

DESIGN OF A PIEZOELECTRIC SHAKER FOR CENTRIFUGE TESTING

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ABSTRACT

Centrifuge testing is a new experimental technique designed to test models of large earthen structures. Using centrifuge modeling to study the effects of earthquake ground motion requires a compact lightweight actuator that can produce large forces at high frequencies.

This paper describes the design of a prototype piezoelectric shaker and its development to date. Although certain design problems remain to be solved, the piezoelectric system shows promise for adaptation to a larger payload system such as the proposed geotechnical centrifuge at the Ames Research Center.

INTRODUCTION

A centrifuge can simulate gravity-induced stresses on foundations and earth structures at a reduced geometric scale through centrifuge loading. Using this technique, large structures of a size that cannot be tested practically in any other fashion can be modeled. In addition, centrifuge modeling is potentially a more accurate testing method and therefore can be used to verify and improve present finite element analysis techniques. In centrifuge model testing, a soil payload experiences increased soil pore pressure due to the normal (centripetal) acceleration produced by the centrifuge rotation. This increased pressure causes the soil to behave as though it was at a deeper level. For example, a 100 foot deep earthen dam can be represented by a 1 foot deep embankment when subjected to a 100 g normal acceleration.

Centrifuge modeling can be used to study the effect of earthquake ground motion on earthen embankments. To correctly simulate real world conditions in an N times g normal acceleration test, the centrifuge model laws require the simulated earthquake motion to have N times the acceleration, N times the frequency and $1/N$ times the duration of the real world motion. With an actual peak acceleration of .25 g 's, a frequency of 4 Hz and a time duration of 10 seconds, a model payload rotating with a 100 g normal acceleration would need to experience a 25 g peak acceleration with a frequency of 400 Hz lasting for 0.1 seconds.

In 1976, at the University of California, Davis, the Department of Civil and Mechanical Engineering, in collaboration with the NASA-Ames Research Center, undertook a study to develop an earthquake simulator system. It was proposed that the feasibility of the system be tested in the Schaevitz 3-meter-arm centrifuge at Davis. The system components include:

1. An actuator capable of producing the desired earthquake motion on a test payload.
2. A payload box (container) to hold model earthen embankments.
3. A mounting and drive system to attach the actuator and payload box to each other and secure them to the centrifuge.
4. A monitoring system to record the earthquake motion and its subsequent effects on the test payload.

Using this system, preliminary data was obtained to check the accuracy of the model laws and to determine the feasibility of developing this system for use in a larger centrifuge.

DESIGN

Discussions between the organizations involved and other geotechnical researchers resulted in the following design requirements:

1. The actuation need only produce motion in one dimension. The desired model factor is $N=100$ (i.e., the test payload will experience a 100 g normal acceleration).
2. The shaking system is to be mounted in the outer, free-swinging bucket presently attached to the centrifuge. Some additional space is available at the center of the centrifuge for equipment mounting.
3. The total useful load mounted inside the outer bucket may not exceed 100 pounds (this includes soil, payload box, actuator, and instrumentation mounted in the outer bucket).
4. The instrumentation will include a videotape camera (mounted in the center of the centrifuge), an accelerometer attached to the payload box, and a pore pressure transducer.
5. The shaker mechanism must be made as light as possible in order to maximize the amount of soil that can be used in a centrifuge test.
6. The soil payload depth will not exceed 5 inches.
7. The shaker should be able to model real world accelerations of .2 to .5 g.
8. The exact simulation of a given earthquake record is desired, but a shaker which can produce sinusoidal motion from 200 to 600 Hz frequency range is considered acceptable.

METHOD CHOSEN

Seven possible actuator methods were considered. These were electrodynamic, rotating counterweights, explosive firing, spark gap (electric arc discharge), electrohydraulic, natural oscillation of a spring-mass system and piezoelectric. The piezoelectric method was found to best satisfy the design requirements. It is simple, light, compact, and can potentially model an exact earthquake record.

A piezoelectric ceramic element is an artificially-polarized wafer which deforms when an electric field is applied across its polarized surfaces. The magnitude of the strain is directly proportional to the magnitude of the electric field applied. To obtain the desired displacements, it is necessary to stack a number of elements in series and pre-stress them in compression. Since the strain is related to the electric field applied, it is possible to control the motion by varying the electrical input signal.

