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# VIBRAFUGE - COMBINED VIBRATION AND CENTRIFUGE TESTING

J. D. Rogers, F. Cericola, J. W. Doggett, and M. L. Young
Sandia National Laboratories
Albuquerque, NM 87185

The combined dynamic environments of vibration and linear acceleration are experienced by a large number of devices. Testing such devices has normally been a two step process in which independent vibration and centrifuge tests were performed. Concerns have existed for some time with possible combined effects from these two dynamic environments which could cause failures in the field which were not predicted from either analysis or the independent tests. This paper describes the design and performance of a testing facility which combines vibration and centrifuge testing in a single operation. The test facility is called the VIBRAFUGE and utilizes Sandia's 29 foot underground centrifuge with an attached electrodynamic shaker.

## INTRODUCTION

The VIBRAFUGE is a combined dynamic environment testing facility. The facility permits the simultaneous simulation of vibration and linear acceleration environments on small test articles, typically on the order of a few pounds. The purpose of the facility is to impose dynamic loadings on the test article in the laboratory which closely replicate those dynamic loads which the article will experience in use. Previously, the dynamic environments of linear acceleration and vibration were simulated in separate tests.

The VIBRAFUGE is not the first combined vibration-linear acceleration test facility to be developed. Sandia [1] developed a similar capability in 1970 and WYLE Labs [2] has recently developed a similar facility. The previous Sandia facility had two shakers and a small slip table on the end of the 29 foot underground centrifuge. The system performed reasonably well but suffered from two difficulties. First, the shakers were liquid cooled; and pumping liquids in the high g-field of the centrifuge was difficult. Second, the system was very large and difficult to place on the centrifuge requiring several days of set-up and tear down time. The new facility design was required to address these problems.

The fundamental concept of the facility is very straight-forward. One simply places a shaker on the centrifuge arm and runs the combined test. The simplicity of the concept is greatly complicated by the difficulties associated with running a controlled vibration test while the centrifuge is rotating. The problems fall into two categories: providing electrical energy and cooling to the shaker, and ensuring that the shaker and its mounting fixtures survive the dynamic environment.

The VIBRAFUGE design goal was to provide the combined dynamic test environment of 50 g's of linear acceleration and 20 $g_{rms}$ vibration in the bandwidth of 10 - 2000 Hz

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to a small test item. The typical test item would be no larger than a 12 inch cube and would weigh no more than 30 pounds. The design program required the selection of an appropriate shaker, designing the required fixturing to mount the shaker to the centrifuge, designing the electrical and cooling connections, and providing a means for vibration perpendicular to the acceleration field. The design program was constrained to provide a functioning facility within slightly less than one year after the program was initiated.

## SHAKER SELECTION

The selection of an appropriate vibration shaker was of paramount importance in the program. A market survey was conducted with the following shaker requirements:

1. Electrodynamic shaker design
2. Minimum of 4000 lbs peak force, sinusoidal
3. One inch stroke, double amplitude
4. Sixty inches/second velocity
5. Test frequency range: 1 - 5000 Hz
6. Air cooled
7. Armature centering capability: 3000 lbs minimum
8. Laboratory quality motion control in 50 g acceleration field
9. Must provide controlled vibration perpendicular to the g-field

No standard commercially available shaker was located which could meet all the above listed requirements. The time constraints on the program were such that a new shaker design project was not possible. In addition, problems with the shaker have historically occurred when it was designed to operate with the centrifuge g-field perpendicular to the shaker's moving element. Elaborate suspension systems using bearings to counteract the acceleration side loads cause large amounts of distortion and create undesirable system resonances. It is less complicated to design a shaker to compensate for the centrifuge loading if the acceleration field is parallel with the shaker moving element axis. A commercially available shaker was located which, with modest modification, would meet the requirements with the exception of motion perpendicular to the g-field. A Multiple Direction Vibration Fixture, discussed in a later section, was designed which transfers motion from the g-field axis to a perpendicular axis so the shaker moving element remains aligned with the g-field.

A series of modifications to the commercial shaker were needed to meet the design requirements. These modifications included: four airbags located under the armature for load centering, extra field coil supports, high strength epoxy potting for field and degauss coils, and DC driver coil supply for additional load centering. These modifications were performed primarily by the shaker/power amplifier manufacturer. Other minor modifications to strengthen the shaker body for the acceleration loading were also performed both by the shaker manufacturer and at Sandia.

Load centering of the shaker armature is accomplished by four airbags located between the bottom of the shaker head and the shaker body and by DC current in the driver coil. The airbags are inflated with nitrogen supplied by an on-board high pressure tank. The armature center position is maintained through centrifuge spin-up by an electro-optical armature position sensor and an electronically controlled pneumatic servo. The electronic servo circuit board and pneumatic valves are located on a panel mounted near the centrifuge center of rotation where the linear acceleration field is quite small. The pneumatic system has a force offset capability which varies with the pressure available from the nitrogen regulator. At a pressure of 155 psig, the pneumatic offset capability is 2200 pounds. This is the maximum safe working pressure for the airbags.

