Active Seat Suspension to Control Low Back Injuries

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16. Abstract (Limit: 200 words) A prototype semi-actively controlled vehicle seat suspension system was designed, built, and tested. The seat suspension system was intended to reduce low back injuries caused by exposure to vertical vibration. The suspension system used a computer controlled continuously variable hydraulic damper and passive spring elements. The system was tested using a truck seat rigidly mounted onto an hydraulically actuate cylinder used for vibration testing; the input signal was actual road test data for a truck. A human model was used to simulate the dynamic properties of a human sitting on the seat. An overall reduction of 16% root mean square acceleration was realized with the controlled seat suspension system when compared to the same system operating in passive mode. Vibration at the human's natural frequency of 5 hertz was reduced as much as 23.3%.

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ABBREVIATIONS

a	Acceleration
C_d	Damping coefficient (Newtons x Second / Meter)
Hz	Hertz (Cycles/Second)
k	Spring constant (Newtons/Meter)
m	Mass
RMS	Root Mean Square
t	Time
v	Velocity
y	Unsprung mass displacement (Meter)
$\dot{y}$	Unsprung mass velocity (Meter/Second)
z	Sprung mass displacement (Meter)
$\dot{z}$	Sprung mass velocity (Meter/Second)

FIGURES

SIGNIFICANT FINDINGS

The fact that vibration at the human's natural frequency of 5 hz (approximately) was reduced as much as 23.3% is very important. Damage to the spine due to vibration most likely occurs from the work performed on the body by the kinetic energy from the vibration. Since kinetic energy is proportional to the square of acceleration, a reduction to 77% of the original acceleration is equivalent to a reduction to 59% of work done to the driver. This represents a significant improvement in the occupational health environment of the driver.

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ABSTRACT

The long term goal of this work is to produce an inexpensive, semi-actively controlled vehicle seat suspension system to prevent low back injuries caused by vertical vibration exposure. This phase of the project involved designing, building, and testing a prototype to prove feasibility. A computer controlled continuously variable hydraulic damper and passive spring elements were used. An overall reduction of 16% RMS acceleration was realized when compared to the same system operating in passive mode. Additionally, a 23.3% reduction in acceleration was measured at 5 Hz, the resonant frequency of a seated human. This represents a significant improvement in seat design and we feel that further improvement can be realized with additional refinement of the system.

INTRODUCTION

The operation of vehicles, e.g. trucks, has been demonstrated to be highly correlated with the occurrence of low back pain and herniated discs. Truck drivers and other vehicle operators typically report two to four times the number of low back pain problems and disabilities as the normal population. The economic and human costs of lower back injuries are enormous. Vehicle-related lower back injuries have been attributed in a large part to vibration-induced stresses in the lumbar spine. In particular, many vehicles have vibration resonances at frequencies that coincide with the 4-5 Hz fundamental resonance of seated individuals. Vibration at resonance routinely damages mechanical structures and could surely be a cause of lumbar spine damage. The development and use of a seat suspension in trucks and other vehicles that suppresses most of the injury-causing vibrations would significantly reduce the number of back injuries and related disabilities. In addition, such a seat would have the added benefit of reducing operator fatigue and possibly accident rates.

Current seat suspension technology is based on the use of passive elements, e.g. springs and shock absorbers. However, the design of a passive seat suspension is a compromise between conflicting requirements of; 1) suppressing vibrations over a wide frequency range, 2) suppressing the response to periodic, random, and shock type loading, 3) preventing bottoming out and other jerk-type events, 4) preventing large scale excursions that may adversely affect the ability of the operator to control the vehicle, 5) responding to a vertical and fore-aft loading environment, 6) accommodating operators and vehicles with different and time-varying mechanical properties, e.g. mass, damping and stiffness, 7) accommodating variations in posture, and 8) accounting for routine engineering concerns such as minimizing cost, maintenance and weight. Incorporating of all of these design requirements into a seat suspension using passive components is impractical. The resulting seat is capable of reducing vibrations somewhat, but is wholly inadequate for the prevention of long-term injury and disability.