SYSTEM COMPONENTS

The design of the piezoelectric actuator system was divided into five areas:

1. Piezoelectric elements.
2. Electrical system.
3. Piezoelectric stack support and insulation.
4. Preload and drive system.
5. Payload box and its support.

Piezoelectric Elements

The piezoelectric material chosen was a lead titanate ceramic called PZT-4. This material has the highest strain per unit length of any available material when operated just below its maximum depoling field. It has a low compliance and can withstand the most compressive stress of the piezoelectric materials manufactured. Its relatively low capacitance combined with a high d_{33} constant (d_{33} is a measure of the strain produced for a given applied electric field) reduces the power necessary to operate it. For PZT-4 operating at low voltage levels d_{33} is approximately 2.89×10^{-10} meters/volt. The largest standard thickness available (0.5 inches) was chosen to reduce the number of electrical connections. To reduce the power requirements, primary tests conducted used a 6-inch stack (12 piezoelectric elements) to drive the payload. The proposed final design is to use an 8-inch stack (16 piezoelectric elements).

Electrical System

Figure 1 shows the basic electromechanical model of a piezoelectric element for a non-resonant system. The leakage resistance R_L is, for this application,

so large that it can be treated as an open circuit. The electrical portion of the model reduces simply to a capacitor with a reactance of $1/\omega C_e$. Because of the large impedance and high voltage requirements of the piezoelectric stack, it is estimated that the power required to operate at maximum amplitude would be near 28 kw (a 6 inch stack operated sinusoidally at 500 Hz). To evaluate the system without the use of a larger power supply the system is incorporated into an LRC resonant circuit. In this circuit the piezoelectric stack behaves as a capacitor while a variable inductor is used to peak the applied electric voltage at different frequencies.

As shown in Figure 2, the piezoelectric elements are connected electrically in parallel by thin shims placed between each element.

Piezoelectric Stack Support and Insulation

To insulate and protect the piezoelectric elements from other components in the system and to hold the stainless steel shims in place, a hollow polyvinylchloride (PVC) cylinder (Figure 2) surrounds the piezoelectric stack. As an additional safeguard to prevent electrical arcing, anticorona coating is sprayed on the outer sides of the elements. To separate the outer edges of the positive shims from the groundwire, and visa versa, each shim has a terminal strip which passes through a slot in the PVC shell and allows connections outside the shell. The positive and negative terminals are connected on opposite sides of the shell. The PVC shell also acts as a support for the piezoelectric elements and helps to keep them aligned.

Piezoelectric Preload and Drive System

The stack is preloaded using disc springs to hold the stack tightly together as it expands and contracts (Figure 2). Using disc springs with a total spring constant much less than the spring constant of the stack drive system allows the higher spring constant to control the resonant frequency. Preloading can be varied from 0 - 6500 pounds. Most tests were conducted with a 4500 pound preload. The piston actuator slides through the hollow center of the disc springs and the preloading bar. The preloading bolts are insulated from the preloading bar by bakelite collars preventing a ground loop which would reduce the current passing through the elements.

The piezoelectric stack and drive system is mounted underneath the baseplate of the payload box as shown in Figure 3. Although mounting the stack to the baseplate of the payload box would provide a more rigid connection, this was not possible due to the dimensions of the outer bucket in the centrifuge. The stack is compressed on one end by a baseplate which in turn is bolted to the outer centrifuge bucket. In addition to compressing the piezoelectric stack the baseplate distributes the force acting on the outer bucket wall, reducing the stress and deflection in this wall. The outer end of the stack is compressed by the piston actuator which in turn is threaded and bolted to a mounting plate attached to the bottom of the payload box.

Payload Box and Its Support System

The payload box is shown in Figures 3 and 4. Weight, size, stress, resonance, and providing a very stiff attaching platform were the five major considerations used in developing the box.

The payload box is constructed of 6061-T651 aluminum alloy. To increase the rigidity between the payload box and the piston actuator, stiffening ribs are welded to the bottom of the box (Figure 3). With this reinforcement, the stiffness of the baseplate is estimated to be between 10 and 100 times greater than that of the four cantilever supports.

