension deflection response for soft type suspension

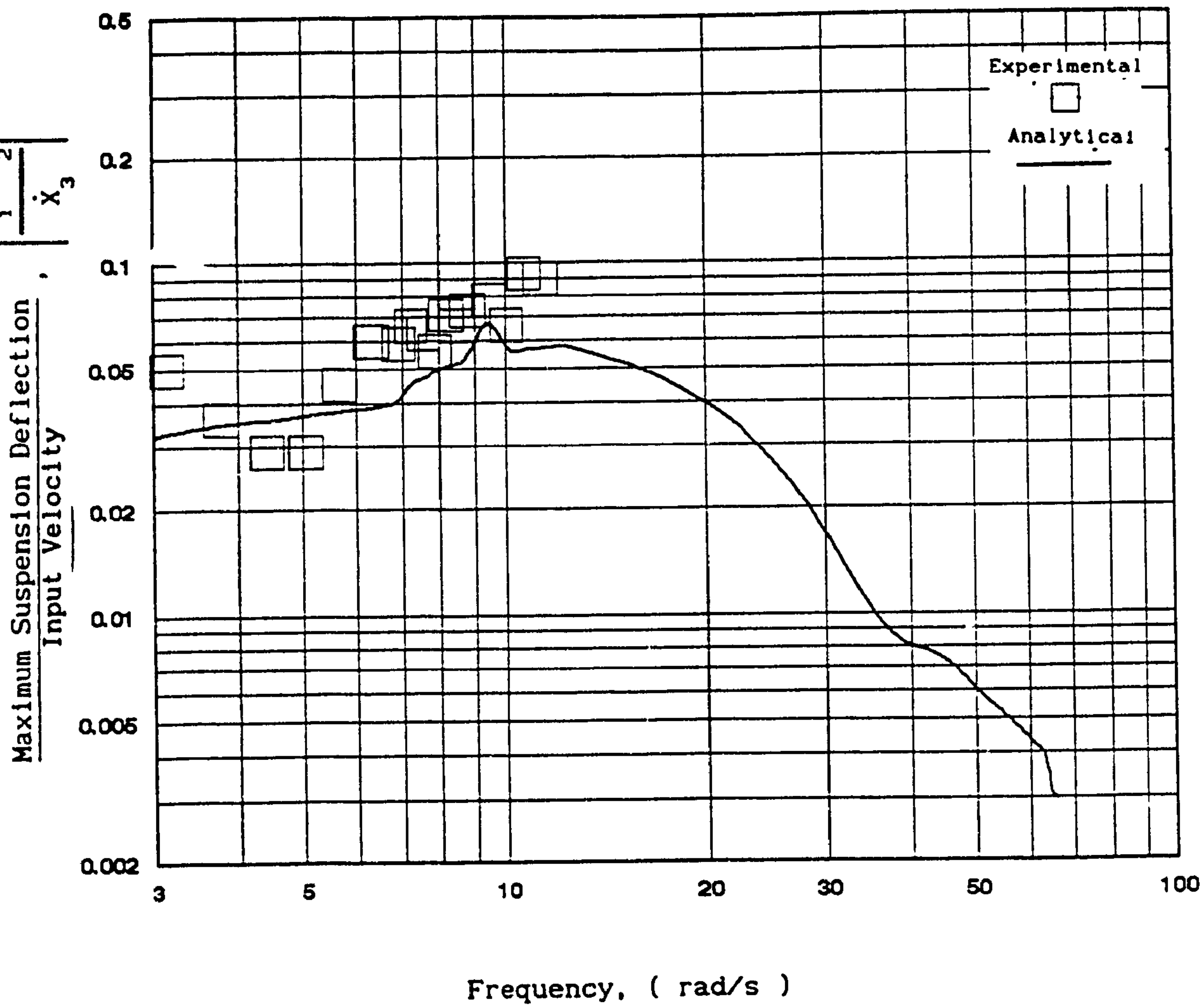


Fig. 4.33 Comparison of maximum suspension deflection response for SA-type 4 suspension

type of suspensions are similar in the low frequencies. However, for frequencies in the range 8-30 rad/s, the response for firm type is higher than normal type which in turn is higher than soft type suspension. Beyond 30 rad/s, all three types have similar characteristics. The response of the SA-type 4 suspension follows closely the firm type suspension in the frequency range 5-12 rad/s.

The tire deflection responses of SA-type 4, firm, normal, and soft type of suspensions using the modified analytical model are evaluated and the results are presented in Fig.4.35. The responses of firm and normal types of suspensions are similar. The SA-type 4 suspension response is lower compared to other types of suspensions except for a small frequency band around 9.5 rad/s.

In Figs 4.36, 4.37, 4.38,and 4.39, the experimental responses of firm, normal, soft, and SA-type 4 suspensions are presented and compared with their analytical responses respectively. The analytical results are in general similar to experimental results. However, the analytically predicted results seem to be higher compared to experimental results throughout the frequency range of interest. The results presented in Fig 4.39 show that the analytically obtained response for SA-type 4 suspension correlates well with the experimental result for all the frequencies where the testing was conducted.

4.5 Simulation of a Typical Vehicle with SA-type 4 suspension

In sections 4.3, and 4.4, a detailed study on the analytical response of various types of suspensions are presented and compared with

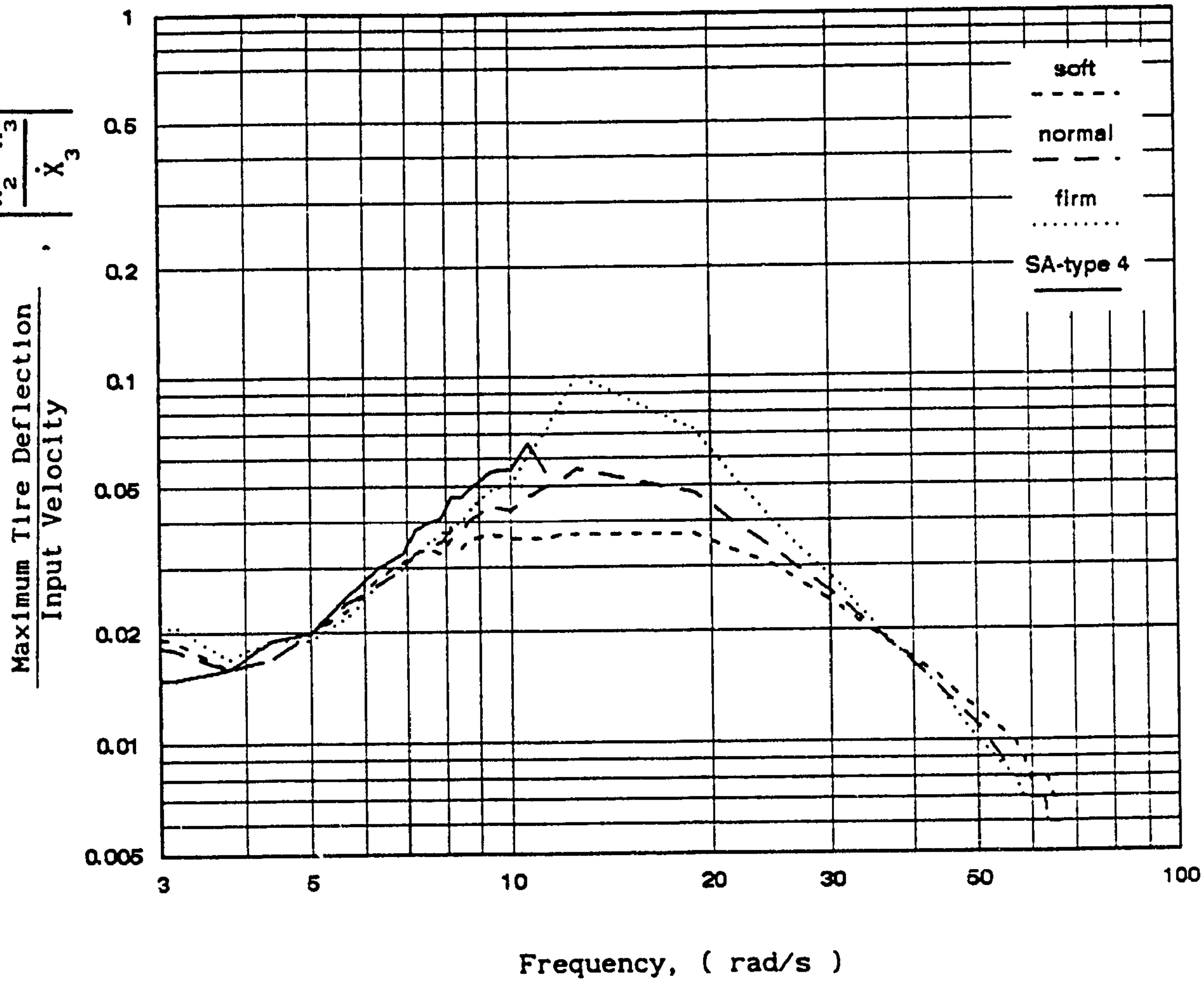


Fig. 4.34 Comparison of experimental maximum tire deflection responses for firm, normal, soft, and SA-type 4 suspensions

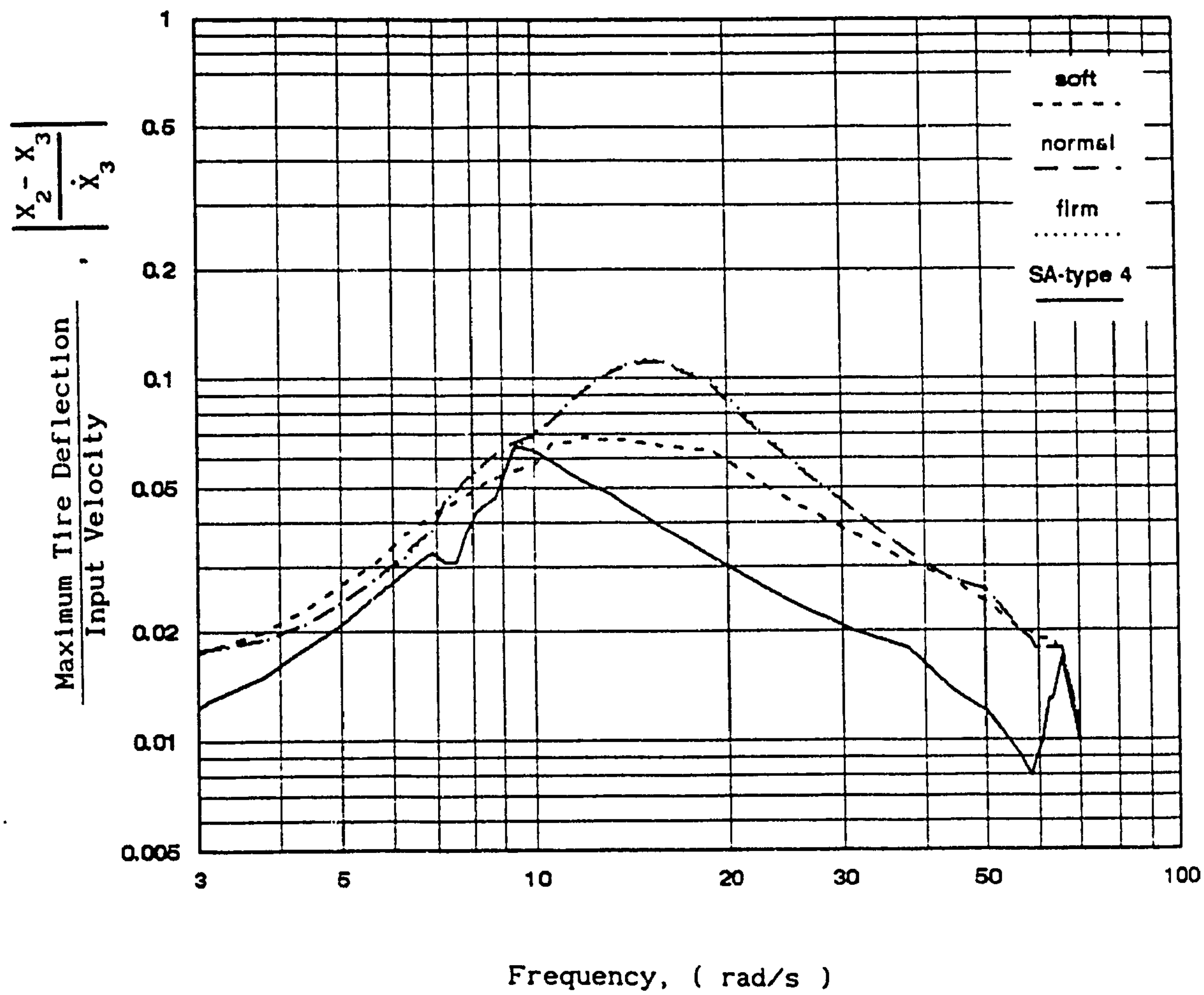


Fig. 4.35 Comparison of maximum tire deflection response for firm, normal, soft, and SA-type 4 suspensions using modified analytical model

