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# Active Vibration Control in Microgravity Environment

Prepared by: Carl H. Gerhold, PHD, PE

Academic Rank: Associate Professor

University & Department: Texas A&M University
Department of Mechanical Engineering

NASA/JSC

Directorate: Engineering

Division Structures and Mechanics

Branch Loads and Structural Dynamics

JSC Colleague: A. Rodney Rocha

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# Active Vibration Control in Micro-gravity Environment

Carl H. Gerhold, PhD, PE

Associate Professor
Mechanical Engineering Department

Texas A&M University
College Station, Texas 77843

The low gravity environment of the Space Station is suitable for experiments or manufacturing processes which require near zero-g. Such experiments are packaged to fit into rack-mounted modules approximately 106.7 cm (42 in) wide x 190.5 cm (75 in) high x 76.2 cm (30 in) deep. The mean gravitation level of the Space Station is expected to be on the order of $10^{-6}$ g ($9.81 \times 10^{-6}$ m/s$^2$). Excitations, such as crew activity or rotating unbalance of nearby equipment can cause momentary disturbances to the vibration-sensitive payload on the order of 0.4 g. Such disturbances can reduce the micro-gravity environment and compromise the validity of the experiment or process. Isolation of the vibration-sensitive payload from structure-borne excitation is achieved by allowing the payload to float freely within an enclosed space. Displacement-sensitive transducers indicate relative drift between the payload and the surrounding structure. Small air jets provide a negative thrust vector which keeps the payload centered within the space. The mass flow rate of the air jets is controlled such that the resultant acceleration of the payload is less than a criterion level of $10^{-5}$g. It is expected that any power or fluid lines that connect the experiment to the Space Station structure can be designed such that they transmit vibration levels within the criterion. A flexible coiled hose such as is used to carry shop air has the requisite compliance.

An experiment has been fabricated to test the validity of the active control process and to verify the flow and control parameters identified in a theoretical model. Zero-g is approximated in the horizontal plane using a low-friction air-bearing table. An analog control system has been designed to activate calibrated air-jets when displacement of the test mass is sensed. The experiment demonstrates that the air jet control system introduces an effective damping factor to control oscillatory response. The amount of damping as well as the flow parameters, such as pressure drop across the valve and flow rate of air, are verified by the analytical model.

NASA Colleague: A. R. Rocha/ES4/x-4393

Active Vibration Control in
Microgravity Environment

## Introduction

Microgravity research experiments, such as crystalline growth in zero gravity, require a static gravitational environment on the order of $10^{-6}$ g. Such a low gravity environment is obtained in space near the centerline of a Space Station. In a low gravity environment, momentary disturbances, such as thruster fire or crew pushoff introduce shocks to the vibration sensitive experiment on the order of $3.0 \times 10^{-3}$ g. [1] Rotating equipment, such as pumps located near the experiment, may transmit steady state vibration of 0.4 g at 10 Hertz. [2] Such disturbances alter the microgravity environment and can degrade the validity of the experiment.

The purpose of this project is to devise and evaluate a method to reduce the vibration transmitted from the Space Station structure to the vibration sensitive payload. The most effective way to decouple the payload is to allow it to float freely in space. Figure 1 is a schematic representation of the payload suspended within an enclosure which is attached to the Space Station structure. Unbalanced forces, whether generated internally or externally, cause the payload to drift toward the enclosure. This relative motion is sensed and used to activate air jets. The jets provide thrust to stop the motion and to return the payload to a central location within the enclosure.

External forces arise from gravity gradient over an orbit cycle and aerodynamic drag-induced deceleration of the Space Station, and result in long period drift between the enclosure and the payload. External forces are also transmitted through connections such as air, fluid, or power lines. The lines act as compliant elements which transfer structural vibration to the payload. Unbalanced forces are generated internally by rotating equipment or fluid motion within the payload.

The air jet control system is required to keep the payload from contacting the enclosure. The criterion for the air jet is that the thrust produced by the jet results in a net acceleration of the payload less than or equal to $10^{-5}$ g.

The sensors used to control the jets respond to velocity and displacement of the payload. This type of control is expected to be sufficient for the following reasons. It is assumed that the internally generated forces are sufficiently small that the acceleration produced by them is less than $10^{-5}$ g. However, these forces may cause the payload to drift, which drift the air jet is intended to control. Gravity gradient is expected to be the major long period external excitation, producing a disturbance on the order to $10^{-5}$ g [1]. The period of this excitation is approximately 90 minutes (one orbit cycle). Since the gravity gradient does not exceed the allowable acceleration criterion, the air jet control which limits long term drift of the payload keeps it essentially neutrally buoyant. Structural vibrations of magnitude 0.4 g at 10 Hertz may be transmitted through hoses and power lines to the payload. Compliant connections, such as self-coiling flexible air lines, can be used to reduce the transmitted vibration so that the steady state acceleration of the payload is on the order of $10^{-5}$ g. The air jet control is used to limit drift resulting from this disturbance.

Preliminary investigations, including computer simulation, indicate that the air jet control system is feasible. However, in order to establish a workable system it is necessary to test the concept experimentally. The experimental setup consists of a mass constrained to move in a one-dimensional simulated low-g environment. The displacement and velocity of the mass are monitored and used to control solenoid-activated air valves. The first phase of the experimental program is intended to establish the feasibility of the air jet feed back control system concept and to identify parameters for the air jet and the feedback control systems.

## Theoretical Background

The basic one-dimensional model is derived from the system shown in figure 1. The one-dimensional model is felt to provide sufficient detail to identify system parameters. It is assumed that the internal forces generated by rotating unbalance or fluid motion within the payload are negligible in comparison to the force transmitted through the compliant coupling to the structure. This couple between the payload and the structure is modeled as a massless spring element. The differential equation describing the motion of the payload is:

$$M\ddot{x} = k(y-x) + F_j \qquad (1)$$

where:

$M$ = payload mass
$k$ = equivalent stiffness of the hose or power line
$x$ = absolute displacement of the payload
$\ddot{x}$ = absolute acceleration of the payload
$y$ = displacement of the structure
$F_j$ = thrust exerted by the jet

1. Jet Thrust

The thrust exerted by the jet is modeled from basic fluid dynamics theory [3] as:

$$F_j = \frac{dm}{dt}(\dot{x} - v_f) \qquad (2)$$

where:

$\frac{dm}{dt}$ = mass flow rate of the air jet

$\dot{x}$ = velocity of the payload

$v_f$ = local flow velocity of the jet

The thrust produced always opposes motion of the mass. When the mass displacement is positive and to the right, the right side jet is activated to produce a left pointing thrust vector. Similarly, when the payload is to the left of the central position and moving toward the left, the left side jet is activated to produce a right pointing thrust vector. This model of the jet thrust has been verified in a static test performed at the Shock and Vibration Laboratory at Texas A&M University. The experiment consisted of a 1.41 kg mass suspended by wire 0.686 m. long. The air jet impinges on the mass and the angle the wire makes with the vertical is measured, as shown in figure 2a and 2b. Summing forces in the vertical and horizontal direction, the expected force balance is:

$$Mg \frac{\sin \theta}{\cos \theta} = \frac{dm}{dt} V_f \tag{3}$$

where:

M = mass of pendulum

θ = static angle

$\frac{dm}{dt} V_f$ = momentum flux of the jet

The jet diameter is 9/32-in. (7.1 mm). The air supplied was compressed air used throughout the building. The flow rate was varied from $2.12 \times 10^{-3}$ $m^3/S$ (4.5 cfm) $7.55 \times 10^{-3} m^3/S$ (16.0 cfm). The results of the experiment are shown in figure 3 in which the angle reached by the pendulum is plotted against flow. The relatively simple theory provides reasonable estimate of flow rate required to produce a thrust force on the mass. The figure shows that as the distance between the jet and the mass increases (angle increases), the flow rate required to maintain equilibrium is greater than the simple theory estimates. Factors contributing to this are experimental error; the fact that the air jet impinges on the mass at an angle of inclination, as shown in figure 2b, the thus the momentum flux is transferred less efficiently as the angle increases; and the loss of momentum flux due to temperature changes as the distance downstream of the jet exit increases. While experimental error and loss of momentum flux transfer due to impingement angle are factors particular to the static experiment, the possible loss of momentum flux due to heat transfer with the surrounding air can affect the thrust in the active control project as is indicated in the following section.