The dimensions of the cantilever supports represent a compromise between the need for low stresses and a high resonance frequency and the desire for low compliance and minimum mass.

Description of Overall System

Figure 5 portrays the current test setup. When the frequency generator is manually triggered, a low voltage AC signal is sent to the power amplifier for a prescribed time interval. The output signal from the power amplifier is fed through the autotransformer and into the inductor. This high voltage sinusoidal signal is fed to the piezoelectric elements producing a sinusoidal motion proportional to the electric field across them. The signal generator also triggers the oscilloscope sweep and sends a signal to the digital counter which records the time interval of the test run.

The voltage across the entire LRC system, the voltage across the piezoelectric elements and the current through the LRC system is monitored by voltage and current probes connected to an oscilloscope. The acceleration produced is measured by an accelerometer rigidly attached to either the piston actuator or the payload box. A video camera attached on the center platform inside the centrifuge is used to observe the payload.

Slip rings on the main shaft of the centrifuge provide electrical connections for dynamic tests. The charge amplifier is mounted inside the centrifuge to reduce the effects of electrical noise generated when the signal passes through the slip rings. A solenoid remotely triggered by a DC power supply grounds the charge amplifier and clears it of transient signals. The bandpass filter screens out 60 Hz noise and other high frequency electrical and mechanical noise.

RESULTS

Static tests (conducted without the centrifuge rotating) produced accelerations at or above the level expected. Mechanical resonance of the shaker payload system occurred at a lower frequency than predicted amplifying the acceleration produced near 500 Hz. This resonance can be observed in Figure 6 where acceleration peaks even though the applied voltage is declining.

Dynamic tests produced acceleration which declined as RPM's increased. Part of this decline is due to the normal acceleration affecting the mechanical resonance frequency. Another part is due to the normal force acting on the mechanical coupling between the piezoelectric stack and the payload bucket. A number of modifications have been tested but none have completely eliminated the decline. Other factors that may contribute to the decline are under investigation.

Current tests were conducted at 40% of the piezoelectric systems rated capacity. The system is being modified to operate at higher power levels in addition to modifications that will hopefully eliminate the decline in acceleration or at least lead to an understanding of why it exists.

CONCLUSIONS

To date the results of this study have shown:

1. The feasibility of producing large forces using the piezoelectric system.
2. The feasibility of easily controlling and varying the frequency, the amplitude of acceleration and the time duration of motion.
3. The system shows promise for adaptation to a larger payload system.

Before a complete evaluation can be made, the system needs to be better understood. Elimination of the decline when the centrifuge is rotating, tests conducted with a "stiffer" system to eliminate the influence of mechanical resonance, and tests conducted at higher power levels are necessary before valid geotechnical studies can be successfully conducted.

