SEMI-ACTIVE FLUID VISCOUS DAMPERS FOR SEISMIC RESPONSE CONTROL

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ABSTRACT

An earthquake directly affects a structure by increasing the energy within the structural system. A significant portion of this energy can be dissipated and/or reflected through the introduction of a passive, active, or semi-active control system. If certain performance criteria are established which require continuous reconfiguration of the structural system, either an active or semi-active control system will generally be required. In the case of semi-active control systems, the control forces are developed by the motion of the structure itself through appropriate adjustment of the stiffness and/or damping characteristics of semi-active control devices. Further, the operation of semi-active control devices requires a minimal amount of external power. Examples of these devices include electro-rheological fluid devices, semi-active friction devices, and semi-active fluidic control devices.

This paper describes semi-active fluidic control devices, the successful use of such, devices in military applications, and the research efforts of the writers in transferring and adapting this technology to the field of earthquake hazard mitigation. The experimental testing of a semi-active continuously adjustable damping device which operates on the principle of fluid orificing is described. Furthermore, mathematical models which describe the behavior of the device are presented.

1. INTRODUCTION

The energy supplied by an earthquake is transferred through the foundation of a structural system and into the superstructure. A significant portion of the energy within the foundation and superstructure can be dissipated through the introduction of a supplemental energy dissipation system placed either within a seismic isolation system or as structural elements within a conventional construction. Although a variety of supplemental energy dissipation systems have been proposed for the purpose of mitigating the harmful effects of earthquakes, all such systems may be categorized under three basic headings: passive control systems, active control systems, and semi-active control systems.

1.1 Passive Control Systems

A number of passive control systems are currently in use for protection of structures against seismic or wind excitation. The term “passive” is used to indicate that the operation of these systems does not require an external power source. Typically, the mechanical properties of these systems can not be modified. Furthermore, a passive damping system utilizes the motion of the structure to produce relative motion within damping devices which, in turn, dissipate energy. Passive damping systems dissipate energy through a variety of mechanisms including the yielding of mild steel, viscoelastic action in rubber-like materials, sloshing of fluid, shearing of viscous fluid, orificing of fluid, and sliding friction. A discussion of the operation and performance of passive energy dissipation systems has been presented by Constantinou and Symans (1992).

1.2 Active Control Systems

Active control systems have been studied extensively and are currently in use in a number of structures in Japan for protection against wind excitation and minor earthquakes (Soong et al. 1991, Soong 1990, Kobori 1990). The term “active” is used to indicate that the operation of these systems requires a significant amount of external power. The mechanical properties of these systems are typically adjusted based on feedback from the structural system to which they are attached. Control forces are generally developed by electro-hydraulic actuators which require a large power source for operation (on the order of tens of kilowatts). Active control systems may also be designated as active energy dissipation systems because the primary effect of these systems is to modify the level of damping in a structure with only minor modification of stiffness (Soong 1990).

1.3 Semi-Active Control Systems

The use of passive control systems and active control systems represents two extremes in the application of control theory to earthquake hazard mitigation. A compromise between these two extremes is available in the form of semi-active control systems which have been developed to take advantage of the best features of both passive and active control systems. The term “semi-active” is used to indicate that the operation of these systems requires a very small amount of external power (on the order of tens of watts). As in an active control system, the mechanical properties are typically adjusted based on feedback from the structural system to which they are attached. As in a passive control system, semi-active control systems utilize the motion of the structure to develop control forces. The control forces are developed through appropriate adjustment of damping or stiffness characteristics of the semi-active control system. Furthermore, the control forces always oppose the motion of the structure and therefore promote stability. Semi-Active control systems are typically considered to be fail-safe in the sense that semi-active devices can be designed to exhibit either prescribed damping or prescribed stiffness characteristics in the event of a complete loss of power.

