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Active Noise and Vibration Control for Vehicular Applications

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Abstract

This is the final report of a one-year, Laboratory Directed Research and Development (LDRD) project at the Los Alamos National Laboratory (LANL). This project investigated semi-active suspension systems based on real time nonlinear control of magneto-rheological (MR) shock absorbers. This effort was motivated by Laboratory interactions with the automobile industry and with the Defense Department. Background research and a literature search on semi-active suspensions was carried out. Numerical simulations of alternative nonlinear control algorithms were developed and adapted for use with an MR shock absorber. A benchtop demonstration system was designed, including control electronics and a mechanical demonstration fixture to hold the damper/spring assembly. A custom-made MR shock was specified and procured. Measurements were carried out at Los Alamos to characterize the performance of the device.

Background and Research Objectives

Advances in digital signal processing hardware and sensor/actuator technology are making active noise and vibration control (ANVC) systems practical for a wide range of vehicular applications. Successful application of ANVC requires a synergistic combination of technical disciplines including structural analysis, mechanical design, adaptive control, digital signal processing, and sensor/actuator design. The ability to model the physical system, simulate its interaction with alternative control strategies, and relate these analyses to real world results is crucial. Los Alamos has all of the ingredients necessary to make a significant technical contribution to vehicular applications of ANVC. This is particularly true in those areas where modeling and simulation can make a difference. The objective of this program development project was to marshal and augment existing Laboratory expertise in order to develop a coherent and demonstrable capability that could be applied to transportation problems. To achieve this end, we focused our efforts on a carefully selected demonstration problem—a small scale “idealized” version of a semi-active vehicle suspension.

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Semi-Active Vehicle Suspensions

A *fully active* ANVC system cancels unwanted vibrations by applying opposing energy of equal magnitude. While this approach offers optimum performance, power requirements make it unattractive in high energy situations such as vehicle suspensions. In these cases, a more attractive alternative is *semi-active* control. A semi-active system does not directly generate oppositional energy. Instead, it selectively modulates the properties of a passive energy absorber to reduce the overall level of vibration. The key to semi-active control is that the modulation is performed on a cycle-by-cycle basis, at a rate equal or greater than the frequency of the unwanted vibration. The resulting control problem is nonlinear.

The fundamental task of a vehicle suspension system is to isolate a mass (the vehicle body) from motions induced by roadway irregularities while still allowing the vehicle to follow the road course. This translates to a suspension response that provides isolation at higher frequencies and coupling at lower ones. Standard vehicular suspensions employ passive components and are commonly based on a parallel combination of a spring and damper (shock absorber). To a first order approximation, the response is that of a passive linear system, as illustrated in Figure 1a. In this response, there is an obvious trade-off between isolation at higher frequencies and performance near the natural frequency of the suspension. A more desirable response is obtained from the idealized passive "skyhook" suspension, shown in Figure 1b. In this case, good high frequency roll off can be obtained without a corresponding resonance response near the natural frequency. While this configuration is physically unrealizable, it represents the basic goal of an active or semi-active suspension.

In a semi-active suspension, the passive damper (shock absorber) is replaced by a dynamically controllable shock absorber. With the proper nonlinear control algorithm, the response of a semi-active suspension can in theory come close to the ideal "skyhook" behavior.

Magneto-Rheological (MR) Shock Absorbers

A limiting factor in semi-active suspension development has been the controllable shock absorbing element. Initial attempts at implementing a controllable shock used hydraulic dampers with adjustable mechanical valves, an extension of tunable shock absorber technology. However, these devices cannot respond fast enough to be controllable on a cycle-by-cycle basis. More successful prototype devices have been demonstrated using electro-rheological (ER) fluids in which the viscosity can be controlled

by an applied electric field. While ER shocks have the necessary response speed, they require the application of very high voltages, are limited in their achievable yield stress, and are very sensitive to impurities in the ER fluid.

The most promising implementations of controllable shock absorbers employ newly developed magneto-rheological (MR) fluids in which the viscosity can be controlled by an applied magnetic field. An MR fluid consists of micron sized magnetically polarizable particles suspended in oil. In the absence of a magnetic field, the particles align randomly and produce little or no shear resistance. When a magnetic field is applied, the particles begin to align and exhibit shear force proportional to the field strength (until all are aligned and saturation is reached). MR devices can develop resistance forces up to two orders of magnitude higher than ER devices, and are current rather than high voltage driven. As such they appear to be very well-suited to vehicular applications.

