

A SUSPENSION SYSTEM FOR SIMULATING UNCONSTRAINED BOUNDARY CONDITIONS

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ABSTRACT

A multipoint suspension system is described for supporting test articles such that their unconstrained modal properties will not be altered by stiffness, inertia, or friction forces from the suspension. Intended primarily for flexible space structures, it is suitable for test items with natural frequencies as low as one Hz. Using a combination of passive pneumatic and active electromagnetic subsystems, the suspension offers a wide payload range, near-zero stiffness, zero static deflection, small added mass, and zero friction. The electromagnetic system can also provide active cancellation of added mass, accurate ride-height control, and controlled disturbance input. The concept and hardware are described, test results are given, and applications experience from several installations is discussed. Current development efforts are described, aimed at testing of payloads such as solar arrays and large beam-pointing experiments.

INTRODUCTION

Unconstrained or "free-free" boundary conditions for test articles are often required in dynamic testing. It may be necessary to compare predicted and measured modal properties under some standard, reproducible boundary conditions, or to obtain measured free-free modal properties for use in component modal synthesis of a multi-component structural system. Perhaps most important, the need can arise when the test article is a spacecraft which actually operates with free boundaries: the so-called zero-G simulation problem. In any case, the practical requirement is to support the weight of the test article without imposing boundary constraints that would change its free-free normal modes.

Free-free conditions can be accurately simulated if the forces of constraint from the suspension system are small compared to internal forces due to stiffness and inertia. A common rule-of-thumb is that the vertical plunge frequency on the suspension should be at least an order of magnitude below the first flexural natural frequency of the test article. For natural frequencies above about 20 Hz, this requirement can often be met by hanging the

test article from simple linear springs. For large or very flexible test articles with first frequencies below about 10 Hz, this simple approach becomes impractical due to static sag and surge modes of the springs themselves. As test article frequencies approach 1 Hz or less, the suspension problem becomes quite challenging.

The spacecraft problem in particular has produced a need for an advanced suspension system for dynamic testing. This paper describes such a system. Begun in 1988 as a research topic under the NASA/LaRC Dynamic Scale Model Technology Program [1] [2], the development has now reached the status of a commercial product in daily use on a number of aerospace programs. Called the pneu-mag system because of its combination of passive pneumatics and active electromagnetics, it provides a unique combination of wide payload range, very low stiffness, low added mass, and zero friction. These allow high-fidelity free-free dynamic testing of low frequency structures.

The following section describes in general terms the requirements and figures of merit for a suspension system for dynamic testing. The operating principle, analytical model, and current hardware implementation of the CSA Engineering pneu-mag system are described. Performance test results are given for comparison to the figures of merit. Finally, some current development efforts for specialized applications are summarized.

SUSPENSION SYSTEM REQUIREMENTS

The essence of the suspension problem is to support the weight of the test article without imposing constraints or dynamic forces at the support points that would change its normal modes. Constraint forces can be produced by stiffness, mass, friction, or resonances of the suspension devices. The figures of merit described below are primarily measures of the degree to which these undesired forces are minimized.

Historically, efforts to develop hardware for zero-G simulation have often concentrated simply on achieving very low stiffness with adequate payload capacity. However, experience reported here has shown that the low-stiffness requirement is only one of several and is not necessarily

the most difficult. A particular example of neglecting other effects in order to minimize stiffness has been the use of helium-filled balloons to suspend a test article. While vertical stiffness is small (zero were it not for density gradients in the surrounding air), it can easily be shown that the mass added by the balloons can never be less than about 16% of payload mass. This added mass will corrupt even the lowest modes of the test article and will fundamentally change the higher modes due to inertia constraint forces. Further, the added mass cannot be accurately included in the analytical model because both it and its coupling to the article are anything but rigid. A systematic, balanced approach is necessary that considers all sources of undesired constraint forces.

