ANALYSIS OF A MODIFIED PASSIVE HYDRAULIC DAMPER WITH VARIABLE DAMPING CHARACTERISTICS

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ABSTRACT

Conventional hydraulic dampers, which are widely employed in mechanical vibration and shock isolation systems, exhibit inherent performance limitations due to the fixed orifice damping characteristics. In this paper, a conventional hydraulic damper is modified to achieve variable damping via simple passive means. The variable damping mechanism is realized passively by limiting the pressure differential across the hydraulic damper piston, using pressure relief valves. The hydraulic damper is modelled as a nonlinear dynamical system incorporating the nonlinearities such as orifice damping, gas-spring and pressure relief mechanism. The damping characteristics of both conventional and the modified dampers are discussed in view of vibration isolation performance. The dynamic response of a vehicle suspension model employing the modified hydraulic damper is investigated via computer simulation. The transmissibility characteristics of the vehicle suspension system are obtained to evaluate the vibration isolation performance of the proposed damper. The shock isolation performance is evaluated in terms of its transient response to a road bump input. The simulation results of the modified vehicle suspension are compared with that of the conventional hydraulic shock absorber system. It is concluded that the vehicle ride performance can be improved considerably using the modified hydraulic damper.

INTRODUCTION

The selection of appropriate spring and damping mechnisms is one of the most important tasks in the design of vibration and shock isolation systems. Conventional passive hydraulic dampers are widely employed in mechanical vibration and shock isolation systems and offer a simple, inexpensive and reliable mean to protect the mechanical systems and human body from vibration and shock disturbances. However, it is well known that the passive vibration isolation systems exhibit inherent performance limitations due to the fixed orifice damping characteristics [1, 2]. A heavily damped passive vibration isolation system tends to reduce the amplitude of vibration response only when the frequency of the base disturbances is around the natural frequency of the system. While the vibration isolation performance of the passive system is deteriorated considerably at higher frequency range. On the other hand, a lightly damped vibration isolation system is desirable when the disturbance frequencies are beyond the natural frequency; however it yields a poor response at the resonance of the system. In order to overcome these inherent limitations of the passive vibration isolation systems, various vibration isolation systems with variable parameters have been proposed, such as active and semi-active vibration isolation systems.

In active vibration isolation systems, the controlled damping and stiffness parameters change with variations in excitation and response characteristics and thus provide superior isolation performance. However, active controlled damping mechanisms, proposed in the literature, require an external energy source, complex feedback control devices and sensors. Thus the general use of active shock and vibration isolation systems has been severely limited due to the associated high costs, complexities and poor reliability [3, 4].

Semi-active vibration isolation systems generate damping forces passively while the damping parameters are modulated using an active control system. Semi-active damping is often realized by modulating the orifice area of hydraulic dampers using various control schemes, such as skyhook [5, 6] and sequential or 'on-off' schemes [2]. Semi-active vibration isolation systems require only low level electrical power for necessary signal processing and can provide improved vibration and shock isolation performance compared with that of the passive vibration isolation systems. However, the semi-active vibration isolation systems still require a comprehensive instrumentation and control devices.

In this paper, a conventional hydraulic damper is modified to achieve a variable damping in vibration and shock isolation systems via simple passive means. The variable damping mechanism is realized passively by limiting the pressure differential across the hydraulic damper piston, using pressure relief valves. The hydraulic damper is modelled as a nonlinear dynamical system incorporating the nonlinearities, such as orifice damping, gas-spring and pressure limiting mechanism. The damping characteristics of both conventional and the proposed dampers are discussed in view of their vibration isolation performance. The concept of the proposed modified hydraulic damper employing pressure limiting valves is discussed through the flow and force balance equations. The dynamic response of a vehicle suspension employing the modified hydraulic damper is investigated via computer simulation. The vibration isolation performance of the proposed damper is discussed in terms of the vibration transmissibility characteristics of the vehicle model. The shock isolation performance is established in terms of its transient response to a road bump input. The simulation results of the modified suspension system are compared with that of the conventioal hydraulic shock absorber system to demonstrate the improved vehicle ride performance of the proposed damper.

DEVELOPMENT OF ANALYTICAL MODEL

Fig. 1 presents the schematic of a conventional passive hydraulic damper. Neglecting leakage flows and seal friction, the total dynamic force f_D generated by the hydraulic damper due to pressure differential across the piston is expressed as:

$$f_D = (p_1 - p_0)A_P - (p_2 - p_0)(A_P + A_R) \tag{1}$$

where p_0 is the hydraulic pressure corresponding to the static equilibrium position, p_1 and p_2 are the instantaneous pressures in chambers I and II, respectively, A_P is the piston area, and A_R is the cross section area of the rod. By letting $p_{ij}=p_i-p_j$, equation (1) can be expressed as:

$$f_D = p_{12}A_P + p_{32}A_R - p_{30}A_R \tag{2}$$

where p_3 is the instantaneous pressure in chamber III. Assuming turbulent flow condition the pressure differentials p_{12} and p_{32} across the piston and cylinder orifices are expressed as [7]:

$$P_{12} = \frac{\rho}{2n^2C_{d1}^2}\left(\frac{A_P}{a_1}\right)^2 |\dot{z}|\dot{z} \tag{3}$$

and

$$P_{32} = \frac{\rho}{2C_{d1}^2}\left(\frac{A_R}{a_2}\right)^2 |\dot{z}|\dot{z} \tag{4}$$

where $\dot{z}$ is the relative velocity across the damper, given by

$$\dot{z} = \dot{x} - \dot{x}_i$$

and n is the number of orifices on the piston, a_1 and a_2 are areas of orifices on the piston and cylinder, respectively, C_{d1} and C_{d2} are discharge coefficients, and ρ is the density of hydraulic fluid. The pressure differential in chamber III can be related to the relative compression/extension of gas column. Assuming polytropic process, the differential pressure P_{30} is be expressed as:

$$P_{30} = - \frac{[(V_0+A_R z)^{\gamma} - V_0^{\gamma}]}{(V_0 + A_R z)^{\gamma}} p_0 \tag{5}$$

where V_0 is the initial gas volume corresponding to the static equilibrium position and γ is the polytropic constant. It is evident from equations (2) to (5) that the total dynamic force generated by the damper comprises of a damping force f_d due to the orifice flow and a restoring force f_a due to the pressurized gas column. The total dynamic force generated by the hydraulic damper can be expressed as:

$$f_D(t) = f_d(t) + f_a(t) \tag{6}$$

where

$$f_a(t) = - p_{30} A_R \tag{7}$$

$$f_d(t) = \alpha p_{12} \tag{8}$$

and

$$\alpha = A_P + \frac{p_{32}}{p_{12}} A_R \tag{9}$$

From equations (3) and (4), it is evident that the ratio of pressure differentials (p_{32}/p_{12}) can be expressed as a constant:

$$\frac{p_{32}}{p_{12}} = \left(\frac{C_{d1}}{C_{d2}}\right)^2 \left(\frac{na_1}{a_2}\right)^2 \left(\frac{A_R}{A_P}\right)^2 = \lambda \tag{10}$$