The development of a better seat suspension requires the incorporation of active vibration control techniques where the suspension senses the vibration environment and adjusts its mechanical properties, including the use of active forces, to counteract the vibrations. The primary advantage of active vibration control is the improved level of performance. Active vibration and noise control systems have just recently become commercial engineering realities due to recent advances in transducer, microprocessor, actuator and algorithm technology (Rogers and Fuller, 1991). The design of an active seat suspension will require; 1) design of the active mechanical components, e.g. active dashpots and actuators, 2) formulation of a robust control algorithm, 3) fabrication and testing of a prototype that includes transducers, active elements and electronic controller.

The significance of this work lies in its ability to minimize one of the numerous factors (anatomic, industrial, environmental) which may influence both the severity of low back complaints, as well as the resultant disability. In particular, the focus on load exposure history as a significant risk factor has particular promise, because to some degree its effects can be minimized through control systems (for example seating design), as opposed to factors which are less controllable (for example spinal anatomy). The importance of this work is apparent when one considers that low back pain is one of man's most important disabling health problems (Kelsey et al, 1979). The medical, industrial and socio-economic consequences of low back pain (LBP) syndromes are staggering (Kelsey et al, 1979). Low back pain is the leading cause of industrial disability payments; the second most common medical cause of work loss in industry resulting in eight million workers being affected. Two million are chronically disabled from these processes. The cost of lost wages alone comes to nearly 4 billion dollars per annum (Frymoyer et al, 1983) and thus can be anticipated to be far more than that in the 1990's. Hence, even partial solutions to the low back pain problem could provide significant societal benefits. Epidemiologic work (Frymoyer et al, 1983) indicates significant association of the low back complaint with exposures to both industrial and nonindustrial (particularly vehicular) vibration. The Centers for Disease Control (1983) concur that back pain and vertebrogenic pain may be associated with whole body vibration. Kelsey et al (1984) found an increased incidence of herniated discs with long term exposure to automobiles and trucks. Gearhart (1978) found that helicopter vibration can cause changes in the musculoskeletal system. Investigators have suspected or have found association between truck or tractor vibration environments and early or unusual degenerative changes in the spine (Christ 1974, Gruber 1974, Rosegger 1970). In a previously reported study of 3,920 patients (Frymoyer et al, 1980), it was determined that the complaint of low back pain was more common in individuals exposed to vibration, e.g., truck and tractor driving and heavy construction equipment operation. Sandover (1981) also found that the seated vibration environment is associated with the production of low back pain. In the New York City Subway system, lateral vibration was associated with a high prevalence of low back pain (Johanning et al, 1991). This corroborates the effects on the spine found in other railroad environments, attributed to vertical and horizontal vibration (Arnautova-Bulot 1979, Louyot et al 1954).

There are several mechanical factors which contribute to stress the posterior portion of the intervertebral disc in the lumbar spine when one sits:

1) Sitting flattens the lumbar lordosis and shifts the line of force of the spine to a point posterior to the effective pivot point of the ischial tuberosities (Chaffin and Andersson, 1984). In a vibration environment, the load transmitted by the spine, applied along the moment arm created by the anterior offset of the ischial tuberosities, may induce an additional rocking motion in the pelvis and may amplify the vibration motion transmitted to the

spine.

2) Sitting also causes an increase in the posterior disc height which may mechanically strain the posterior and postero-lateral collagen fibers of the annulus fibrosus where they are thinner and fewer in number (Farfan 1973, Krag et al 1987, Panjabi and White 1978, Galante 1967).

3) Additional strains are created in forward flexed motion segments because the facets disengage and allow an increase in the anterior-posterior translation compliance (Panjabi et al 1977, Schultz et al 1979, Tencer et al 1982).