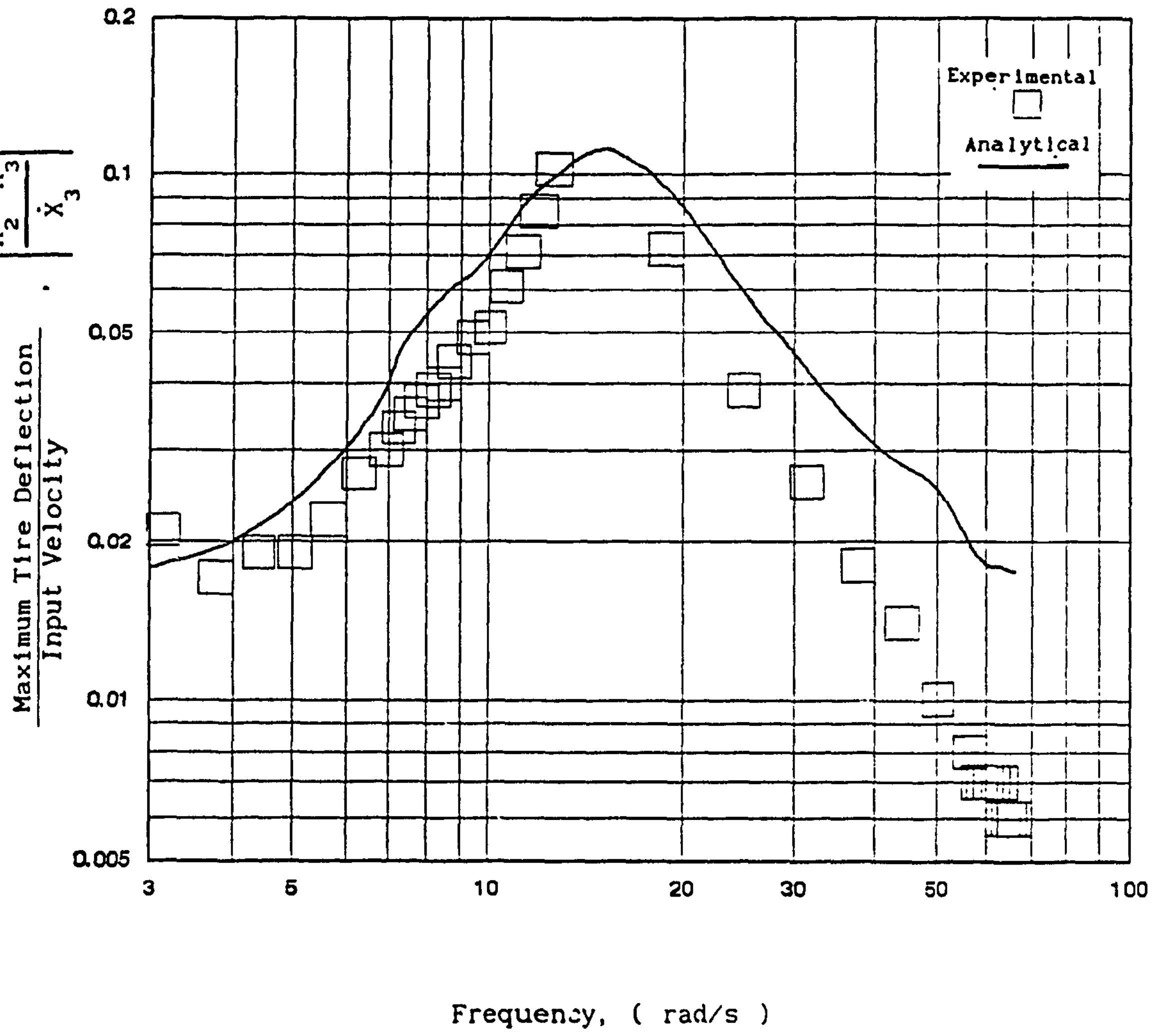


Fig. 4.36 Comparison of maximum tire deflection response for firm type suspension

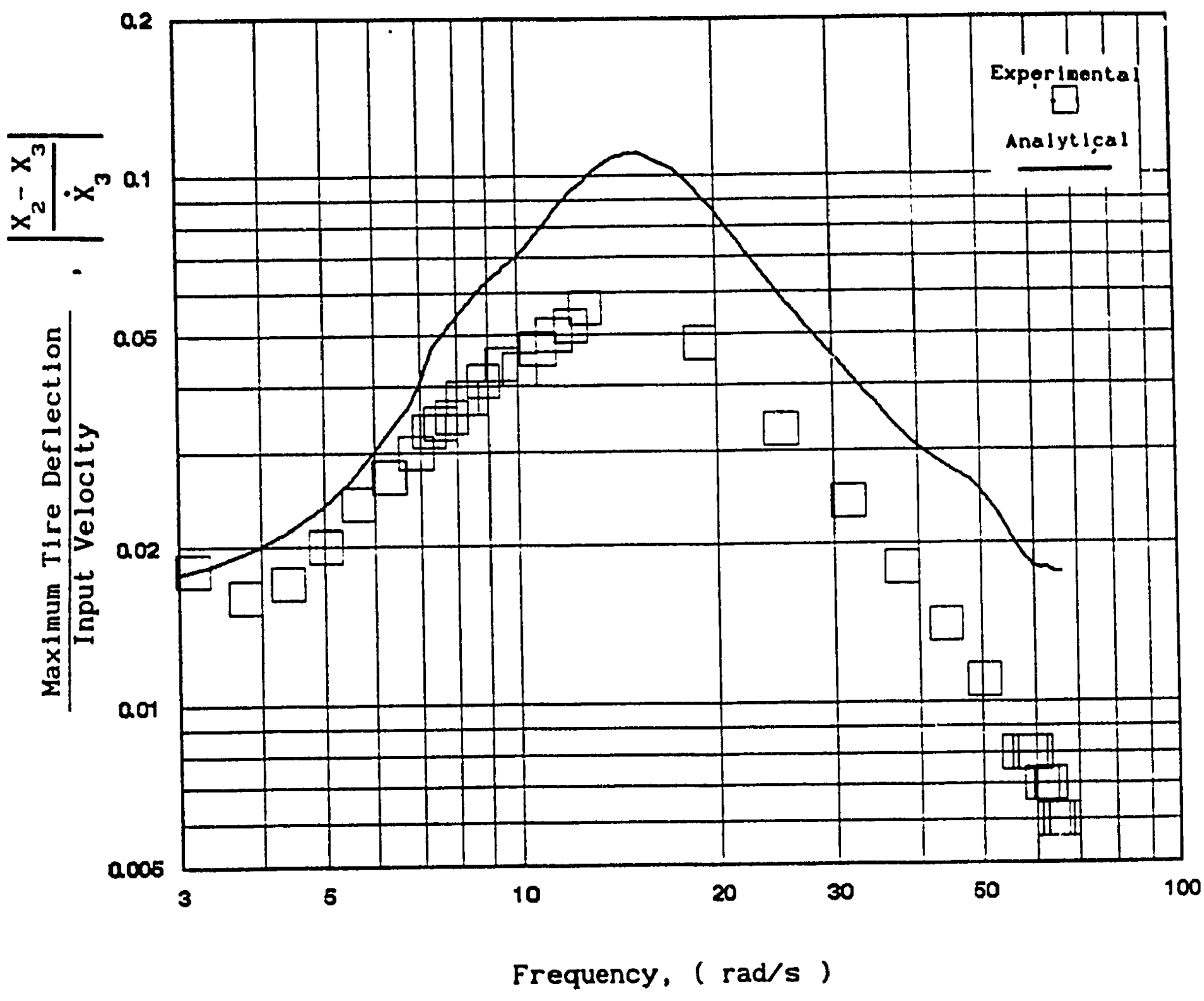


Fig. 4.37 Comparison of maximum tire deflection response for normal type suspension

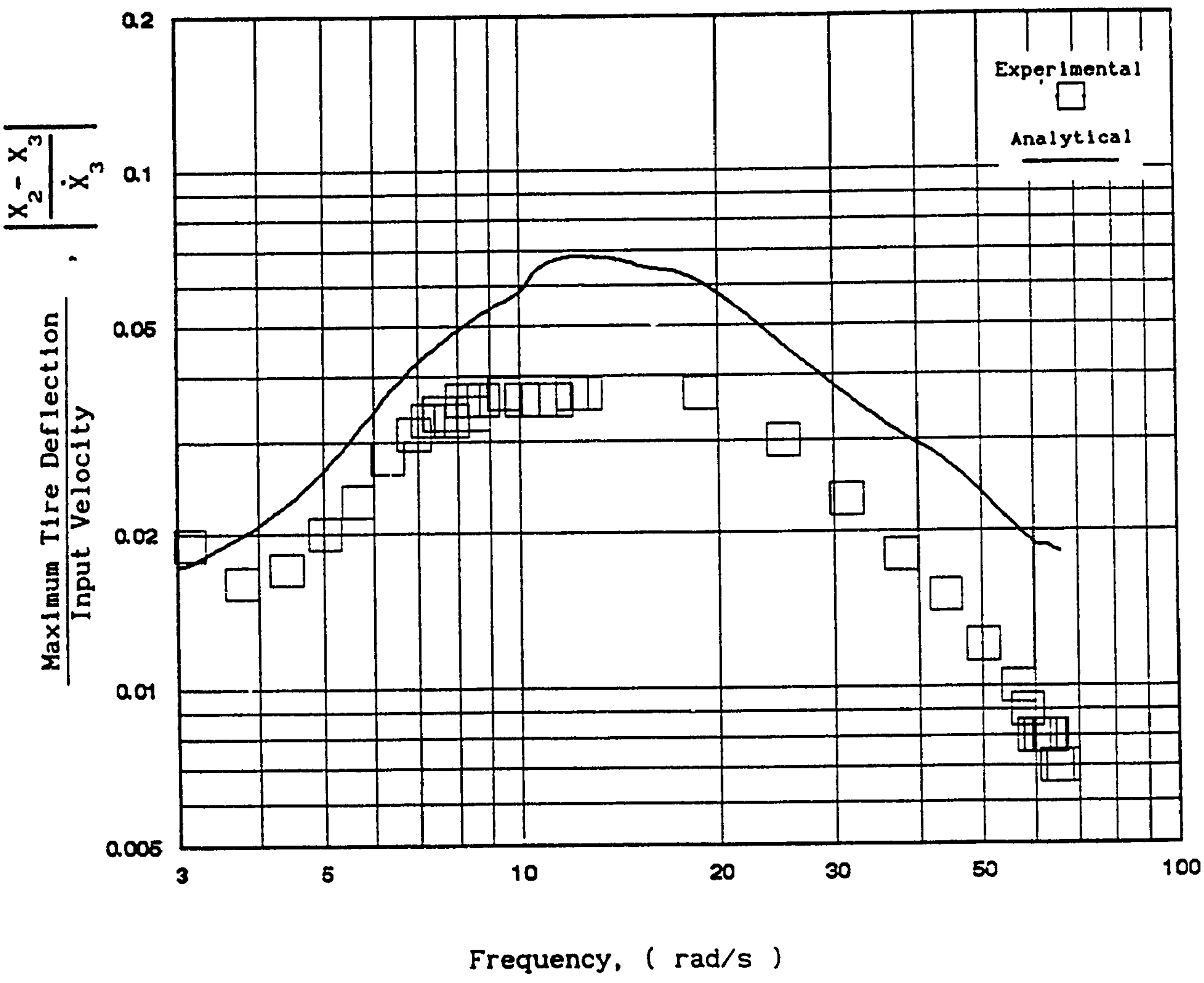


Fig. 4.38 Comparison of maximum tire deflection response for soft type suspension