To achieve the required 3000 pounds of force offset, a DC current must be supplied to the driver coil. The system is designed to deliver up to 100 amperes of current to the driver coil, resulting in an additional force offset of 800 pounds. The DC driver coil current is controlled by the operator at the power amplifier panel. Operator panel lights indicate HIGH-CENTER-LOW for the armature location to allow the operator to select the proper driver coil current with a ten turn potentiometer. Thus, the combined pneumatic-electrical force offset system provides the required 3000 pounds of armature load centering. The moving element of the shaker has an effective mass of 38 pounds so at the design level of 50 g's a test item weighing 22 pounds can be load centered. This is slightly less than the design goal of a 30 pound test item; but, it was determined to be adequate for the anticipated test items.

## SHAKER UTILITIES

The shaker requires electrical energy and cooling air to operate. The electrical requirements include the field and driver coil signals along with power for the various safety interlocks. This amounts to a requirement of 330 amperes driver coil current and 150 amperes field coil current along with the low power protection circuits. The cooling required is room temperature air supplied through an 8 inch pipe at a pressure of approximately 20 inches of water column. Since the shaker is located on the rotating centrifuge arm, a rotating connection was required.

The shaker utilities are supplied from the stationary power amplifier and blower to the rotating centrifuge by the Rotary Joint Assembly. An overall view of the system is shown in Figure 1. The rotary joint assembly is shown in detail in Figure 2.

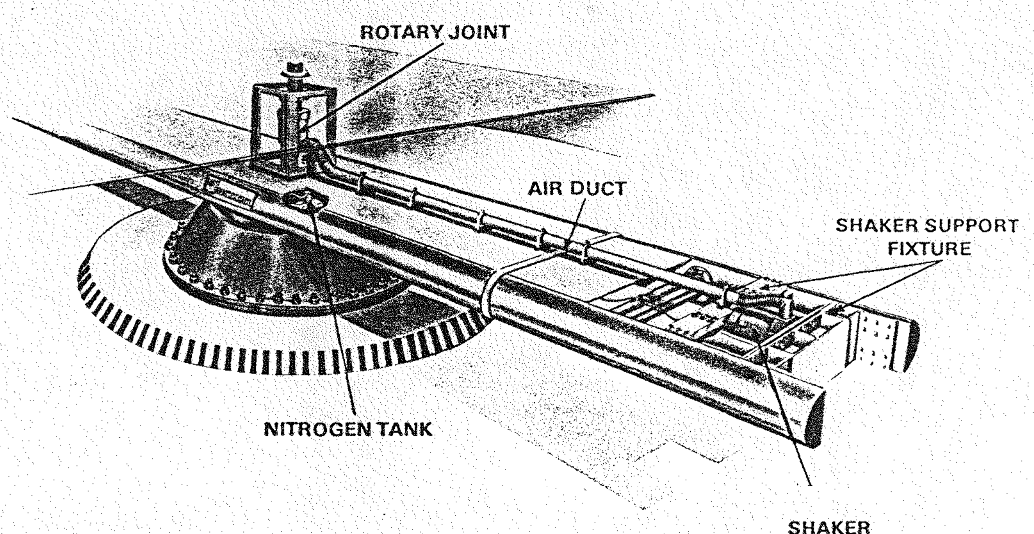


Figure 1. Overall view of the VIBRAFUGE.