Suspension systems for dynamic testing have usually taken the form of support-from-above cable arrangements with a soft spring in series with each cable. Pendulum action gave a simple, effective soft restraint in the horizontal direction, given adequate cable length. The version of the pneu-mag device described here uses this overhead arrangement. Another variant now in development will support its payload from below, thus removing the head-room requirement.

The remainder of this section describes specific requirements and how they have affected the development of the pneu-mag system.

Payload Applications to date have required from 15 to 600 lb per support point (i.e., per suspension device) for overhead suspension and as much as 3000 lb per device for support-from-below devices. While there is obviously no limit to what might be required, most applications seem to fall between 30 and 400 lb per support point. Equally important for a general-purpose system is the range of payloads that can be supported. The pneu-mag devices described in later sections can accommodate a weight range of about 6/1 by simple adjustments (no changing of parts) and about 20/1 with simple changes.

Static Deflection Linear springs, regardless of their implementation, are usually ruled out for low frequency applications by the contradictory requirements for low stiffness and small static deflection. A primary feature of pneumatic springs, shown later, is that they decouple the local tangent stiffness from static deflection. Pneu-mag devices use this feature to operate at zero static deflection.

Stiffness Allowable stiffness is usually specified in terms of the vertical rigid-body plunge frequency on the suspension. For a first flexural frequency of 1.0 Hz and plunge frequency of 0.1 Hz, this translates to approximately 0.1 lbf/inch of stiffness per 100 pounds of payload. Linear springs are obviously excluded; the static deflection would be 1000 inches. Figure 1 shows a more payload-

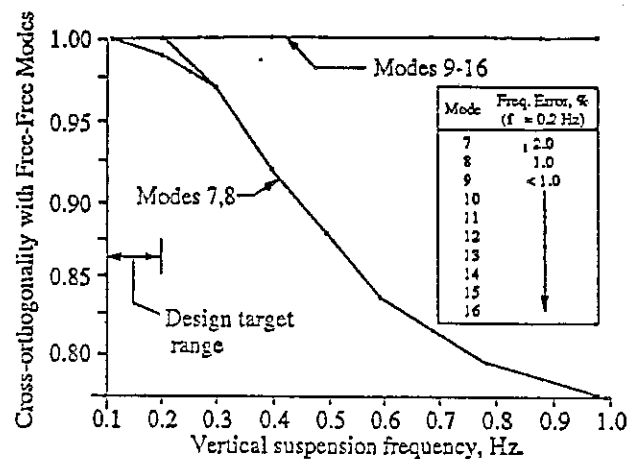


Figure 1: Effect of suspension stiffness on cross orthogonality between unconstrained and suspended modes of a mass-loaded truss structure

specific method of determining suspension stiffness effects. It shows the gradual loss of cross-orthogonality between true free-free modes and modes including suspension stiffness as the suspension stiffness is increased. The example structure is a dynamic scale model of an early configuration of Space Station Freedom. Physically, it is a number of rigid masses held together by light truss-work. Modes 7 and 8 are the first global flexing modes at slightly over 1 Hz. The example shows that the 10/1 frequency separation criterion is, in this case, probably conservative. It also shows that the effect of suspension stiffness varies greatly from mode to mode. The pneu-mag devices have been designed from the outset to have a variable suspension frequency with a range extending down to 0.1 Hz for most payloads.

Moving Mass Any suspension device adds some amount of mass to the payload, and can thus change its properties regardless of suspension stiffness. For light, flexible payloads in particular, it often occurs in low frequency testing that inertia constraint forces due to moving suspension mass exceed stiffness constraint forces for frequencies over a few Hz. Allowable added mass for the example structure described above was determined by finite element analysis to be about 5%. The pneu-mag device easily meets this for payloads towards the upper end of its range. For very light payloads, an active mass canceling feature has been developed. Described in a later section, it has demonstrated the ability to cancel about 85% of the 3.4-lbm moving element of a pneu-mag device with good stability margin. This allows testing a 20-lb payload with less than 3% added mass.