Importance to LANL's Science and Technology Base and National R&D Needs

Vibration and noise abatement is a key area of interest for the Partnership for a New Generation Vehicle (PNGV). PNGV is a government/industry consortium involving several federal agencies, including the Department of Energy (DOE), and the US automobile industry. Industry's US Council for Automotive Research (US CAR) is supportive of research in this area. In March 1996, the First Invitational Simultaneous Vehicle and Infrastructure Design (SVID) Workshop was held in Dearborn, MI, under the auspices of US CAR and the Alliance for Transportation Research. Participants at this workshop included representatives from the auto industry, federal transportation agencies, and the DOE national laboratories. At the workshop, noise abatement and the interaction of vehicle suspensions and pavement surfaces was identified as one of four major areas in which SVID could make a difference.

There is additional potential in the application of ANVC to payload isolation—minimizing the roadway induced forces on sensitive or high value cargo. This is directly applicable to critical DOE interests in the transportation of hazardous waste and nuclear material.

The Department of Defense is also actively pursuing related technology through such programs as the Advanced Research Projects Agency Active Control program. Their thrust is to address both defense and domestic problems in areas such as vehicle stealth, precision machining, and smart weapons.

Scientific Approach and Accomplishments

Efforts were focused on the investigation of semi-active suspension systems based on real time nonlinear control of magneto-rheological (MR) shock absorbers, modeling and simulation of a simplified one-degree-of-freedom system, the design of a benchtop demonstration, and the evaluation of a commercial prototype MR device. The following tasks were completed.

- Background research and literature search on semi-active suspensions and controllable MR shock absorbers [1-20].
- Numerical modeling and simulation of alternative nonlinear control schemes and adaptations for their use with an MR shock absorber. Three control algorithms described in the literature were investigated.
- Specification, procurement, characterization, and evaluation of a custom-built commercial prototype MR shock. While the measured performance was satisfactory, the device was repeatedly plagued by MR fluid leakage and was unsatisfactory for extended use.
- Design of a benchtop demonstration system, including control electronics and the mechanical fixture to hold the damper/spring assembly. The benchtop demonstration system was not implemented due to continuing technical problems with the MR shock unit and because of the mid-FY96 decision to discontinue this effort in FY97.
- Attendance at the March 1996 SVID Workshop in Dearborn, MI.

Numerical Modeling and Simulation

System and control algorithm simulations were carried out with the Mathworks Simulink simulator running in a Matlab environment. Three control approaches taken from the literature were adapted to incorporate MR shock characteristics. Each approach was simulated and its response characteristics compared to both the passive and idealized responses.

The response for a basic one degree of freedom (1DOF) linear passive suspension model for a spring/shock absorber combination is $m\ddot{y} = -c(\dot{y} - \dot{x}) - k(y - x)$ (Figure 1a). The response for the idealized "skyhook" suspension is $m\ddot{y} = -c\dot{y} - k(y - x)$ (Figure 1b). An ideal active suspension can generate force in either direction without limitation. Since the $-k(y - x)$ term can be implemented with a passive spring, the model used for both

active and semi-active 1DOF suspensions was $m\ddot{y} = -k(y - x) + f$. For the *active* case, f is unrestricted, so to implement the skyhook, $f = -c\dot{y}$. In the *semi-active* case, f can only act in the direction that is in opposition to a change in the relative displacements of x and y . In other words, $\text{sgn } f = -\text{sgn}(\dot{y} - \dot{x})$.

The first approach to semi-active control (SA-1) is to approximate the skyhook. In this case the virtual damping is applied when the required force is compatible with the limitation of the semi-active element and turned off when it is not. The force relation for SA-1 is

$$f = \begin{cases} -c\dot{y} & \text{sgn } \dot{y} = \text{sgn}(\dot{y} - \dot{x}) \quad \text{or } \dot{y}(\dot{y} - \dot{x}) > 0 \\ 0 & \text{otherwise} \quad \text{or } \dot{y}(\dot{y} - \dot{x}) \leq 0 \end{cases}$$

A disadvantage of this approach is that it requires the measurement of the absolute velocity $\dot{y}$.