4) Lumbar intradiscal pressures are significantly greater in the seated posture. This has a tension increasing effect on the disc collagen fibers analogous to increasing the tension in a taut wire by pushing on it from the side (Andersson 1974, Nachemson and Morris 1964, Okushima 1970, Schultz et al 1979, Belytschko et al 1974).

5) Polymers such as rat and human medial collateral ligaments (Fung 1981) have been shown to become softer and weaker due to vibration loading (Weisman, Pope and Johnson, 1980), a fatiguing type response (Hertzberg and Manson 1980, Ridell, Koo and O'Toole, 1966).

6) Physical changes and disc herniations have been caused in motion segments by exposure to cyclic loading (Adams and Hutton 1983, Brown et al 1957, Liu et al 1983, Wilder et al 1982)

7) Whole body vibration studies have established that the seated human has a resonant frequency close to those frequencies produced in common working and vehicular environments. A structure vibrating at its resonant frequency is more likely to fail (Bastek et al 1977, Pope et al 1987, Seidel et al 1980, Wilder et al 1982). In vivo whole body vibration studies have established the motion characteristics of the lumbar region during vibration exposure (Panjabi et al 1986, Hagena et al 1985, Kaigle et al 1992).

8) A vehicle driver's disc is at risk for mechanical damage while unloading the vehicle after driving as back muscles fatigue with some vibration exposure (Wilder et al 1984, Magnusson et al 1988), lumbar balance point locations shift posteriorly following sustained sitting, thereby decreasing the erector spinae moment arm, and increasing the load requirements of the erectors and increasing the imposed loads on the disc (Wilder et al 1987), trunk fatigue results in increased coupled (out of plane) torques (Parnianpour 1988) and muscles overcompensate for unexpected loads (Marras et al 1987); all behaviors which would enhance the motion segment's tendency to buckle after seated vibration exposure (Wilder et al 1988).

9) The combination of imposed lateral bend vibration and the flexion-extension response due to vertical vibration poses the most severe mechanical environment for the lumbar disc (Wilder et al 1990, Wilder et al 1982), due to the greater potential for stretching the postero-lateral region of the disc with its subsequent mechanical fatigue.

The successful design and eventual routine use of an active vibration suppression seat suspension has the potential to reduce significantly the number of lower back injuries experienced by truck drivers and other vehicle operators. Concentrating on seat suspensions rather than the entire vehicle suspension has the advantage that existing vehicles can be retrofitted without changing the vehicle suspension and that the power requirements are minimized.

The current truck seat manufacturing industry consists of several small specialized companies that sell seats as installable components for trucks. Active seat suspensions have not been designed for commercial use to date. Many of the technologies that are required to implement such a system are available in various forms and would require modification/adaption. The market is such that a small company with a superior product could find a profitable niche. The magnitude of the economic and human costs of low back pain are very high. A product that reduces lower back injuries, such as the proposed active vibration suppressing truck seat, could find a large, profitable market.

METHODS

I. Control Strategy

The successful design of a semi-active suspension system must foremost have an effective control scheme as well as components capable of carrying out those control functions. There exist at least three different types of control strategies.

The first is simply an on-off scheme. Here the damping force is either on or off, depending upon the directional relationship between the absolute velocity of the sprung mass and the relative velocity of the sprung to unsprung masses.

The second method is discrete damping control. The damping force is either soft or firm. Input criteria includes the absolute velocity of the sprung mass (above or below a threshold value) and whether the mass is in compression or rebound mode.

Continuous control represents the third level of control. For this strategy the damping force is continuously adjusted to maximally damp vibration to the sprung mass. It uses relative position and velocity information of the sprung to unsprung masses as control input.