2. Jet Flow Equations

The expressions for $\frac{dm}{dt}$ and $V_f$ are derived based on the assumption that the momentum flux is constant throughout the flow field. It is required to know the mass flow rate and flow velocity separately because the thrust term in equation 2 depends on the relative velocity between the jet and the mass. The derivations for the terms are shown in Appendix A. For a one-dimensional, isothermal jet, the mass flow rate is:

$$\frac{dm}{dt} = 0.234\ x\ (M_o \varrho)^{1/2} \tag{4}$$

where:

$x$ = distance downstream of jet exit
$M_o$ = momentum flux at the jet exit
$\varrho$ = density of air

The momentum flux is assumed constant and equal to the momentum flux at the jet exit, $M_o$, where, for a round jet:

$$M_o = \rho \pi U_o^2 R^2 \tag{5}$$

where

$U_o$ = air velocity at the jet exit (assumed uniform)
$R$ = jet radius

The mass flow rate is evaluated at a downstream location using equation 4, and since the momentum flux is constant, the equivalent uniform flow speed is calculated.

3. Expected Air Consumption Parameters

The thrust required to give a 45 kg mass an acceleration of $10^{-5}$ g is estimated to be generated by a flow rate of $2.24 \times 10^{-5}$ $m^3/S$ (0.048 cfm) through a jet of diameter 0.79 mm (1/32 in.). The pressure drop across the valve required to produce this flow is calculated [4] to be $3.45 \times 10^3$ $N/m^2$ (0.5 psi). An example of a compressor that can supply such a system is a 12 V DC, 187W (1/4 HP) oil free, piston compressor rated a $3.30 \times 10^{-4} m^3/S$ (0.7 cfm) at $6.90 \times 10^5$ $N/m^2$ (100 psi). This is not expected to be a prohibitive power requirement, even if the control system operates continuously.

The type or frequency of excitation in the Space Station is not known. However, in order to assess the expected performance of the air jet system, the following example is investigated, based on the differential equation of motion:

$$M\ddot{x} = k\,(y-x) + \frac{dm}{dt}\,(\dot{x} - V_f)$$

where the terms are defined in the previous section.

The payload mass is 45 kg and the jet uses air at $21^{o}C$ ($70^{o}F$) and $1.0 \times 10^{5}$ $N/M^{2}$ (14.7 psi), with a flow rate of $2.24 \times 10^{-5}$ $m^{3}/S$ through a jet of 0.79 mm diameter. The spring stiffness is 5.0 N/m (0.34 lb/ft). This is the measured stiffness of a 95.3 mm (3.75 in) coil diameter spiral, self-coiling, flexible hose. The structural displacement, y, is assumed to be sinusoidal with period of 0.1/sec. The magnitude is calculated from the acceleration magnitude of 0.4 g. The jets are spaced such that the excursion of the mass is limited to $\pm$ 25.4 mm ($\pm$ 1.0 in). The control mechanism thresholds are set such that the jet is activated when the mass displacement exceeds 2.54 mm (0.10 in) or the velocity exceeds $1.0 \times 10^{-4}$ m/S ($3.94 \times 10^{-3}$ in/S).

The mass is displaced 10.0 mm (0.40 in) and released from rest at t = 0. Without the air-jet control systems, the mass oscillates with small amplitude at the driving frequency superimposed on a vibration at the natural frequency of the spring-mass system. The peak acceleration of the payload due to transmissions of the 0.4g structural acceleration through the spring is calculated to be $1.13 \times 10^{-5}$ g. The natural frequency vibration component has a period of 18 seconds, based on the mass and stiffness values, and amplitude equal to the initial displacement. In the absence of any control, this oscillation continues indefinitely.

With the air jet control, the response is shown in figure 4. The resultant motion of the mass is a sinusoid that decays linearly in time. The period of oscillation is 18 seconds, which corresponds to the natural period based on the spring and mass. The decay rate corresponds to an equivalent viscous damping factor of 0.0282. The time required for the mass to reach a steady state vibration about the central location is approximately 120 seconds. Of that time, the jet is on for 48.3 seconds, or 40 percent.

Experimental Program

The first phase of the experiment is intended to demonstrate the feasibility of the air jet control and to establish the jet flow parameters.

The experimental setup consists of the following major elements: a. sensor, b. electronic control system, c. air jets, and d. test mass. The electronic control system and air jets are designed for Space Station application. The sensor is a commercially available Linear Variable Differential Transforms (LVDT). This transducer limits allowable displacement to one dimension and thus is not applicable to Space Station application where three degrees of translation and three degrees of rotation are possible. The test mass is supported by an air-bearing table and is constrained to move in the horizontal plane. The air-bearing facility simulates zero gravity in the horizontal plane. The experimental set up is shown schematically in figure 5.

1. Control Algorithm

The LVDT produces a voltage which is proportional in magnitude and in sign to the displacement of the test mass. The voltage output from the LVDT is differentiated, producing a voltage signal proportional to the velocity of the test mass. The displacement and velocity proportional signals are each compared to thresholds values. The purpose for the threshold is to permit a dead band in which no control is activated. If the threshold is exceeded, a +15 volt signal is output on the line corresponding the sign of the input voltage, and a -15 volt signal is output on the other line. The outputs of the threshold detectors are combined in the comparator circuit. If the combined voltage on one of the lines is large (30v) and positive, this indicates that the mass is displaced from the center and moving further away. The comparator opens the relay to activate the appropriate jet. At the same time, the timer circuit is activated which limits the duration of the air jet pulse. If the combined voltage at the comparator is large and negative, this indicates either that the mass is within the dead band or that the mass is displaced from the central position but is tending toward it. In either of these cases, no air thrust is required.

## 2. Experiment Parameters

The air-bearing table is intended to provide friction-free horizontal motion. Any friction at the air bearing surface will add damping to the system which degrades the validity of the air jet efficiency determination. In an effort to quantify the air bearing equivalent damping, an experiment was run using an air bearing pad on a laboratory quality marble slab at the Shock and Vibration Laboratory at Texas A&M. The pad was connected to ground by springs, loaded with 90.72 kg (200 lb) and set into free vibration. From the logarithmic decrement, the damping coefficient, c, was measured to be 0.342 Ns/m ($1.95 \times 10^{-3}$ lb-s/in). The experiment was repeated vertical plane to eliminate the air friction. The damping coefficient was again found to be 0.342 N-s/m. Thus, the air film damping is negligible in comparison to the internal damping of the springs. The expected valve of the viscous shear damping coefficient is calculated from:

$$c_{exp} = A\mu/h$$

where:

A = contact area of the bearing surface
$\mu$ = dynamic viscosity
h = film thickness

The bearing film thickness is on the order of 0.051 mm (0.002 in). The expected damping coefficient is $c_{exp}$ - $8.38 \times 10^{-3}$ N-s/m ($4.79 \times 10^{-5}$ lb-s/in) for a bearing 0.152 m x 0.152 m (6 in x 6 in). The theoretical coefficient is 2.5 percent of the measured coefficient, which includes the springs and air bearing together. The theoretical value does not account for turbulent flow or surface roughness. Thus, as a first approximation, it is assumed that the air bearing damping is 10 percent of the measured coefficient, or $3.42 \times 10^{-2}$ N-s/m. The system defined in a previous section consists of a 45 kg mass connected to a spring with stiffness 5.0 N/m. The air jet control introduces an equivalent damping factor of 0.0282. The expected damping factor for three air bearing is $1.14 \times 10^{-3}$. Thus, the damping introduced by the air bearing is expected to be negligible in comparison to the damping introduced, based on the damping coefficient assumed above by the control system.