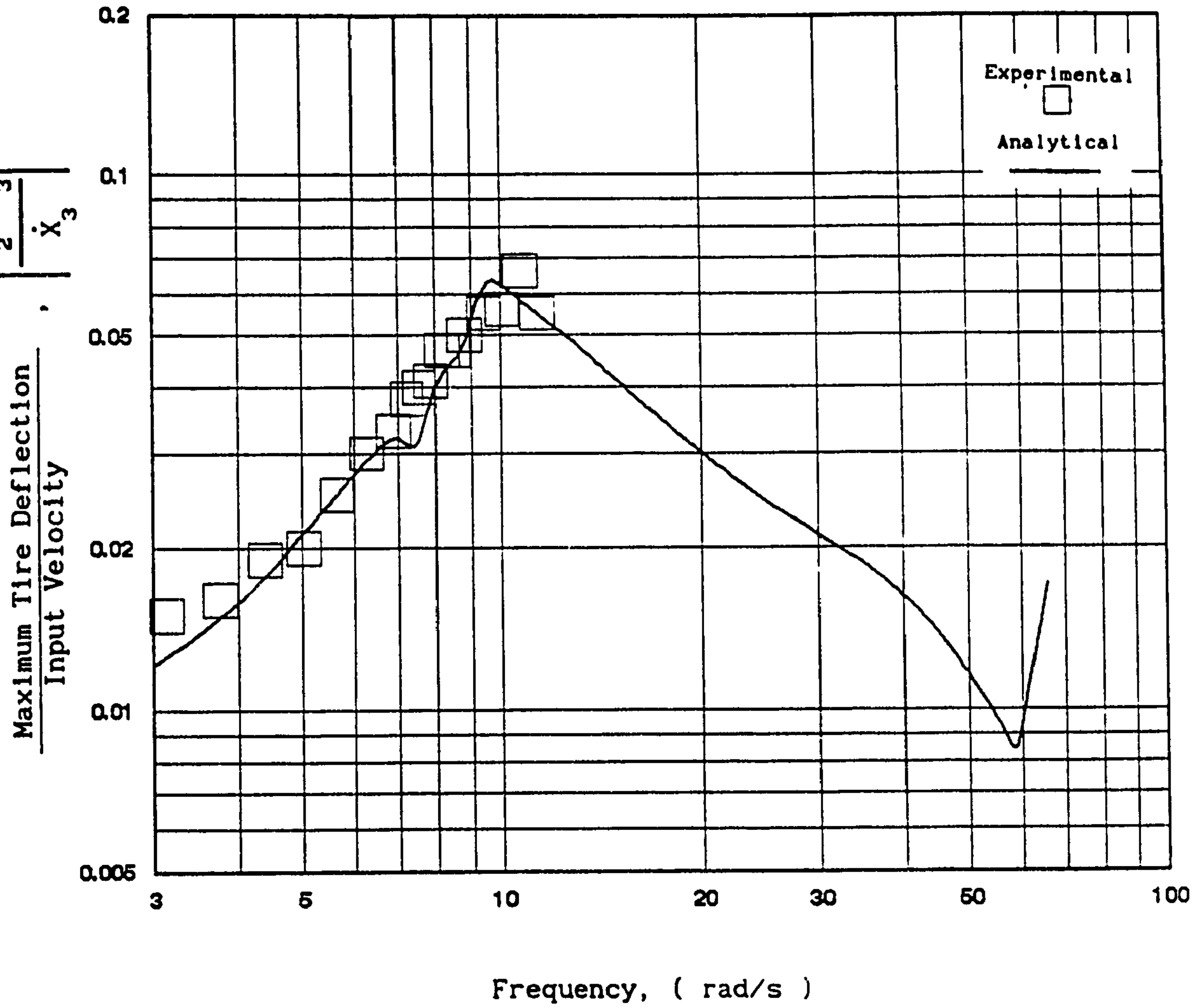


Fig. 4.39 Comparison of maximum tire deflection response for SA-type 4 suspension

experimental results. In order to evaluate the response behavior of a typical road vehicle with different types of suspensions, simulation is carried out for a typical vehicle using a quarter-car model and the results are presented. The parameters used in this study are: M_1 = 264. kg ; M_2 = 44. kg ; K_1 = 24500. N/m ; K_2 = 175000. N/m ; The damping values for the simulation are selected corresponding to the experimentally estimated values of firm, normal, and soft types described in previous sections.

The bounce acceleration response for the SA-type 4 suspension is presented in Fig 4.40 and compared with passive suspensions in the firm, normal, and soft modes. The results show that the SA-type 4 has an optimal response with minimum acceleration responses both at the sprung and unsprung mass frequencies. In Fig 4.41, the maximum suspension deflection response of SA-type 4 suspension is plotted and compared with the other three suspension types. The results indicate that the response of SA-type 4 lies between soft and other two suspension types around the sprung mass natural frequency. However, its response is higher than the passive systems in the high frequency range.

The maximum tire deflection response of SA-type 4 suspension is presented in Fig 4.42 along with the responses for passive suspension with soft, normal, and firm modes. The SA-type 4 suspension response is lower than passive suspensions for all frequencies greater than the sprung mass natural frequency. However, for frequencies lower than the sprung mass natural frequency, the SA-type 4 response is considerably higher than the others.

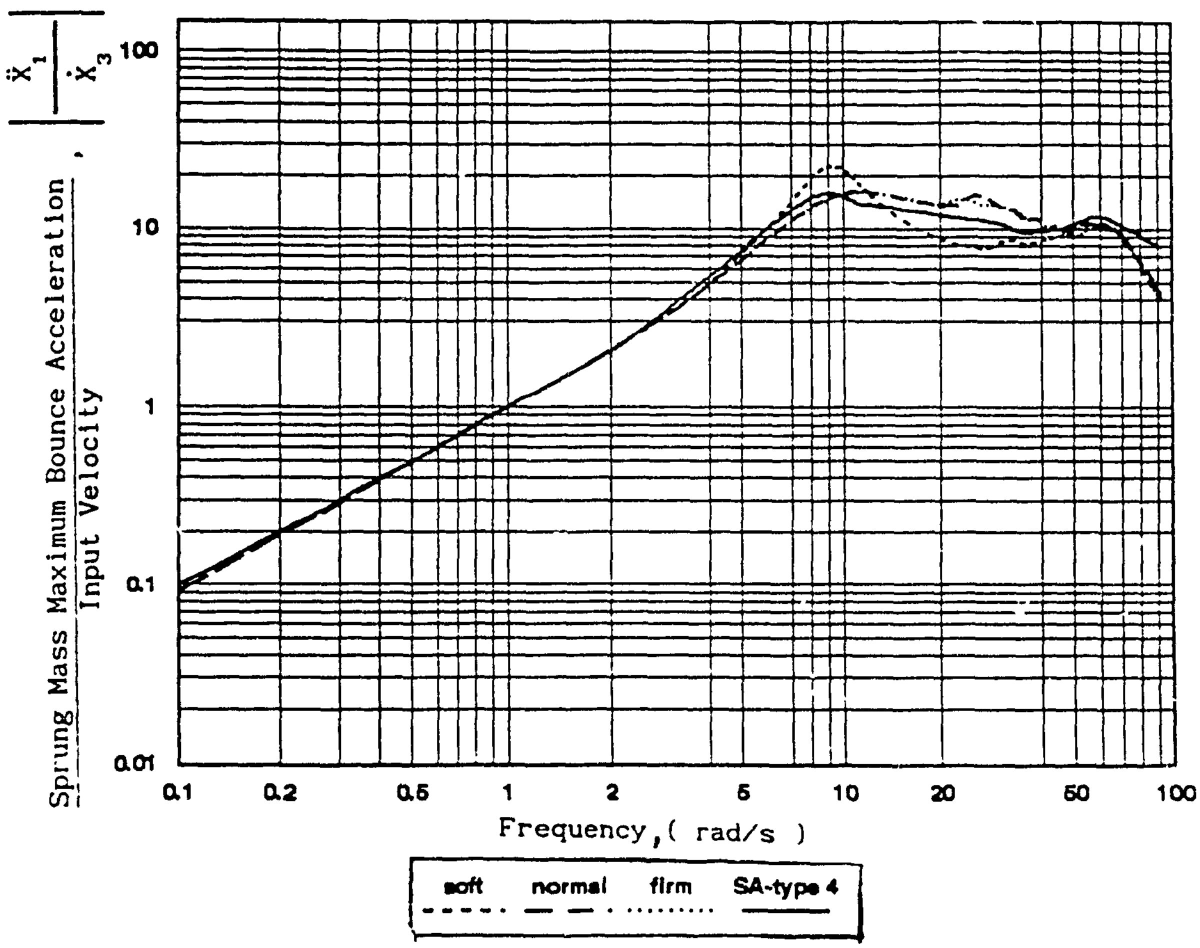


Fig. 4.40 Comparison of maximum bounce acceleration response for SA-type 4 suspension with firm, normal, and soft type suspensions using a quarter-car model of a typical road vehicle