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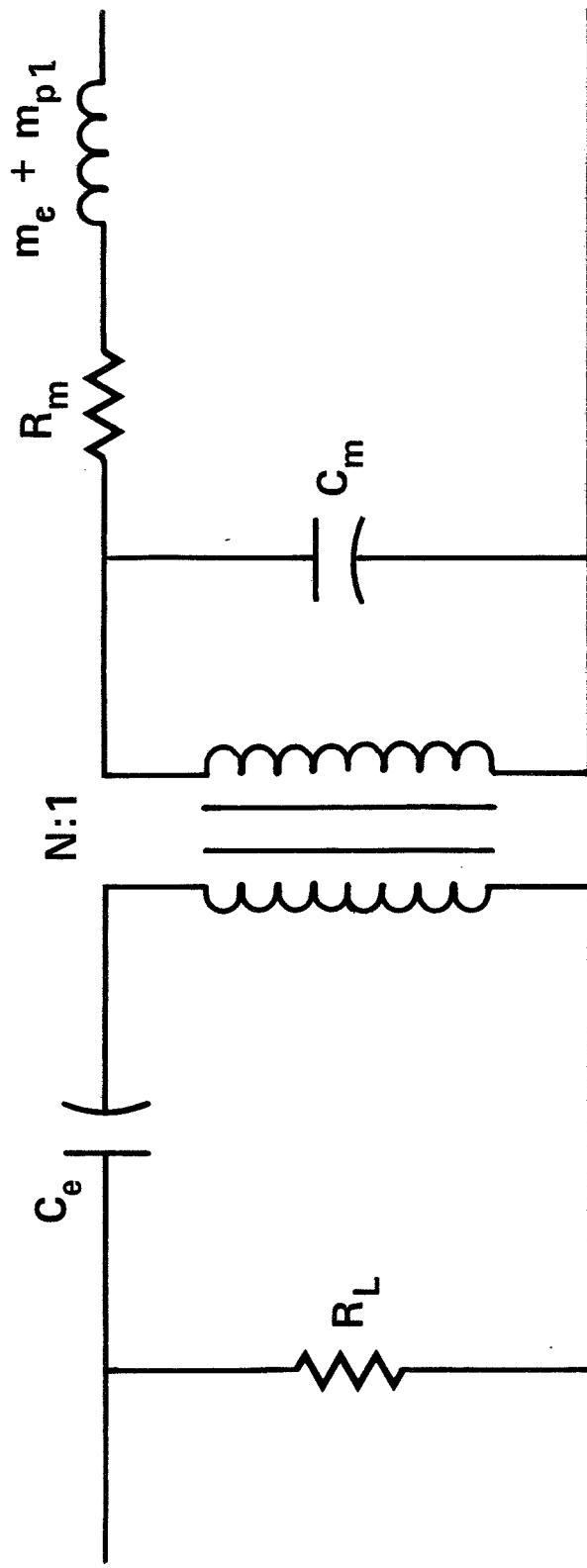


FIGURE 1 - Electromechanical model of a piezoelectric element. R_L is the leakage resistance, C_e the electrical capacitance, N the transformation ratio, C_m the mechanical compliance, m_e the effective mass, R_m the mechanical resistance, and m_{pL} the mass of the payload.