We originally began by computer modelling the system and eventually came to the conclusion that modelling each of these control strategies would easily use up the allotted labor costs. At about this same time a report (J.Y.Wong et al, 1992) was found that had experimentally compared these three control strategies using a quarter-car model. Each control method was compared to that of a passive damper system with a damping ratio of 0.7. Acceleration RMS was used as their outcome measure. For two different vehicle speeds, 50 and 100 km/h, the continuous control strategy was found to be more effective in reducing sprung mass vibration than the other two methods. This result led us to develop a system which would use the continuous control strategy. This system is also capable of implementing either of the other two strategies if desired, although these have not yet been tested.

The continuous control strategy can be described as follows:

If $(\dot{z}-\dot{y})(z-y) > 0$, then minimum damping ($Cd=0$) is required.
If $(\dot{z}-\dot{y})(z-y) < 0$, then the desired damping is calculated from
$C_d = -k(z-y)/(\dot{z}-\dot{y})$, where

z and y are the displacements of the sprung and unsprung masses, respectively; $\dot{z}$ and $\dot{y}$ are the velocities of the sprung and unsprung masses, respectively; k is the suspension spring rate; and C_d is the desired damping coefficient.

This control strategy indicates that if the spring force and damping force exerted to the sprung mass are in the same direction, to reduce the sprung mass acceleration, the damping force should be zero. On the other hand, if the spring force and

damping force are in the opposite direction, then the damping force should be made equal to the spring force in order to cancel it and produce zero acceleration of the sprung mass.

II. Mechanical Design

A prototype was constructed by modifying an existing spring-damper equipped seat (Sears Manufacturing, Davenport,Iowa). The sprung portion of the seat has a mass of 27.2 kg. The original seat spring had a spring constant (k) of 7364 N/M and was incorporated into the current design. The passive damper was replaced by a standard hydraulic cylinder with a 7/8 in. bore by 4 in. stroke (Air Inc., Franklin, Mass.). The ports at each end of the cylinder are connected by tubing to a hydraulic valve. Variable damping is accomplished by varying the orifice size that the fluid must pass through as the shaft of the cylinder extends or retracts. A digitally controlled servo valve (Olsen Controls, Bristol, CT) with a rated flow of 45 gpm at 1000 psi is used as the orifice for the variable damper. This variable damper system was tested and calibrated on a load test frame (MTS Inc., Minneapolis, MN) to determine the curve of damping coefficient (C_d) vs. valve position. An equation containing both linear and exponential terms was fit to the curve and then used in the control software to calculate valve position for the desired C_d (see Figure 1.). A linear potentiometer (ETI Systems, Carlsbad, CA) records relative displacement between the seat (sprung mass) and seat base (unsprung mass).

The control operations run on a 80386 based PC using a digital to analog board to aquire position data of the seat at a sampling rate of 400 hz. Seat position data is differentiated and filtered to provide relative velocity information. This information is then fed into the control equations in order to calculate the desired C_d.

III. Testing

The truck seat was rigidly mounted onto an hydraulically actuated cylinder (MTS model 248.10, MTS Inc., Minneapolis, MN) which is specifically setup for vibration testing. Actual road test data previously recorded at the floor of a truck was used as the input signal.

To simulate the dynamic properties of a human sitting on the seat a simulated human model was built. It is a mass-spring-mass system with equal masses, a total mass of 82 kg, and a resonant frequency of 4.5 hz. The base, or lower mass, of the model was placed on the seat during dynamic testing.

Accelerometers (PCB Piezotronics Inc., Depew, NY) were attached to the seat base and to the lower mass of the human model. This acceleration data was also recorded at 400 hz and provided the data for comparative analysis of the system. A signal analyzer (Wavetek-Rockland, Rockleigh, New Jersey) was used to analyze the data (see Figure 2.).

Two test conditions were performed. The first was with semi-active control on. The second run was with active control off (passive damping) and the damper set to a constant damping ratio of approximately 0.7.