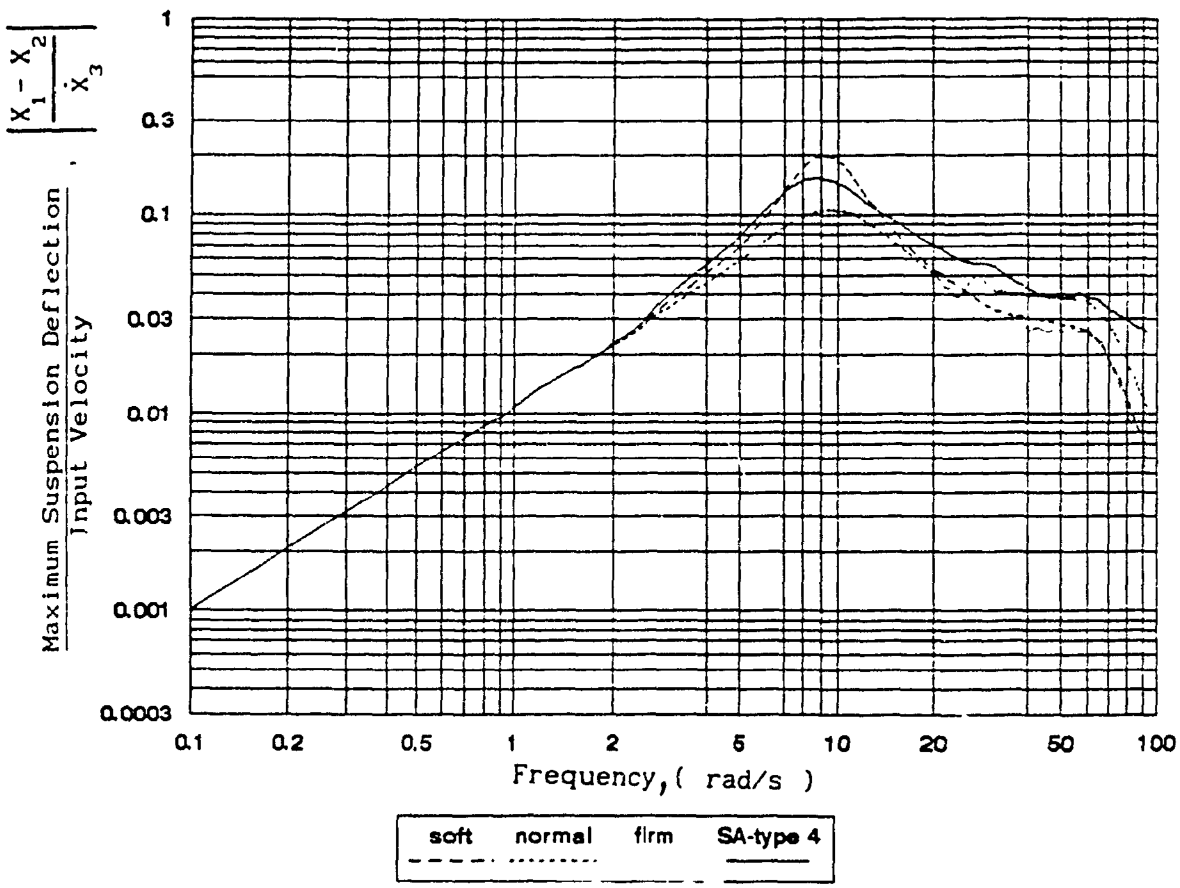


Fig. 4.41 Comparison of maximum suspension deflection response for SA-type 4 suspension with firm, normal, and soft type suspensions using a quarter-car model of a typical road vehicle

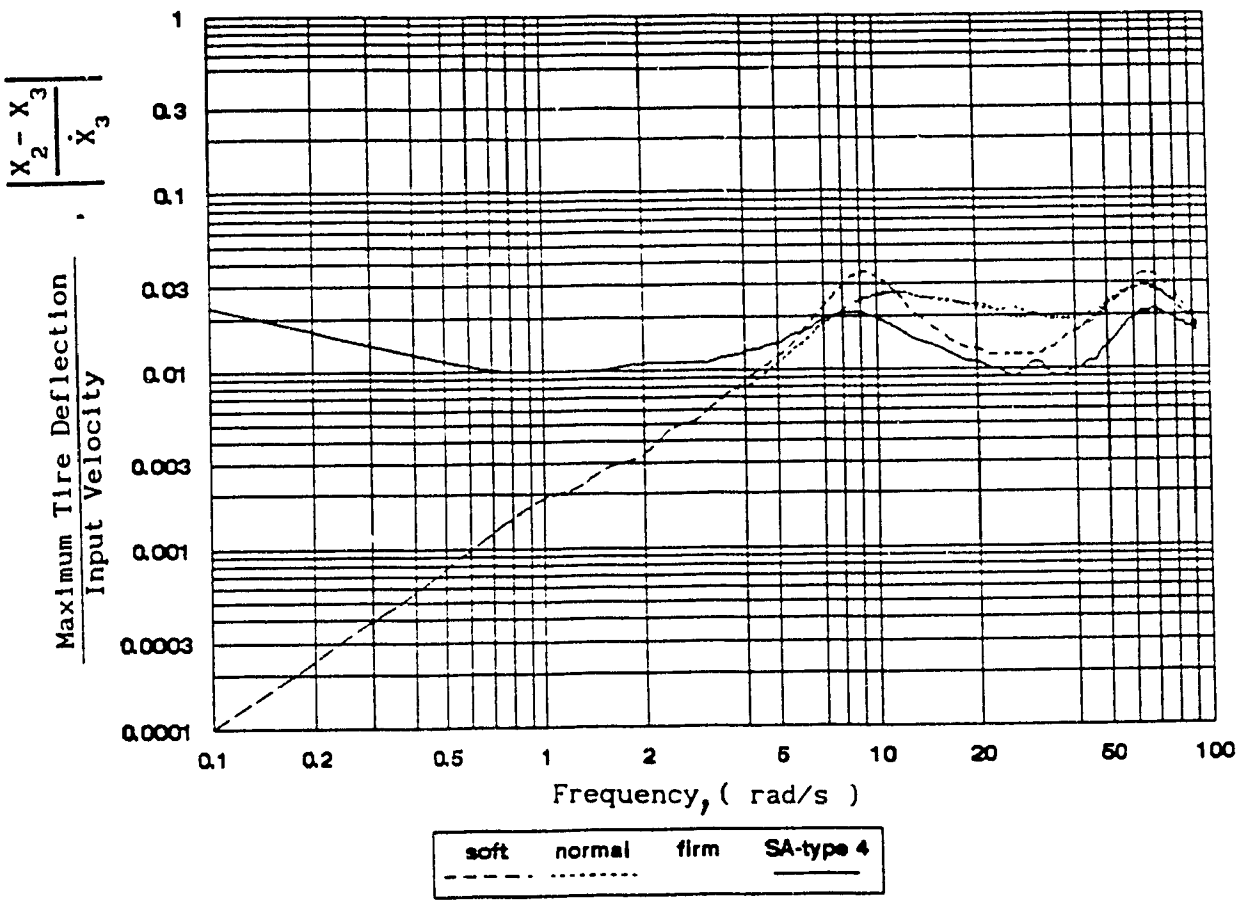


Fig. 4.42 Comparison of maximum tire deflections response for SA-type 4 suspension with firm, normal, and soft type suspensions using a quarter-car model of a typical road vehicle

4.6.Summary

Analytical and experimental studies are conducted on an SA-type 4 suspension system. For this purpose a quarter car model of the suspension system is built and experimental testing is conducted. An adjustable shock absorber used in commercial vehicle, Nissan 300ZX, is selected as the damper in the experimental set-up. A micro-processor control is conceived to automatically switch the shock absorber between firm and soft modes. Preliminary testing are conducted to assess damping characteristics of the shock absorber.

From the analytical study it can be seen that the system is highly damped. Both peak resonance responses corresponding to the two natural frequencies are replaced by a single peak around the frequency characterized by the oscillation of the two masses together with respect to the tire. The response plots for the firm, normal, soft, and SA-type 4 suspensions show that the system behaves like a 2 d-o-f system only when the damping is at 25% and 50 % of the full capacity.

The experimental results of each type of suspensions are compared with their analytical responses. The experimental acceleration response is poor for firm type of suspension and better for soft type suspension. The response of normal type suspension lies in between the firm and soft types of suspensions. The response of SA-type 4 suspension is very similar to the soft type suspension at lower frequencies but inferior for frequencies up to 10.7 rad/s (1.7 Hz), and is better at 11.3 rad/s (1.8 Hz).

It can also be seen that the experimental suspension deflection (rattle space requirement) is more in case of the soft suspension, compared to the firm and normal types of suspensions. The suspension deflection response for SA-type 4 suspension is small in the low frequency range up to 5 rad/s and after this frequency, the deflection response is similar to that of the soft type suspension. The analytical responses in general have similar behavior as observed in experimental testing and have good correlation at low frequencies.

The experimentally obtained maximum tire deflection response for firm, normal, and soft type of suspensions are similar in the low frequencies and for frequencies in the range 8-30 rad/s, the response for firm type is higher than normal type which in turn is higher than soft type suspension. Beyond 30 rad/s, all three types have similar characteristics. The response of the SA-type 4 suspension follows closely the firm type suspension in the frequency range 5-12 rad/s. The results presented show that the analytically obtained response for SA-type 4 suspension correlates well with the experimental result for all the frequencies where the testing was conducted. However, the analytically predicted tire response seems to be higher compared to experimentally found results throughout the frequency range of interest.

In general, the analytical responses at lower frequencies correlate well with experimental results. In the higher frequencies, analytical responses are little higher in value than the experimental responses.

This may be due to the reason that in the analytical mode, damping value is based on the peak force measured.

CHAPTER 5

CONCLUSION

5.1 OverView

A review of the past work and the state-of-the-art survey on vehicle suspension is presented. A detailed discussion on passive, semi-active and active suspensions is also provided. Since for a conventional passive system, the spring rates and damping values are constants, it is shown that the performance of passive systems cannot be improved beyond a certain level. Active suspensions on the other hand, have superior performance compared to passive systems and require an actuator which replaces or acts in parallel with the passive components. Energy, usually a significant amount, may be put into or taken out of the system and the operational frequency bandwidth is defined by the frequency response characteristics of the actuator and the control components. Since the fully active suspension systems are complex, expensive and hard to maintain, semi-active (SA) suspensions which are a compromise between passive and active suspensions are presented. SA-suspensions are shown to have either a continuous or an on-off control for the damping orifice. The SA-suspension systems with on-off type orifice control, are passive damping devices that modulates flow through an orifice area, according to a particular control algorithm.

A detailed presentation on the various types of semi active vehicle suspension models studied in the past are outlined. Based on these SA-models, new types of SA-suspension schemes are proposed. A discussion is also presented on the solution methodology for solving the discontinuous system equations. The occurrence of system lock-up and the necessity to find a zero-crossing to treat for the lock-up are explained.

A Two D.O.F. quarter car bounce model with semi-active suspension systems is examined for vibration isolation properties. The performance of the SA-systems are evaluated and compared with passive suspension. The discontinuous equations of semi-active suspension systems are solved in conjunction with a zero-finder. A detailed insight is made to study the response behavior and configuration of the SA-systems through steady-state time plots. The steady-state time responses are presented with detailed discussions on the response behavior and the nature of the control logic of the system. The analytical investigation of vibration isolation peroformance of three types of SA-systems are carried out. In this thesis, efforts are made to provide a comprehensive analytical study on SA-suspensions with an experimental validation of the performance of one of the types of SA-systems.

The comparative studies on SA-systems [22,23,25,29] reported earlier lacks in-depth knowledge and comprehensive understanding of the SA-system behavior based on both analytical and experimental verification.

5.2 Highlights of the Thesis

It is observed from this investigation that the SA-type 1 and SA-type 3 systems undergo a lock-up mode for certain parts of the vibration cycle due to the relative velocity response crossing a zero value at lower frequencies. Though this cannot be identified explicitly in the time plot, it can be seen that there is a sharp spike in the acceleration response when the system breaks away from a lock-up mode. The relative velocity, the condition function, and hence the damping force responses have fluctuations due to the tendency of the system to switch back to its original course of motion at lock-up condition. The treatment of lock-up is facilitated by finding out the zero of the condition function using a zero-finder.

From the high frequency time histories, it is observed that for the SA-type 1 system, the value of the sky-hook condition function remains positive for most of the time in one period of oscillation, requiring the semi-active damping force (F_d) to be "on" for a long period of time. Consequently, the duration of time for the negative value of the condition function is almost zero in the same one period. This implies that the damping force (F_d) is to be switched instantaneously back to "on" position. Though this switching behavior of the condition function and the damping force is identified at high frequencies with an instantaneous breaking of the lock-up mode, difficulties may be encountered in practice while implementing this scheme.

