A NEW SEMI-ACTIVE PIEZOELECTRIC BASED FRICTION DAMPER

Memet Unsal, Christopher Niezrecki, Carl D. Crane III

University of Florida, Department of Mechanical Engineering
Gainesville, Florida, 32611, USA
e-mail address of lead author: memetu@ufl.edu

Abstract

A new way to perform vibration control on a single-degree-of-freedom system using a piezoelectric friction damper is developed. The damper consists of an actuator, which is based on a piezoelectric stack with a mechanical amplifying mechanism that provides symmetric forces within the isolator. The advantages of such an actuator are its high bandwidth, actuating response and its ability to operate in vacuum environments such as in space. The damper is constrained to move using an air bearing that produces a virtually ideal single-degree-of-freedom spring-mass system. Within this work, the actuating ability of the friction-based actuator is characterized.

INTRODUCTION

Vibration isolation and control has traditionally been performed using passive techniques (damping material, viscoelastic mounts, etc.). With the recent advances in computer technology, active vibration control has become realizable and researchers have investigated using active control techniques for vibration control. Including an active component in the design allows a system the flexibility to tailor its dynamic characteristics depending upon the loading. However active control systems also have significant limitations that prevent their implementation in real world systems (i.e. actuator power requirements, instability).

Semi-Active Vibration Control

The concept of using semi-active control for vibration suppression was originally proposed by Karnopp, *et al*. They suggested varying the force in a viscous damper by controlling the orifice area [1]. Since then, semi-active vibration control has been researched utilizing hydraulics [2-3], electromagnetics [4], magneto-rheological fluids [5-6], electro-rheological fluids [6-8], and friction based devices [9-19]. Semi-active vibration control has been previously applied to civil structures, automobile suspensions, bicycle shock absorbers, etc.

Semi-active control offers the enhanced performance of active methods without overly complicating a vibration control system. The dynamic properties (stiffness, damping, etc.) of a structure is continuously adjusted by a controller that has the potential to modify the structure's vibrating properties. Including an active component in the design allows the system the flexibility to tailor its dynamic characteristics depending upon the loading. Because the semi-active system is responsible for only adjusting the amount of mechanical dissipation a structure has, it does not require an overwhelming amount of power or weight (as required by an active system), while providing enhanced performance (as compared to a purely passive system).

Semi-Active Friction Actuators

Because the actuator only needs to change the normal force exerted onto the vibrating element, it requires very little actuating displacement and mechanical power. The active element is not required to generate a displacement having the same order of motion as the mounts. Therefore, the amount of work done by the control actuator is significantly smaller than that required of a purely active control actuator. Also, since the friction actuator only dissipates energy from the system, (assuming the system was originally stable) it cannot cause instabilities to occur [20, 21, 22].

Modern electromagnetic actuators are well suited to provide rotational motion (electric motors), however their use as linear actuators is limited. Although they are capable of generating sufficient force and displacement, the large size, weight, electrical demands and cost of these actuators makes them impractical. Because of their high force and bandwidth capability, piezoelectric actuators appear to be a natural candidate as friction dampers. However until only recently, the maximum (freely loaded) mechanical strain of these devices did not exceed 0.1%. This means that an actuator 1 inch long could only deflect 0.001 inch (significantly less under load). As a result, the development of a practical frictional damper has been hampered. A flextensional mechanical amplifier can be used to increase the displacement of the actuator. Such an actuator has recently been developed (FPA-1700, Dynamic Structures and Materials, LLC.) that can generate a 1.7 mm displacement and is the heart of the damper being investigated. Within this work, the actuating force characteristics of the piezoelectric based friction actuator is investigated experimentally. This actuator has potential application to space environments in which other viscous dampers as well as electro-rheological and magnetorheological dampers are not suitable, civil engineering structures, parallel platform mechanisms, large space structures, and vehicle suspensions. The purpose of this work is to characterize the frictional force capabilities of the new actuator design.