From equations (3) and (8) it is clear that the pressure differential p_{12} and thus the damping force are dependent upon the square of relative velocity across the damper. The magnitude of the damping force becomes predominant at high excitation frequencies and thus yields poor vibration isolation performance. The magnitude of damping force at high excitation frequencies can be lowered in a manner similar to the sequential semi-active dampers by limiting the pressure differential across the piston. However, the pressure limiting and thus variable damping force can be realized via passive means using pressure relief valves. The pressure relief valves are selected to limit the magnitude of pressure differential across the piston to a preset value $(p_{12})_o$ by modulating the flow through compression and rebound relief valves across chambers I and II, as shown in Fig. 2. When the magnitude of the pressure differential p_{12} is less than a preset limiting value $(p_{12})_o$ of the relief valves, the relief valves remain closed and thus the damper acts as a conventional passive hydraulic damper. However, when the magnitude of the pressure p_{12} exceeds the preset value $(p_{12})_o$, the relief valve opens. The damping force is then reduced considerably by permitting the fluid flow pass through the opened relief valve. Neglecting dynamics of the relief valves, the sequential damping force due to the modified passive damper can be expressed as:

$$f_d = \begin{cases} \alpha p_{12}, & |p_{12}| < (p_{12})_o \\ \alpha (p_{12})_o \operatorname{sgn}(p_{12}), & \text{otherwise} \end{cases} \tag{11}$$

where

$$\operatorname{sgn}(*) = \begin{cases} +1, & (*) > 0 \\ -1, & (*) < 0 \end{cases}$$

The sequential damping via pressure limiting can be realized completely passively, and it does not require the instrumentation and control package and external power required by the active and semi-active systems.

DAMPING CHARACTERISTICS OF THE MODIFIED DAMPER

A damping parameter β is defined as the ratio of the damping force to the critical damping force of a viscously damped vibration isolation system:

$$\beta = f_d/(2\sqrt{mk}\ \dot{z}) \tag{12}$$

From equations (3), (8) and (12), the damping parameter of the conventional passive hydraulic damper is obtained as:

$$\beta = \frac{\alpha}{2\sqrt{mk}} \left(\frac{\rho}{2n^2 C_{d1}^2}\right) \left(\frac{A_P}{a_1}\right)^2 |\dot{z}| \tag{13}$$

It is obvious that the damping parameter of the conventional hydraulic damper is proportional to the magnitude of relative velocity across the damper. The damping parameter of the modified hydraulic damper can be obtained from equations (11) and (12):

$$\beta = \begin{cases} \dfrac{\alpha}{2\sqrt{mk}} \left(\dfrac{\rho}{2n^2 C_{d1}^2}\right) \left(\dfrac{A_P}{a_1}\right)^2 |\dot{z}|, & |p_{12}| < (p_{12})_o \\[2ex] \dfrac{\alpha (p_{12})_o}{2\sqrt{mk}|\dot{z}|}, & \text{otherwise} \end{cases} \tag{14}$$

Equation (14) reveals that for $|p_{12}|<(p_{12})_o$ the damping parameter of the modified damper is proportional to the magnitude of the relative velocity response as in the same case of the passive damper. However, the damping parameter is inversely proportional to the magnitude of the relative velocity response, when the pressure differential p_{12} exceeds the limiting value of $(p_{12})_o$.

In view of vibration isolation performance, it is desirable to achieve a high value of damping parameter around the resonant frequency so that the resonant peak can be appropriately controlled. On the other hand, a low value of damping parameter is desirable at high frequencies to achieve the improved vibration isolation performance. The modified hydraulic damper can provide the desirable damping characteristics expressed in equation (12) when $(p_{12})_o$ is appropriately selected. In order to control the resonant peak response, the minimum value of the limiting pressure is estimated by using the relationship of damping parameter and resonant amplitude of a linear vibration system [8]. A suitable value of the preset limiting pressure $(p_{12})_o$ is then expressed as:

$$(p_{12})_o = \nu(kX_i/\alpha) \tag{15}$$

where X_i is the amplitude of excitation, and ν is the pressure limiting factor of the modified hydraulic damper. The pressure differential characteristics of a base excited single degree-of-freedom mechanical system employing the modified damper is compared with that of the conventional damper as shown in Fig. 3. The pressure differential p_{12} across the conventional damper piston increases rapidly with the increase of the excitation frequency. However, the pressure differential p_{12} of the modified damper increases with the excitation frequency and then settles down to the preset value $(p_{12})_o$. The value of pressure differential p_{12} and thus the damping force at high excitation frequencies are dependent upon the limiting factor ν as shown in Fig. 3. The damping characteristics of the modified damper as well as the conventional hydraulic damper are presented in Fig. 4. It is observed that at low excitation frequencyies the damping parameter of the modified damper is identical to that of the conventional hydraulic damper. However, the damping parameter of the modified damper decreases at higher frequencies when p_{12} exceeds $(p_{12})_o$.

MODELLING OF A VEHICLE SUSPENSION WITH MODIFIED DAMPER

A vehicle suspension equipped with the modified hydraulic damper, as shown in Fig. 5, is modelled and analysed, in order to evaluate the vibration and shock isolation performance of the modified damper. The vehicle is modelled as a two-degree-of-freedom dynamic system, often refered to as 'quarter vehicle model' [9]. The vehicle mass is represented by a sprung mass m_s, and the wheel and axle assembly is modelled as an unsprung mass m_u. The primary vehicle suspension model comprises of a linear spring of stiffness k_s and a modified nonlinear hydraulic damper D. The total force generated by the modified damper includes a restoring force due to the gas-spring and a dissipative force due to the orifice flow. The tire is modelled as a linear spring of stiffness k_t, assuming point contact with the terrain. The hysteretic properties of the tire is assumed to be small. The equations of motion of the two-degree-of freedom vehicle model are expressed as:

$$\left\{\begin{array}{ll} m_s\ddot{x} + k_s z + f_a(t) + f_d(t) = 0 & (16) \\ m_u\ddot{x}_1 - k_s z - f_a(t) - f_d(t) + k_t x_1 = k_t x_i & (17) \end{array}\right.$$

where $z=x-x_1$, $f_a(t)$ is the gas-spring force derived in equations (5) and (7), and $f_d(t)$ is the modified damping force expressed in equations (3) and (11).

The simulation parameters of the vehicle model and the modified hydraulic damper are selected as follows:

$m_s=240$ kg, $m_u=36$ kg, $k_s=16000$ N/m, $k_t=160000$ N/m, $\rho=797.18$ kg/m^3, $C_{d1}=0.7$, $C_{d2}=0.7$, $A_P=2.513\times10^{-3}$ m^2, $A_R=3.1416\times10^{-4}$ m^2, $\gamma=1.4$, $V_0=1.9\times10^{-4}$ m^3, $p_0=13.7\times10^5$ Pa, and $a_1=3.1416\times10^{-6}$ m^2, $a_2=3.1416\times10^{-6}$ m^2, and X_i = 1.86×10^{-3} m.

RESULTS AND DISCUSSION

The vehicle model incorporating nonlinearities due to gas-spring, orifice damping and pressure relief valve mechanism is simulated using the numerical integration technique. The dynamic ride performance of the vehicle suspension employing the modified damper is evaluated through the vibration and shock isolation characteristics. The vibration isolation characteristics of the modified hydraulic suspension are expressed in terms of vibration transmissibility of the suspension system. The shock isolation performance of the modified damper is evaluated in terms of its transient response to a road bump input.

Vibration Isolation Performance

The vibration isolation performance of the conventional and modified suspensions is evaluated for harmonic displacement excitations at the tire-road interface. The vibration transmissibility is obtained by computing the ratio of the steady state response amplitude to the excitation amplitude for each excitation frequency.