For SA-type 3 system, in one cycle, the condition function switches periodically between positive and negative values for all the frequencies. Because of this behavior, this SA-system unlike the SA-type 1 system does not require instantaneous switching at higher frequencies. The acceleration response of the SA-type 3 system remains zero for certain parts of the vibration cycle when the damping force is "on". This is in conformity with the objective of the control logic.

The ride performance in terms of acceleration response for types 1, 2^*, and 3 SA-systems is better than the passive system at the sprung mass natural frequency and inferior at the unsprung mass natural frequency. The response of SA-type 1 system is superior at the sprung mass natural frequency, however, the response of SA-type 3 system is better in the frequency range between the two natural frequencies. The SA-type 2 suspension system is unstable at certain frequency ranges and hence a modified system, SA-type 2^*, having a small value of passive damping in parallel to the SA-type 2 damping is studied. The response of SA-type 2^* system is better than the passive system, though it is marginally inferior between the two natural frequencies. On the overall basis, the response of SA-type 2^* system is a good compromise at both the natural frequencies. Hence SA-type 2^* can be considered to provide an optimal ride performance among the three SA-systems.

The suspension deflection responses show that the response of SA-type 2^* system is better than the other two SA-systems and the passive system with an exception at very low frequencies. The suspension deflection of SA-type 3 system, however, is inferior at lower frequencies.

The tire deflection responses of the SA-type 1 and SA-type 3 systems are inferior at the lower frequencies, and unsprung mass natural frequency. The SA-type 2^* system provides a compromise performance for the tire deflection, however, though the response is marginally inferior between the two natural frequencies. Since the tire deflection is inferior for SA-systems at lower frequencies, heavier pavement load is expected, resulting in more wear and tear of the tire.

An effort is made to study the effect of a zero-finder in the solution methodology. Though, the study on the SA-systems without using a zero-finder in the solution methodology shows significant variations on the system responses for all the frequencies, the influence is dominant at the natural frequencies. This is due to the fact that the magnitude of the forces are high at these frequencies and hence solving for the discontinuous equations of SA-systems without proper initial conditions at the instant of switching could cause a dominant and obvious deviation in the responses. The SA-type 4 system is analytically studied and compared with laboratory testing for validation. The laboratory testing is carried out using an adjustable shock absorber fitted on a road vehicle, Nissan 300ZX, which has three modes of damping corresponding to firm, normal, and soft type suspensions. A preliminary testing was carried out to find out the damping characteristics for firm, normal, and soft type suspensions and then to verify the manufacturer supplied damping specifications. The vibration isolation performance of firm, normal, soft and SA-type 4 suspensions are then experimentally evaluated and compared with each other. Discussions are provided on the behavior of SA-type 4 suspension system based on both analytical and experimental studies. The vibration isolation performance of a typical road vehicle (Nissan 300ZX) based on a quarter-car model is verified analytically using the SA-type 4 system and compared with the available firm, normal, and soft type suspensions.

Detailed conclusions are provided on the response behavior.

The SA-type 4 suspension offers improved vibration isolation properties such as ride performance and a compromising suspension and tire deflection responses over the firm, normal, and soft type of suspensions when analyzed for the typical quarter vehicle of Nissan 300ZX automobile.

5.3 Recommendations for Future Study

Based on this, the following recommendations are made on both analytical and experimental investigations:

1. The investigation of SA-systems could be extended using higher degrees of freedom with a detailed modeling of the full vehicle.

2. The response behavior of other, SA-control logics could be studied using detailed steady-state time response analysis. Their feasibility in implementation and the requirement on switching, can be observed and their vibration isolation performance can be evaluated.

3. The SA-systems, types 1, 2, and 3 shall be evaluated using quadratic type damping instead of viscous damping and the vibration isolation performance be compared with SA-type 4 system.

4. All SA-systems be evaluated for typical random terrain input and their relative performance be compared against passive and active

suspensions.

5. In the experimental analysis of SA-systems, noisy response of the condition function may cause frequent switching of the damping forces. Since the commercially available shock absorbers have limitations in their switching response a phase lag may occur while enforcing the control law due to this frequent demand on switching. A detailed experimental investigation be carried out by filtering high frequency signals in the response variables prior to its use in the control function.

6. Other SA-type suspension systems, especially SA-type 2^* system be experimentally investigated.

7. Modeling of the shock absorber response could be done in detail taking into account the influence of the seal friction and gas spring.

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APPENDIX 1

EXPERIMENTAL EVALUATION OF DAMPING CHARACTERISTICS OF THE SHOCK ABSORBER

In this appendix, an experimental evaluation of the damping characteristics of the shock absorber used in the laboratory testing is presented. The adjustable shock absorber has the choice of three types of suspension damping properties corresponding to the firm, normal, and soft modes. The type of damping characteristic for a given mode depends on the position of the orifice opening that governs the flow of damping fluid, within the shock absorber [70] as shown in Fig A.1. The size of the orifice opening in the firm mode is small and hence it allows less fluid flow, resulting in higher magnitudes of damping force. In the soft mode, the orifice opening is wide and hence it allows more fluid flow resulting in low magnitude of damping force. In the normal mode, the orifice opening is between the other two modes and hence the magnitude of the damping force is also between the firm and soft modes.

The specifications on the damping force vs velocity relationship supplied by the manufacturer [70] is presented earlier in Fig 4.3b. It is intended to carry out a testing of the shock absorber using a test rig as shown in Fig. A.2 to verify the manufacturer supplied specifications. The testing is conducted by mounting the shock absorber between a fixed inertial frame and mounting platform. All the three modes are independently selected using the electronic control and the testing is conducted to establish their damping characteristics. From the preliminary testing it is observed that the shock absorber response time is 180 ms to switch between each mode.

The damping characteristics of the firm, normal, and soft modes are tested at various mid-stroke positions. The initial position of

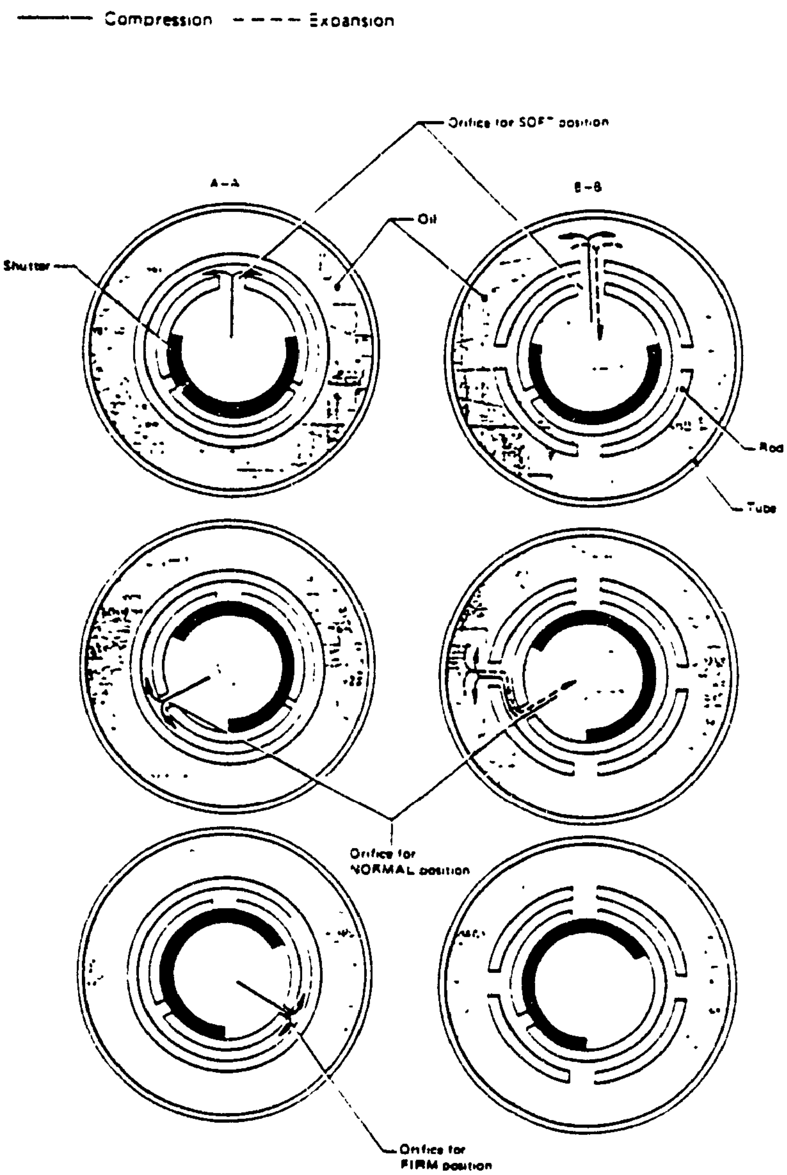


Fig. A.1 The relationship between the shutter and the orifice opening of the firm, normal, and soft type suspension using the Nissan 300ZX adjustable shock absorber [70]

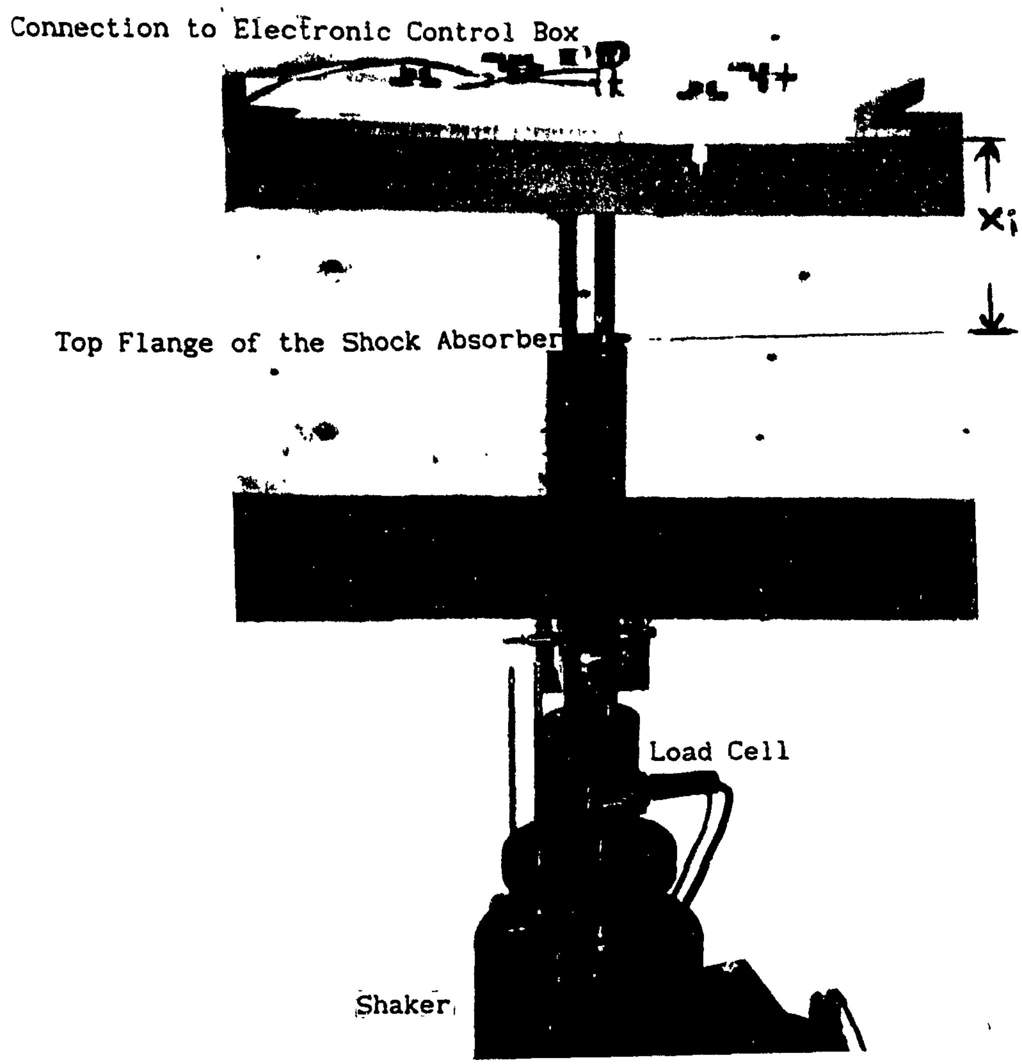


Fig.A.2. A pictorial view of the experimental setup.

operation X_i, is identified as a reference by measuring the distance between the top inertial frame and the top flange of the shock absorber as mentioned in Fig A.2. This distance is a direct measure of the free length of the piston rod when initially compressed into the cylinder due to the pre-load from the hydraulic shaker. This is also similar to the initial deflection due to the static load in the automobile. The testing was conducted at various initial positions of operations (X_i = 40.6 mm (1.6 in.), X_i = 101.6 mm (4. in.), X_i =139.7mm (5.5 in.) to study for their influence on the behavior of the shock absorber. The results indicate no significant variation in the response of the shock absorber. Harmonic input excitation is selected with peak-to-peak amplitudes of 12.7 mm (0.5 in.) at discrete frequencies in the range of 0.1 - 15.0 Hz. The damping characteristics presented in this work correspond to the initial position of X_i = 101.6 mm (4. in.).