2. SEMI-ACTIVE FLUIDIC CONTROL DEVICES

Figure 1 shows the construction of a semi-active fluidic (fluid + logic) control device whose operation is similar to that of a passive fluid viscous damper except that, based on the status of the control valve, it can deliver damping at two distinct levels (two-stage) or over a wide range between an upper and lower bound (continuously adjustable). Its potential for use as a semi-active two-stage damper in seismic energy dissipation systems has been explored by Shinozuka et al. (1992). The device of Figure 1 can be modified to allow for the development of stiffness through removal of the accumulator. Furthermore, a semi-active damping and stiffness device which can modify both its damping and stiffness characteristics can be developed by including a control valve connected to an external accumulator. In fact, semi-active damping and stiffness devices have been used in numerous applications within the U.S. military. Examples of applications include the suspension system of armored vehicles, the suspension system of self-propelled Howitzers, and the Sikorsky Flying Crane Helicopter. During the 1960's, the Sikorsky Flying Crane Helicopter used a semi-active stiffness fluid device to isolate the lifted load from the airframe. The device accepted sensor inputs and altered its output to suit different loads and load environments. This device was entirely successful and many of these helicopters are still in service with the Air National Guard and various commercial firms. In the late 1960's, the U.S. Navy experimented with an isolated ship deck utilizing a semi-active damping and stiffness device. The intent of the isolation system was to allow high speed patrol craft to operate in severe sea states without injury to the crew. An experimental patrol boat with the semi-active isolators proved highly successful, but was never produced in quantity.

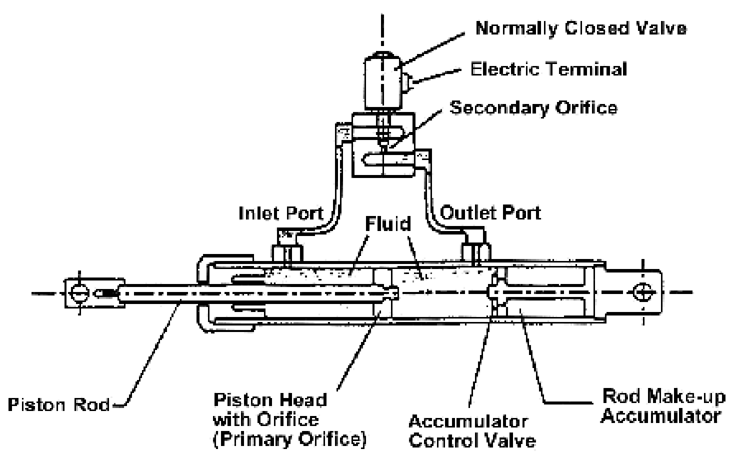

FIGURE 1
SCHEMATIC OF SEMI-ACTIVE FLUID DAMPER

3. DESCRIPTION AND MECHANICAL PROPERTIES OF SEMI-ACTIVE FLUID DAMPERS

Both a two-stage and a continuously adjustable semi-active fluid damper have been developed and experimentally tested. However, due to space limitations, only the continuously adjustable semi-active damper will be described in this paper. The semi-active damper consisted of a passive fluid viscous damper in combination with an external control valve (see Figure 1). The passive fluid viscous damper portion of the semi-active damper has been studied extensively by Constantinou and Symans (1993a, 1992) and Constantinou et al. (1993b) for seismic energy dissipation and seismic isolation. Physical characteristics of the semi-active damper include a weight of 79.5 N, a rated force output of 8.9 kN, a stroke of about ±75 mm, and a length of 257 mm (fully extended piston rod).

The passive portion of the semi-active damper consists of a stainless steel cylinder containing a piston with a bronze orifice head and an accumulator. It is filled with a thin silicone oil (kinematic viscosity = 100 cSt). The orifice flow is compensated by a passive bi-metallic thermostat that allows stable operation of the device over a wide temperature range (-40 degrees C to 70 degrees C). The force generated by the passive fluid damper is a result of a pressure differential across the piston head. The orifice within the passive device utilizes specially shaped passages to alter flow characteristics with fluid speed such that the force output is proportional to the velocity of the piston head relative to the damper housing. This results in essentially linear viscous behavior.