The velocity transmissibility response of the sprung mass of the vehicle employing modified and conventional passive dampers is shown in Fig. 6. The damper piston with two orifices is considered for the conventional as well as modified damper ($n=2$). A unit pressure limiting factor $\nu=1$ is selected for the modified hydraulic damper. It is observed that the transmissibility of the conventional hydraulic damper system yields two peaks corresponding to the resonant frequencies of the sprung and unsprung masses of the vehicle, respectively. The second peak corresponding to the resonance of the unsprung mass m_u, is mainly due to the high value of orifice damping produced by the conventional passive damper. The transmissibility characteristics of the vehicle model employing the modified hydraulic damper, is identical to that of the conventional suspension system at low exciting frequency. The modified hydraulic damper continues to dissipate energy identical to that of the conventional damper around the first resonant frequency. However, as the excitation frequency and thus the relative velocity response increase, the pressure differential p_{12} is held around $(p_{12})_o$ by the pressure relief mechanism to reduce the damping force generated by the modified hydraulic damper. Thus the velocity transmissibility peak corresponding to the unsprung mass resonance is reduced significantly as shown in Fig. 6. A comparison of the transmissibility characteristics of the modified and conventional dampers reveals that the vibration isolation performance of the modified hydraulic damper system is considerably superior to that of the conventional damper at higher excitation frequencies.

The vibration transmissibility of the conventional passive damper, corresponding to the unsprung mass resonance, can be reduced considerably via increasing the number of orifices, as shown in Fig. 7. However, the peak transmissibility response, corresponding to the the sprung mass resonant frequency, increases considerably with light damping. A comparrison of the displacement transmissibility characteristics of the conventional dampers ($n=2$ and $n=4$) with the modified damper ($n=2$ and $\nu=1$) reveals that the modified damper can provide an appropriate control of the peak response corresponding to the low as well as high frequencies. A comparison of velocity and displacement transmissibilities in figures 6 and 7 also confirms that for nonlinear dynamical systems the transmissibiities of velocity and displacment are no longer identical as in the case of linear systems.

The velocity transmissibility characteristics of the modified hydraulic dampers with different preset pressure limiting factors, $\nu=$ 0.7, 1.0 and 1.5, are shown in Fig. 8. The influence of limiting factor ν and thus the pressure limiting value $(p_{12})_o$ on the vibration transmissibility is observed by comparing the response due to the different ν values. A low value of pressure limiting factor ν yields a further improved vibration transmissibility at the higher excitation frequencies as compared with that of a higher value of ν factor. However, a lower value of ν factor with a lower preset pressure limiting value $(p_{12})_o$ results in a early opening of relief valves, that may produce a very large resonant peak due to the insufficient damping at the resonance. A higher value of ν factor results in a late opening of the relief valves and yields a higher transmissibility value at higher excitation frequency. It is obvious that the suspension system equipped with the modified hydraulic damper with the pressure limiting factor, $0.7<\nu<1.5$, exhibits superior vibration isolation performance.

Shock Isolation Performance

The shock isolation performance of the conventional as well as modified dampers is investigated for bump excitation, as shown in Fig. 9. The bump represent a half round obstacle of radius h. The vehicle is assumed to travel at a constant speed v. The instantaneous coordinate of the vehicle u(t) is expressed as:

$$u(t) = u_0 + vt \tag{18}$$

where t is the time, u_0 is the initial possition of the vehicle away from the centre point of the bump. The instantaneous excitation due to the road bump can be expressed as:

$$x_i = \begin{cases} \sqrt{h^2 - u^2}\,, & -h < u < h \\ 0\,, & \text{otherwise} \end{cases} \tag{19}$$

The shock isolation characteristics of the suspension system equipped with the modified hydraulic damper are established through computer simulation, with the following parameters: h=0.1524 m, u_0=−0.2524 m, and the vehicle speed is selected as v= 5 m/s.

The transient displacement response of the sprung mass m_s of the vehicle suspension system with a conventional passive damper and a modified hydraulic damper, together with the history of the input displacement x_i, is shown in Fig. 10. Where the total orifice areas of both conventional and modified hydraulic dampers are identical (n=2), and the pressure limiting factor for the modified damper is selected as ν= 1. Fig. 11 shows the transient velocity response of the sprung mass with the conventional and modified hydraulic damper systems. Figs. 10 and 11 reveal that the peak displacement and velocity response of the modified hydraulic damper are considerably smaller tant that of the conventional damper. Moreover, the transient response of the modified damper is less oscillatory than that of the conventional damper. The transient displacement response of the unsprung mass m_u, $x_1(t)$, for the same systems is shown in Fig. 12. The maximum transient response of the modified hydraulic damper is slightly larger than that of the conventional damper system, however it yields less oscilations than that of the conventional system. Since the main purpose of the suspension design is to reduce the oscilation of the sprung mass, the modified hydraulic damper can, therefore, be used to improve the ride performance.

The transient displacement response of the sprung mass m_s with a lightly damped suspension system (n=6) both the conventional and modified hydraulic dampers, together with the bump input history, is shown in Fig. 13. The conventional passive damper system with much small damping yields an improved transient response as compared with that of the conventional system with a higher damping in Fig. 10. The maximum amplitude of the response is slightly lower and the oscilations are much less than those of the highly damped system. However, the modified hydraulic damper yields a further improved transient response, in terms of both the maximum amplitude and the munber of oscillations, as compared with that of the lightly damped conventional system. Moreover, it should be pointed out that a lightly damped conventional system yields a high peak value of vibration transmissibility of the sprung mass at the resonance as shown in Fig. 7.

CONCLUSIONS

A conventional hydraulic damper is modified to achieve a variable damping in vibration and shock isolation systems. The damping characteristics of the modified hydraulic damper are discussed in view of the vibration isolation performance. A vehicle suspension model employing the modified hydraulic damper is analysed via computer simulation. The shock and vibration isolation performance of the suspension due to the modified passive hydraulic damper with the variable damping is evaluated and compared with that of the conventional hydraulic damper system. The vibration isolation performance of the modified damper is presented in terms of its transmissibility caharacteristics. The shock isolation performance is evaluated in terms of its transient response to a road bump input. From the results of both the vibration transmissibility and the transient respons, it is concluded that the vehicle ride performance can be improved considerably by using the modified hydraulic damper.