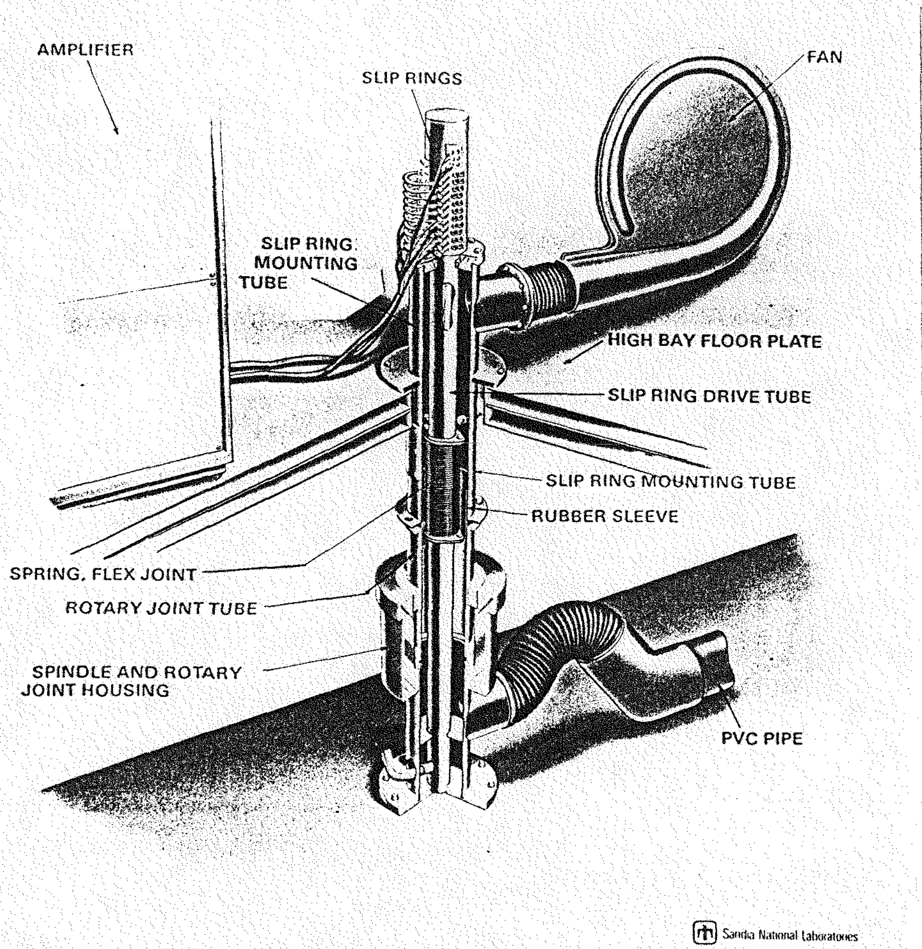


Figure 2. The rotary joint assembly.

The rotary joint assembly is designed to transmit cooling air from a stationary blower to the rotating centrifuge, to transmit rotary motion to the slip ring assembly, and to carry instrumentation cables from the slip rings to the centrifuge hub. The rotary joint assembly is essentially a slip ring stack and the associated conduit required to carry electricity and air from the fixed amplifier and blower to the rotating shaker.

The electrical energy is transmitted from the rotary joint to the shaker through a terminal board and cabling tied to the inside of the centrifuge arm. The air is transmitted from the rotary joint to the shaker through eight inch PVC pipe attached to the top of the centrifuge arm. The pipe is attached to the centrifuge arm with unistrut pipe clamps. Shoulders are cemented and bolted onto the pipe to absorb the centrifugal force of the pipe on the clamps. Failure to use these shoulders in an early test spin resulted in the PVC pipe sliding free from the clamps at about 60 rpm (30 g's at the test location).

In designing the rotary joint assembly, it was assumed that the spindle and rotary joint housing would rotate eccentrically with respect to the slip ring mounting tube (see Figure 2). The eccentric rotation results from run out in the centrifuge bearings and inaccuracies in the construction of the rotary joint assembly. The internal spring and slip ring drive tube were designed to transmit rotary motion to the slip rings. The spring section of the slip ring drive was designed to absorb any eccentricities in the rotation while transmitting torque. The outer tube of the rotary joint was designed as an air passage onto the centrifuge that did not transmit torque. The rotary joint tube was suspended from the stationary slip ring mounting tube into the brass bushings of the rotary joint housing. The rotary joint housing was designed to rotate around the rotary joint tube. A rubber sleeve was installed between the rotary joint tube and the slip ring mounting tube to contain the air flow as the rotary joint tube moved laterally as a result of the eccentric rotation. This design resulted in satisfactory transmission of torque to the slip rings and satisfactory air flow while adequately decoupling the rotational misalignment from the slip rings.

The slip ring design has one special feature which should be discussed. The driver coil supply is a high power alternating current which is susceptable to inductance losses. To minimize this loss, the shaker manufacturer bundles the driver coil cables with wires carrying opposite polarity currents adjacent to one another so that inductance losses are reduced by partial inductance cancellation. By the same principal, wires carrying opposite polarity currents should share a common port through any metal plate. Passing wires of opposite polarity through adjacent holes in a metal plate would cause the metal between the holes to act as a shorted turn of a transformer and would result in high energy losses in the form of heat. The aluminum rotor of the slip ring assembly, into which the wires pass from the rings, is slotted so that wires carrying opposing polarity currents are paired and brought through the common hole. The slotting of the rotor was a special modification done by the slip ring manufacturer.

## MULTIPLE DIRECTION VIBRATION FIXTURE

The multiple direction vibration fixture (MDVF) is an important element in the design concept of the VIBRAFUGE. By transmitting vibration to a direction perpendicular to the shaker, vibration perpendicular to the g-field is obtained without rotating the shaker. The MDVF provides this capability while maintaining laboratory quality vibration control.

The MDVF is essentially a gusseted 6061 aluminum angle with a tight journal bearing. The fixture was designed to provide a random vibration input to a small component (up to 5 pounds) of up to 30 $g_{rms}$ from 20 Hz to 3000 Hz in a 50g linear acceleration field. The design criteria for this fixture do not differ from those of any good vibration fixture design. The fixture uses a flex hinge on one side of the angle that attaches to the vibration exciter and tight journal bearings with nylatron bushings about which the fixture rotates and transmits the vibration perpendicular to the input at the other side of the angle (see Figure 3). This rotation introduces a transverse component of vibration at the test item mounting location on the fixture, but it is always less than the desired motion. The flex hinge was designed so that the first resonance through the web is over 10 kHz. It was also designed for an infinite fatigue life with a maximum of 0.5 inch double amplitude displacement at the payload end.

