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NIST Torsion Oscillator Viscometer Response: Performance on the LeRC Active Vibration Isolation Platform

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PERFORMANCE ON THE LeRC ACTIVE VIBRATION ISOLATION PLATFORM

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SUMMARY

Critical point viscosity measurements are limited to their reduced temperature approach to T_c in an Earth bound system, because of density gradients imposed by gravity. Therefore, these classes of experiments have been proposed as good candidates for "microgravity" science experiments where this limitation is not present.

The nature of these viscosity measurements dictate hardware that is sensitive to low frequency excitations. Because of the vibratory acceleration sensitivity of a torsion oscillator viscometer, used to acquire such measurements, a vibration isolation sensitivity test was performed on candidate "microgravity" hardware to study the possibility of meeting the stringent oscillatory sensitivity requirements of a NIST torsion oscillator viscometer. A prototype six degree-of-freedom active magnetic isolation system, developed at the NASA Lewis Research Center, was used as the isolation system. The ambient acceleration levels of the platform were reduced to the noise floor levels of its control sensors, about 1 microgravity in the 0.1 to 10 Hz bandwidth.

INTRODUCTION

Undesired vibrations, often a problem for Earth-bound experiments, are of special concern for Space Shuttle microgravity experiments. Although the Shuttle's dc accelerations are indeed microgravity, the ac levels, for frequencies above 0.01 Hz, can be 100 times greater than a "quiet" 1-g laboratory environment, see table I (refs. 1 and 2). In an effort to correct this problem, NASA Lewis Research Center (LeRC) has recently developed a prototype active vibration isolation platform. This prototype system, with full digital electronic control of all six rigid-body degrees of freedom (DOF), has the potential to reduce Shuttle experimental vibration levels to near noise floor levels of its input accelerometers.

The torsion oscillator viscometer developed at NIST, to perform critical point viscosity measurements is limited to it's reduced temperature approach to T_c in an Earth bound system because of the density gradient imposed by the gravity field. Therefore, it has been proposed to operate the laboratory viscometer as a "microgravity" flight experiment to measure the viscosity of xenon two orders of magnitude closer, in reduced temperature, to T_c than can be achieved on Earth. However, since this viscosity measurement device is also inherently limited by vibratory disturbances, to successfully operate as a Shuttle experiment, will require vibration isolation beyond that offered by present commercial systems. During a series of successful 1-g viscosity measurements, the NIST group used a custom passive isolation platform to reduce vibration levels below those of the laboratory. However, this type of passive isolation platform used in the laboratory would not be compatible with an orbital environment.

In order to investigate the possibility of meeting the torsion viscometer's acceleration sensitivity needs in the orbital acceleration environment with integratible isolation hardware an initial, and admittedly severe, test of the newly constructed active isolation platform against a "real-world" challenge was studied. The combination of the NASA Lewis isolator loaded by the NIST torsion oscillator and its thermostat were tested. This report describes the results of the testing conducted during 8-12 April 1991.

DESCRIPTION OF THE ACTIVE SIX DOF ISOLATION PLATFORM

The NASA Lewis prototype active inertial vibration isolation platform was developed under an Advanced Technology Development (ATD) Program. This Microgravity Science and Application Division (MSAD) program was initiated in 1987 to develop enabling technologies which will be needed for microgravity experimentation. The prototype hardware was designed and fabricated as a proof-of-concept design for a development phase of the ATD Vibration Isolation Technology project. This prototype hardware was the design verification for a Vibration Isolation Technology demonstration testbed to be flown in the NASA Lewis low gravity aircraft.

The subject hardware was developed to demonstrate the inertial isolation of a platform from external disturbances while achieving microgravity acceleration levels. This hardware consists of a platform which is suspended in the 1-g environment by three attractive electromagnets and actively controlled in all dimensions. The laboratory system consists of 12 control sensors (6 relative and 6 inertial), a control computer, and 9 programmable linear power amplifiers which drive the 9 magnetic actuators for the control of the suspended mass.

For the tests conducted with the NIST torsion oscillator, the horizontal DOF were uncoupled from the vertical DOF. Figure 1 shows the physical layout of the system where the hexagonal platform is shown internal to the actuator pod locations. The hexagonal platform was fabricated out of 0.635 cm (0.25 in.) thick carbon steel angle, where the platform material was used as the ferromagnetic attractive core. The actuator pods housed three attractive electromagnets in the configuration shown. In the prototype system only nine electromagnets were utilized, and gravity was used as the restoring force in the vertical dimension. The control of the platform was uncompromised while the control algorithm did not request an acceleration in the vertical dimension greater than 1-g. Relative sensors are used to resolve the six rigid-body DOF of the platform while inertial sensors are used to resolve the six DOF of the support structure. With these data a feedforward/feedback algorithm is used to control the platform and reject disturbances from its dynamic environment.