RESULTS/DISCUSSION

The seat exhibited a frequency response primarily in the 0 to 20 Hz range, with little or no transmitted energy above 20 Hz. Therefore, our analysis was focused on frequencies in this range only. A power spectral analysis revealed a decrease in total acceleration RMS of 16.0% from the semi-active suspension to that of the passive suspension. Specifically, at 5 Hz an analysis of the transfer function between the seat base and human base shows a reduction in gain of 23.3%. The greatest reduction in acceleration appears to be in the 2 to 10 Hz range (see Figure 3.). Interestingly, the gain at 1.4 Hz actually increased slightly, although the bandwidth of the resonance curve at this frequency narrowed.

The overall reduction in energy transferred to the seated human model is very encouraging. Where Wong et al, 1992 showed a decrease of around 40% for their quarter car suspension model, our overall reduction of 16% leads us to believe that with a little fine tuning of the system we can further reduce the transmitted vibration.

The fact that vibration at the human's natural frequency of 5 Hz (approximately) was reduced as much as 23.3% is very important. If one considers that the damage to the spine due to vibration occurs from the work performed on the body by the kinetic energy from the vibration, two things become very clear. In the simplest form, the work performed on the body is equal to the kinetic energy applied to the body. The equation for that kinetic energy is $1/2\ mv^2$ (work = kinetic energy = $1/2\ mv^2$). Because velocity = acceleration x time ($v = at$), solving that equation in terms of acceleration and time yields an equation where kinetic energy equals $1/2\ m(a^2t^2)$. Using that formulation, it is then apparent that the work on the body from the kinetic energy of vibration can be reduced to the square of the reduction in the acceleration applied to the body. So if an acceleration applied to the body is reduced by roughly 23% to 77% of its original amount, then, all other things being equal, the amount of energy, and hence work being done on the body, is at only 59% of the original work with the higher acceleration. This is significant because small changes in acceleration level can lead to large changes in the total work performed on the body or the total energy imparted to the body.

The power spectral analysis shows that a large portion of the energy occurs in the 8 to 14 Hz range. Preliminary load frame testing of the servo-valve controlled damper showed it to be capable of achieving full range of effective damping in about 50 ms, or 20 Hz. Our control scheme requires the damper to instantaneously open or close as often as it requires a graded position of the valve. This hardware limitation may be preventing the system from attenuating these higher frequencies. The valve manufacturer (Olsen Controls) has available a servo motor that produces higher torque than the current design. They indicate that this motor should substantially reduce response time of the valve. This improvement should allow vibration attenuation at these higher frequencies and greater attenuation at the lower frequencies as well.