Stroke Modal testing of even large spacecraft typically requires only a modest amount of vertical suspension stroke, on the order of one inch. The real requirement is set by measurement of local modes of very flexible appendages such as solar arrays, or by other dynamic tests involving rigid-body rotations of the payload. To meet these requirements, the current generation of pneu-mag devices has a vertical stroke of 6.0 inches.

Friction This is perhaps the most difficult requirement and has been an important driver in developing the pneu-mag concept. Unlike forces due to suspension stiffness or moving mass, friction is inherently nonlinear, often not repeatable, and always expensive to model in structural dynamics. A high priority was therefore given to reducing friction to the point where it had no measurable effect. This leads to a requirement that can be formulated in two ways. Either the friction force in the device must be small compared to its stiffness force, or friction force must be too small to produce a measurable change in payload acceleration based on typical accelerometer noise floor levels. Either leads to an allowable friction force on the order of 0.01% of payload weight. The nominal specification for the pneu-mag devices was set at 0.005%.

Controllability The projected large, flexible spacecraft applications that motivated the pneu-mag suspension system would have required 10-20 support points to avoid excessive weight-induced stresses. Even then, weight distribution between devices would have to be carefully controlled by tuning each device ride height. Testing efficiency required that this be done remotely, probably by a computer. While no such grand-scale application has actually materialized, the need for remote control was basic to the pneu-mag concept. The feature has proven to be valuable in day-to-day testing of smaller test articles. It has become a de facto requirement that a single operator be able to adjust from one location all devices of a multipoint suspension system in order to obtain a desired weight distribution.

Local Modes Accurate modal testing demands that the suspension devices contribute no local modes of their own within the test frequency band. Failing this, the suspension becomes part of the test article, a serious complication that the test engineer would much rather avoid. The pneu-mag devices are designed such that their only moving part is relatively light and stiff, with its first free-free mode well over 100 Hz.

PNEUMATIC-MAGNETIC PRINCIPLE

Figure 2 shows the basic operating principle of the pneu-mag device. Two parallel subsystems, one pneumatic and one electromagnetic, give the device its name. Figure 3 shows a simplified schematic diagram.

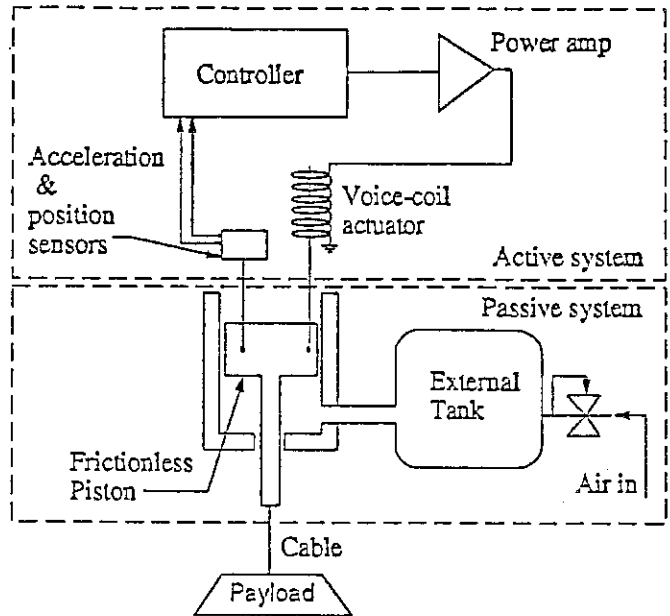


Figure 2: Principle of pneu-mag suspension device

The entire payload weight is borne by a special frictionless air piston operating vertically in a closely fitted cylinder. By porting the cylinder to an external volume through a large diameter line, the stiffness of the air spring thus formed can be made very small while retaining a large payload capacity. A precision pressure regulator maintains the mean cylinder pressure and supplies makeup air. For small volume changes, the regulated air spring behaves like a linear spring in series with a dashpot (Figure 3). The spring stiffness can be determined from elementary thermodynamics by assuming isentropic pressure changes. The result, expressed in terms of vertical suspension frequency, is as follows.