The theorectical approach to this systems isolation design is published in NASA Technical Paper 2984 (ref. 3). This publication explains some of the details behind this and other active control approaches. The theoretical transmissibility function for inertial feedforward control, in the one-dimensional case, can be approximated by the following equation:

$$T = \left| \frac{X}{U} (j\omega) \right| = \left\{ \frac{\left(1 - B_p\right)^2 + \left(2\xi \frac{\omega}{\omega_n} \left(1 - B_v\right)\right)^2}{\left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\xi \frac{\omega}{\omega_n}\right)^2} \right\}^{1/2} \tag{1}$$

space. As shown by the transfer function (eq. (1)), if one chooses the appropriate scaling parameters of the system, one can theoretically negate any excitation transmission from the support structure. However, in practice, because of hardware specific and noise floor limitations one is not able to negate all excitations, but attenuation roll-off can be increased substantially.

DESCRIPTION OF THE VISCOMETER

The NIST torsion oscillator viscometer was developed to measure the viscosity of fluids near their critical point. The physics of such fluids limits the oscillator's frequency to about 1 Hz and its shear rate to about 1 s^{-1}, resulting in a visually unobservable oscillation amplitude of 1 milliradian (0.06°). Typical laboratory vibrations, while not disturbing the nearly critical sample itself, greatly perturb the oscillator's small movements and thereby obscuring the coherent oscillations whereby the viscosity is measured. These perturbations were quantified and modeled in a report to NASA Lewis in 1989 (ref. 5) which showed that, for an adequate signal-to-noise ratio, vibration levels near the oscillator's resonant frequency of about 1 Hz must be kept below 4×10^{-8} g/$Hz^{1/2}$.

Previously published scientific articles (refs. 6 to 8) and the earlier report to NASA Lewis mentioned above (ref. 5), describe the torsion oscillator viscometer and its operation in detail. As shown in figure 2, the oscillator consists of a disk about 1 cm thick and 4 cm in diameter attached to a vane of comparable mass below and to a thin quartz torsion fiber above. It is suspended inside a three-shelled thermostat, the outermost vacuum-tight shell being the heaviest. Capacitative measurements of the vane's position monitor the torsional motion of the oscillator, which decays exponentially following an electrostatic "kick." The viscosity of the sample contained in the disk is inferred from this exponential decrement.

For the tests described in this report, the oscillator was a solid body containing no fluid. Its dissipation, determined by the viscosity of air and the geometry of the gap between the oscillator vane and the stationary capacitor electrode, was comparable to that of the NIST group's xenon measurements (ref. 5). The natural resonant frequency was 2.52 Hz, and the decrement was 2.6×10^{-3}, corresponding to a Q of 1200.

THE VIBRATION TESTS

The platfrom accelerations and the resulting motion of the torsion oscillator were measured with the platform levitated and unlevitated. Horizontal accelerations, which were measured with Sundstrand QA-2000 accelerometers, are emphasized in this report due to their importance to the viscometer's signal-to-noise ratio.

Figures 3 and 4 show the noise levels from 0.01 to 10 Hz and from 0.1 to 100 Hz when the platform was unlevitated. The lowest accelerations recorded, which happened to fall in the range of typical torsion oscillator resonant frequencies from 0.5 to 3 Hz, were 1 to 10×10^{-7} g/$Hz^{1/2}$. Turning off all electronic

QA-2000 accelerometers, are emphasized in this report due to their importance to the viscometer's signal-to-noise ratio.

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The torsion oscillator's 2.52 Hz resonant can be seen in figure 3(c). Its amplitude is in approximate agreement with that expected from the relation (ref. 6)

$$\Delta V = \frac{\partial V}{\partial \theta} = \frac{a_\theta}{\omega^2}\left[\frac{\pi\omega}{2D}\right]^{1/2}, \tag{2}$$

where $(\partial V/\partial\theta)$ is the sensitivity to angular displacement, a_θ is the spectral density of angular accelerations, ω is the resonance frequency, and D is the oscillator decrement. (The radius of platform torsion was assumed to be 0.2 m.)