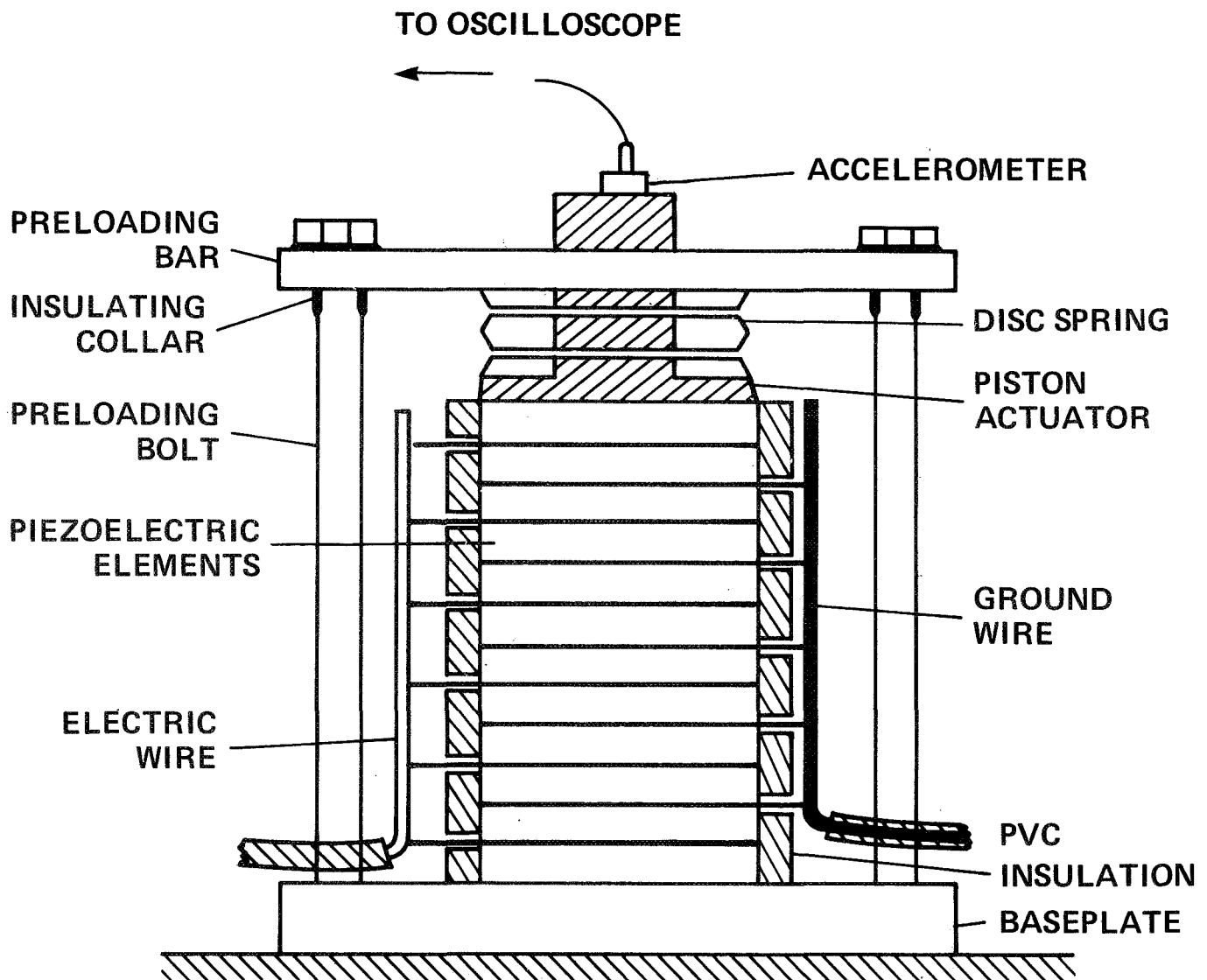


FIGURE 2 - Piezoelectric stack shown with preloading system and electrical wiring. The piston actuator shown is used to measure acceleration of the system when no payload bucket is attached.

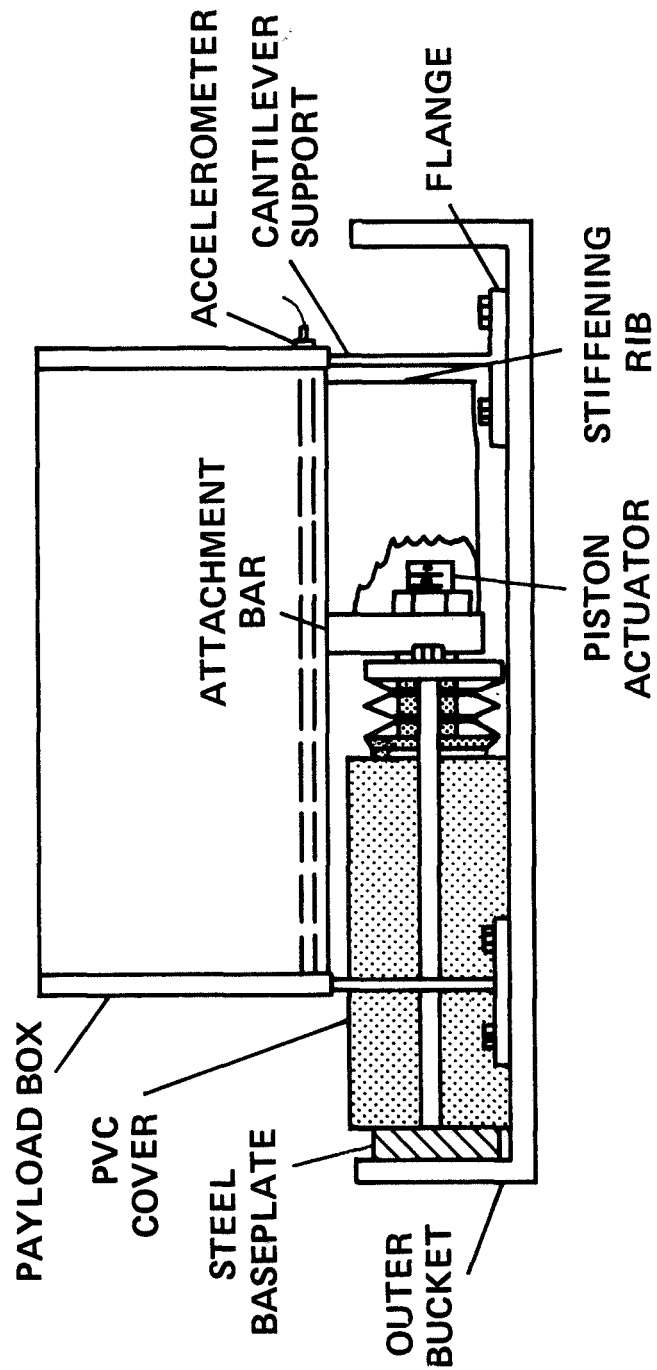


FIGURE 3 - Side view of piezoelectric system showing attachment of the piston actuator to the payload bucket. Electric wiring is not shown.