The air-jet momentum flux, $M_0$, required to produce an acceleration of $10^{-5}$ g of a 45 kg mass is $4.41 \times 10^{-3}$ N. The jets used in this experiment are commercially available solenoid operated air valves fitted with plugs in which a 0.799 mm (0.03125 in) hole has been drilled. The force exerted by the flow from the jet was measured statically by impinging the flow on a scale. The force versus pressure ratio across the jet curve is shown in figure 6. It is found that the force decreases linearly as the distance from the jet to the scale decreases. This indicates that the assumption of constant momentum flux is incorrect. However, the overall percent difference from the lowest to the highest force is approximately 25 percent. Thus, the constant momentum flux assumption is a valid first approximation. The curve of expected force is shown in the figure. The expected curve is derived from sharp-edged orifice flow theory [4], and is found to provide a good estimate of the force. The transient response of the jet was measured using a hot-wire anemometer. When the switch activating the solenoid is closed, the response shows second order characteristics with approximately 1.5 percent overshoot reached at 220 m-sec. This response time is a combination of the air-jet response and the anemometer response. The anemometer response time was measured separately and found to be approximately 150 m-sec. Thus, the response of the anemometer is dominant in the total measured system response. As a first approximation, the response time of the valve is assumed to be the difference, or 70 m-sec. The air-jet, when activated, will pulse for a predetermined time period of 0.50 sec. Thus, the response time is expected to have a negligible effect on the active control system performance.

## Results

The one-dimensional test set up has been fabricated and assembled on the precision air-bearing floor in the Technical Services Facility at NASA-JSC. The total mass of the air-bearing cart is 62 kg (137 lb). Compliant coupling is simulated by 2.33 mm (0.090 in) diameter wire arranged as four cantilevered beam elements. The effective stiffness of the springs is 19.34 N/m.

It was found that the total damping in the system, including auxiliary spring elements used to reduce rotation and lateral translation of the mass and the friction in the LVDT and its pulleys, was greater than the force exerted by the air jets. The free vibration of the mass is shown in figure 7. The natural frequency of the system is 0.089 Hz, and the damping factor is 0.084. The air jet identified in the previous section produces an equivalent damping factor of 0.0282. Since the damping in the experimental set up is three times the damping introduced by the air jet, the effect of the control system is expected to have negligible effect on the vibratory response.

In order to demonstrate that the control system, the following parameters are used. The air jet diameter is increased to 2.38 mm (3/32 in) and the pressure drop across the jet is increased to $8.28 \times 10^4$ N/m$^2$ (12.0 lb/in$^2$).

The vibrational response of the mass is shown in figure 3. It is seen that the air jet produces an equivalent damping, which increases the damping of the system by 0.015. In this plot, the jet, when activated, was pulsed for 0.5 second before reset. The analytical model is amended to reflect the modified jet parameters. Figure 9 shows the estimated free vibration response of the test mass. This plot correlates well with the measured free vibration shown in Figure 7. Figure 10 is the estimated response with the air-jet controller on. Again, the estimated response compares favorably with the experimental plot of Figure 8. It is found that the estimated effect of the air jet controller is strongly dependent on pulse time. The plot in figure 10 is obtained with a jet pulse of 0.1 second.

This is much less than the 0.5 second pulse set on the timer of the experimental controller. The discrepancy indicates that the air jet does not go to full flow at the instant that the solenoid is activated.

## Conclusion and Recommendations

An experimental facility incorporating air jet active vibration control has been fabricated. The facility has been used to show that the air jet controller effectively damps oscillations. An analytical model has been developed which estimates the effect of the air jet controller. The model can be used as a design tool to quantify parameters such as pressure drop, flow rate and net acceleration of the mass under combined air jet and spring-transmitted excitation. The model has shown that the solenoid dynamics limit the thrust produced by the jet to 20 percent of the thrust from an ideal valve which produces full flow when activated.

Continued work in this project will be in (1) sensor development, (2) extension to general plane motion control, and (3) model development.

(1) Sensor development. The current LVDT will be replaced by a non-contacting probe, such as accelerometer, ultrasonic tracker, or laser tracker. Such a transducer eliminates the need for pulleys and thereby reduces system friction. The transducer also permits extension of the system to general plane motion with both translation and rotation.

(2) Extension to general plane motion. The control system will be expanded to three-degrees-of-freedom. Sensors will be developed which respond both to translation and to rotation of the mass. The analog control circuitry and the air jet configuration will also be modified.

(3) Model development. The analytical model will first be refined to resolve the discrepancy between measured and estimated jet thrust noted in the previous section. The model will then be expanded to the general plane motion case. The modified model will be used as a tool in the design of the experiment.

It is expected that the same control algorithm which is obtained for plane motion control will be applicable to the more general six-degree-of-freedom application. Thus, the control system developed in the laboratory can be adapted for use in the Space Station.

Acknowledgments

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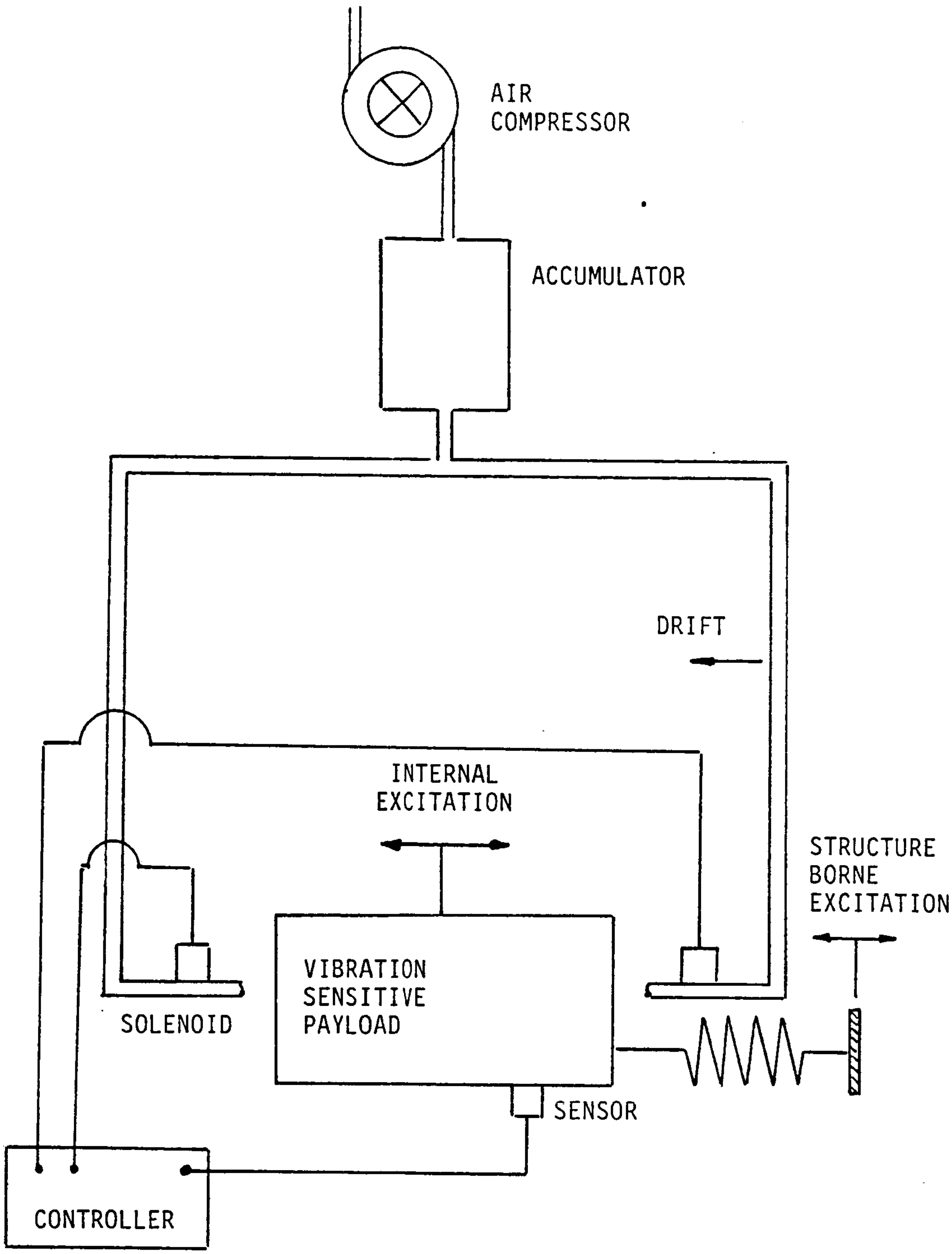