To convert the passive fluid viscous damper to a semi-active fluid viscous damper, an external path for fluid flow is created by drilling two ports in the cylindrical housing and connecting them with steel tubing and a control valve (see Figure 1). The amount of fluid which can pass through the external path is determined by the orifice opening within the control valve. High damping is achieved when the control valve is closed and low damping is achieved when the control valve is open. An intermediate level of damping can be achieved by positioning the control valve at a position between open and closed. In general, the force output of the system is given by

$$F = C(\xi)\,\dot{u} \tag{1}$$

where $C(\xi)$ is the damping coefficient which is a function of command voltage, ξ, and is the relative velocity of the piston head. Note that, in general, $C(\xi)$ is bounded by a maximum (C_{max}) and a minimum (C_{min}) value and may take on any value within these bounds.

The tested system utilized a Direct Drive servo-valve for control of fluid flow through the external loop. The Direct Drive servo-valve was originally developed for control of the primary flight control servo-actuation system on the U.S. Air Force B-2 Stealth Bomber. The valve was designed to replace the conventional hydraulic amplifier pilot stage with a high-force, high-response drive motor acting directly on the valve spool. The Direct Drive servo-valve is ideally applicable to semi-active fluidic control in that it affords electrical control of high flow valve elements without the need for a source of hydraulic pressure to operate a pilot stage.

The Direct Drive servo-valve is a normally closed valve and therefore offers fail-safe characteristics in that the loss of power to the device causes the valve to close which in turn causes the semi-active damper to behave as a passive device with high damping characteristics. Furthermore, the Direct Drive servo-valve requires a peak power of 3.5 W and can therefore operate on the power of batteries which is critical during an earthquake when the main power source of a structure may fail.

A series of tests were performed to determine the mechanical properties of the semi-active damper. The mechanical properties were determined for tests run over a wide range of frequencies (0.5 to 10 Hz) and peak velocities (48.8 to 379.3 mm/s). In these tests, an electro-hydraulic actuator was used to impose a sinusoidal or sawtooth displacement history to the piston head and the force required to maintain this motion was recorded. The command signal supplied to the control valve for some of the tests and associated experimental results are shown in Figure 2.

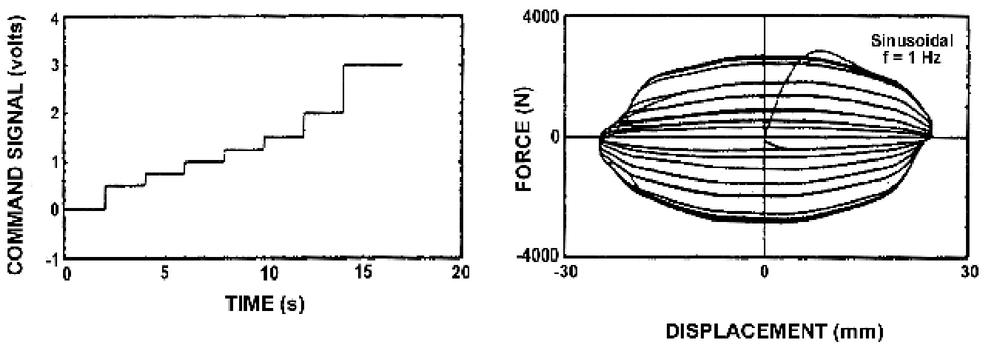

FIGURE 2
TYPICAL EXPERIMENTAL RESULTS FOR SEMI-ACTIVE DAMPER COMPONENT TESTS

The damping/stiffness characteristics of semi-active control devices are generally a function of command voltage. This is demonstrated for the semi-active damper in Figure 3 which shows experimental data for the case of valve opening under a command signal from 0 to 3 volts. The values of damping coefficient were determined for each command voltage by dividing the peak force by the peak velocity of the input motion. An analytical expression for the damping coefficient has been fit through the experimental data and is given by

$$C(\xi) = C_{\min} + (C_{\max} - C_{\min})\exp(-\mu\xi^{\eta}) \quad (2)$$

where C_{min} = 2.56 N-s/mm is the damping coefficient at the full open valve position, C_{max} = 17.94 N-s/mm is the damping coefficient at the full closed valve position, μ = 0.4 volts$^{(-3)}$ and η = 3 are constant parameters. Results similar to Figure 3 are obtained when the valve moves from the full open to the closed position (valve closing). This indicates that, if the command voltage is known, Equation (2) can be used to predict the damping coefficient.