$$f_s = \frac{1}{2\pi} \sqrt{\frac{\gamma g A}{V}} \sqrt{1 + \frac{p_{amb}}{p_g}} \quad (1)$$

where

- f_s = suspension frequency, Hz
- γ = specific heat ratio for air (1.40)
- A = piston area, in^2
- V = tank volume, in^3
- p_{amb} = ambient pressure, psia
- p_g = gage pressure in air spring, psig

Eq. (1) and Figure 2 illustrates two well-known advantages of air springs. For p_g of several atmospheres or more, f_s is nearly independent of payload weight. That is, the suspension stiffness automatically changes in proportion to payload. Secondly, by adjusting the cylinder pressure to equilibrium with the the payload weight, there is effectively no static deflection of the spring.

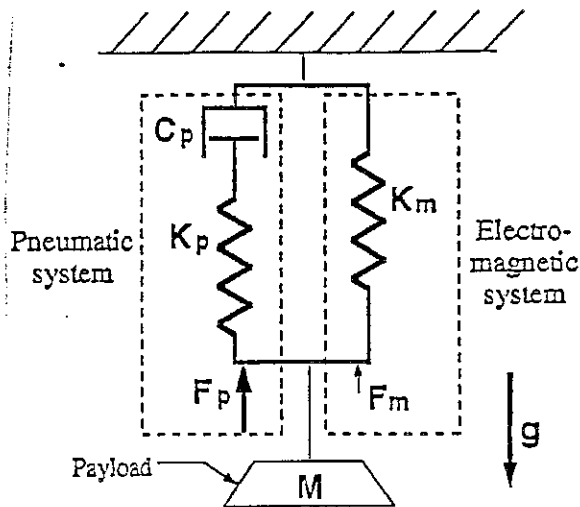


Figure 3: Simplified schematic of pneu-mag device

The series dashpot is the result of the action of the regulator. It vents air in or out of the external tank, attempting to keep the pressure constant in spite of piston movement. The spring-dashpot combination has a frequency-dependent stiffness with a first-order, high-pass break frequency determined by the size of the external tank and the characteristics of the regulator. The regulated pneumatic spring has no static stiffness at all: it can be in equilibrium with the payload weight at any vertical position of the piston within the cylinder. This property can be a disadvantage since the payload can float to the end of the stroke and strike the limit bumper. Thus arises the need for the parallel magnetic system.

The active part of the system includes a long-stroke, voice-coil magnetic actuator, a displacement sensor, controller, power amplifier, and (optionally) an acceleration sensor. A negative-feedback, low-gain displacement loop is implemented to serve as a noncontact "spring" with a very small, variable stiffness. It keeps the moving element of the device near the center of the working stroke, thus preventing it from striking the limit bumpers when excitation is applied to the payload. A secondary use of the displacement loop is to provide fine-trim control of ride height. A small, variable DC offset voltage can be injected into the loop. It serves the same purpose as adjusting the pressure regulator set point but does so with much faster response and much greater resolution.

The active system also provides other useful features. A positive-feedback acceleration loop can be added to cancel most of the moving mass of the suspension device. This has improved the fidelity of the zero-G simulation for small payloads, on the order of 20 pounds, where the added mass can have a significant effect [3]. An external dynamic signal can be summed into the loop to provide

a controlled vertical excitation to the payload. This is useful in situations where the payload is very flexible and would be affected by the rotational constraint of a shaker pushrod. A modest multipoint-excitation modal test can be performed using only the actuators and sensors built into the pneu-mag devices. The DOF's are, of course, limited to the vertical direction at the hang points.