FIGURE 1. SCHEMATIC REPRESENTATION OF FEEDBACK CONTROLLED SYSTEM USING COMPRESSED AIR AS FORCE PRODUCER

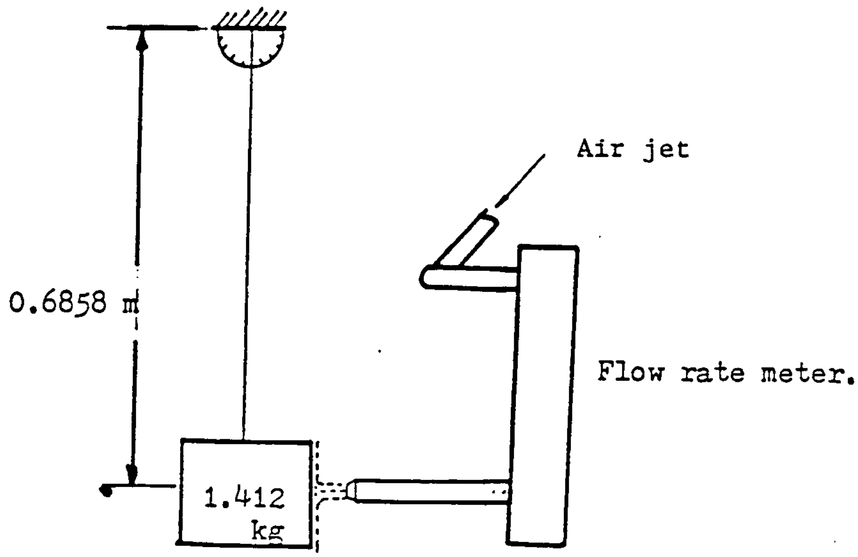


Fig. 2a. Schematic of experiment.

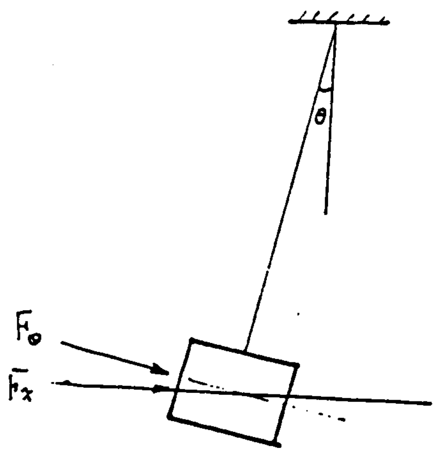


Fig. 2b. Equilibrium of Pendulum due to Impinging Air Flow.

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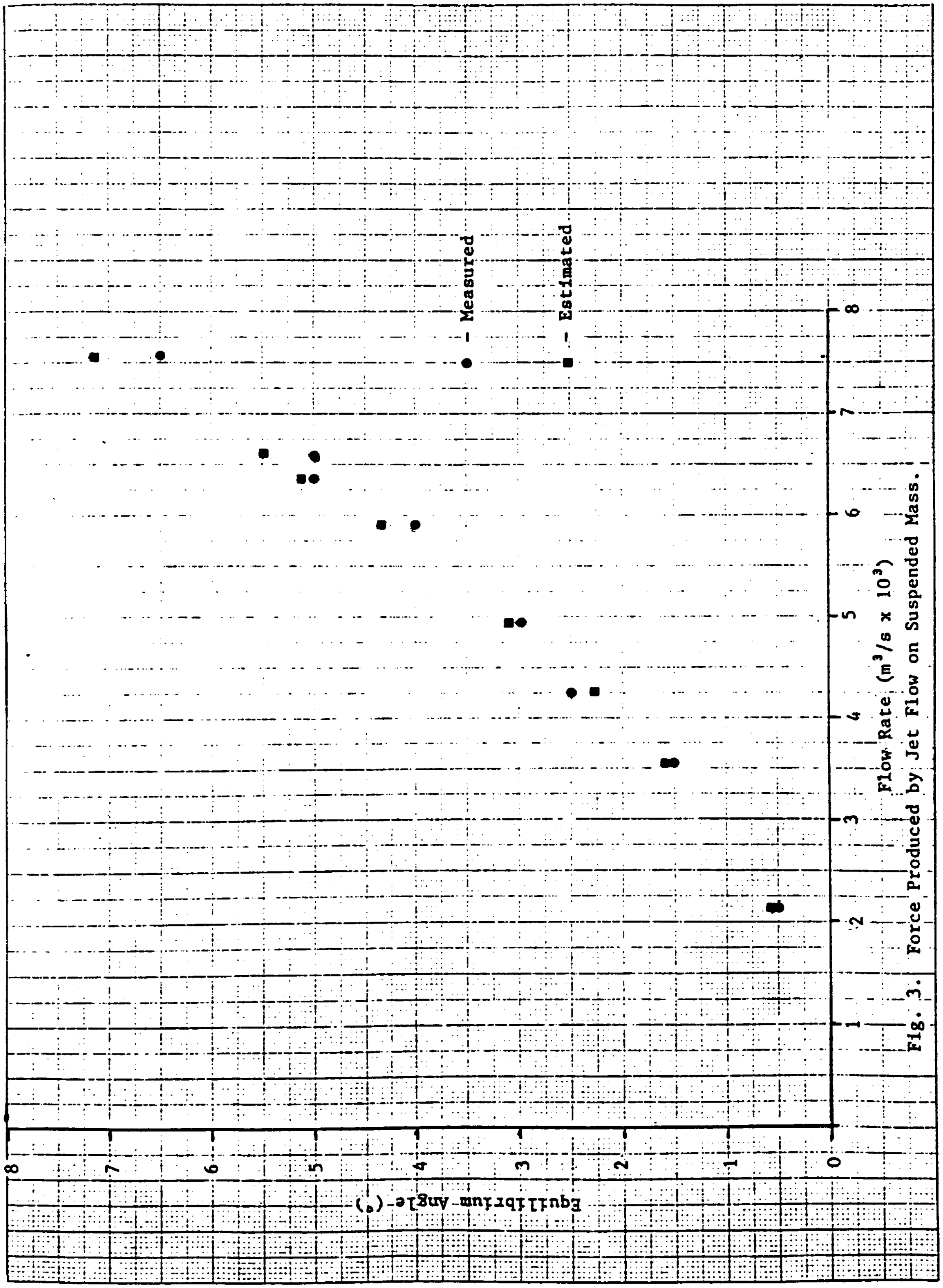


Fig. 3. Force Produced by Jet Flow on Suspended Mass.