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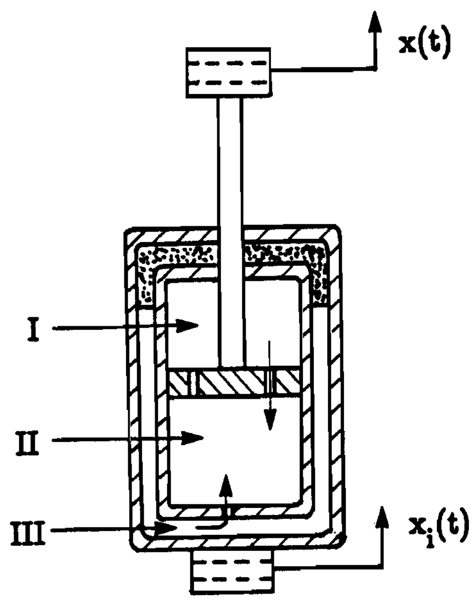

Fig.1 Schematic of a conventional hydraulic damper

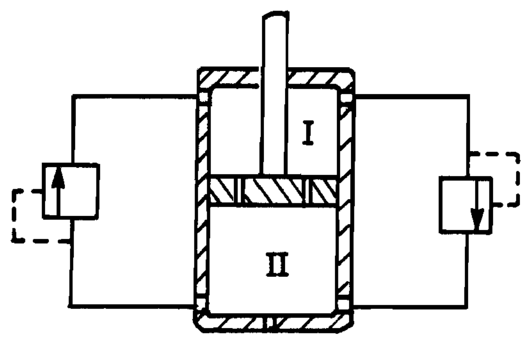

Fig.2 Schematic of a modified hydraulic damper with two relief valves

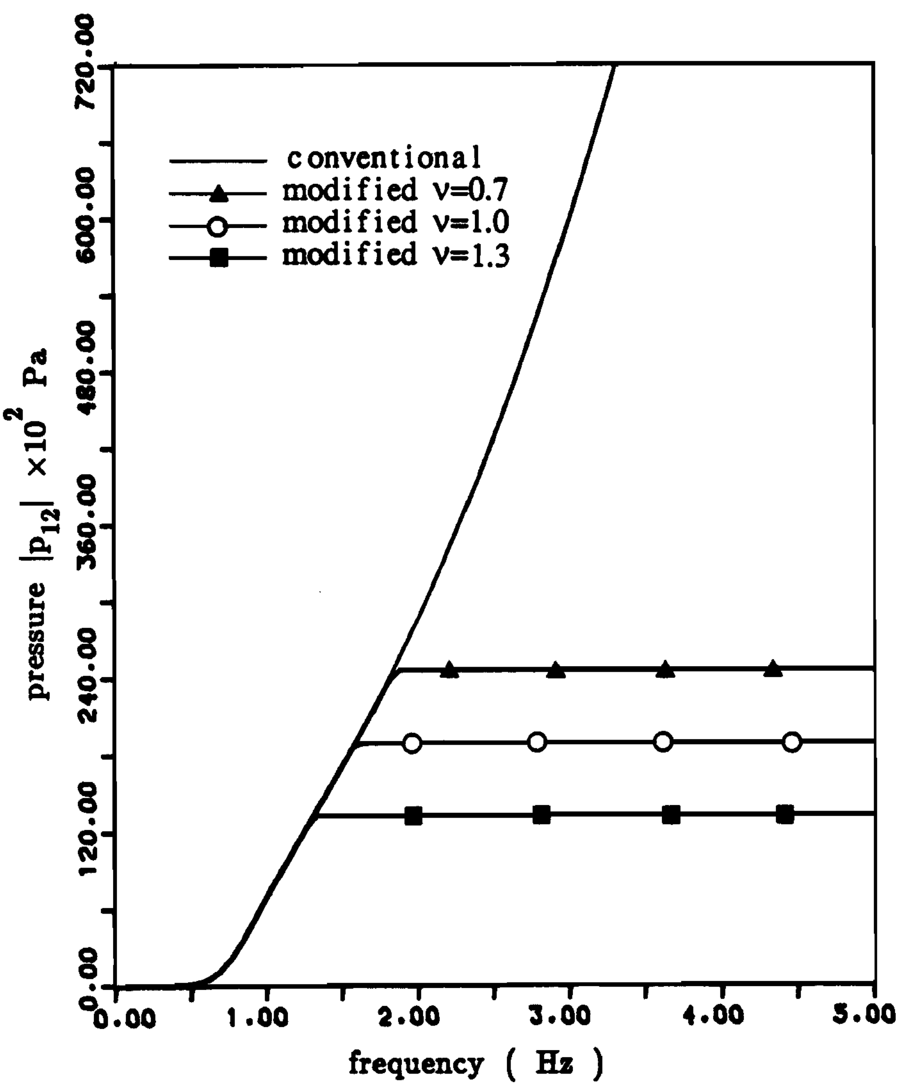

Fig.3 Pressure differential characteristics of the conventional and modified shock absorbers (ν = 0.7, 1.0 and 1.3)

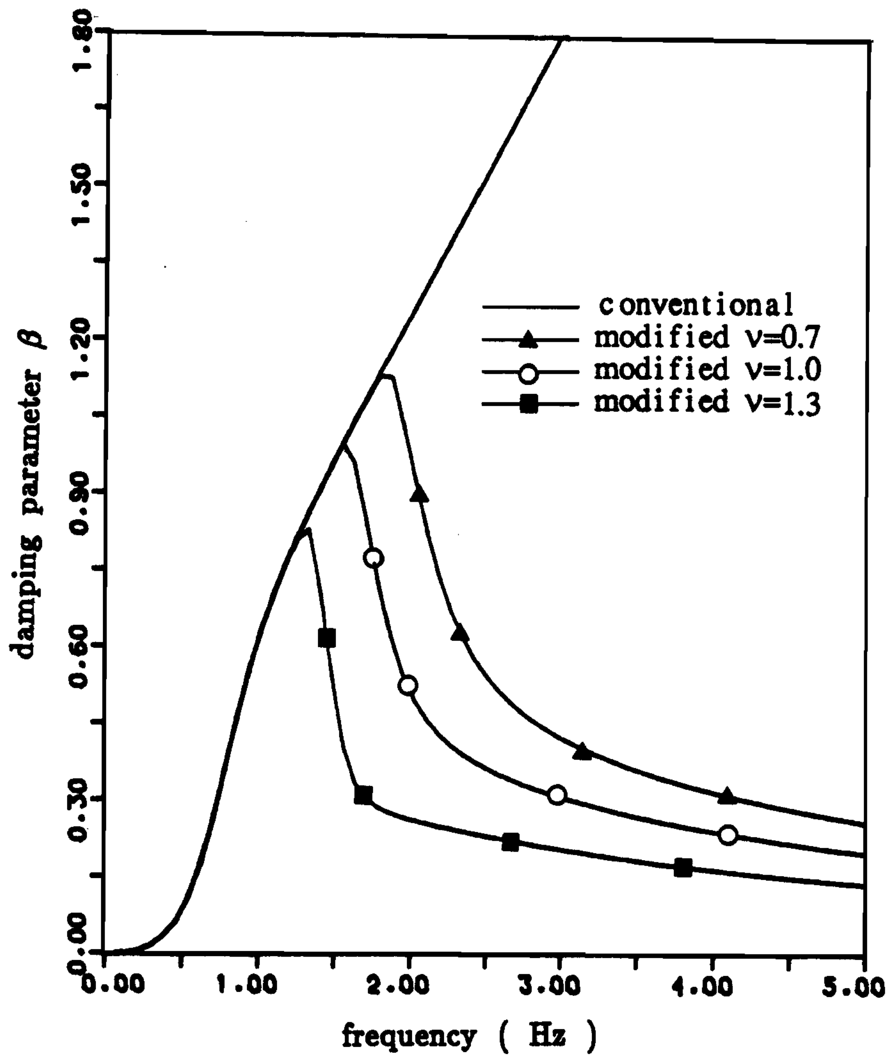

Fig.4 Damping characteristics of the conventional and modified shock absorbers (ν = 0.7, 1.0 and 1.3)