Special versions have been built with a lowpass filter in the displacement loop such that DC stiffness could be increased for improved centering without imposing constraining stiffness at higher frequencies. Negative velocity feedback was necessary to stabilize the loop in the presence of the phase lag of the filter. The arrangement was not entirely satisfactory because the velocity feedback slightly increased the apparent damping of the lowest modes of the test article [4]. The eventual solution was a basic redesign with much longer stroke (described below) which eliminated the need for enhanced DC stiffness. Frequency-dependent stiffness is one of many features that have been investigated in a research context since the first pneu-mag device was built in 1988.

CURRENT HARDWARE

Figures 4 and 5 show two views of the most recent pneu-mag version along with nomenclature. Figure 6 shows an example of this third-generation design which is currently in use at NASA/LaRC with the CSI Evolutionary Model. Figure 7 shows a pneumatic control panel and electronic chassis for a three-device system.

Referring to Figures 4 and 5, the pneu-mag device is composed of a carriage that moves vertically in a rectangular box frame on four air journal bearings. The moving carriage is composed of a single main rail down the center with a horizontal crossmember fixed to it about one-third of the way from its upper end. Two identical frictionless piston/cylinder sets are used, one on either side of the main rail, with each piston lifting against one end of the crossmember via a connecting rod. Both cylinders are ported through the bottom of the box frame to hoses connected to a common accumulator tank. The main rail runs in two air bearings, one near the top of the frame and one near the bottom. An alignment rod extends upwards from either end of the crossmember, through a small air journal bearing fixed to the frame. The alignment rods prevent the carriage from rotating in the main journal bearings, thus keeping the connecting rods coaxial with the cylinders and preventing side-load on the pistons. The single-main-rail, twin-piston geometry is used to avoid bending stress in the rail and allow a high payload capacity (large piston area) with minimum carriage mass. The design replaced an earlier single-piston configuration with shorter stroke [4][5].

The payload is connected to the carriage by a cable that

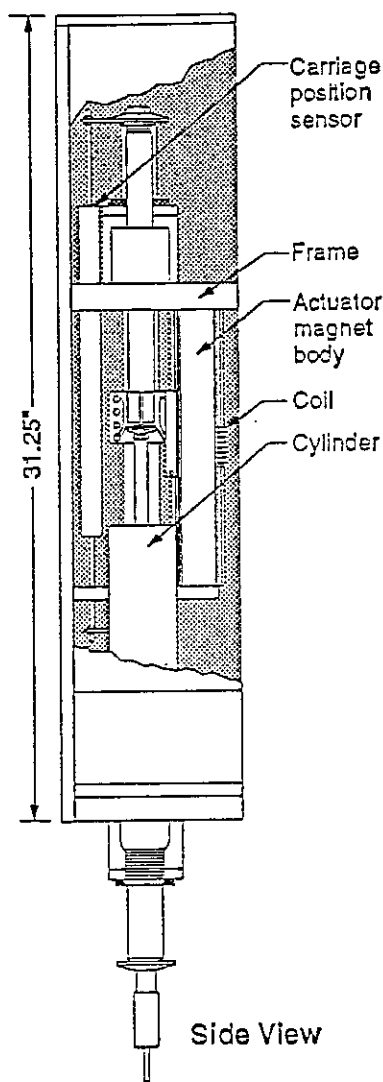


Figure 4: Side view of pneu-mag suspension device

attaches to the lower end of the main rail. An optional load cell may be interposed between the cable and carriage to sense cable tension.

The magnetic voice-coil actuator is a custom, long-stroke design that uses a coil connected to the carriage crossmember and a magnet body mounted to the frame. High-energy NdFeB magnets are used for maximum force capacity. The magnet structure is designed to produce a high flux density in the air gap with minimal variation in the stroke direction.