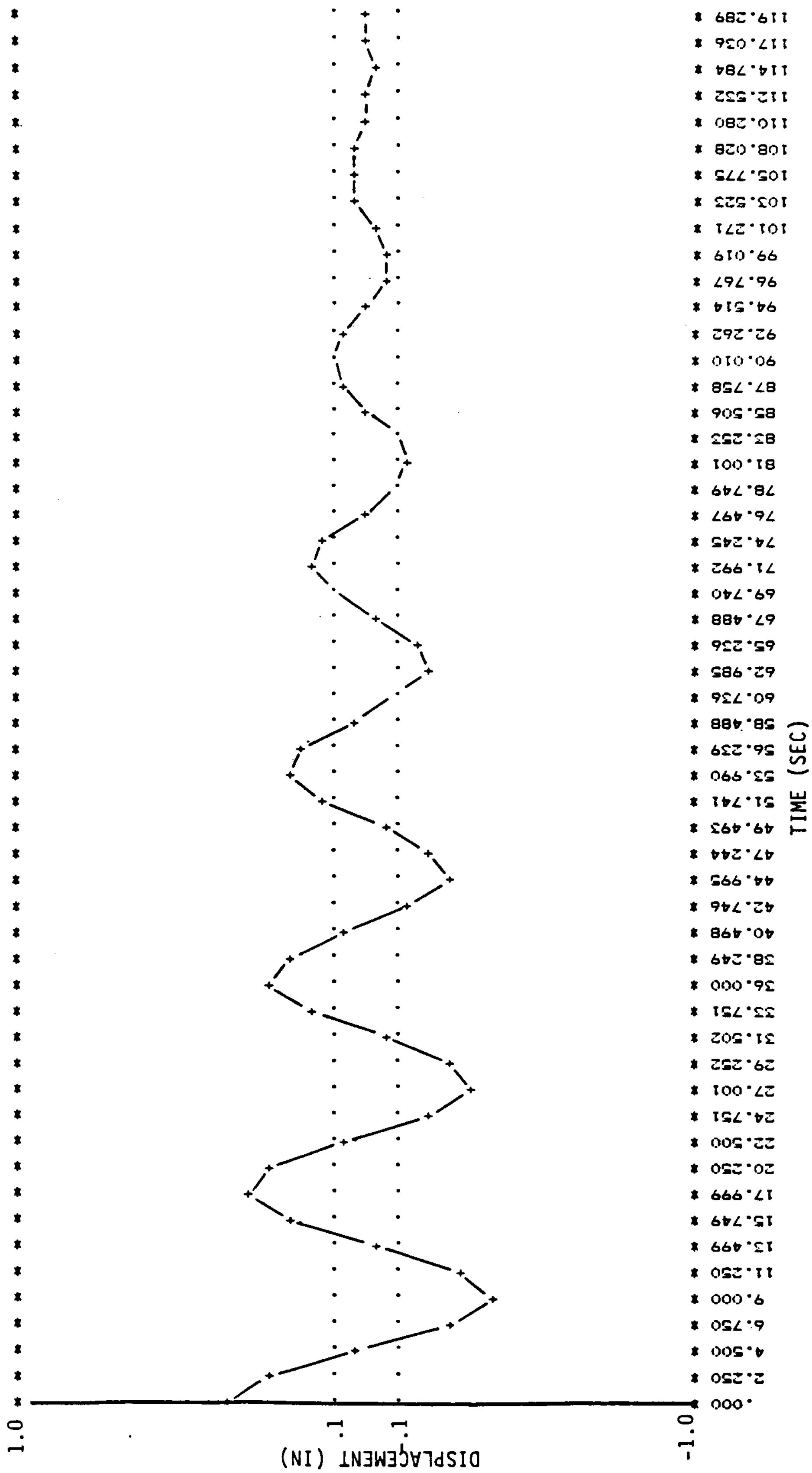


FIGURE 4. EXPECTED RESPONSE OF SPRING-MASS SYSTEM WITH AIR-JET CONTROL

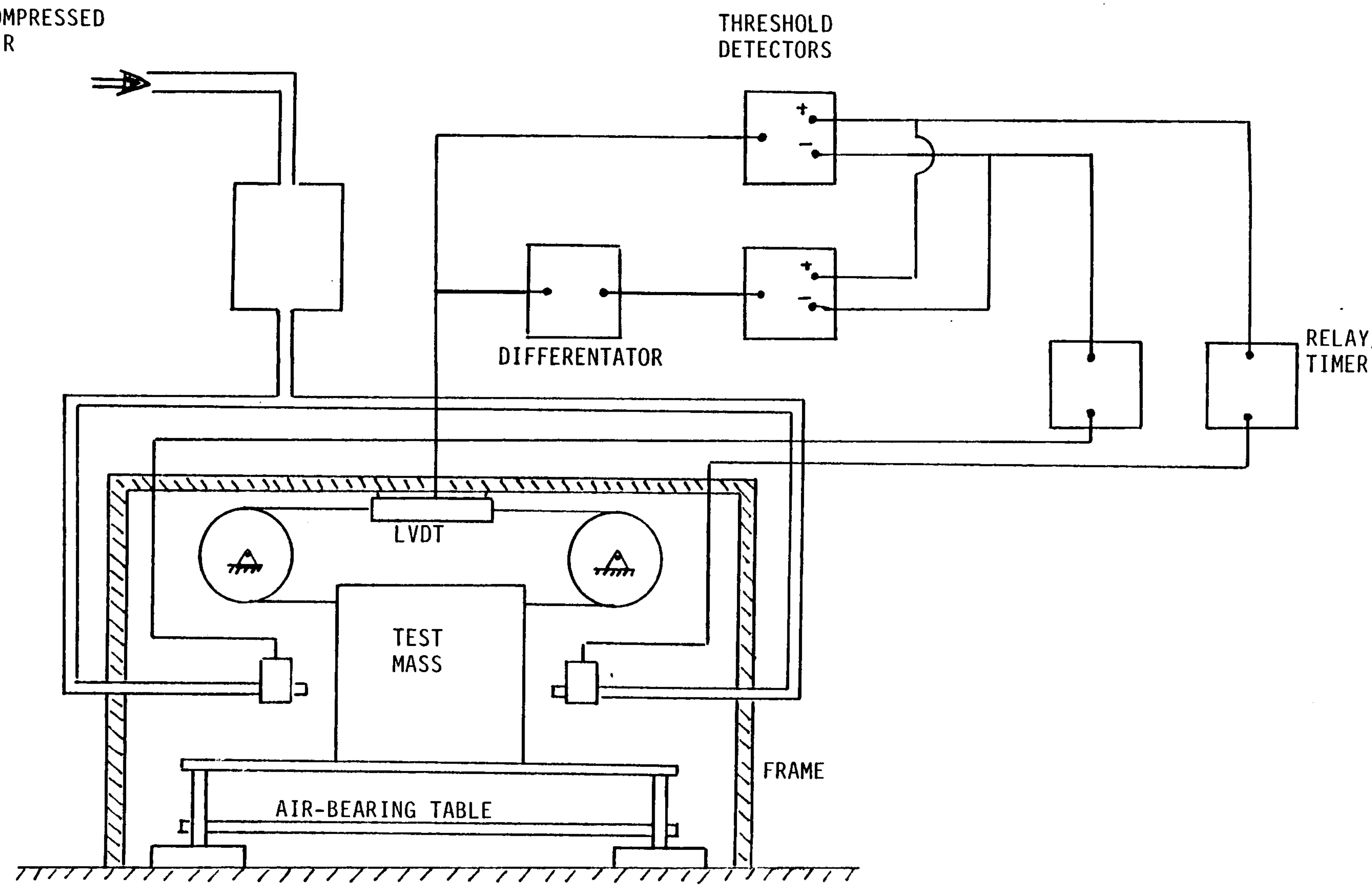


FIGURE 5. AIR JET CONTROLLER SYSTEM EXPERIMENT SET-UP