When the platform was fully levitated and controlled in all six degrees-of-freedom, platform vibrations at the torsion oscillator's resonance of 2.52 Hz were increased by as much as a factor of 7, and, as shown in figure 5, this resulted in a comparable increase in the vibration-excited amplitude of the torsion oscillator. At least some portion of the increase in this vibration was due to the platform's resonant frequency of 0.8 Hz, determined by the particular choice of control loop parameters, listed in tables II and III.

A large portion of the increase in acceleration response of the levitated platform was due to the horizontal balancing of the platform. A few milliradians of tilt causes significant coupling between vertical and horizontal motions. The finite dynamic range of the platform's motion sensors exacerbates this problem, leading to low amplitude motions at the natural frequency of the system. The magnitude of this disturbance can be estimated from the digitization resolution of the analog circuitry and the analog-to-digital (A/D) converters. The 12 bit A/D converter was set at ±2.5 V, full scale range, giving a bit error of ±1.22 mV. Thus, the control code cannot resolve any relative displacement smaller than 0.39 μm. Any drift of the platform's tilt will eventually cause a horizontal displacement exceeding this threshold. When this occurs, the control code causes a sudden acceleration of about 1.0 μg or 1.1 μg/Hz$^{1/2}$ at the natural frequency. This calculation assumes one-dimensional motion and no coupling of the horizontal DOF which simplifies the calculation and predicts a best case result. However, the result agrees within a factor of 10 to the measured results of figure 5. In making predictions as to the cause of this discrepency, but assuming minimum coupling between axes, one can argue that the controller cannot respond adequately until a few bit changes are detected. This would result in the previous calculation showing better agreement with the experimental result. It is expected that once a 16 bit converter is utilized for the control sensors the noise floor or bit error will be improved by a factor of 2^4.

Between 6 and 10 Hz the levitated platform reduced vibrations below ambient by a factor varying from 1 to 30. From 10 to 40 Hz (fig. 6) the platform performance was uneven, with both increases and decreases in vibration levels. Above 20 Hz levitation increased platform vibrations by an order of magnitude. We believe this latter increase in magnitude above 20 Hz is due to phase lags caused by bending modes in the platform itself (i.e., nonrigid-body motion).

Without manual assistance, the isolation platform successfully raised itself and the 5.7 kg load of the torsion oscillator and thermostat from rest to levitation in a smooth manner. However, directly clamping the thermostat to the isolator platform caused an instability of the feedback loop at about 50 Hz. Holding and gently hitting the thermostat revealed the existence of an internal mechanical mode of the thermostat of comparable frequency; this was apparently the origin of the instability. To eliminate this instability, 5 mm thick damping pads were clamped between the thermostat and platform for all subsequent tests.

The transmissibility of the isolation platform was tested, when loaded by the torsion oscillator and thermostat, by vibrating the table on which it was mounted, recording the accelerations both on the table and on the levitated platform, and taking the ratio of these accelerations. Although in general there are 36 transmissibilities from the six rigid-body modes of the table to those of the isolation platform, only the horizontal translational accelerations were measured and only the X/X and Y/Y transmissibilities were recorded. Further improvements, such as decoupling the horizontal and vertical table motions, directly measuring the rotational accelerations, and providing higher amplitude vibrations, will allow meaningful interpretation of the other transmissibilities in the future.

Figures 7 and 8 show the X/X transmissibilities without inertial control (the Y/Y plots were similar but with a lower signal-to-noise ratio). The plots show the character of a low-Q oscillator, having a transmissibility of 1 at low frequencies, a resonance at about 0.8 Hz, and a $1/f^2$ rolloff at higher frequencies. Comparing figures 9 and 7 show that, for the chosen set of control parameters, inertial control had little effect. This was because the control parameters chosen at the time of this data measurement corresponded to a 180° out-of-phase condition of the inertial signals.

In order to demonstrate the advantages of inertial referencing, figure 10 compares inertial referencing control of the platform versus noninertial control, taken with the appropriate control parameters, listed in table IV. For inertial referencing, the roll-off is approximately 100 dB of attenuation per decade, an increase of about 2.5 times that of the noninertially referenced payload. The lower bound of approximately -40 dB in figure 10 was a result of test hardware limitations and the digital resolution of the control system, during the time these measurements were taken. In order to make predictions on the performance of prototype hardware, equation (1) was used to generate the theoretical predictions of figure 11. This plot was generated with some modeling corrections made for the specific system parameters listed in table IV and for nonidealities such as sensing errors, resolution, and testing hardware limitations. Figure 11 shows the increase in roll-off by inertially referencing the payload, as demonstrated in the experimental data shown in figure 10. The jagged shapes of the three curves below the natural frequency, in figure 11, are caused by the spacing of the frequencies where the function was calculated and are not physically meaningful.