In some of the constant velocity tests, the control valve was operated midway through the tests in order to determine the response time of the semi-active damper. For example, Figure 4 indicates the response time during a constant velocity test with the valve status changing from full open (3 volts) to closed (0 volts). The response time is measured from the point at which the command signal is sent to the control valve to the point at which the damping coefficient reaches its target value. To eliminate the effect of filters on the measurement of response times, all signals were unfiltered during these tests. The response time is a non-zero quantity as a result of a combination of valve dynamics and hydraulic system dynamics and can be separated into two distinct parts. The first part of the response time is designated as t_1 and is measured from the point at which the command signal is sent to the control valve to the point at which the damping coefficient begins to change. The length of time t_1 is related to the time it takes to energize or de-energize the valve (i.e., to build-up or collapse the electro-magnetic field of the coil). The second part of the response time is designated as t_2 and is measured from the point at which the damping coefficient begins to change to the point at which the damping coefficient reaches its target value. The length of time t_2 is related to the dynamics of the valve and the dynamics of the hydraulic system. The two parts of the response time can be measured in a constant velocity component test. In this case, the force output is directly related to the damping coefficient (see Equation 1). For the tested semi-active damper, the total response time is about 15 msec for the system operating from 0 volts (closed valve) to 3 volts (full open valve). In the opposite direction (full open to closed), the total response time is about 25 msec.

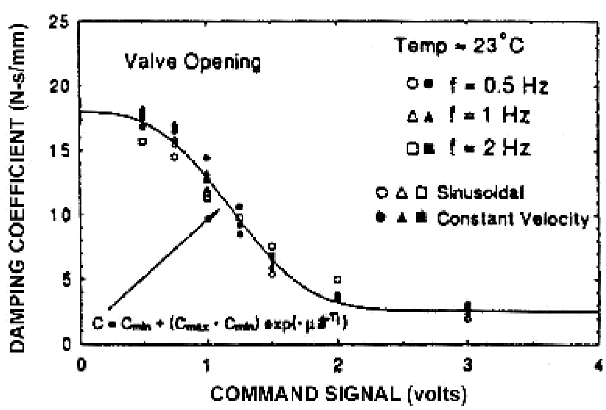

FIGURE 3
COMPARISON OF EXPERIMENTAL AND ANALYTICALLY DERIVED VALUES OF DAMPING COEFFICIENT AS A FUNCTION OF VALVE COMMAND SIGNAL

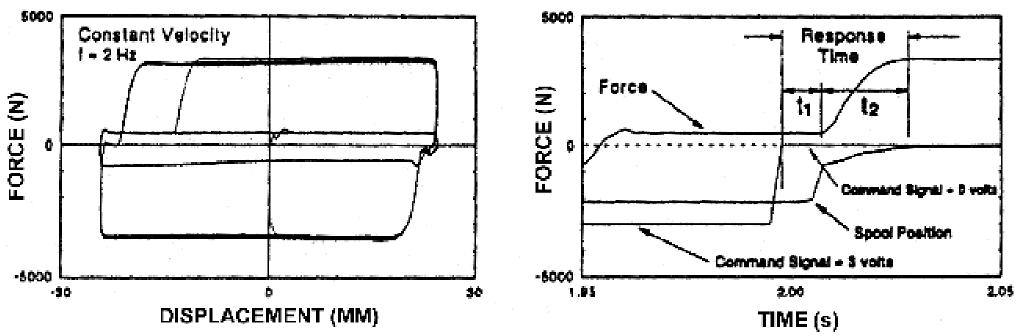

FIGURE 4
TYPICAL CONSTANT VELOCITY TEST USED TO EVALUATE SYSTEM RESPONSE TIME

4. MATHEMATICAL MODEL OF SEMI-ACTIVE FLUID DAMPERS

A general fluid mechanics model for describing the dynamic behavior of semi-active fluid dampers has been developed. A schematic of a semi-active damper used for generating the analytical model is shown in Figure 5. Accounting for boundary deformations, fluid compressibility, and assuming a mean fluid density throughout, one may obtain the mass flow rate continuity equation for chamber i (i = 1, 2) (Watton 1989)

$$\frac{\mathrm{dV_i}}{\mathrm{dt}} + \frac{\mathrm{V_i}}{\beta_\mathrm{i}}\frac{\mathrm{dP_i}}{\mathrm{dt}} = Q_{IN} - Q_{OUT} \tag{3}$$

where V is the fluid volume, β is the fluid bulk modulus, P is the pressure within the fluid, and Q_{IN} (Q_{OUT}) is the flow rate into (out of) the chamber.