A second approach (SA-2) is to control the force of the semi-active element so that it only opposes the spring force and never augments it. When the working direction is opposite of the spring, the force is proportional to the spring. When the working direction is in concert with the spring, no force is applied. This is in effect a minimization of the force applied to the mass. The force relation for SA-2 is

$$f = \begin{cases} \alpha k(y - x) & \text{sgn}(y - x) = -\text{sgn}(\dot{y} - \dot{x}) \quad \text{or } (y - x)(\dot{y} - \dot{x}) < 0 \\ 0 & \text{otherwise} \quad \text{or } (y - x)(\dot{y} - \dot{x}) \geq 0 \end{cases}$$

where α is a gain term between 0 and 1. The effect of the SA-2 approach is to produce an undamped sprung system in which the spring constant switches between k and $(1 - \alpha)k$. An advantage of this approach is that only the relative quantities $\dot{y} - \dot{x}$ and $y - x$ have to be measured. A disadvantage is poor performance near the natural frequency due to the lack of damping.

A third approach (SA-3) tries to improve natural frequency performance of SA-2 by adding damping in parallel to the semi-active force element. The force relation for SA-3 is

$$f = \begin{cases} \alpha k(y - x) + C_x(\dot{y} - \dot{x}) & \text{sgn}(y - x) = -\text{sgn}(\dot{y} - \dot{x}) \\ C_x(\dot{y} - \dot{x}) & \text{otherwise} \end{cases}$$

If the damping term $C_x(\dot{y} - \dot{x})$ is implemented by placing a passive damper in parallel with the spring and active element, then the SA-3 control law is identical to that of SA-2.

Numerical simulations were used to compare the performance of MR-based semi-active suspensions with both the linear passive suspension and the ideal linear fully active suspension. In the simulations, the semi-active suspension models were driven with sinusoidal inputs and the ratio of the output RMS velocity to the input RMS velocity was computed. These simulations were repeated at 100 frequencies, logarithmically distributed over the range of 0.2 to 100 Hz. The result is an RMS transmissibility graph that can be compared with the system responses of the two linear systems. Since the semi-active systems are nonlinear, these transmissibility graphs are not system responses, but rather just responses to a particular class of input, namely sinusoids.

These results confirmed that the SA-1 approach produces a reasonably good approximation of the ideal skyhook (or fully active) response, over the complete frequency range. The SA-2/SA-3 response is similar to the passive linear suspension above the natural frequency. The semi-active control only affects the response below the natural frequency by reducing the resonance peak height for lightly damped systems. In this way, high frequency performance can be gained by using a more lightly damped system without suffering the penalty of excessive response near the natural frequency.

The most common semi-active element implementation is the variable damper. In that case, with the control variable represented by $D(t)$, the element force is $f = -D(t)(\dot{y} - \dot{x})$.

However, the magneto-rheological damper is a variable friction device. In the MR case, with the control variable represented by $F(t)$, the element force is $f = -F(t)\text{sgn}(\dot{y} - \dot{x})$.

The following table relates the variable damper and variable friction element control variables to the three semi-active control approaches.

Control Approach	Variable Damping $f = -D(t)(\dot{y} - \dot{x})$	Variable Friction $f = -F(t)\text{sgn}(\dot{y} - \dot{x})$
SA-1: $f =$ $\begin{cases} -c\dot{y} & \dot{y}(\dot{y} - \dot{x}) > 0 \\ 0 & \dot{y}(\dot{y} - \dot{x}) \leq 0 \end{cases}$	$D(t) =$ $\begin{cases} \frac{c\dot{y}}{\dot{y} - \dot{x}} & \dot{y}(\dot{y} - \dot{x}) > 0 \\ 0 & \dot{y}(\dot{y} - \dot{x}) \leq 0 \end{cases}$	$F(t) =$ $\begin{cases} c \dot{y} & \dot{y}(\dot{y} - \dot{x}) > 0 \\ 0 & \dot{y}(\dot{y} - \dot{x}) \leq 0 \end{cases}$
SA-2/SA-3: $f =$ $\begin{cases} \alpha k(y - x) & (y - x)(\dot{y} - \dot{x}) < 0 \\ 0 & (y - x)(\dot{y} - \dot{x}) \geq 0 \end{cases}$	$D(t) =$ $\begin{cases} \frac{-k(y - x)}{\dot{y} - \dot{x}} & (y - x)(\dot{y} - \dot{x}) < 0 \\ 0 & (y - x)(\dot{y} - \dot{x}) \geq 0 \end{cases}$	$F(t) =$ $\begin{cases} -k y - x & (y - x)(\dot{y} - \dot{x}) < 0 \\ 0 & (y - x)(\dot{y} - \dot{x}) \geq 0 \end{cases}$