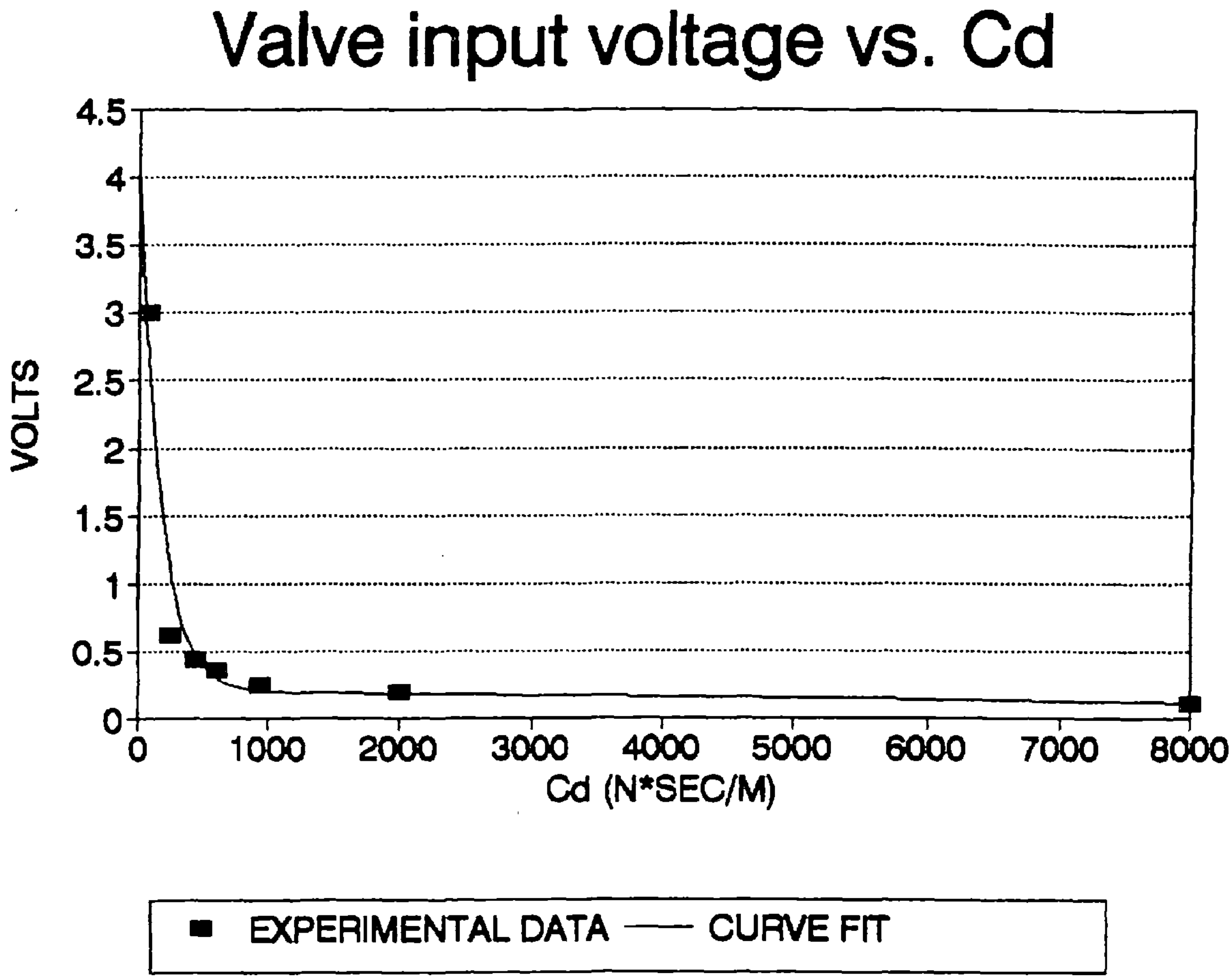
The truck seat which we modified was originally designed to resonate at about 1.4 Hz. This was done to shift resonance away from 5 Hz, the resonance of a seated human, and is the current state of the art in truck seating. We observed that our system slightly increased the transfer gain at this frequency, and why this occurs is not clear. This phenomenon will be investigated further in ongoing development.

Although the results are not available at this time we are currently in the process of incorporating a load cell into the variable damper in order to record the forces transmitted by it to the seat. This will facilitate calibration of the valve during operation, rather than in a load frame. Not only will this allow fine tuning of the damper, we will also be better able to understand the response characteristics and requirements of the system.