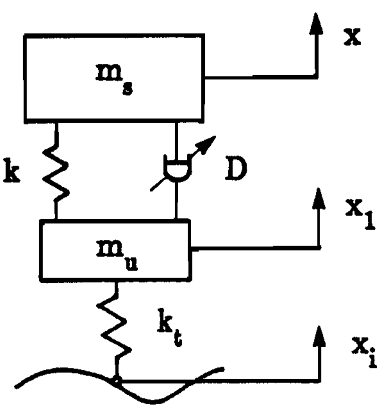

Fig.5 A quarter vehicle model

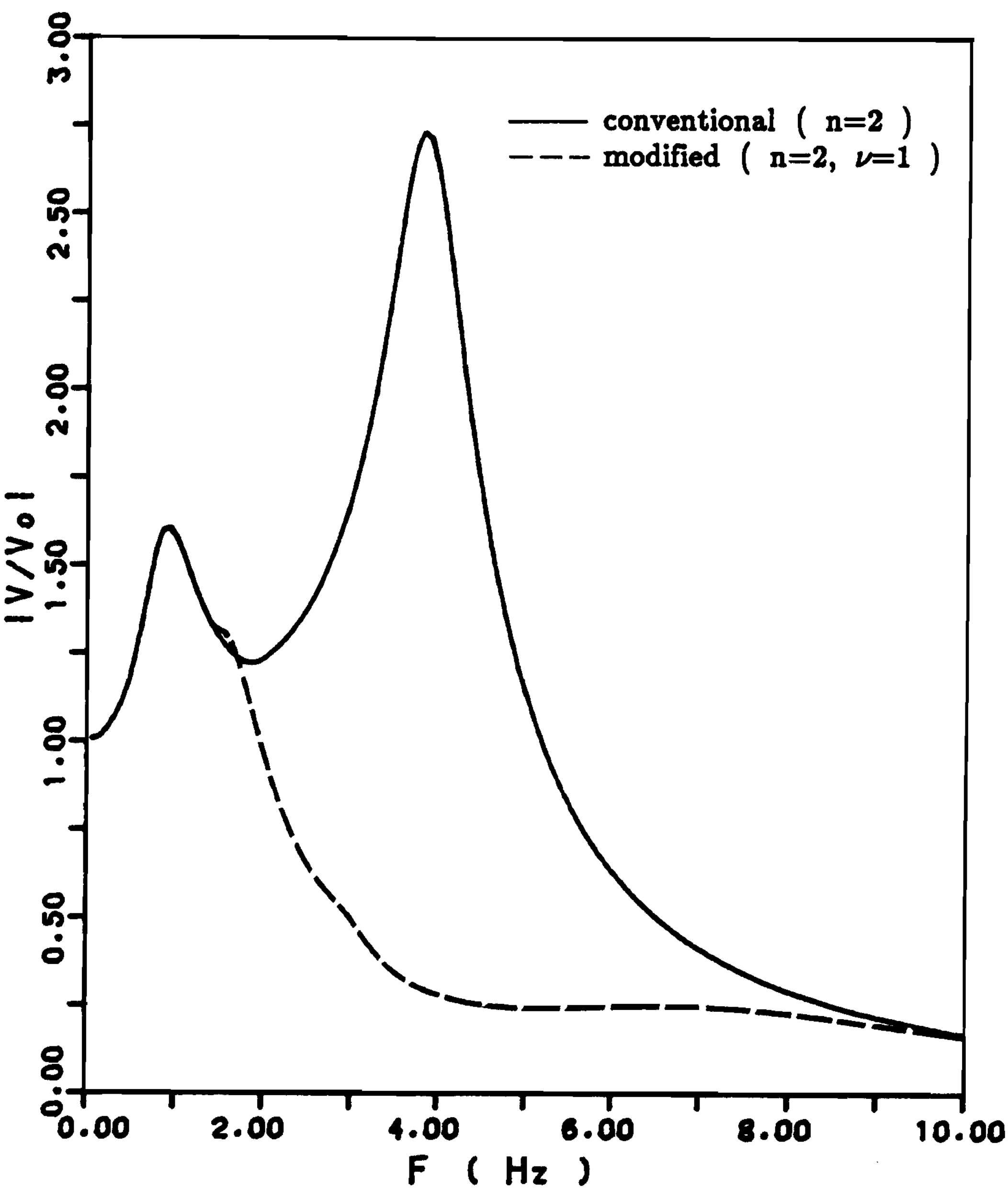

Fig.6 Velocity transmissibility characteristics of the vehicle suspension with conventional and modified dampers

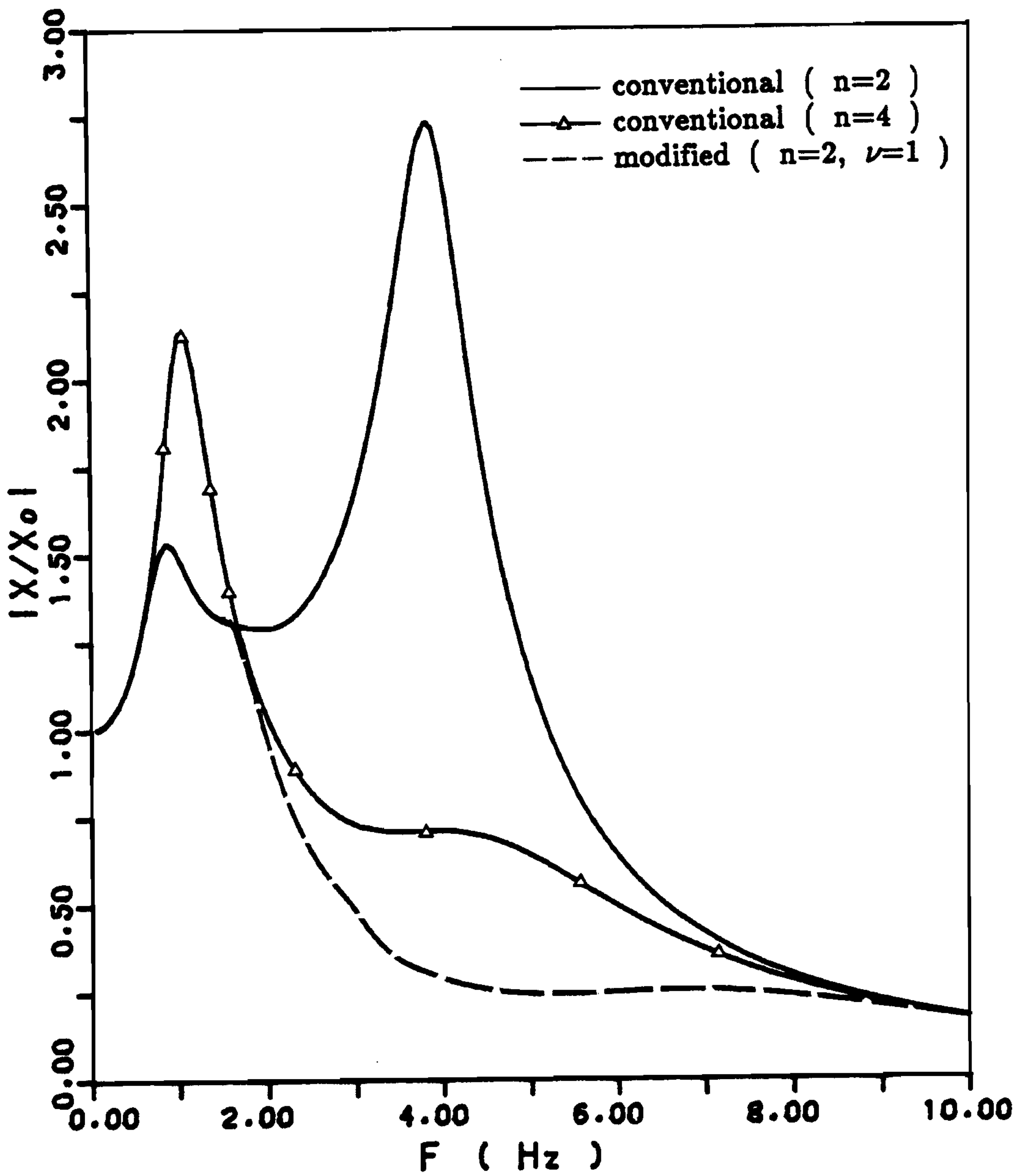

Fig.7 Displacement transmissibility characteristics of the vehicle suspension with two conventional dampers (n=2 and n=4) and a modified damper (n=2, ν=1)