The carriage displacement sensor is a non-contacting LVDT mounted behind the carriage main rail. A piezoresistive (DC-coupled) accelerometer is mounted to the carriage for the mass-canceling loop. A pressure sensor trans-

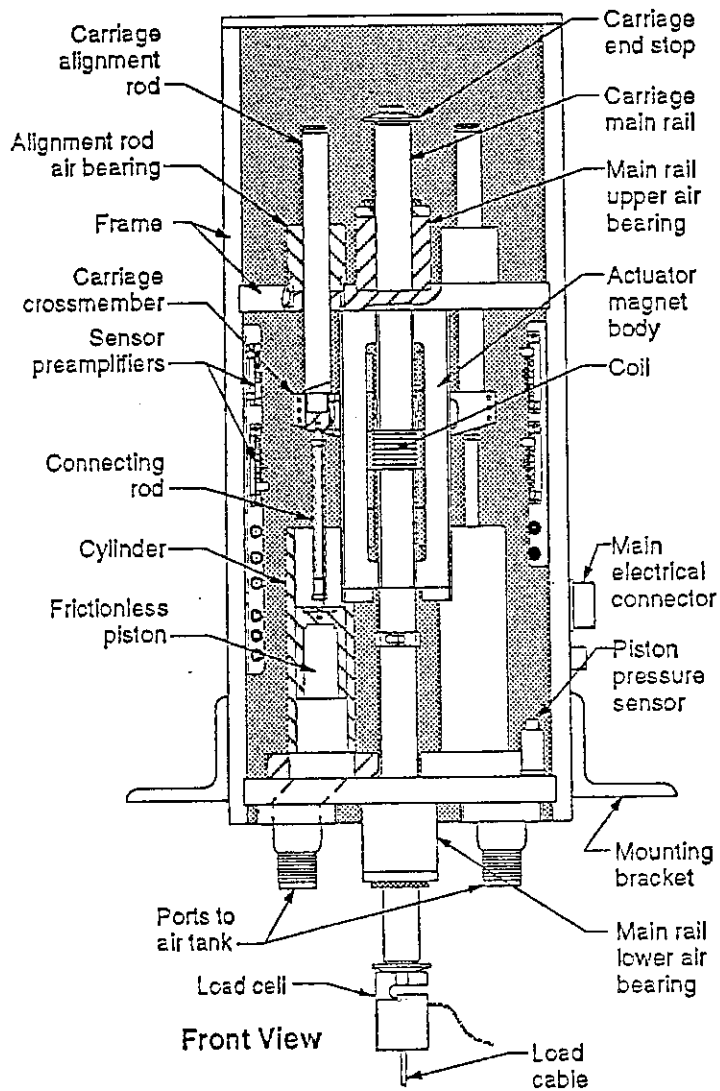


Figure 5: Front view of pneu-mag suspension device

duces the cylinder pressure to obtain a continuous analog reading of piston force. Each suspension device connects to the control panel by a single electrical cable and two compressed air lines.

The key to the performance of the device is the fact that the load is supported completely on air. The combination of air bearings, noncontacting actuator and displacement sensor, and floating pistons eliminates any source of friction. The only connections between the moving and fixed parts of the device are the small wires carrying current to the actuator coil and (optionally) the small signal cable from the accelerometer.

Nominal specifications for the device as shown are: maximum payload (at 80 psig): 500 lb; active stiffness: vari-