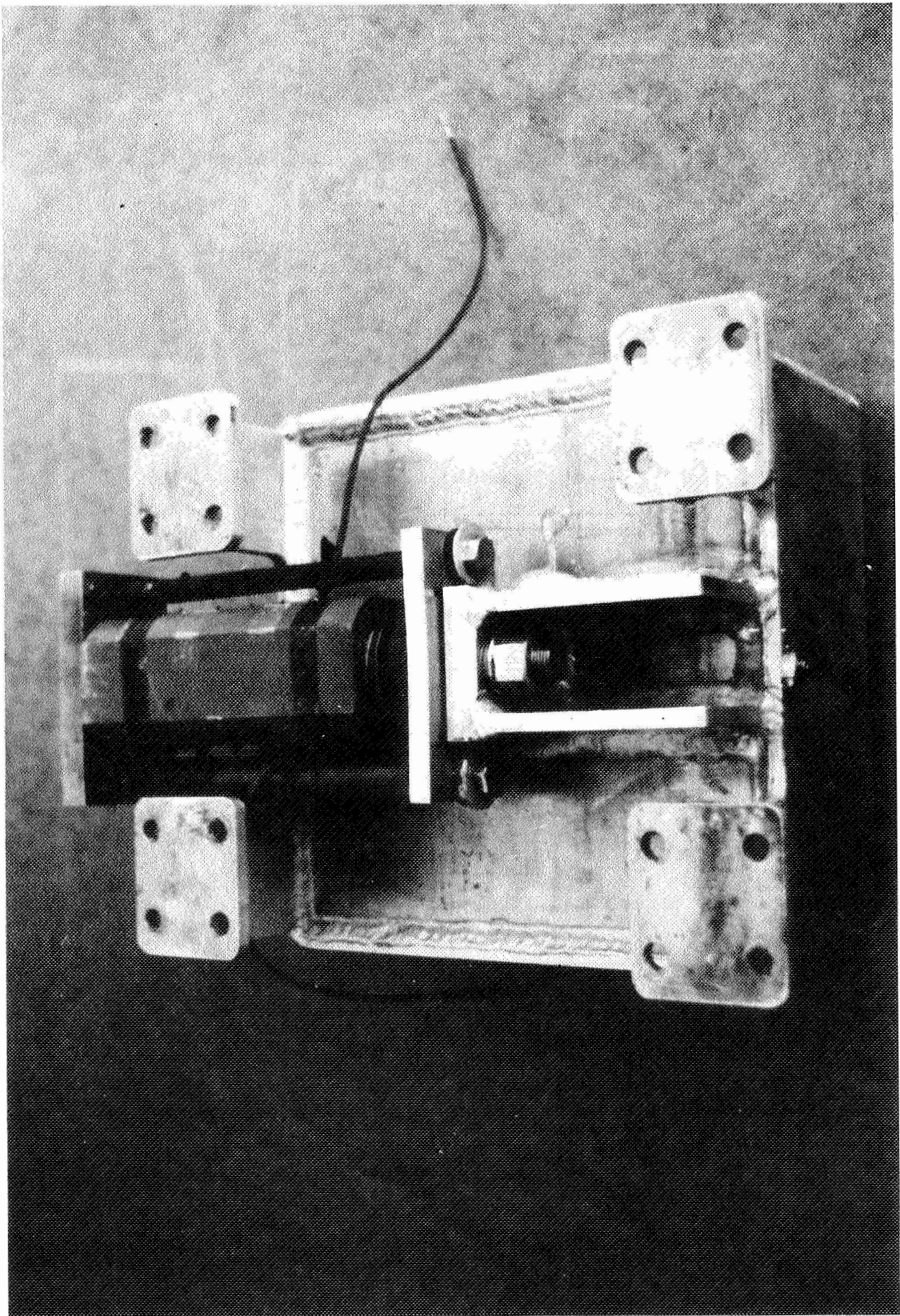


FIGURE 4 - Bottom view of payload bucket with piezoelectric stack and driving system attached.