It is evident from this table that in the 1DOF suspension application a variable friction device, such as an MR shock absorber, has significant advantages over a variable damping device. In all three control approaches, passive damping behavior—the proportionality of the force derived to $(\dot{y} - \dot{x})$ —is not desired. In the variable damper, this behavior is eliminated by a control law inversely proportional to $(\dot{y} - \dot{x})$, a condition leading to instability when $\dot{y} \approx \dot{x}$. In contrast, the control law for the variable friction device only requires an absolute value.

Benchtop Demonstration System Design

The mechanical design of the benchtop demonstration and the idealized situation it represents is shown in Figure 2. This design simulates single degree of freedom linear dynamic motion for study of the MR shock and dynamic control algorithms. The design is very versatile, easily accommodating assorted springs, fixed masses, and shocks. The system natural resonant frequencies are readily tuned by choosing the appropriate spring and associated fixed mass. The design is easily upgradable with additional control arms. The system is excited with a servo motor and eccentric cam. Variable motor speed offers fully variable input frequency. Different cams of varying eccentricity are utilized for different sinusoidal input amplitudes. A separate cam follower with uniquely shaped cams may be utilized to supply non-sinusoidal input as well.

The design for the control electronics is based on an Innovative Integration PC31 digital signal processing board running on a host PC486 computer. The PC31 includes a Texas Instrument TMS320C31 DSP processor and associated analog interface circuitry. This system supports software (C) implementation of the required real time nonlinear control algorithms. Acceleration, velocity, and position sensors are used to provide feedback to the control processor on the state of the system.

MR Shock Absorber

A custom-made MR shock was specified and procured from MR Technologies. (MR shocks are still a developing technology and are not readily available.) Measurements were carried out at Los Alamos to characterize and evaluate the performance of the device. The results of this characterization are shown in Figure 3, which plots measured force verse displacement for the experimental MR shock. Force was measured while actuating the shock at a constant velocity of 4.375 in./sec. Data was collected for six different piston winding voltages.

The data shows, for a given voltage and velocity, that the generated force remains constant, regardless of displacement. Minor non-linearity is encountered at the ends of each curve as the stroke direction is reversed. The compression force for a given voltage is greater than the corresponding extension force due to particular piston geometry. This characteristic is easily compensated for with the control system. Magnetic saturation of the MR fluid is evident at the higher voltages depicted on the graph. The higher voltage curves approach the limiting extension and compression forces that may be generated with this MR fluid and this particular piston design. This data was collected after cycling the device for many cycles to raise the fluid temperature to its nominal operating temperature of 135°F. Slight differences in performance were noted when the shock was operated cold. This characteristic is attributed to the temperature dependence of MR fluid behavior.

While the performance measured was satisfactory, the MR shock was plagued by excessive leakage of the MR fluid and intermittent rough motion of the piston. The shock was repaired by MR Technologies, but subsequently began to leak again. The excessive leaking may be a problem with elastomer compatibility with the MR fluid. The rough motion is probably due to poor fit and finish of mating components and possibly particulate contamination of the MR fluid.

The few linear MR shocks that are available commercially are prototype designs, manufactured in very small quantities. Incorporation of these prototype devices into larger experimental systems is very difficult. The existing designs suffer from poor mechanical reliability and inconsistent performance. Further design and engineering development is necessary to provide reliable MR shocks to further study ANVC systems incorporating these devices. Discussions with other organizations performing comparable research indicates that they have experienced similar problems with reliability.

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Figure 1: (a) Single degree of freedom (1DOF) passive suspension model and its response.
 (b) Ideal 1DOF "skyhook" suspension model and its response.

Figure 2: (a) Semi-active suspension model. (b) Mechanical design of the benchtop semi-active MR suspension demonstration.

Figure 3: Measured force verse displacement for the experimental MR shock. Force was measured while actuating the damper at a constant velocity of 4.375 in./sec. Data was collected for six different piston winding voltages.