The journal bearings provide for an adjustable interference fit so that there is no separation that would introduce high frequency contamination to the control vibration input. The side gussets provide stiffness to the angle plates to keep the cantilever resonances and plate modes as high in frequency as possible. The length of the angle plates also dictate the frequencies of these resonances. A nylatron cylinder with viscoelastic damping pads is preloaded between the gussets to damp out the gusset plate modes. Bolted joints were also used instead of weldments to provide additional damping to the structure. All bolted joints and attachments were designed to operate in a 50g field with safety factors greater than four. The total weight of the MDVF is 62 pounds including the adapter plate.

The MDVF greatly simplified the shaker selection task but placed additional constraints on the shaker mounting fixture. The mounting fixture is required to hold the shaker in place in the acceleration field and must maintain a precise location for the MDVF with respect to the shaker. Two schools of thought exist for performing this task. Either the shaker and the MDVF must be mounted to massive structures so that the inertial masses of the separate structures maintain the appropriate relative positions or the shaker and the MDVF must be tied together with a very stiff structure maintaining the relative positions. The solution selected in the design program was a combination of these approaches. The mounting fixture, shown in Figure 4, is both quite massive and very stiff. The weight of the fixture is approximately 8000 pounds. The MDVF is attached by the pillow block journal bearing to the shaker mounting angle which is, in turn, attached to the shaker. The

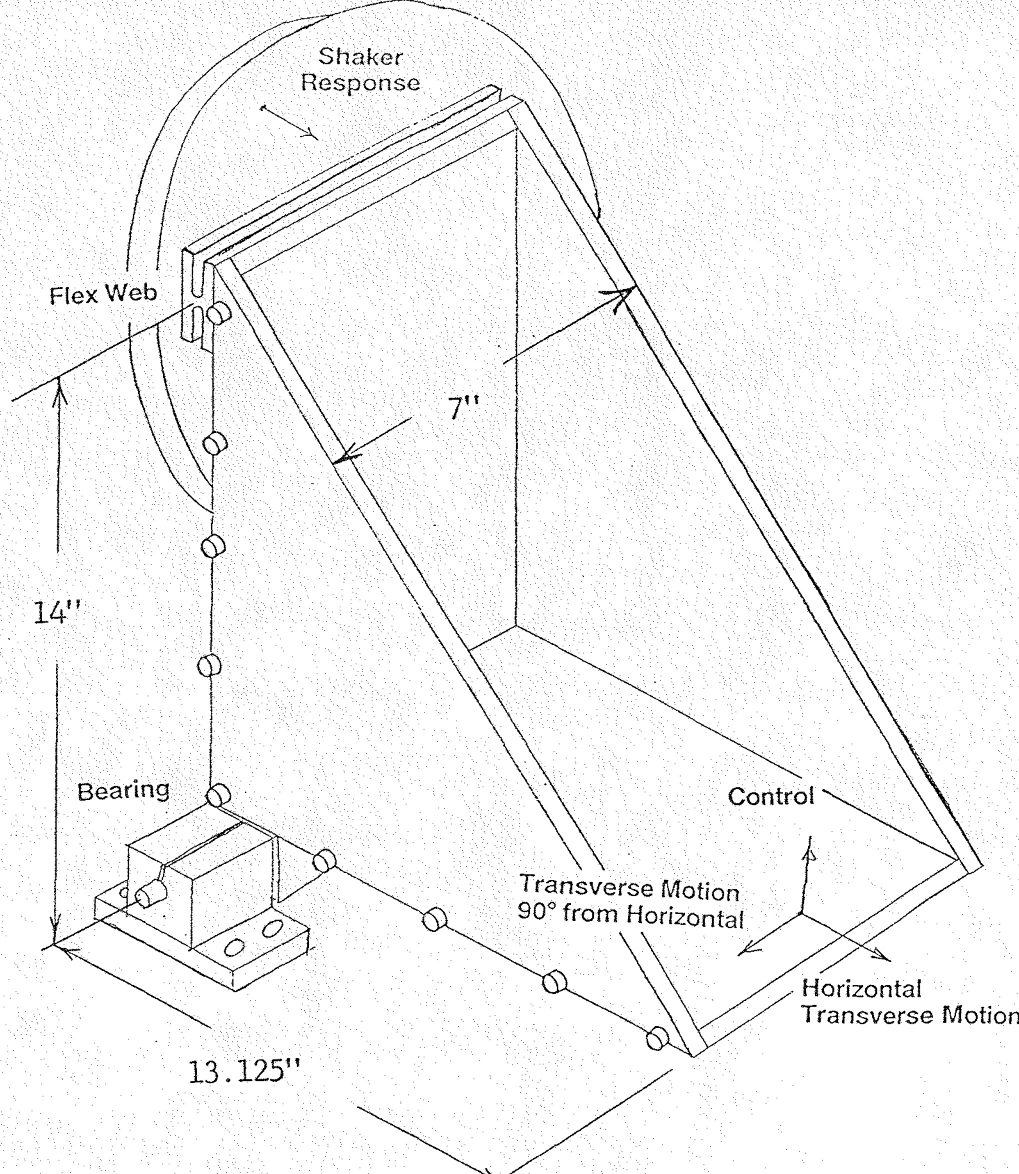


Figure 3. The Multiple Direction Vibration Fixture.

inertial loads from the g-field are reacted by a box beam which was static load tested to 700,000 pounds and is supported by large shear blocks.