CONCLUSIONS

To the best of our knowledge, this is the first fully levitated platform with active digital control of all six rigid-body degrees of freedom. Below 10 Hz the platform was fully stable and behaved as a nearly critically damped harmonic oscillator with a resonance at 0.8 Hz. The platform was stable above 10 Hz also but isolation was degraded, due to bending modes of the platform.

Mechanical resonances in the load can destabilize the isolation platform. Therefore, predicting the loaded performance of this, or any, active isolation system requires a model of the dynamic characteristics of the load to verify that only rigid-body modes of the system exist in the bandwidth of operation.

The ultimate performance of any active isolation system is limited by the accelerometer's noise floor and the digital resolution of the controller. For the system, during these measurement tests, figure 3 indicates that, near 1 Hz, the Sundstrand QA-2000 accelerometer's noise floor is near the 10^{-7} g/Hz$^{1/2}$ level needed for the torsion oscillator viscometer. However, sufficient isolation against Shuttle vibrations requires a dynamic range of four decades in the acceleration measurements and thus 14 bit digitization of the accelerometer outputs. However, the protype system used only 12 bits; this resolution limit is an explanation for the noise floor of about 1×10^{-6} g/Hz$^{1/2}$ near 1 Hz seen for the levitated platform.

Assuming a Shuttle spectral noise density of 10^{-3} g/Hz$^{1/2}$ near 1 Hz, the torsion oscillator viscometer requires an isolator with a transmissibility of about 10^{-4}. The NASA Lewis active isolation platform prototype hardware did not provide such attenuation (nor does any other isolator that we know of). The isolation performance of the prototype system has been improved since the time of the measurements by reducing the horizontal degrees-of-freedom converters full scale range and thus, increasing the digital resolution of the accelerometers by a factor of two. In addition, a Learjet demonstration system has also been built with 16 bit resolution, increasing the attenuation capabilities of the original system by 2^4.

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TABLE I. - TYPICAL ACCELERATION DISTURBANCES

Source	Acceleration, g/g_o	Frequency, Hz
Quasi-steady or constant cycle		
Aerodynamic drag	10^{-7}	0 to 10^{-3}
Light pressure	10^{-8}	0 to 10^{-3}
Gravity gradient	10^{-7}	0 to 10^{-3}
Periodic		
Thruster fire (orbital)	2×10^{-2}	9
Crew motion	2×10^{-3}	5 to 20
Ku-band antenna	2×10^{-4}	17
Nonperiodic		
Thruster fire (attitude)	10^{-4}	1
Crew pushoff	10^{-4}	1

TABLE II. - CONTROL PARAMETERS FOR THE HORIZONTAL DEGREES OF FREEDOM

Control parameters		Agitation control	
cgain8	1.00000		
cgain9	0.95000	Vertical freq (Hz)	0.00000
xybias	70.00000	Vertical amp (mils)	0.00000
onoff	1.00000		
Forgetting factor	0.99500	X rotation freq (Hz)	0.00000
Accelerometer gain	0.70000	X rotation amp (mils)	0.00000
theta stiff, kpth	0.90000		
theta damp, dthh	20.84005	Y rotation freq (Hz)	0.00000
Extra	0.95000	Y rotation amp (mils)	0.00000
Horz Stiffness, Kph	0.65000		
Horz damping, Dh	30.63000	Pulse in x dir	0.00000
Intergal gain horz, Igh	0.00000	Pulse in y dir	0.00000
Horz iner damp const, idh	0.00000	Pulse in theta dir	0.00000
Horz iner stif const, ikh	13.00000		
Increment =	1.00000	Looptime (ms) =	1.06400

TABLE III. - CONTROL PARAMETERS FOR THE VERTICAL DEGRESS OF FREEDOM

Control parameters		Agitation control	
Stiffness, Kp	1.05000	Vertical freq (Hz)	0.00000
Damping, Dv	18.00000	Vertical amp (mils)	0.00000
Intergral gain, Ig	0.00005		
		X rotation freq (Hz)	0.00000
Offset	0.00000	X rotation amp (mils)	0.00000
Forgetting factor	0.99500		
Accelerometer gain	0.70000	Y rotation freq (Hz)	0.00000
Position gain	4.50000	Y rotation amp (mils)	0.00000
Inertial damp const, id	0.20000		
Inertial stiff const, ik	0.00000		
Increment =	1.00000	Looptime (ms) =	1.76900