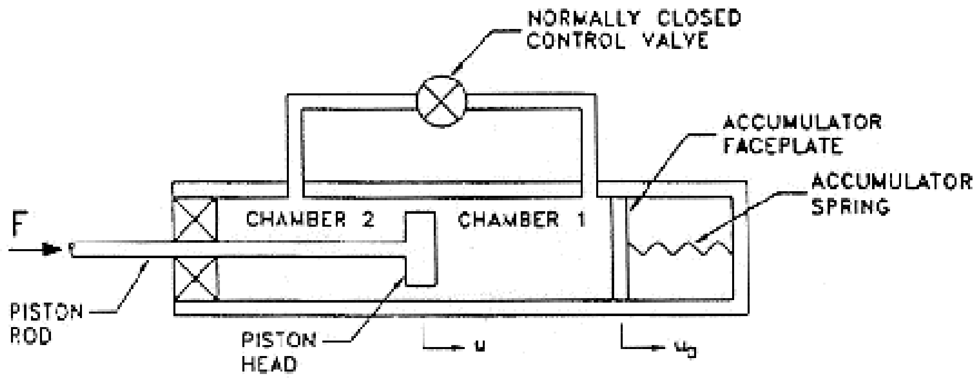

FIGURE 5
SCHEMATIC OF SEMI-ACTIVE DAMPER USED TO GENERATE ANALYTICAL MODEL

Utilizing conservation of energy, mass flow rate continuity, and assuming an incompressible, inviscid fluid, it can be shown that the flow rate of fluid through a small orifice is related to the pressure drop, ΔP, across the orifice by

$$Q = kA\left(\frac{2\Delta P}{\rho}\right)^{\frac{1}{2}} \tag{4}$$

where k is a general orifice discharge coefficient, A is the orifice area, and ρ is the mean fluid density. This relationship is applicable to the continuously adjustable (ADJ) orifice contained within the control valve. However, the primary (PRI) orifice across the piston head does not follow this relation. Recall that the piston head orifice is shaped in a special way so as to obtain a force output which is linearly related to the relative velocity of the piston head. For this reason, the following empirical relationship was utilized for the primary orifice

$$Q_{PRI} = k_{PRI} A_{PRI}\left(\frac{2\Delta P}{\rho}\right)^{1} \tag{5}$$

where the exponent of unity is explicitly shown for emphasis. The mass conservation equation for chamber i (i = 1, 2) may now be written as

$$\frac{dV_i}{dt} + \frac{V_i}{\beta_i}\frac{dP_i}{dt} = (-1)^i k_{PRI} A_{PRI} \left(\frac{2(P_1 - P_2)}{\rho} \right)^1 + (-1)^i k_{ADJ} A_{ADJ} \left(\frac{2(P_1 - P_2)}{\rho} \right)^{\frac{1}{2}} \qquad (6)$$

Physically, fluid enters the accumulator through an orifice and compresses a cylindrical foam element. This has been accounted for by assuming that the fluid in chamber one is in direct contact with an accumulator face plate supported by a linear elastic spring. After some work, Equation (6) may be rewritten as

$$\frac{dP_1}{dt} = \frac{A_p \dot{u} - k_{PRI} A_{PRI} \left[\frac{2(P_1 - P_2)}{\rho} \right]^1 - k_{ADJ} A_{ADJ} \left[\frac{2\,|\,P_1 - P_2\,|}{\rho} \right]^{\frac{1}{2}} \operatorname{sgn}(P_1 - P_2)}{\left[\left(L_1 - u + \frac{P_1 A_f}{K_a} \right) \frac{A_p}{\beta_1} + \frac{A_f A_p}{K_a} \right]} \qquad (7)$$