The force vs displacement responses are presented in the form of lissajous plots and are plotted in Figs. A.4a, A.4b, A.5a, A.5b, A.6a, and A.6b. for firm, normal, and soft modes of the shock absorber. In addition, the steady-state time response plots are also presented. The steady-state responses, show the presence of seal friction and gas spring in the shock absorber. A precise estimate of these factors could not be obtained using the present experimental set-up. The piston speed is obtained using the value of the input displacement and the excitation frequency. The magnitudes of peak damping force on extension and compression strokes are obtained from the steady-state responses as explained in the schematic in Fig A.3. The damping force, velocity responses are presented for the firm, normal, and soft modes in Tables A.1, A.2, A.3, and A.4. It should be noted that the magnitude of damping force measured represents the net force realized in the shock absorber including the forces due to the seal friction and gas spring. From the experimental lissajous plots, the damping force vs velocity relationship of the adjustable shock absorber for the firm, normal, and soft modes are evaluated. The results are presented in Figs. A.7, A.8, and A.9, and compared with the manufacturer supplied specifications [70]. The damping characteristics evaluated from the testing correlate well with the manufacturer data in the low velocity region. However, the responses have discrepancies for higher velocities. The experimentally evaluated results are found to be higher in these high velocity regions. The reasons for this variation can be attributed to the fact that the experimentally evaluated damping force is based on the peak damping force which includes the influence of seal friction.

Using the experimentally evaluated damping forces at discrete frequencies, the damping characteristics for the firm, normal, and soft modes are obtained by fitting a straight line and the results are presented in Table A.4. These damping values are used in the analysis in chapter 4.

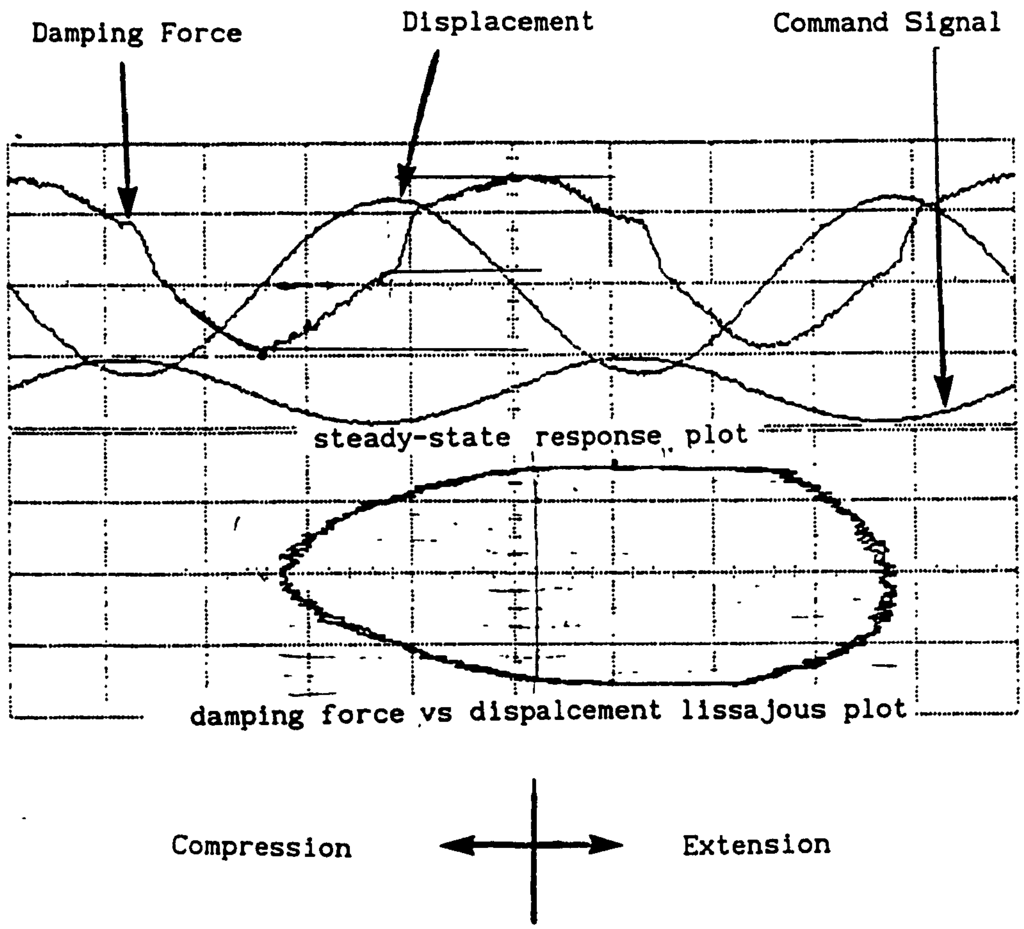
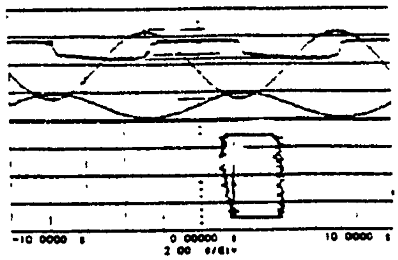


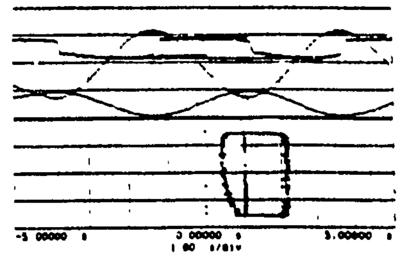
Fig. A.3 A schematic of a typical steady-state response plot and damping force vs dispalcement lissajous plot

a.
b.



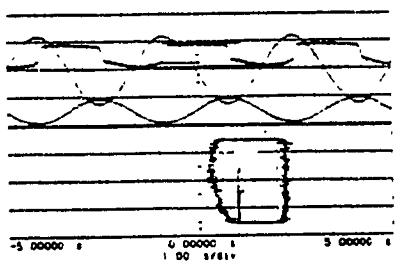
0.1 Hz

a.
b.



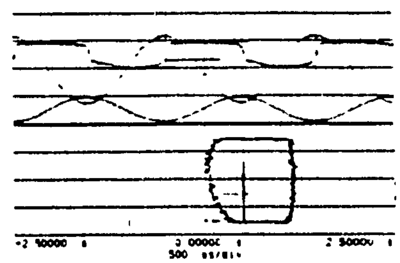
0.2 Hz

a.
b.



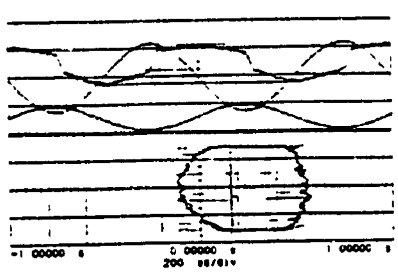
0.3 Hz

a.
b.



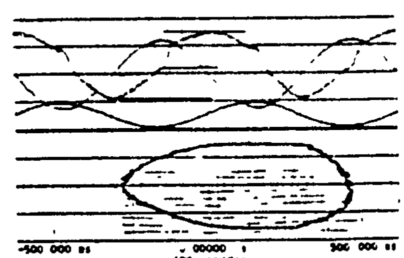
0.5 Hz

a.
b.



1.0 Hz

a.
b.

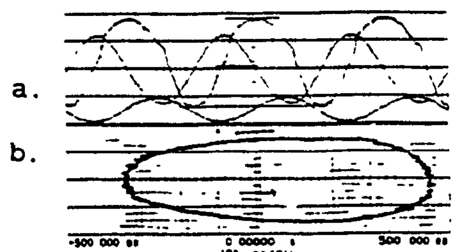


2.0 Hz

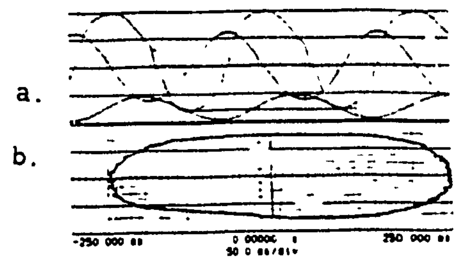
a. Steady-state time response

b. Lissajous plot

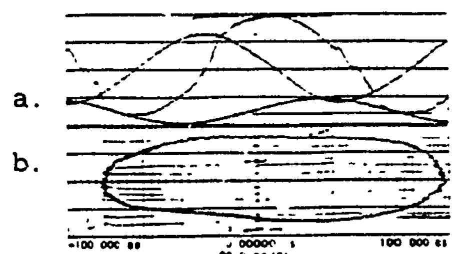
Fig. A.4a. Steady-state time response plots, and damping force vs displacement lissajous plots for firm mode (0.1 to 2.0 Hz)



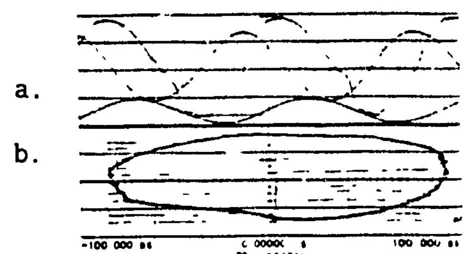
3.0 Hz



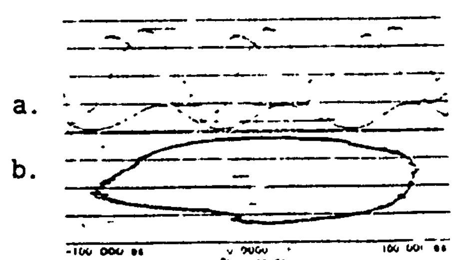
5.0 Hz



7.0 Hz



11.0 Hz



15.0 Hz

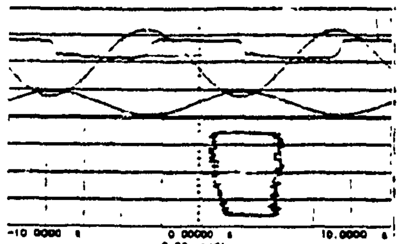
a. Steady-state time response

b. Lissajous plot

Fig. A.4b Steady-state time response plots, and damping force vs displacemet lissajous plots for firm mode (2.0 to 15.0 Hz)

a.