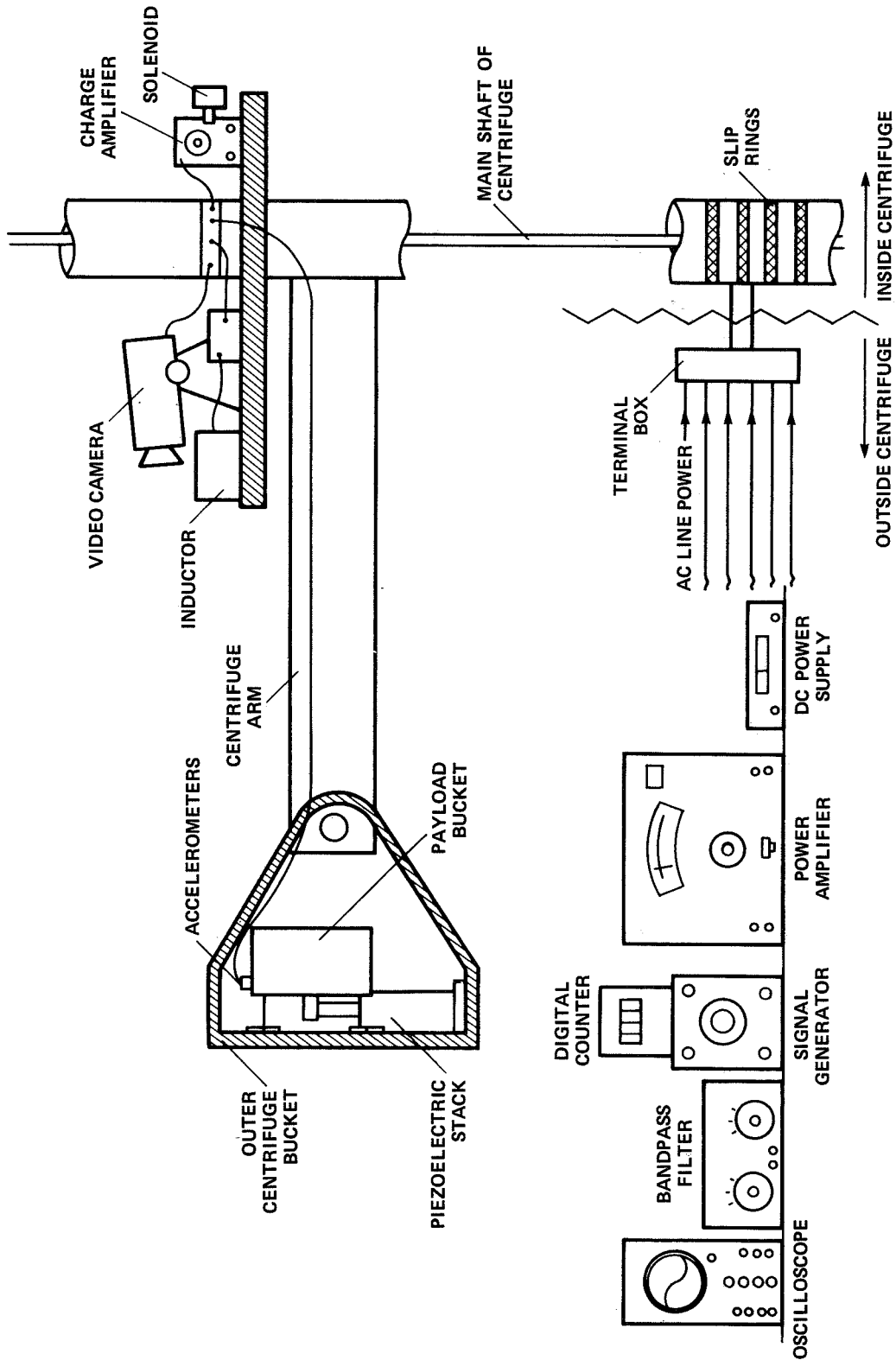


FIGURE 5 - Geographic location of piezoelectric system components.