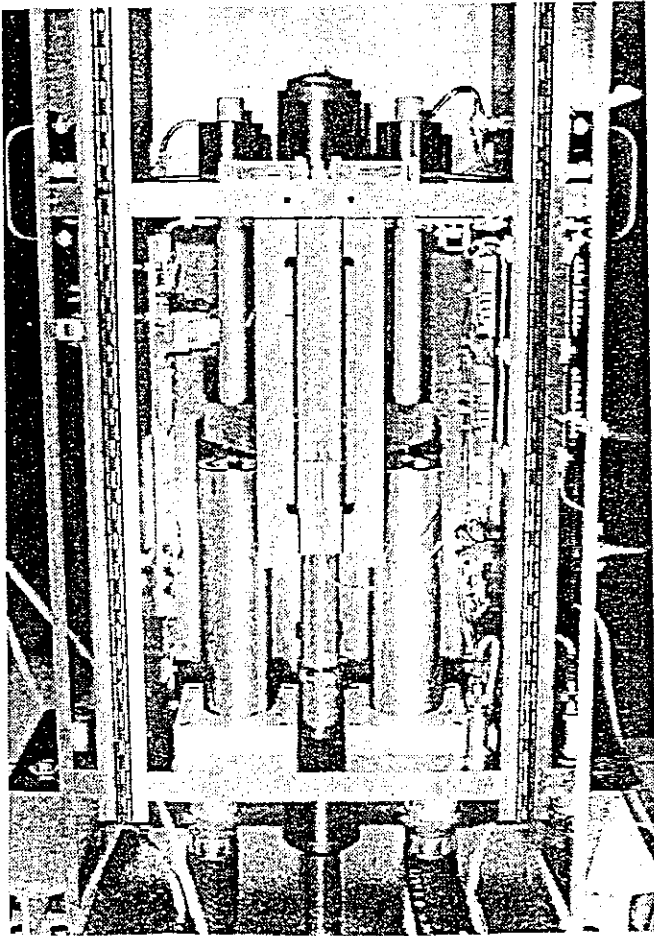


Figure 6: Pneu-mag suspension device currently in use at NASA/LaRC

able over 0.05-2.0 lbf/inch; vertical suspension frequency: 0.1-0.2 Hz; breakaway friction: under 1 gram; stroke: 6.0 inches; carriage mass: 6.0 lbm.

PERFORMANCE TEST RESULTS

This section presents some typical test results for pneu-mag suspension devices. In all cases, a rigid test payload was used.

Figure 8 shows the measured vertical displacement, force FRF with two different payloads. The displacement loop was closed with a loop gain of about 0.2 lbf/inch. The internal actuator and displacement sensor were used to allow FRF measurements at the very low frequencies involved. The series-dashpot effect of the pressure regulator becomes progressively more evident as the payload is increased. At 350 lb, the suspension resonance has disappeared due to regulator action. This is exactly what is needed in a suspension device; the load is virtually floating with negligible vertical constraint.

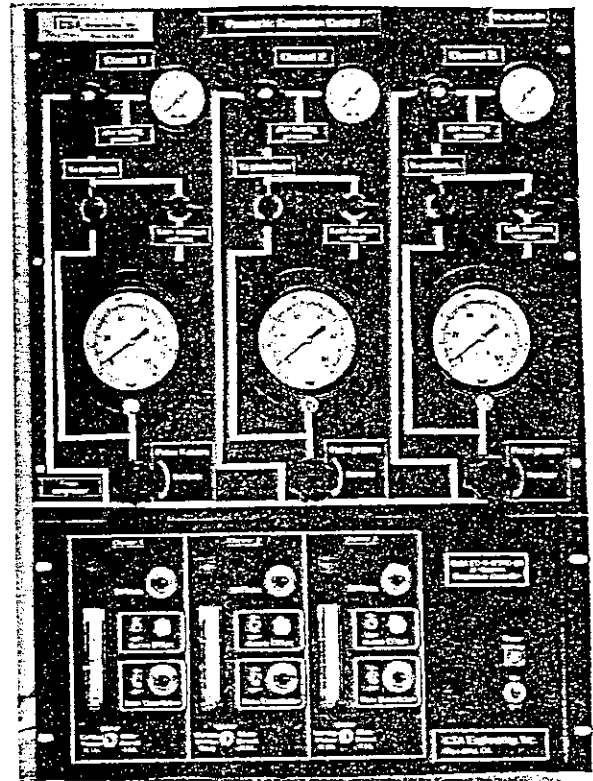


Figure 7: Pneumatic and electronic control panels for 3-device pneu-mag system

Figure 9 shows a typical breakaway friction test result. The pneu-mag device was loaded to 150 lb (68 kg), active loop stiffness was set to 1.5 lbf/inch, and a 1.1 gram weight was carefully lifted from the carriage while monitoring the position sensor output. Transient motion of the carriage, small but clearly visible against the quiet background, shows that even this tiny load step was more than sufficient to cause motion. Friction, if present at all, is too small to measure.