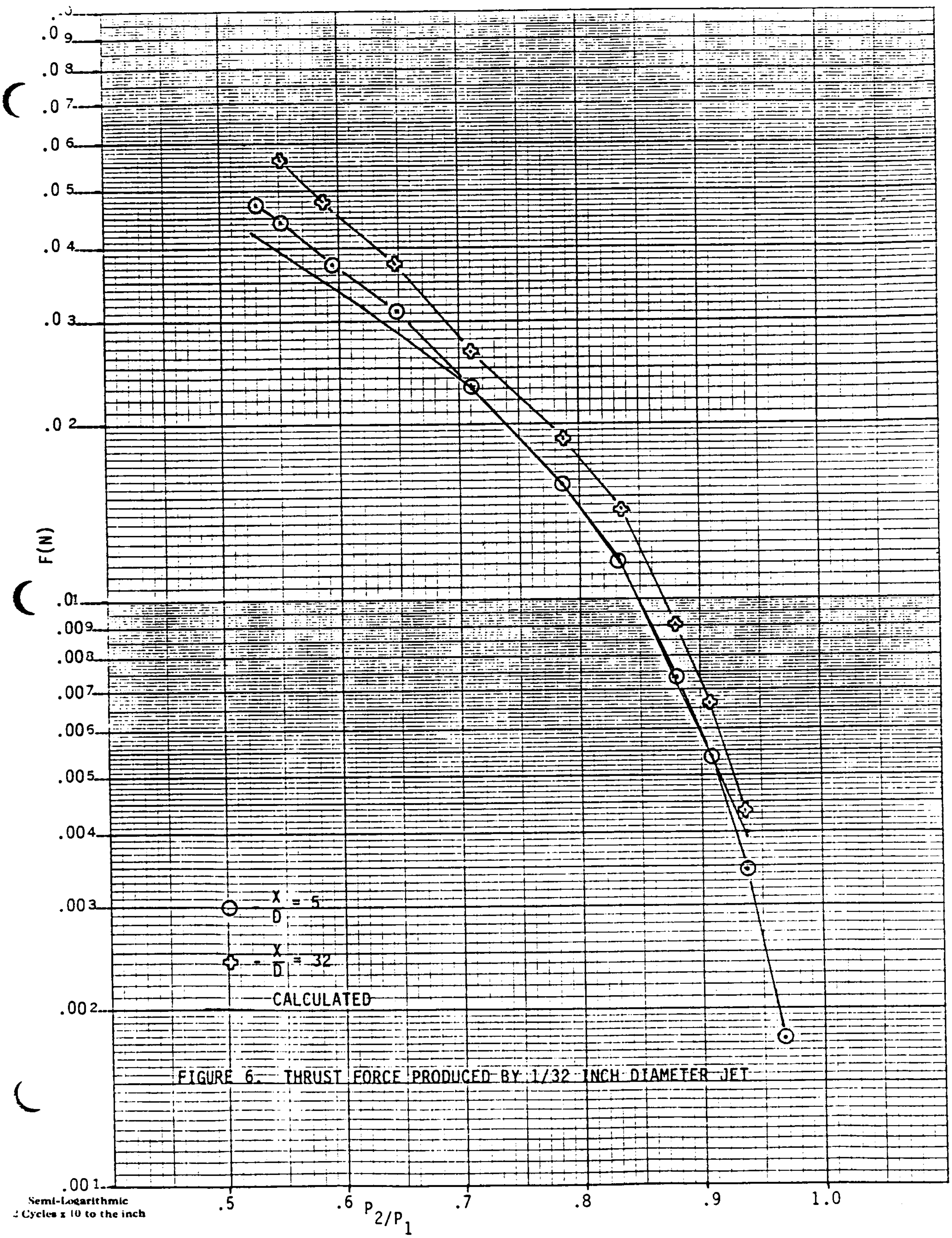


FIGURE 6. THRUST FORCE PRODUCED BY 1/32 INCH DIAMETER JET

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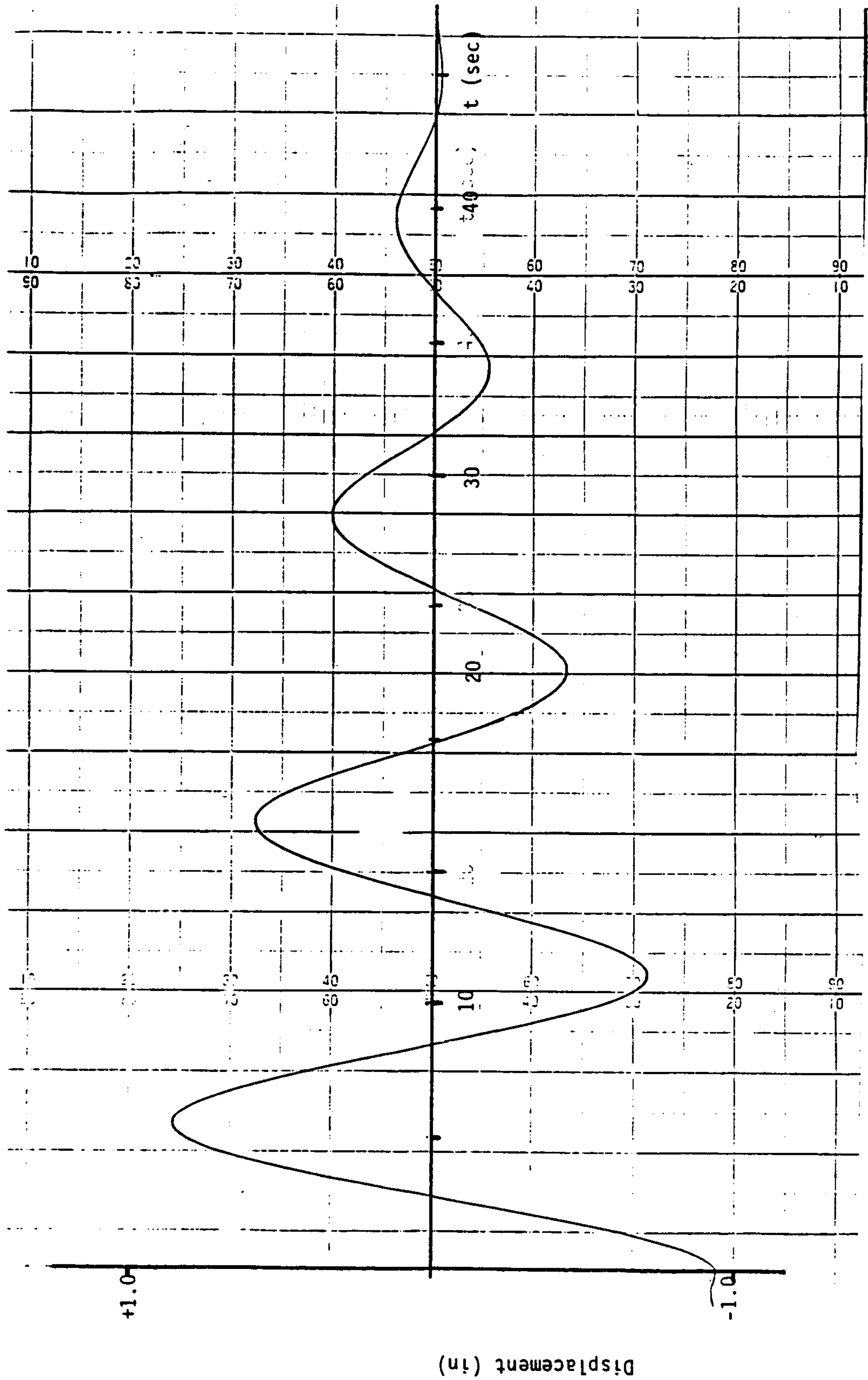