In order to handle this very large fixturing, a large centrifuge was required. The Sandia National Laboratories 29 foot underground centrifuge [3] is sufficiently large to easily accommodate the shaker and the mounting fixture. The centrifuge is capable of accepting a 16,000 pound static load at a 29 foot radius and has a dynamic force rating of 1.6 million g-pounds. Thus, the centrifuge can accelerate the maximum payload to 100 g's of linear acceleration. The rotating member has a total length of 51 feet and a total rotating weight of nearly 38 tons. The centrifuge is hydraulically driven with 16 hydraulic motors supplied by five 350 Hp pumps. Clearly, the centrifuge capabilities are not a limitation for the VIBRAFUGE.

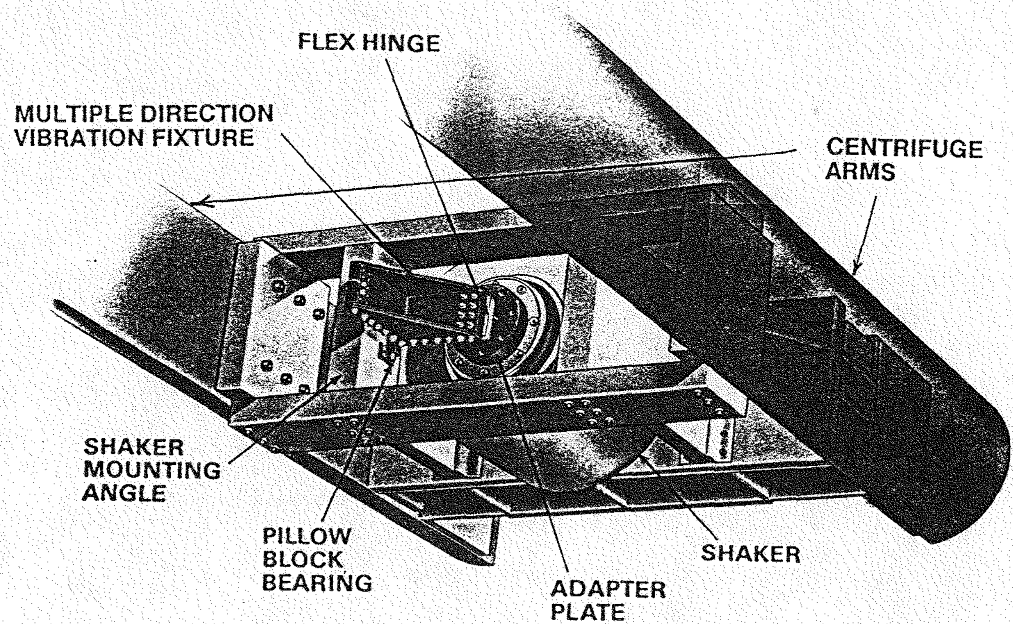


Figure 4. The MDVF and shaker mounting fixtures on the centrifuge.

## PERFORMANCE

The shaker performance was checked at the factory and after receipt at Sandia. In each case, the shaker performed quite well. The total harmonic distortion and cross-axis motion were well within specifications. The distrotion was generally less than 5 percent with only a narrow distortion peak of 12.4 percent at 2027 Hz. The cross-axis motion was always less than 10 percent of the control motion at all frequencies. The shaker was then attached to the centrifuge and spun-up to 50 g linear acceleration without anything attached to the armature. Sine sweeps were performed at the full rated force of 4000 pounds which corresponded to 105 g's peak. The total harmonic distortion was not changed from the static factory check-out tests. The load centering capability of the pneumatic and DC driver coil systems were also checked in this test. The pneumatic servo and the manual DC driver coil systems worked as expected.

After the successful "bare table" tests were completed, the MDVF was attached to the armature and check-out tests were performed. Due to the effective weight of the MDVF on the armature (31.5 pounds), the moving element could not be load centered in acceleration fields greater than 40 g's so the check-out was performed at 40 g's. The vibration environment was controlled on the MDVF at the control location shown in Figure 3. As noted in Figure 3, transverse motions and the shaker motion were also monitored. Random vibration tests and sine sweep tests were conducted. The results of a typical random vibration test are presented here.