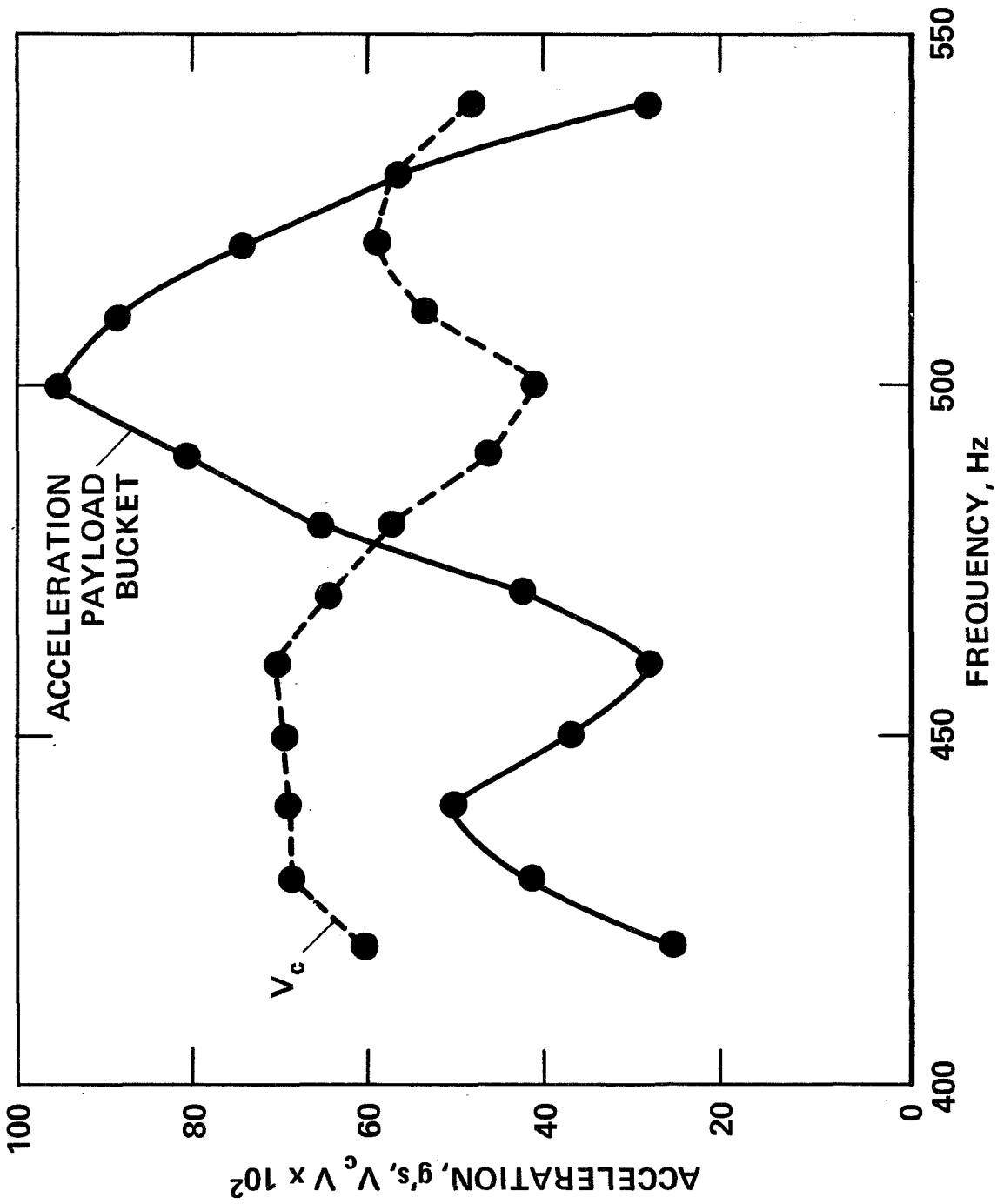


FIGURE 6 - Acceleration and Voltage Versus Frequency for 50 lb_m Soil Payload.