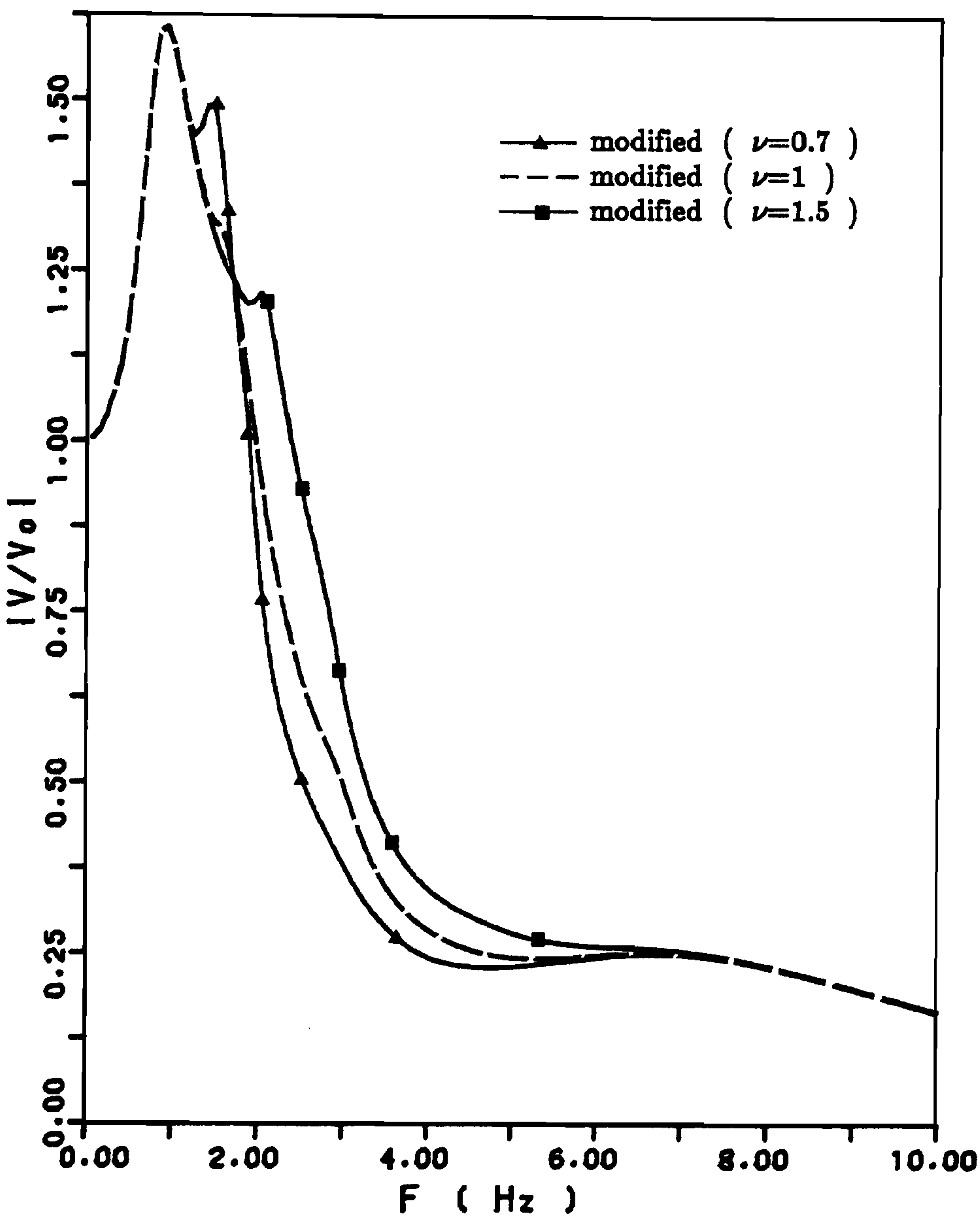

Fig.8 Velocity transmissibility characteristics of the vehicle suspension with modified damper (n=2; ν=0.7, 1.0 and 1.5)