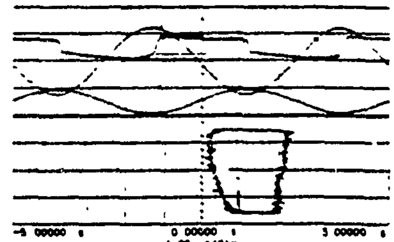
b.



0.1 Hz

a.

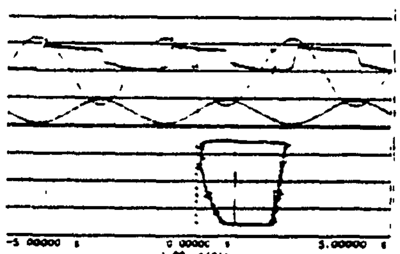
b.



0.2 Hz

a.

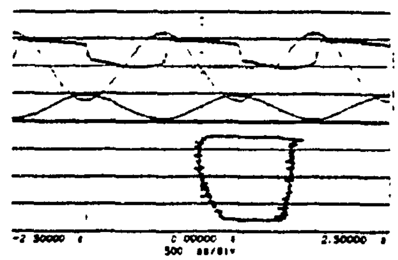
b.



0.3 Hz

a.

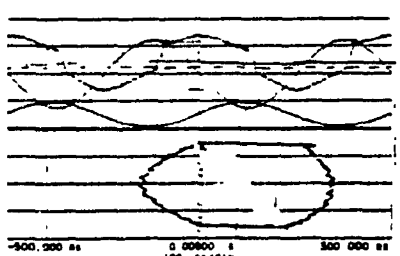
b.



0.5 Hz

a.

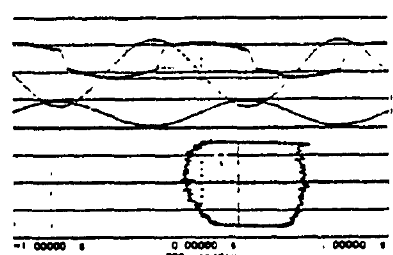
b.



1.0 Hz

a.

b.



2.0 Hz

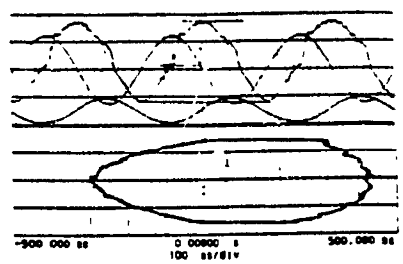
a. Steady-state time response

b. Lissajous plot

Fig. A.5a Steady-state time response plots, and damping force vs displacement lissajous plots for normal mode (0.1 to 2.0 Hz)

a.

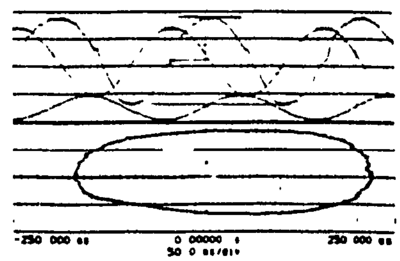
b.



3.0 Hz

a.

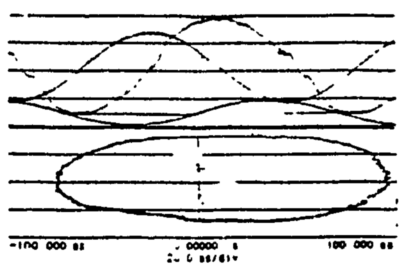
b.



5.0 Hz

a.

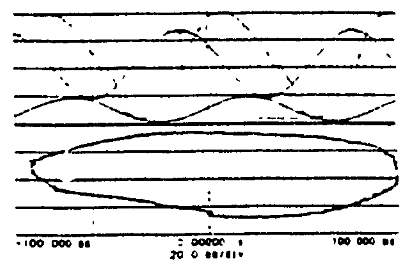
b.



7.0 Hz

a.

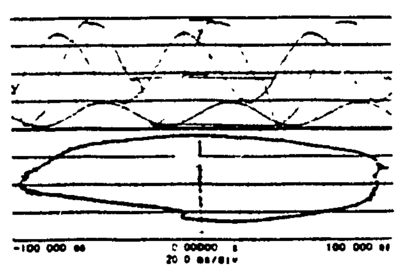
b.



11.0 Hz

a.

b.

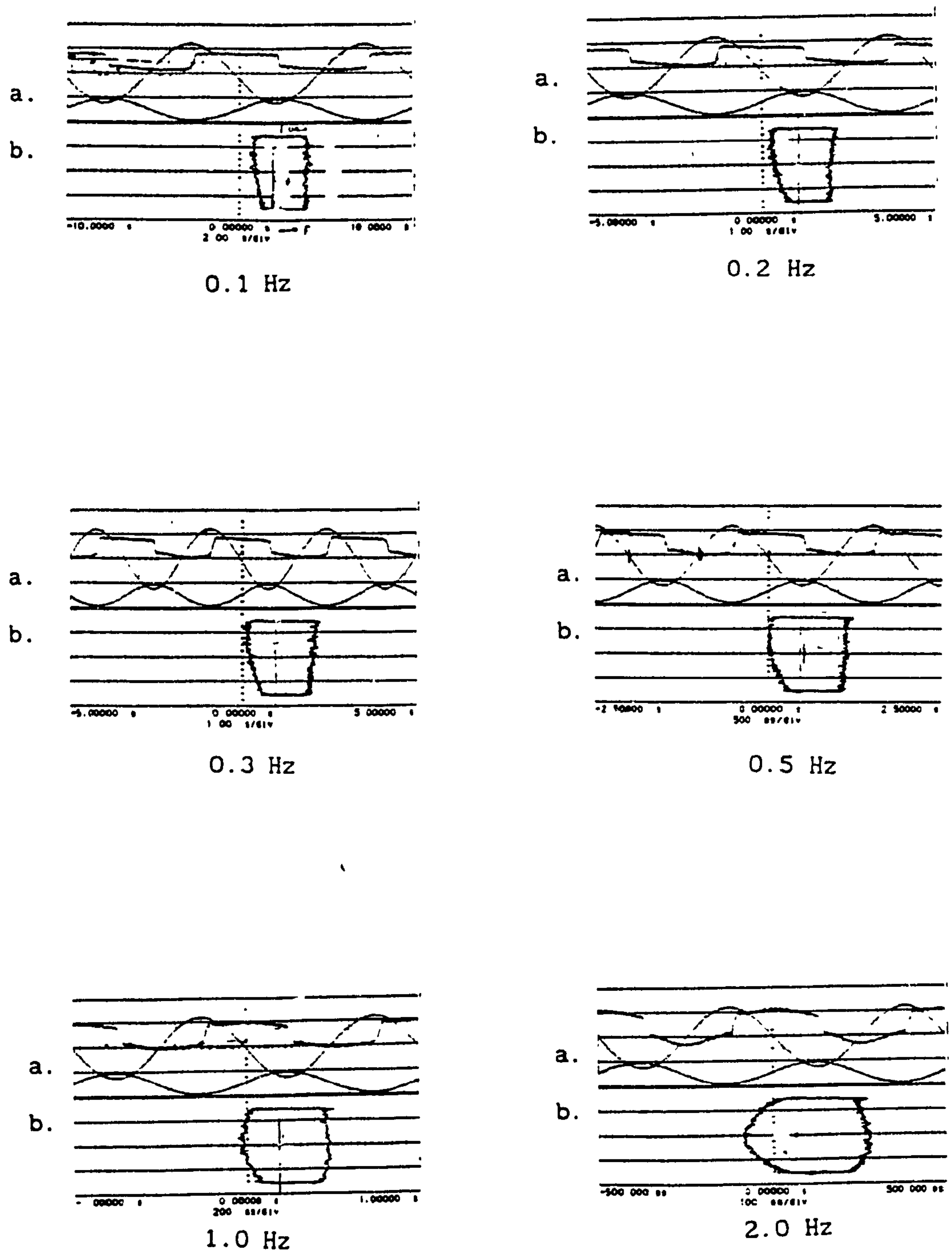


15.0 Hz

a. Steady-state time response

b. Lissajous plot

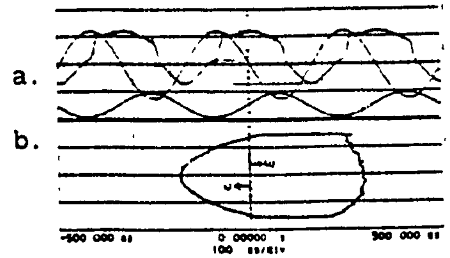
Fig. A.5b Steady-state time response plots, and damping force vs displacement lissajous plots for normal mode (2.0 to 15.0 Hz)



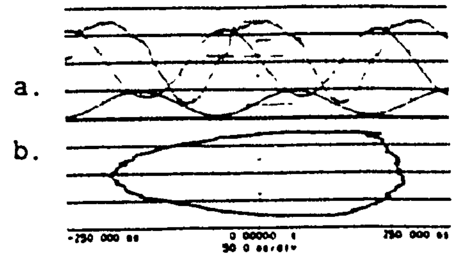
a. Steady-state time response

b. Lissajous plot

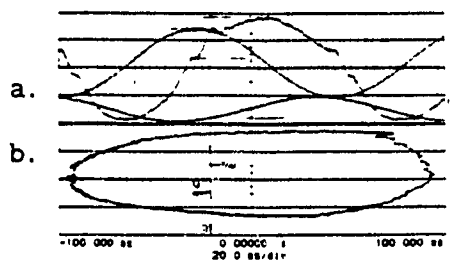
Fig. A.6a Steady-state time response plots, and damping force vs displacement lissajous plots for soft mode (0.1 to 2.0 Hz)



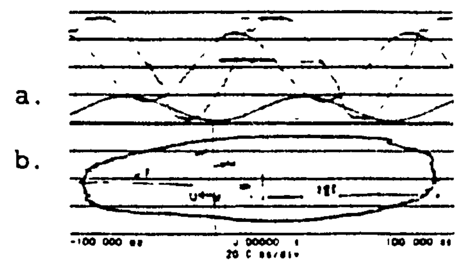
3.0 Hz



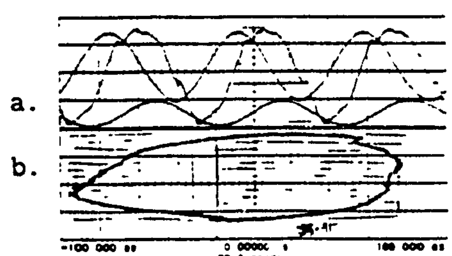
5.0 Hz



7.0 Hz



11.0 Hz



15.0 Hz

a. Steady-state time response

b. Lissajous plot

Fig. A.6b Steady-state time response plots, and damping force vs displacemnt lissajous plots for soft mode (2.0 to 15.0 Hz)

TABLE A.1 Experimental damping force and velocity responses for firm mode

FIRM

f Hz	V (m/s)	F Ext(N)	F Comp(N)
0.1	0.008	170	50
0.2	0.016	175	100
0.3	0.023	200	135
0.5	0.039	235	155
1.0	0.079	325	225
2.0	0.157	575	400
3.0	0.235	700	550
5.0	0.394	860	775
7.0	0.544	1020	945
11.0	0.833	1400	1380
15.0	1.017	1825	1575

TABLE A.2 Experimental damping force and velocity responses for normal mode

NORMAL

f Hz	V (m/s)	F Ext(N)	F Comp(N)
0.1	0.008	175	100
0.2	0.016	215	125
0.3	0.023	225	165
0.5	0.039	235	200
1.0	0.079	290	250
2.0	0.157	450	350
3.0	0.235	650	550
5.0	0.394	908	655
7.0	0.544	1105	835
11.0	0.833	1500	1375
15.0	1.017	1810	1750