The control specification for the random vibration test was a 20 $g_{rms}$ white noise spectrum in the bandwidth of 20 Hz - 3000 Hz. The control autospectral density is shown in Figure 5. The test was controlled very well within the 3dB tolerences.

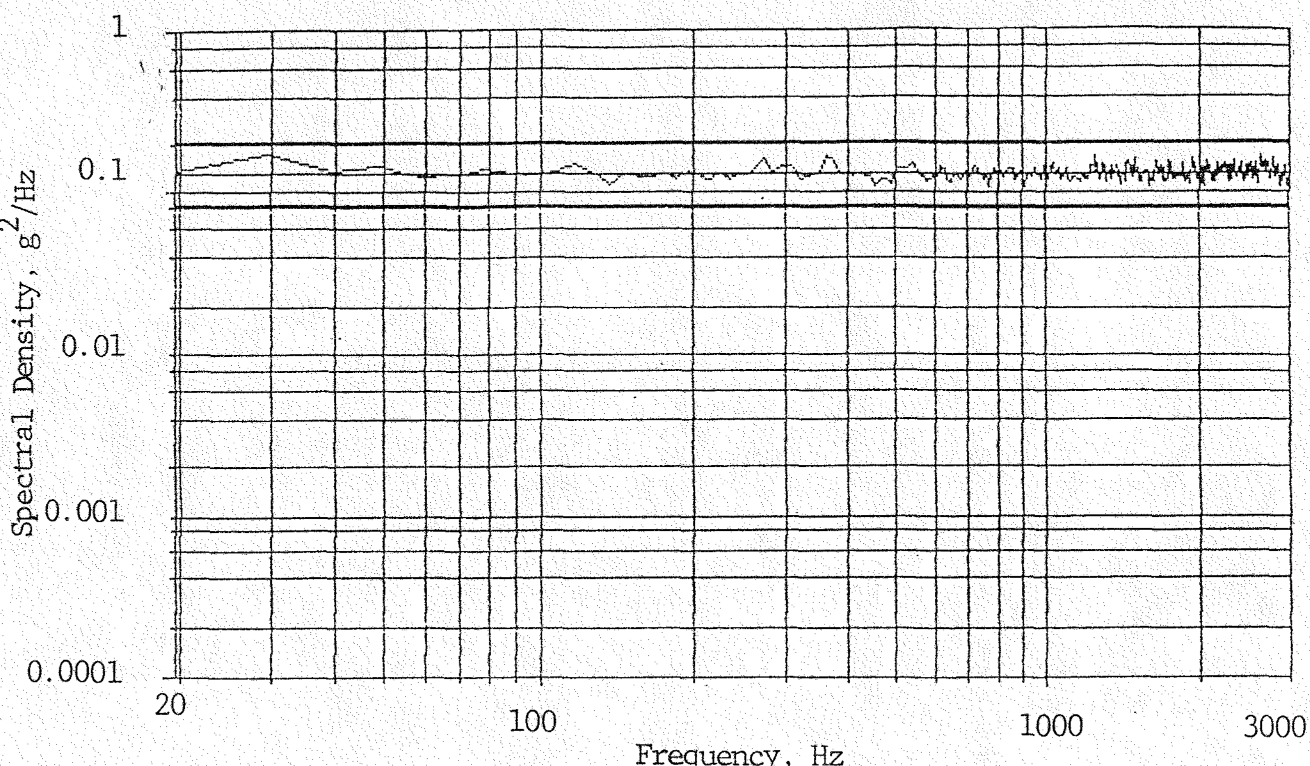


Figure 5. MDVF control autospectral density, 20 $g_{rms}$.

The shaker head motion autospectral density is shown in Figure 6. The notch and peak between 300 Hz and 400 Hz is a function of the tightness of the journal bearings which was optimized for the tests. The other notches in the shaker head autospectral density correspond to resonances in the MDVF and the shaker armature resonances. Figures 7 and 8 show the transverse motion autospectral densities on the MDVF at the control location. The transverse motion is less than the control motion at each frequency line of the analysis; therefore, the MDVF is deemed to be a good vibration fixture. Peaks are visible in the transverse motion spectral densities in the 300 Hz to 400 Hz range and are associated with the journal bearing tightness. The optimization referred to earlier with regard to journal bearing tightness meant the tightness which minimized the peaks in the transverse motion spectral densities.