CONCLUSIONS

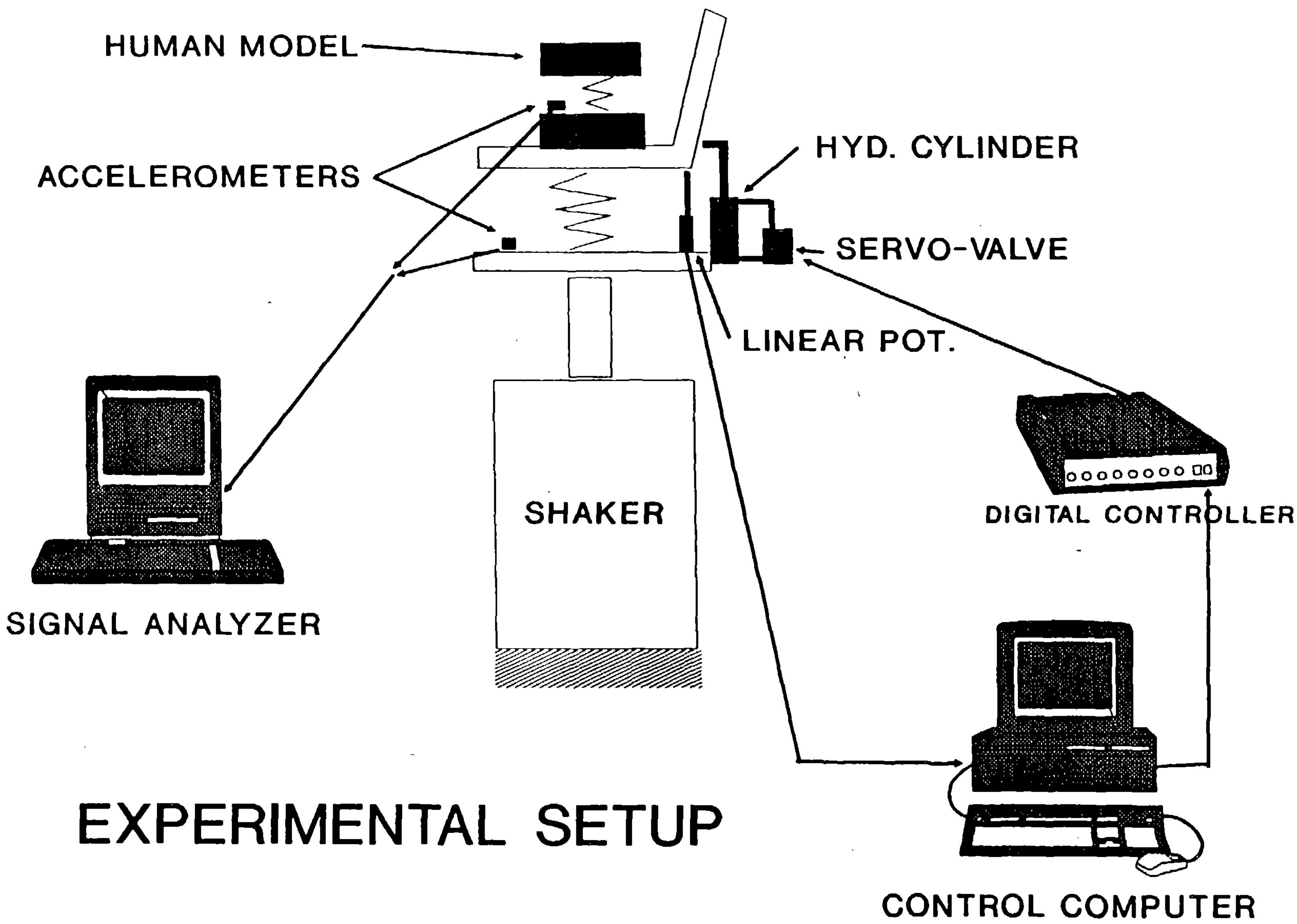
The prototype for a semi-active vibration damping truck seat was successfully designed, built, and tested. Preliminary testing showed the system to be effective in reducing overall vibration transfer to the simulated driver by 16%. Equally as important, vibration at the resonant frequency of a seated human (5 hz) was reduced by 23.3%, which corresponds to a 41% reduction in the kinetic energy transmitted to the occupant of the seat being vibrated. It is our belief that these promising results and the vehicle industry's need for such a seat warrants further development of this system and that even better performance can be obtained.

FIGURE 1.