$$\frac{dP_2}{dt} = \left\{ (A_r - A_p)\dot{u} + k_{PRI} A_{PRI} \left[\frac{2(P_1 - P_2)}{\rho} \right]^1 + k_{ADJ} A_{ADJ} \left[\frac{2\,|\,P_1 - P_2\,|}{\rho} \right]^{\frac{1}{2}} \operatorname{sgn}(P_1 - P_2) \right\} \left\{ \frac{\beta_2}{(A_p - A_r)(L_2 + u)} \right\} \qquad (8)$$

where A_P is the piston head area, A_r is the piston rod area, A_f is the accumulator faceplate area, K_a is the accumulator spring stiffness, L_1 and L_2 are the lengths of chambers one and two (modified to account for the fluid contained within the external steel tubing) when the piston head is at its center (null) position, and u is the piston head displacement relative to the damper housing. Finally, the force output of the semi-active damper is primarily a result of a pressure differential across the piston head and is given by

$$F = P_1 A_p - P_2 (A_p - A_r) + F_f \operatorname{sgn}(\dot{u}) \qquad (9)$$

where F_f is the magnitude of the force required to overcome the friction between the piston rod and seals. Knowing the displacement history, u(t), and the adjustable orifice area history, $A_{ADJ}(t)$, Equations (7) and (8) may be solved for the pressure in each chamber and Equation (9) is then used to evaluate the time history of force within the device. Based on Equation (2), the adjustable orifice area is obtained intuitively as

$$A_{ADJ}(\xi) = A_{\max} \left[1 - \exp(-\gamma \xi^{\zeta}) \right] \qquad (10)$$

where A_{max} is the maximum adjustable orifice area and γ and ζ are constant parameters. Note that Equation (10) neglects the effect of spool overlap and the dynamics of the control valve. Furthermore, A_{ADJ} (ζ) is implicitly a function of time as required by

Equations (7) and (8). All parameters of the model were either measured or determined through analytical calibration.

A comparison of experimental and analytical results is shown in Figure 6 for the fluid mechanics model. Furthermore, a simplified fluid viscous dashpot model was used to obtain analytical results in good agreement with the results of the fluid mechanics model. As demonstrated in Figure 6, the fluid mechanics model appears to adequately describe the semi-active damper behavior.

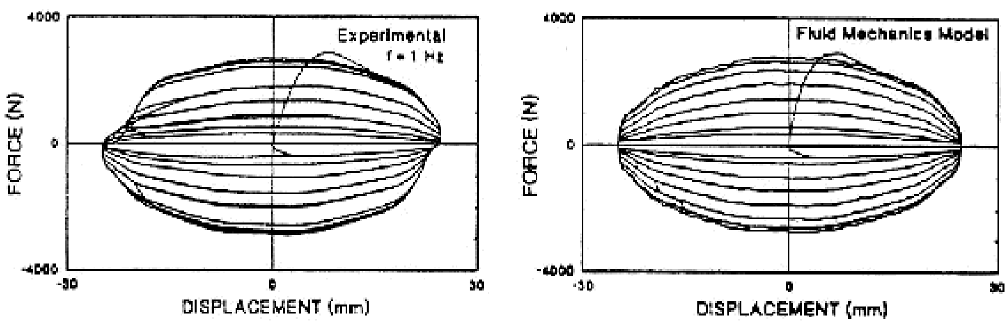

FIGURE 6
COMPARISON OF EXPERIMENTAL AND ANALYTICAL RESULTS FOR SEMI-ACTIVE DAMPER COMPONENT TEST

5. CONCLUSIONS

Semi-Active fluid viscous dampers have been investigated for use as supplemental seismic energy dissipation devices. The mechanical properties of a continuously adjustable semi-active damper were experimentally determined and the device was shown to be capable of delivering two distinct levels of damping (high and low) as well as any prescribed level in between. Furthermore, a detailed mathematical model based on fluid mechanics principles and a simplified fluid viscous dashpot model were developed to describe the dynamic behavior of the semi-active fluid damper. Both models were found to adequately predict the experimental results.

6. ACKNOWLEDGMENTS

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