THEORETICAL DESCRIPTION

The system tested in this work can be modeled as shown in Figure 1. The forces acting on the moving mass are the spring force, the friction force and the external force applied by the shaker. The sum of the these forces provide the inertial force.

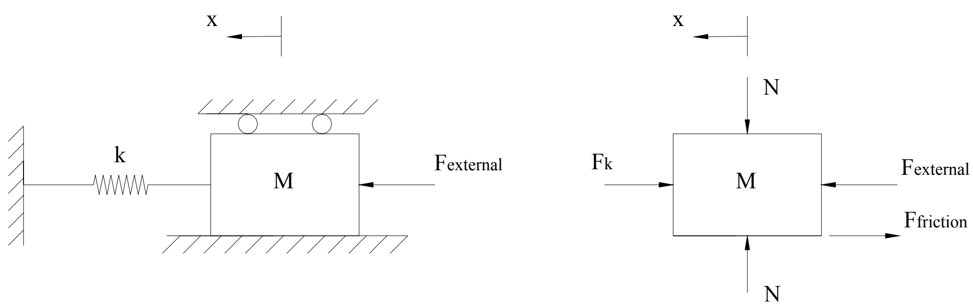

Figure 1 Schematic Drawing and Free Body Diagram

$$\sum F = m\ddot{x} = -F_k - F_{friction} + F_{external} \tag{1}$$

In this experiment, the spring is removed. The acceleration and the applied external force are measured using an accelerometer and a force transducer, respectively. By rearranging Equation 1, it is possible to measure the friction force generated by the actuator. Therefore, the friction is given by,

$$F_{friction} = -kx + F_{external} - m\ddot{x} \tag{2}$$

EXPERIMENTAL DESCRIPTION

The friction damper consists of several moving and stationary components as shown in Figure 2. A 0.75" diameter shaft is fixed to the base of the damper. Mounted to the shaft is the flextensional mechanical amplifier. The moving components consist of the outer housing and the air bearing. The outer housing also comes in contact with the friction pads as it vibrates. The normal force provided between the friction pads and the outer housing induces a frictional load which retards the motion of the outer housing. Within this damper, there is also a spring which connects the moving housing to the stationary base as shown in Figure 3. For these tests, the spring was removed. With the frictional pads not engaged, the air bearing provides a relatively frictionless contact surface. As a result, the damper is essentially an ideal SDOF system.

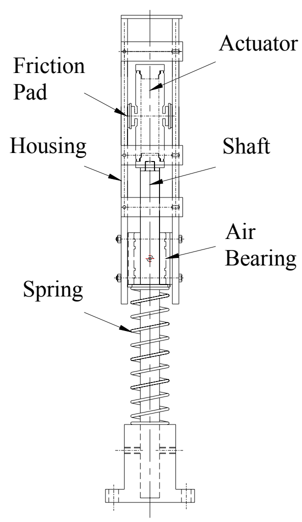

Figure 2 Friction Damper (Front View)

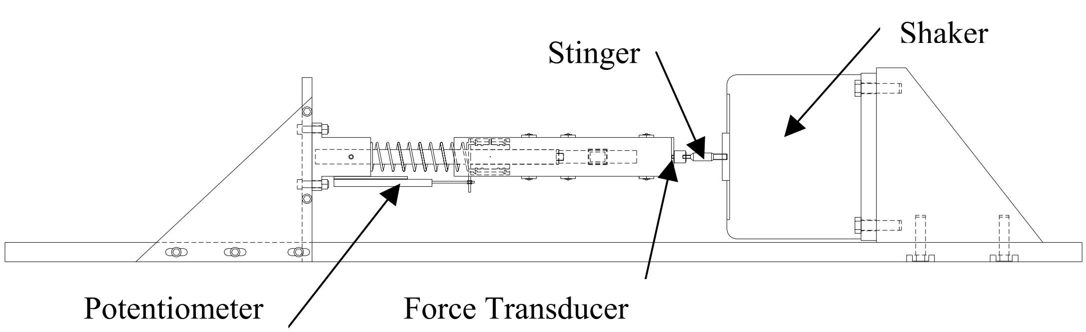

Figure 3 Friction Damper with Shaker and Transducers (Side View)