Valve input voltage vs. Cd
VOLTS
4.5
4
3.5
3
2.5
2
1.5
1
0.5
0
0
1000
2000
3000
4000
5000
6000
7000
8000
Cd (N*SEC/M)
EXPERIMENTAL DATA
CURVE FIT

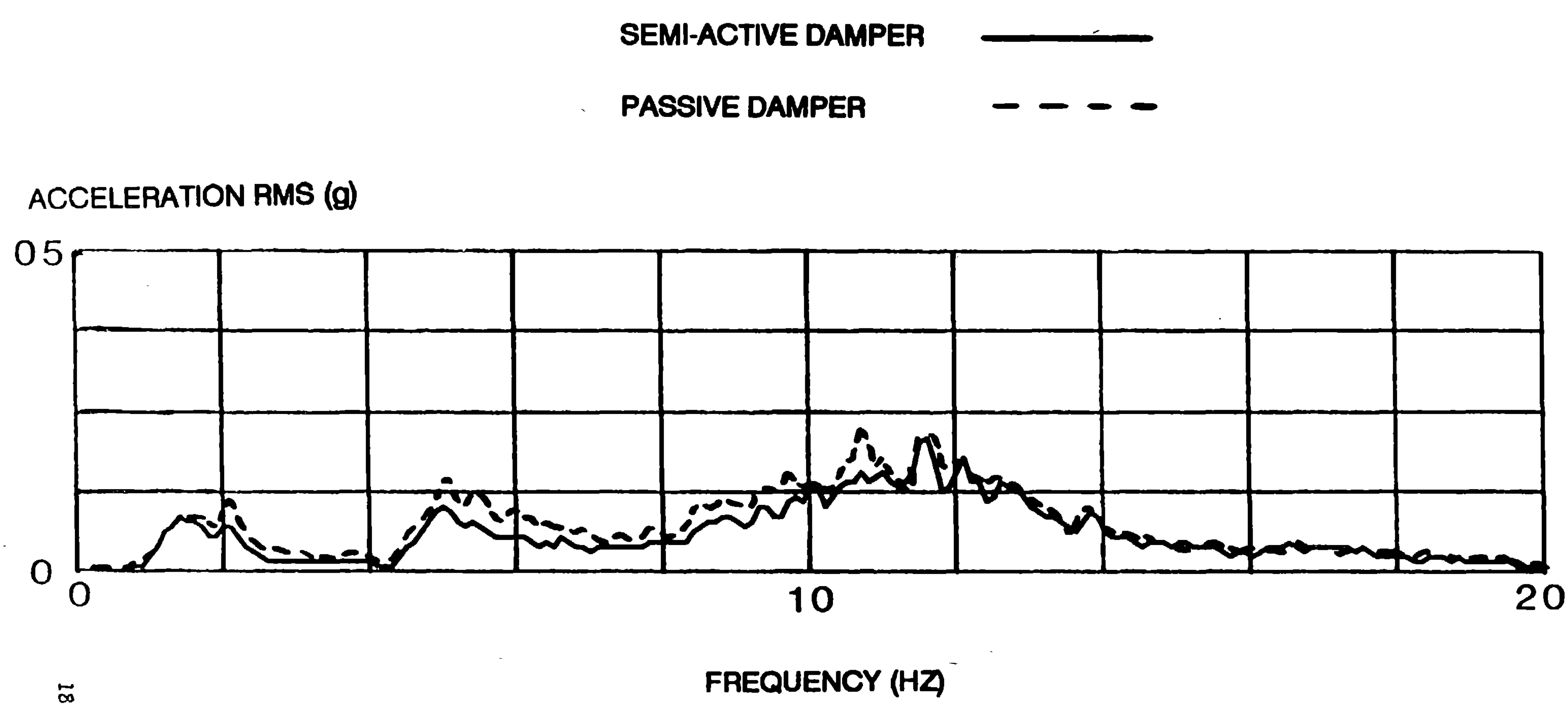
FIGURE 2.



EXPERIMENTAL SETUP

FIGURE 3.

POWER SPECTRAL DENSITY



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PUBLICATIONS

No publications have been submitted, and will not be until the patent licensing process allows it.