Figure 7. Free Vibrational Response of Test Mass

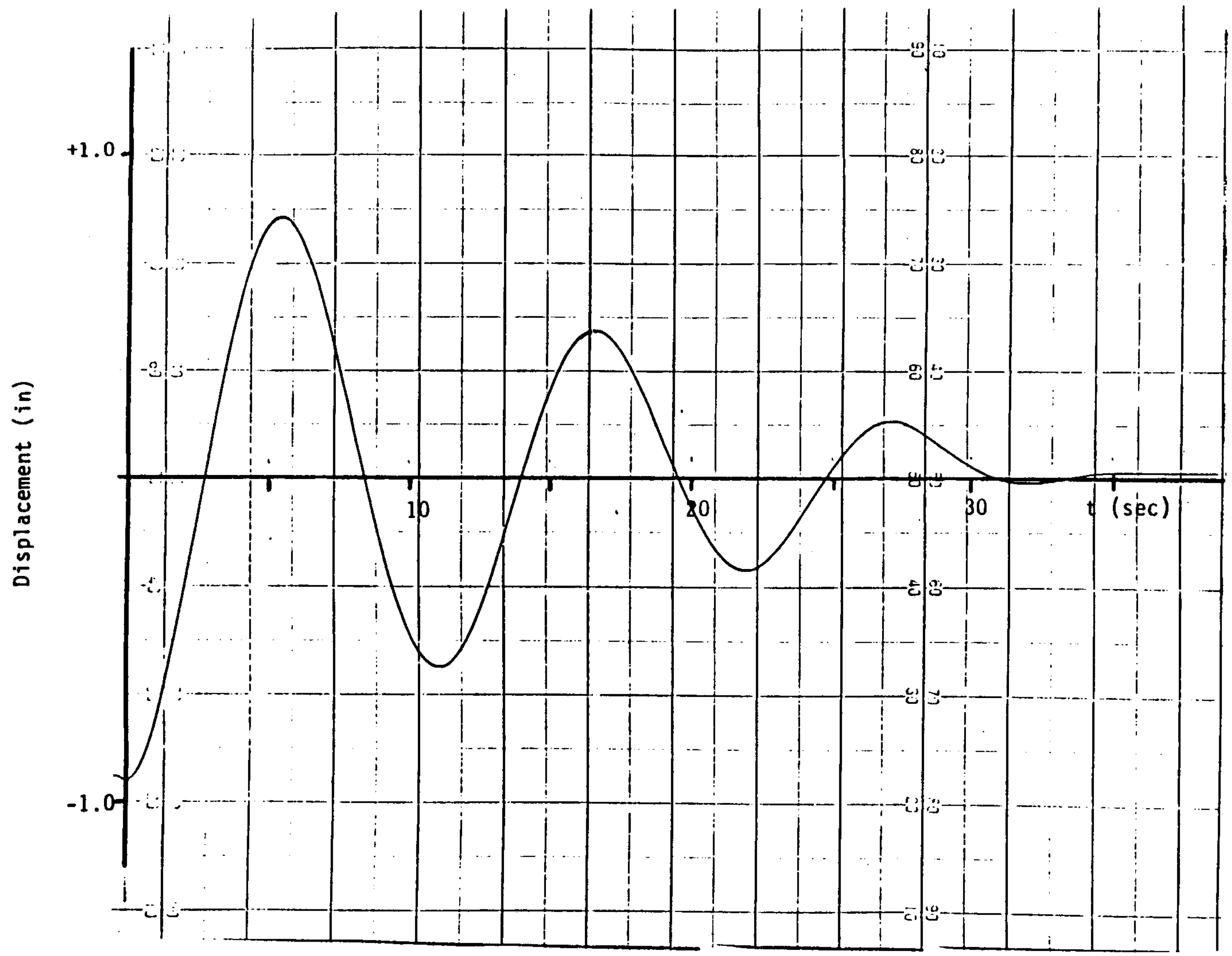


Figure 8. Test Mass with Air Jet Controller Activated
(0.5 Second Pulse)

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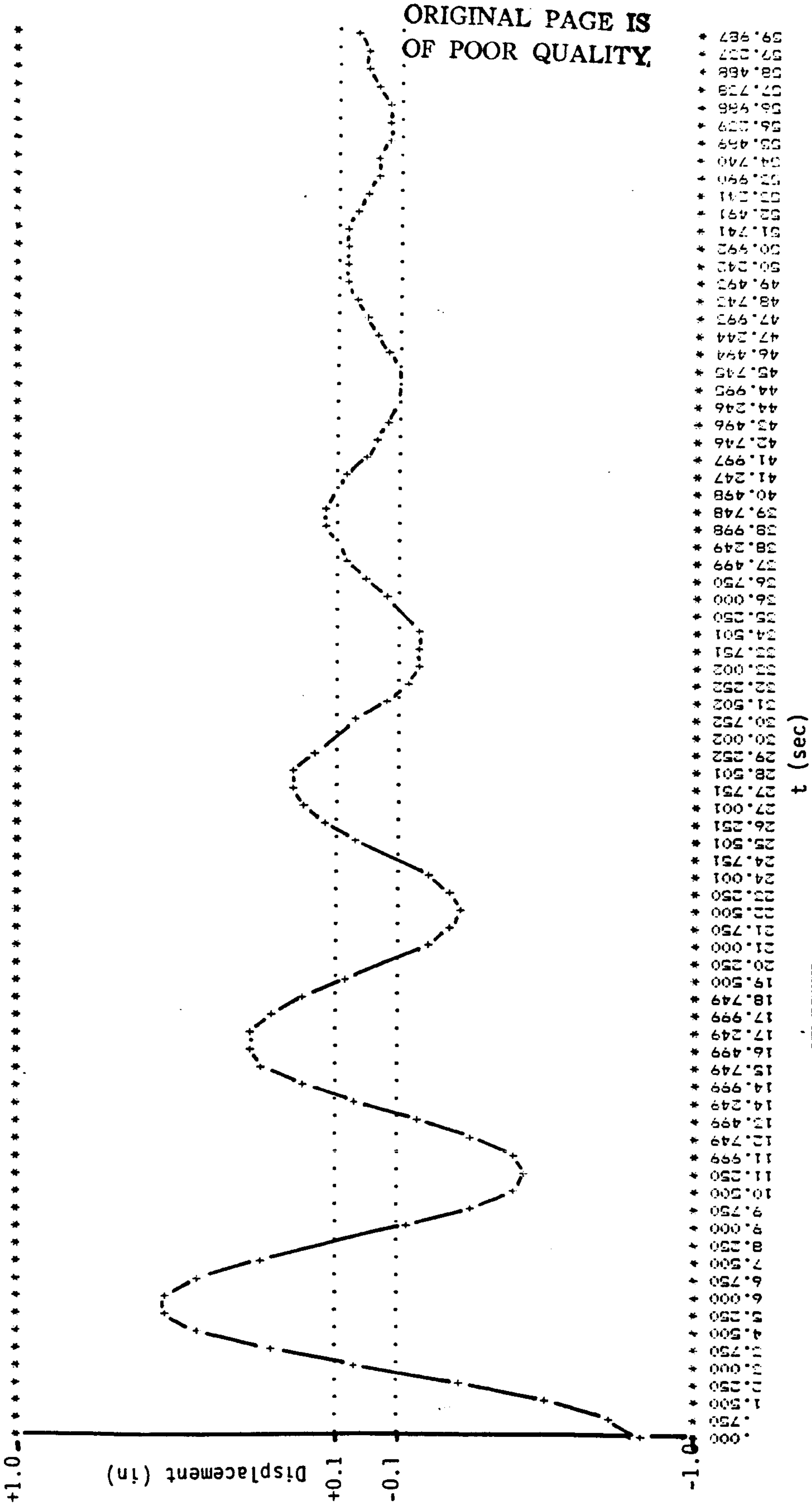