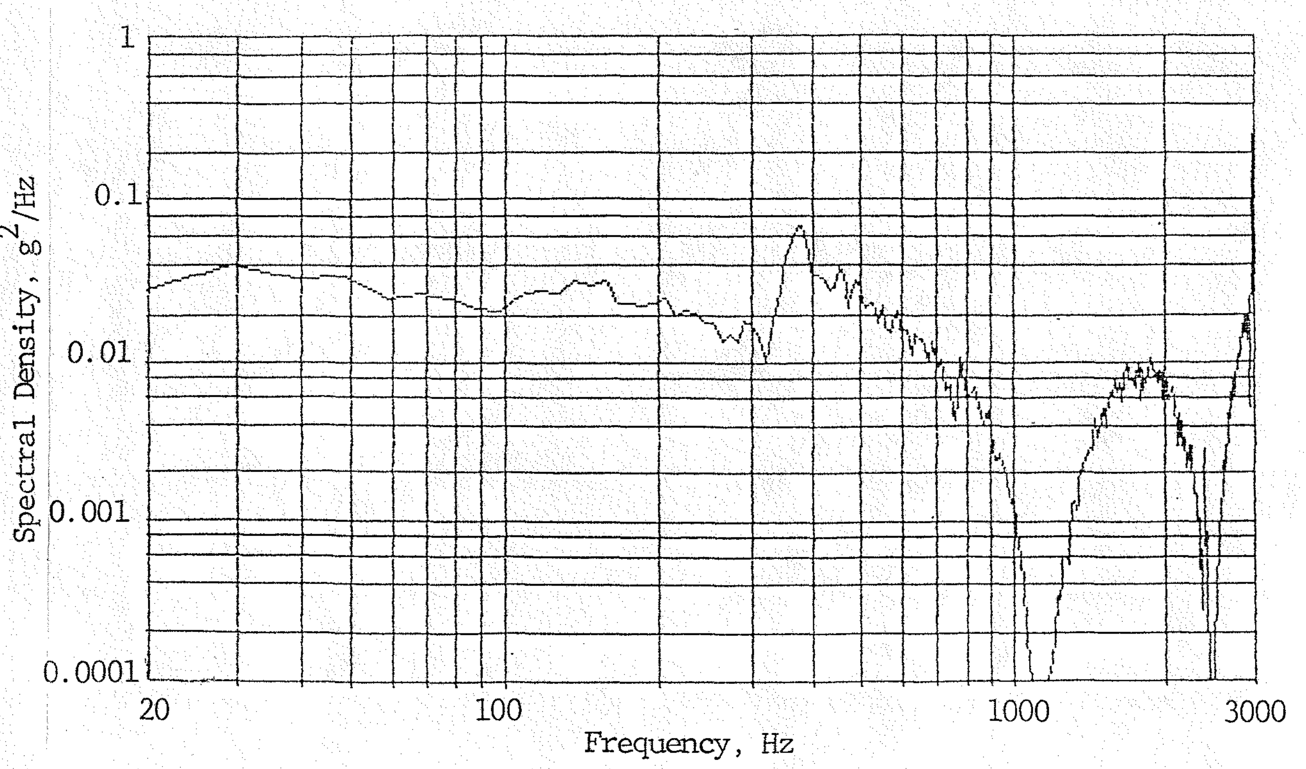


Figure 6. Shaker motion autospectral density, 18.8 $g_{rms}$.