ONGOING WORK

The pneu-mag device shown above is the third-generation design. All were designed primarily around the requirements of modal (i.e., small displacement) testing of low frequency structures in a typical laboratory setting. Several more specialized versions are now in development at CSA Engineering.

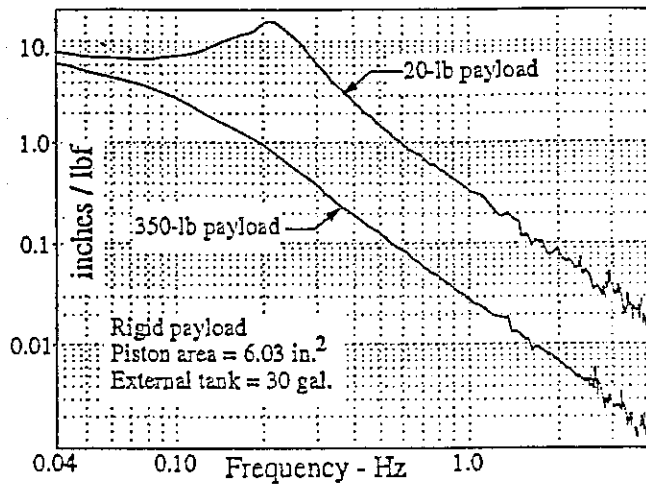


Figure 8: Measured displacement/force frequency response functions with large and small rigid payloads

In-Vacuum Testing of Solar Arrays

Structures with very large area/mass ratios can have modal properties that are affected by ambient air. Solar arrays or large, ultralight antennae for space are important examples. Testing in a vacuum is sometimes necessary to obtain modal properties characteristic of in-space operation. The need to operate in vacuum and to test structures with extremely low internal stiffness makes this application probably the most challenging to date. A pneu-mag variant is being developed to meet the unique requirements. The critical components necessary for operation in vacuum have been demonstrated.

Zero-G Simulation with a Large Space Truss

A three-device, support-from-below, pneu-mag system is nearing completion for use in ground experiments with a simulated large space optical system. The payload weighs approximately 7000 pounds and will be supported by three identical pneu-mag devices. Each device will allow unconstrained, frictionless motion of about 10 inches in all three directions. The system, the only one of its kind, will be used to simulate zero-gravity in rigid-body precision pointing and tracking experiments.

CONCLUSIONS

A suspension system has been described for producing near-unconstrained boundary conditions in modal testing of flexible structures. Called the pneu-mag system, it provides a unique combination of high payload, low stiffness, low added mass, and zero friction. It has reached the status of a commercial product and is in daily use on several current aerospace programs. Work is continuing to develop new versions for specialized applications.

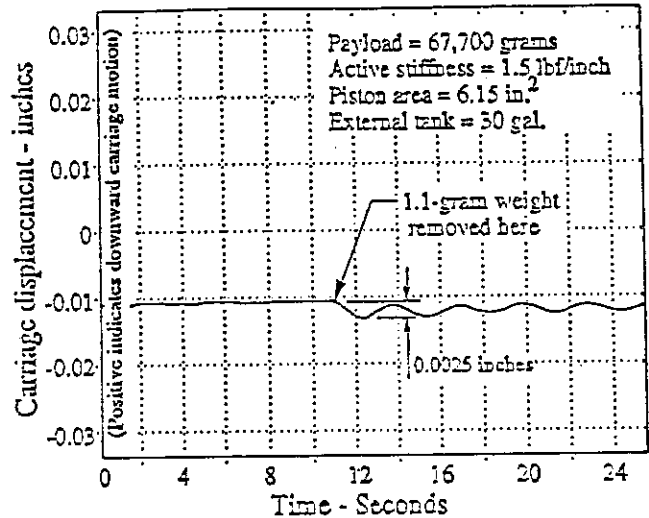


Figure 9: Breakaway friction test

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