Several experiments were carried out in order to characterize the friction induced by the actuator. For three shaker voltages (3, 5 and 7 V), three shaker frequencies (10, 15 and 20 Hz) and 8 different actuator voltages, the results of these experiments have been compared. The applied voltage to the actuator is sinusoidal and the frictional force is determined using Equation 2, by measuring the acceleration (PCB 333B42) and force (PCB 208C02). The friction pads are kevlar bike disc brake pads (CODA® QPDPAD/BLU) and the housing is steel.

RESULTS

Shown is Figure 4 are the results for a shaker voltage of 3 V, and actuator voltage of 48 V and an excitation frequency of 15 Hz. This set of data is displayed because it is representative of typical results for the various test cases. The frictional force generally has the shape of a square wave. This is expected since the normal force does not change with time for a fixed actuator voltage. Additionally, at the transition between motion in the positive and negative direction, an oscillation occurs in the calculated force signal. The frequency of this oscillation is 714 Hz for all test cases and is a result of stick-slip phenomenon. The additional test cases are displayed in Figure 5 for an actuator voltage of 60 V and 72 V, having a shaker input voltage of 3 V, 15 Hz.

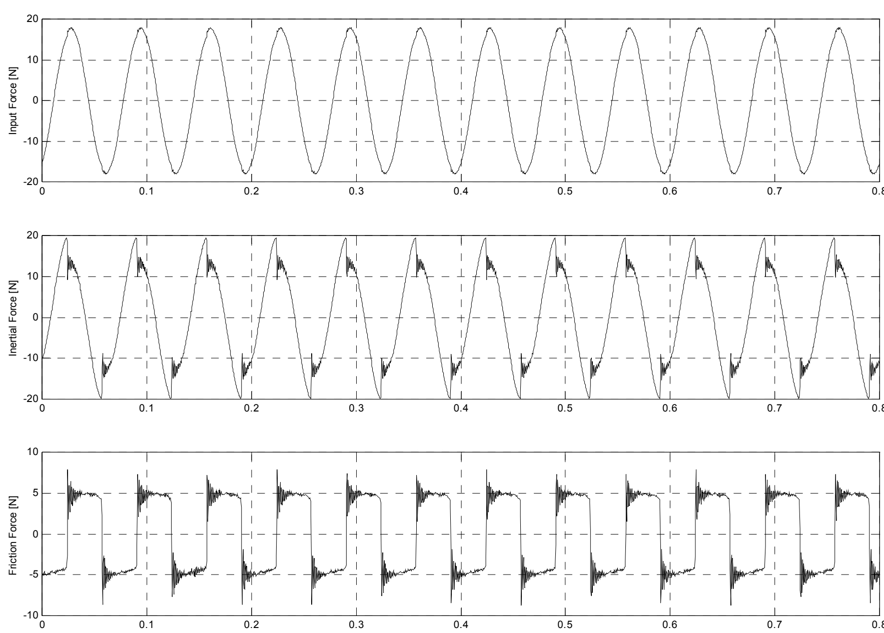

Figure 4 Input Force, Inertial Force and Friction Force

(actuator voltage=48 V, shaker voltage=3 V at 15 Hz)

The results from all of the test conducted are shown in Figure 6. The average friction force amplitude is determined by averaging the peak to peak friction force and dividing by two. The values of the friction force amplitude were calculated for actuator voltages ranging from 48 to 150 V. For some of the test cases, the static friction force exceeded the input force of the shaker. Measurements for these data points have been omitted.