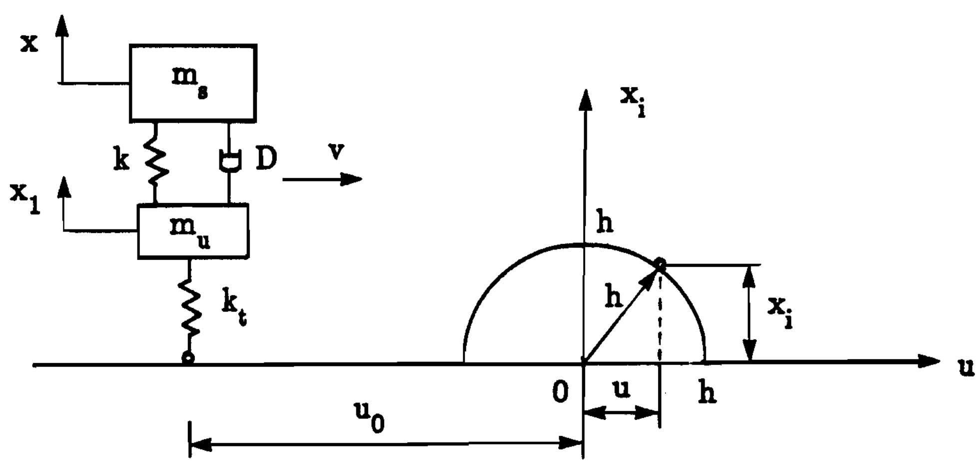

Fig.9 Representation of a bump input

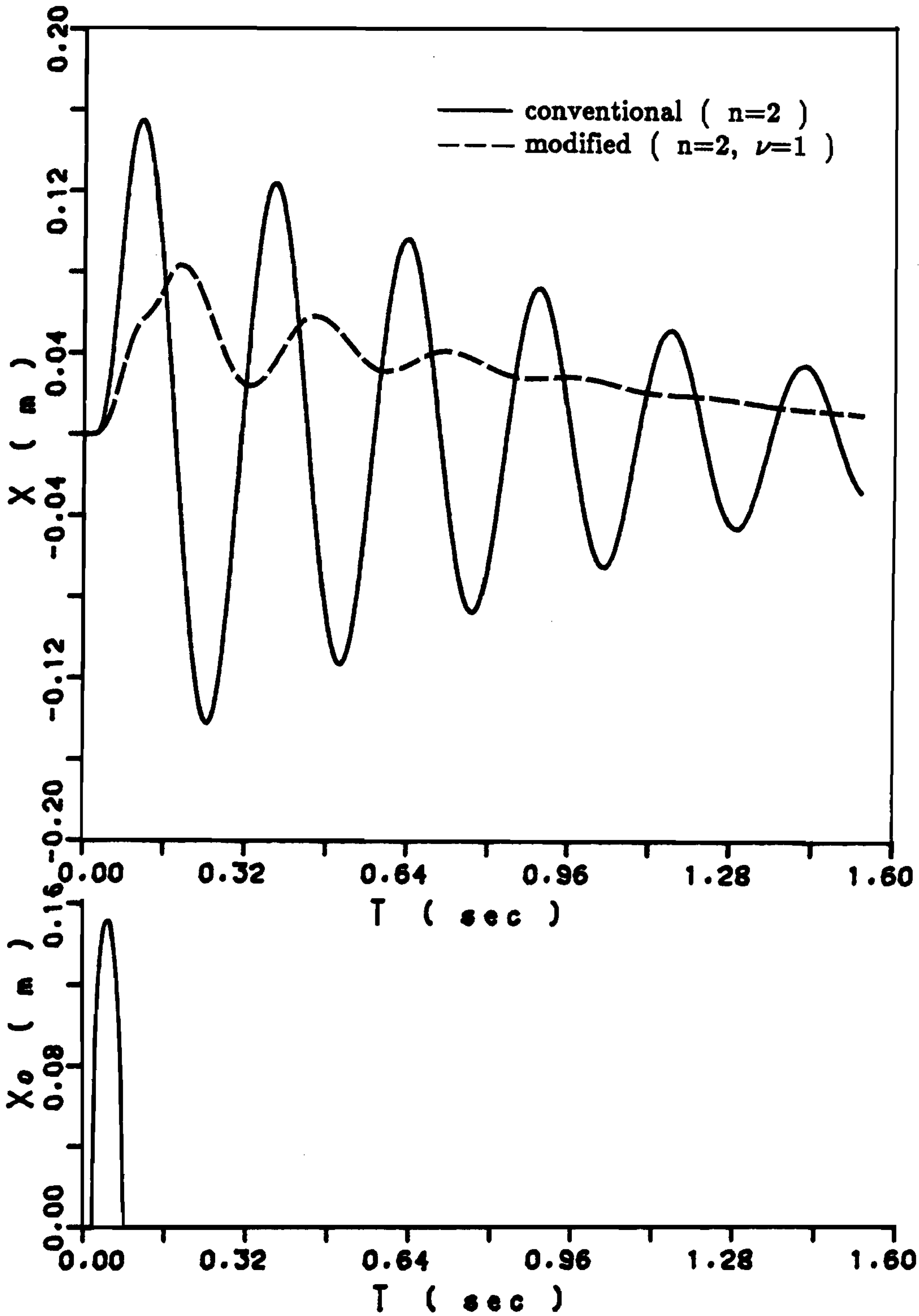

Fig.10 Transient displacement response of the sprung mass with conventional and modified dampers

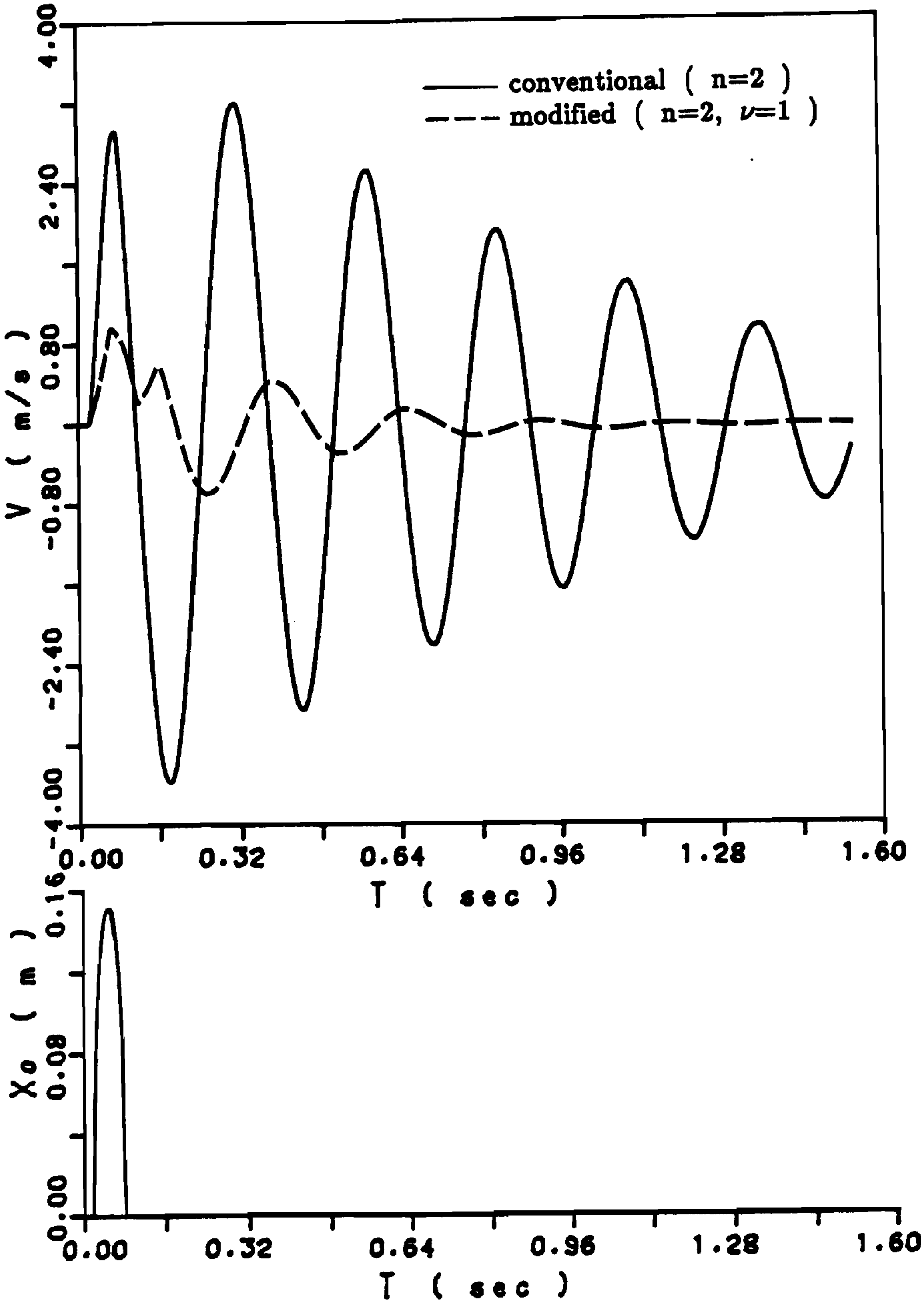

Fig.11 Transient velocity response of the sprung mass with conventional and modified dampers