TABLE A.3 Experimental damping force and velocity responses for soft mode

SOFT

f Hz	V (m/s)	F Ext(N)	F Comp(N)
0.1	0.008	150	75
0.2	0.016	175	100
0.3	0.023	200	125
0.5	0.039	225	150
1.0	0.079	250	200
2.0	0.157	325	250
3.0	0.235	475	275
5.0	0.394	750	375
7.0	0.544	950	525
11.0	0.833	1400	700
15.0	1.017	1725	1100

TABLE A.4 Experimentally evaluated damping values for

Nissan 300ZX Adjustable Rear Shock Absorber

[70]

	soft		normal		firm	
COMPRESSION						
V	< 0.235	> 0.235	< 0.160	> 0.160	< 0.20	> 0.20
C_e	1250.0	1578.0	2400.0	1818.0	2500.00	1107.0
EXTENSION						
V	< 0.15	> 0.15	< 0.180	> 0.180	< 0.16	> 0.16
C_c	1666.0	1666.0	3000.0	1428.0	3000.0	1428.0

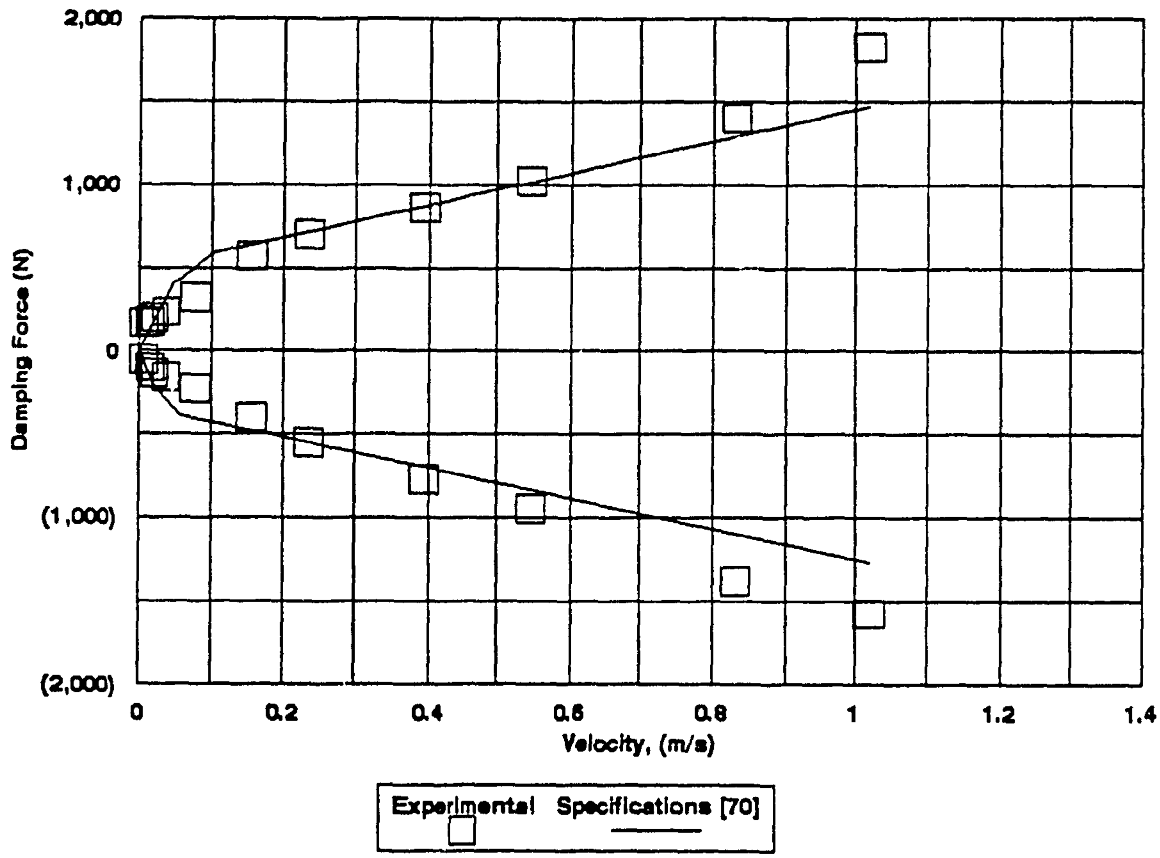


Fig. A.7 Comparison of damping force vs velocity relationship for Nissan 300ZX rear adjustable shock absorber [70] using firm mode

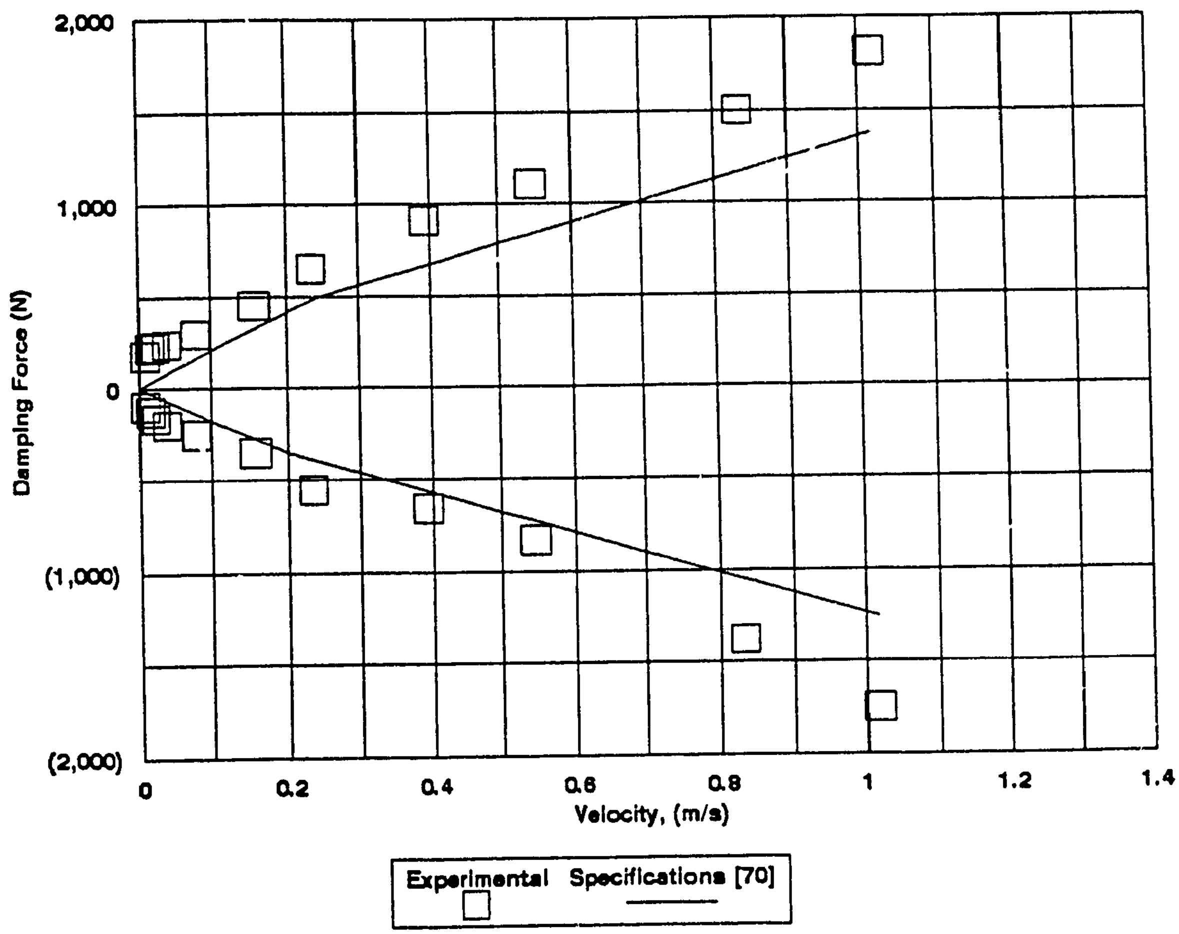


Fig. A.8 Comparison of damping force vs velocity relationship for Nissan 300ZX rear adjustable shock absorber [70] using normal mode

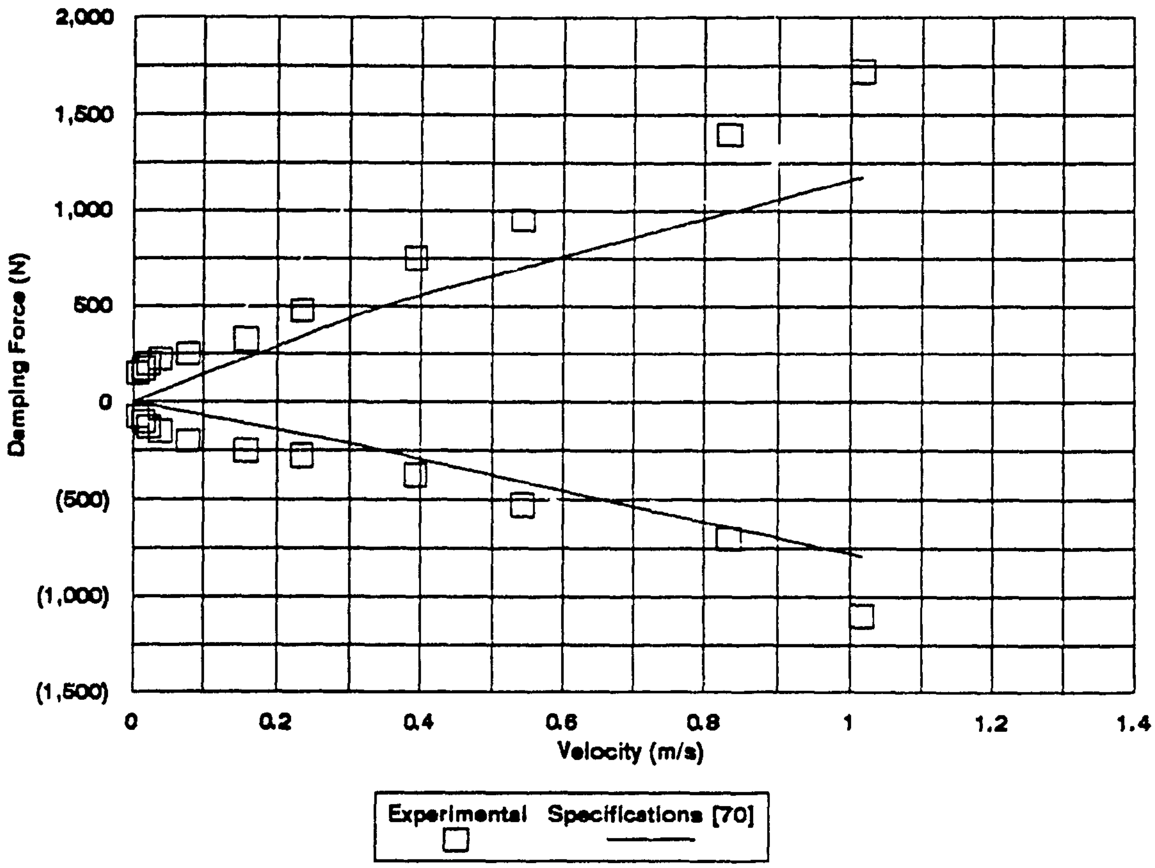


Fig. A.9 Comparison of damping force vs velocity relationship for Nissan 300ZX rear adjustable shock absorber [70] using soft mode