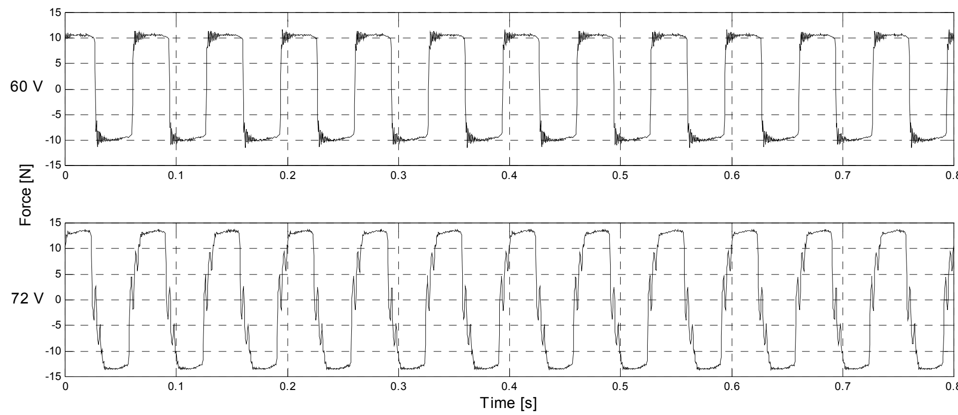

Figure 5 Actuator Friction Force at Higher Voltages

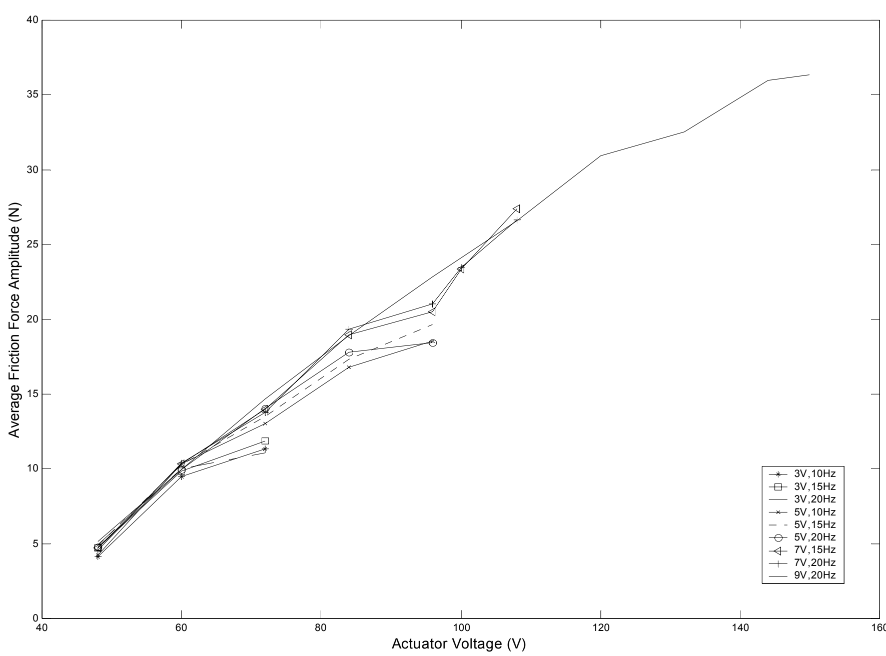

Figure 6 Friction Force Amplitudes

The results indicate that the excitation frequency does not strongly influence the amplitude of the friction force. The frictional force of the actuator is essentially proportional to the voltage applied to the actuator. Based on the empirical results, it can be estimated that the maximum frictional force capability of this actuator with the frictional pads used is approximately 36 N. The frequency of the stick slip oscillation appears to be independent of the normal force and the excitation frequency.

CONCLUSIONS

A new type of semi-active friction based damper has been developed and has the potential to be used in several active vibration control applications. The heart of the damper is a piezoelectric stack with a mechanical amplifying mechanism. Within this work the frictional force capabilities of the actuator have been characterized experimentally. It was determined that the maximum force capability of the actuator is approximately 36 N and that the frictional force amplitude is not strongly dependent on frequency and is approximately linear with applied voltage.

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