Figure 9. Estimated Free Vibration of Test Mass

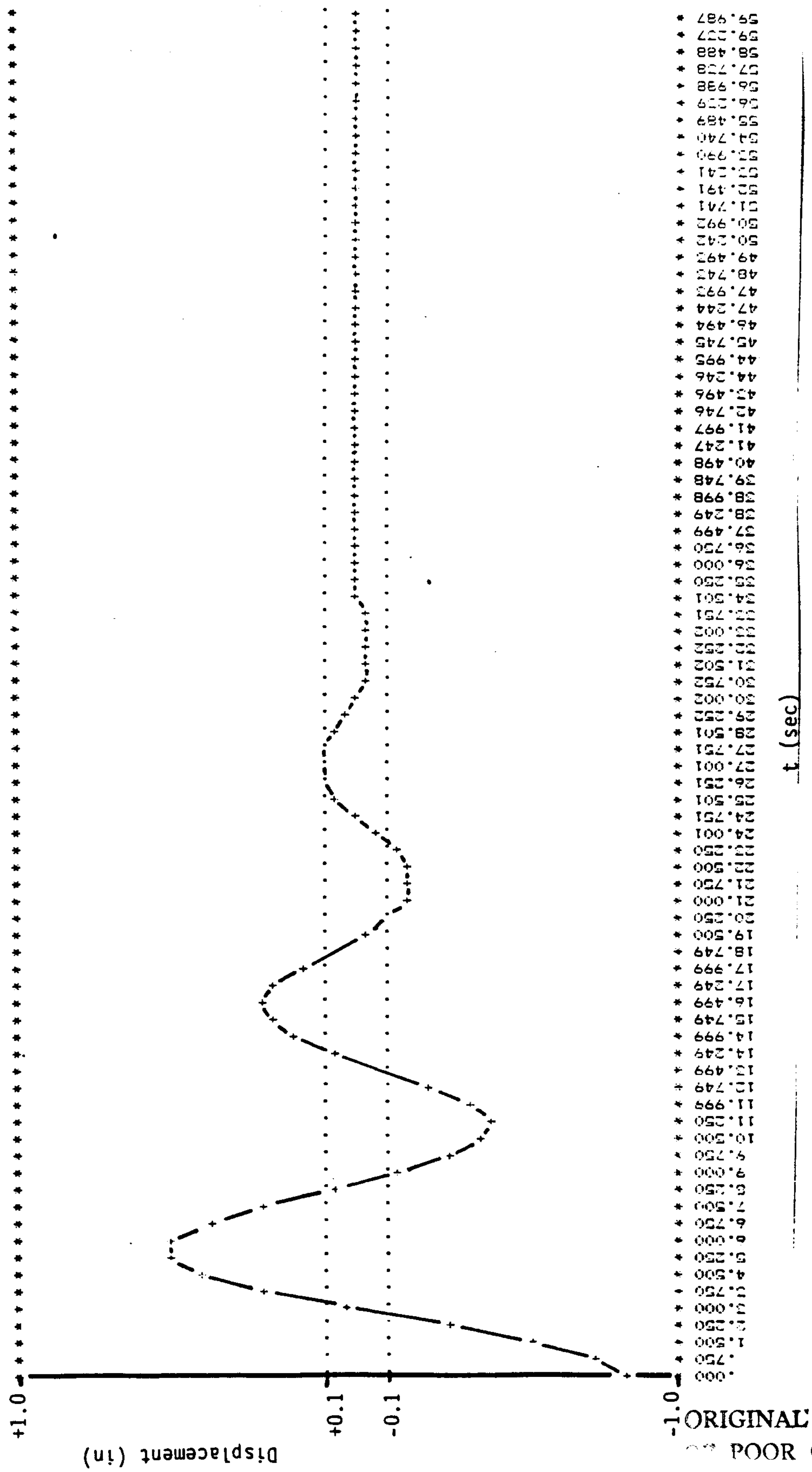


Figure 10. Estimated Vibration of Test Mass With Air Jet Control

(0.1 Second Pulse)

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## Appendix A. Derivation of Jet Flow Equations

It is assumed that the momentum flux, $\frac{dm}{dt} V_f$, for a one-dimensional, isothermal jet is constant downstream of the jet, and equal to the momentum flux at the jet exit, $M_o$. For a round jet:

$$M_o = \rho \pi U_o^2 R^2 \tag{A-1}$$

where:

$\rho$ = fluid density
R = jet radius
$U_o$ = fluid velocity at the jet exit (assumed uniform)

Downstream of the jet exit, the jet widens, the centerline velocity decreases and the velocity profile across the jet takes on a Gaussian distribution [5]:

$$\frac{U(y)}{U_{℄}} = e^{-K\eta^2} \tag{A-2}$$

where:

$U_{℄}$ = centerline velocity at downstream station, x

$$\eta = \frac{y}{x+x_o}$$

y = radial distance from the centerline
$x_o$ = virtual origin of the jet, assumed to be at the jet exit.
K = constant

The constant, K, is empirically derived from the jet widening rate, which forms an included angle of $9.8^{0}$ to the radial location at which the flow velocity is 0.5 the centerline velocity [6]. The value of K is 94.

In the fully developed region, beyond approximately 8 diameters downstream of the jet, the centerline velocity decreases at [6]:

$$\frac{U_{\mathbb{C}}}{U_{o}} = \frac{6.2}{(X/D)} \qquad \text{(A-3)}$$

where:

D = jet exit diameter

As the jet widens and the flow speed decreases, air surrounding the jet is entrained, such that the momentum flux remains constant.

Using the relationships obtained above in the mass flow rate equation:

$$\frac{dm}{dt} = \int_{0}^{\infty} 2\pi \rho y \, U(y) \, dy$$

it is found that, for an isothermal jet:

$$\frac{dm}{dt} = 0.234 \times (M_{0} \rho)^{1/2} \qquad \text{(A-4)}$$

Since the momentum flux is constant and equal to $M_{0}$, the equivalent uniform flow speed $V_{f}$ can be calculated from

$$V_{F} = \frac{Mo}{\frac{dm}{dt}} \qquad \text{(A-5)}$$

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