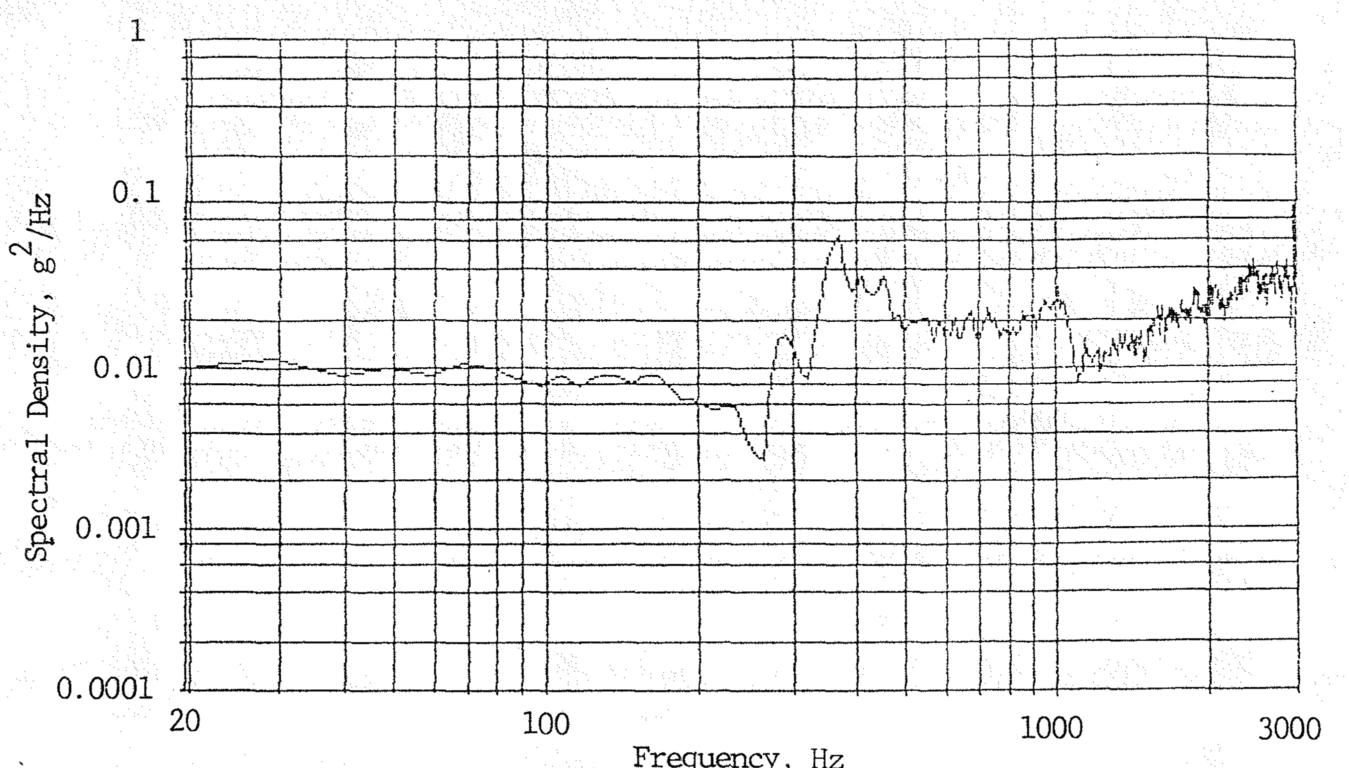


Figure 7. Horizontal transverse motion autospectral density, 8.2 $g_{rms}$.

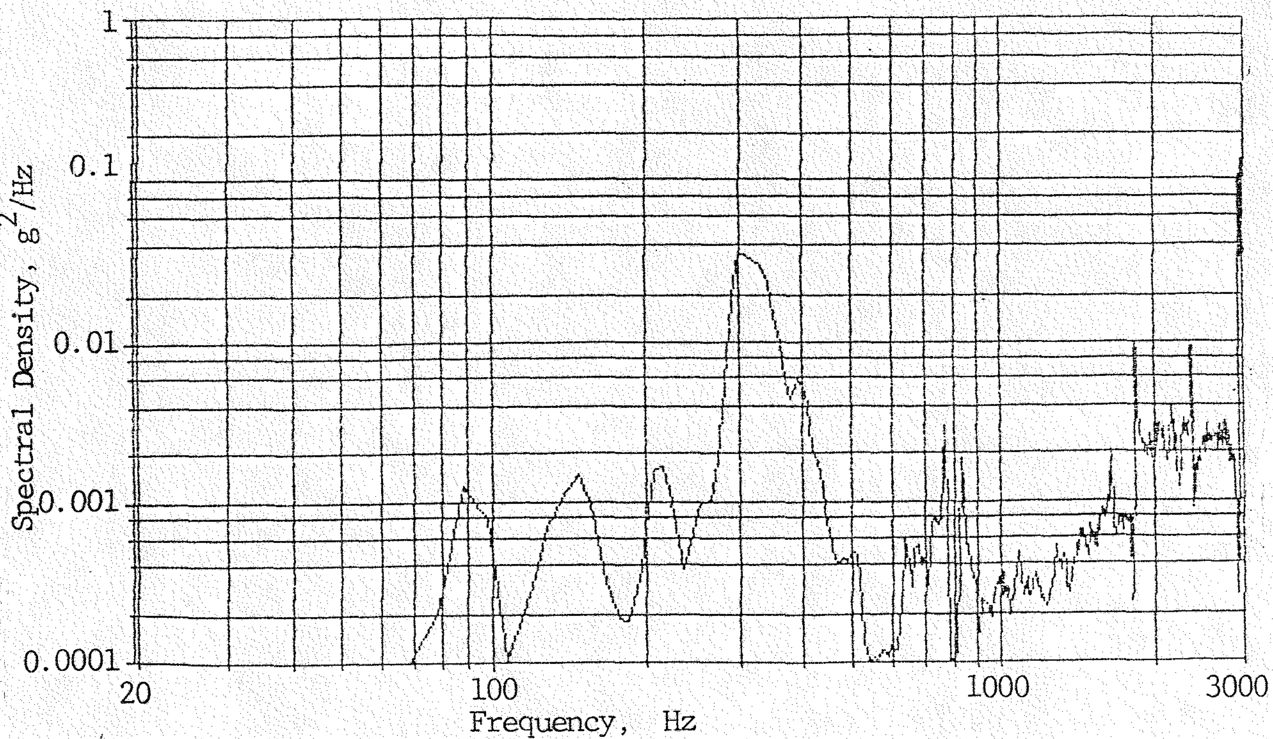


Figure 8. Transverse motion 90° from horizontal autospectral density, 3.1 $g_{rms}$.

## CONCLUSION

The VIBRAFUGE provides a combined dynamic environment testing capability which meets the current testing requirements. The program was completed within the time constraint while requiring only slight relaxation of some of the design goals. It provides a laboratory quality vibration environment in the presence of a linear acceleration field to a small component in either of two axes, parallel with or perpendicular to the acceleration field. A unique device, the MDVF, was developed which allows the shaker to remain oriented parallel with the acceleration field while providing vibration perpendicular to the acceleration field.

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