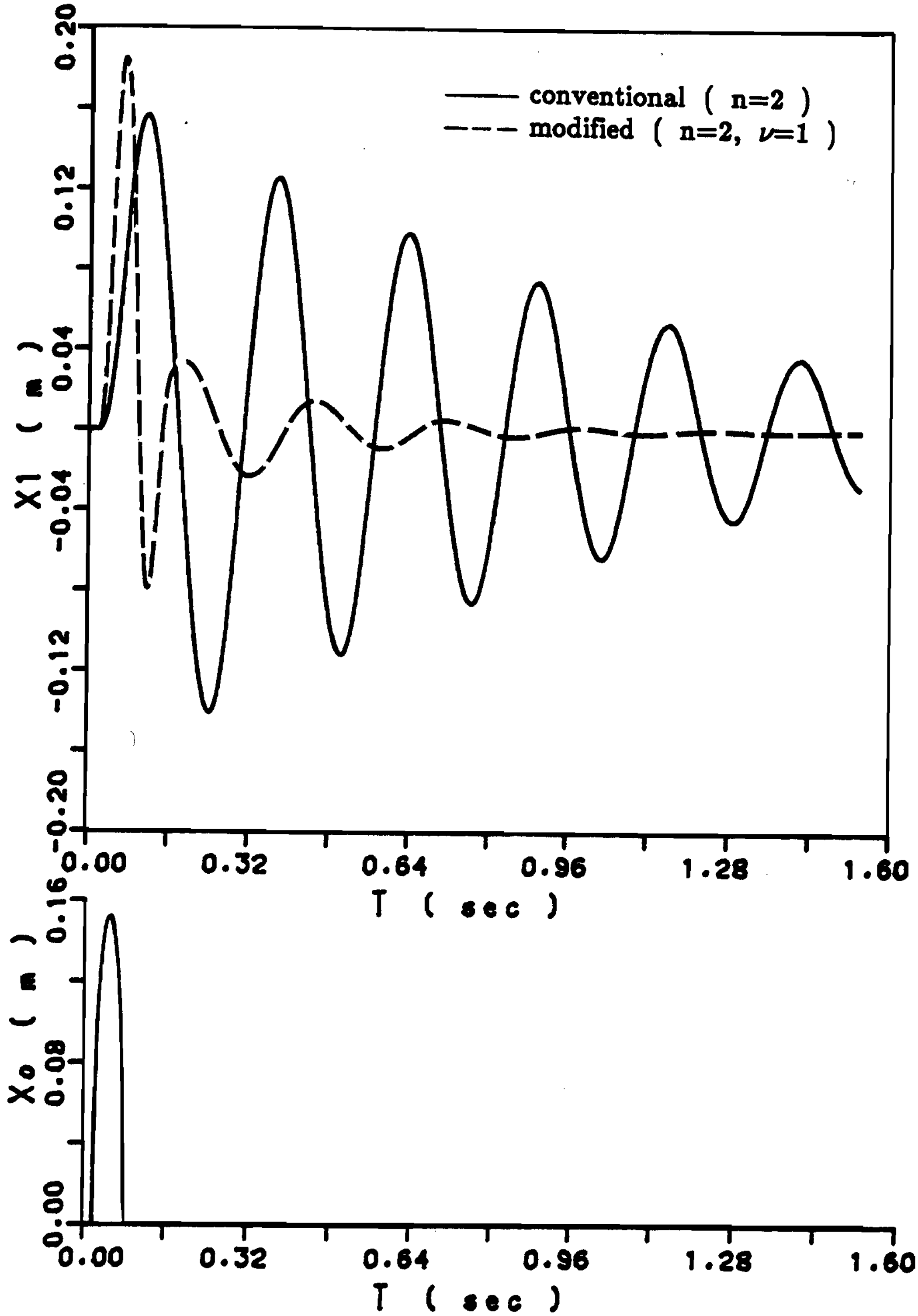

Fig.12 Transient displacement response of the unsprung mass with conventional and modified dampers

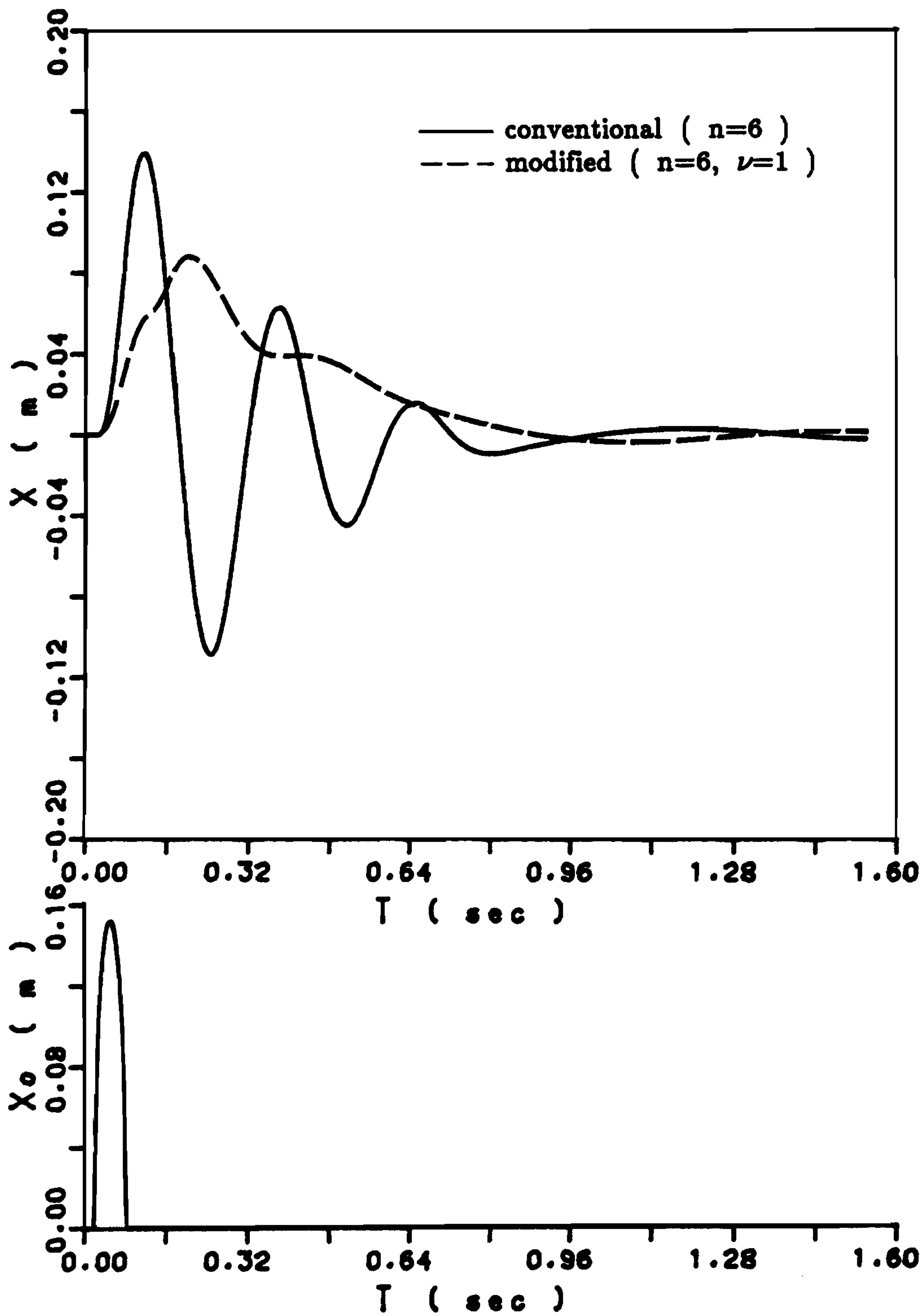

Fig.13 Transient displacement response of the sprung mass with conventional and modified dampers (n=6)