TABLE IV. - CONTROL PARAMETERS FOR THE INERTIAL RESPONSE MEASUREMENTS TAKEN IN FIGURE 10

Control parameters		Agitation control	
cgain8	1.00000	Vertical freq (Hz)	0.00000
cgain9	0.95000	Vertical amp (mils)	0.00000
xybias	70.00000		
onoff	-1.00000	X rotation freq (Hz)	0.00000
Forgetting factor	0.99500	X rotation amp (mils)	0.00000
Accelerometer gain	0.70000		
theta stiff, kpth	0.50700		
theta damp, dthh	41.84005	Y rotation freq (Hz)	0.00000
Extra	0.95000	Y rotation amp (mils)	0.00000
Horz stiffness, Kph	0.32300		
Horz damping, Dh	41.63000		
Intergal gain horz, Igh	0.00000	Pulse in x dir	0.00000
Horz iner damp const, idh	0.00000	Pulse in y dir	0.00000
Horz iner stif const, ikh	20.00000	Pulse in theta dir	0.00000
Increment =	10.00000	Looptime (ms) =	1.08700

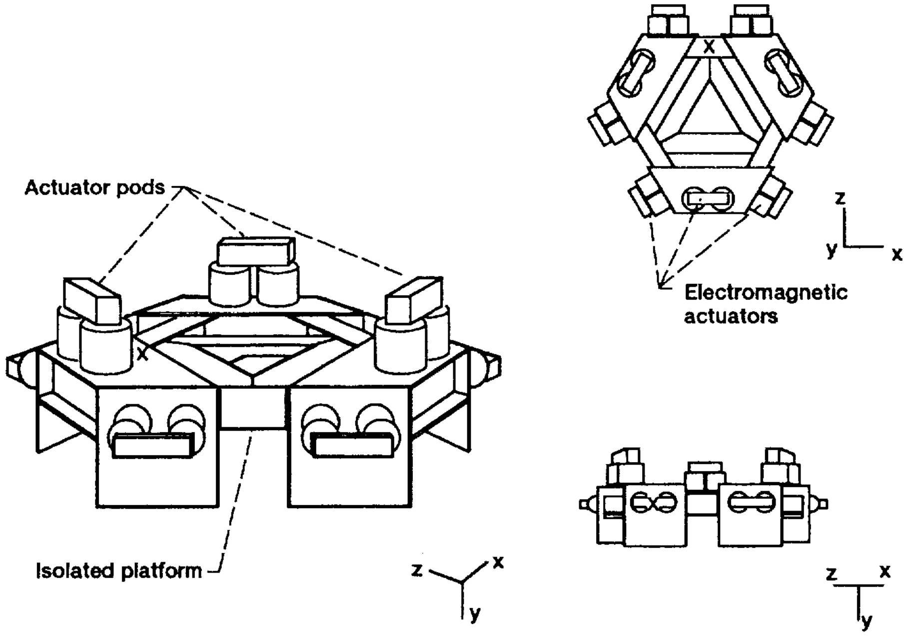


Figure 1.—Active vibration isolation system.

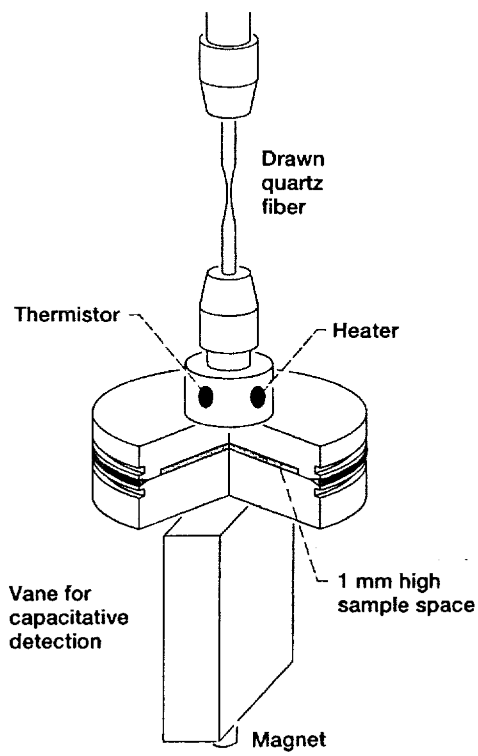


Figure 2.—Torsion oscillator used for this test was similar to that used for the CO_2 and xenon measurements but contained no sample.

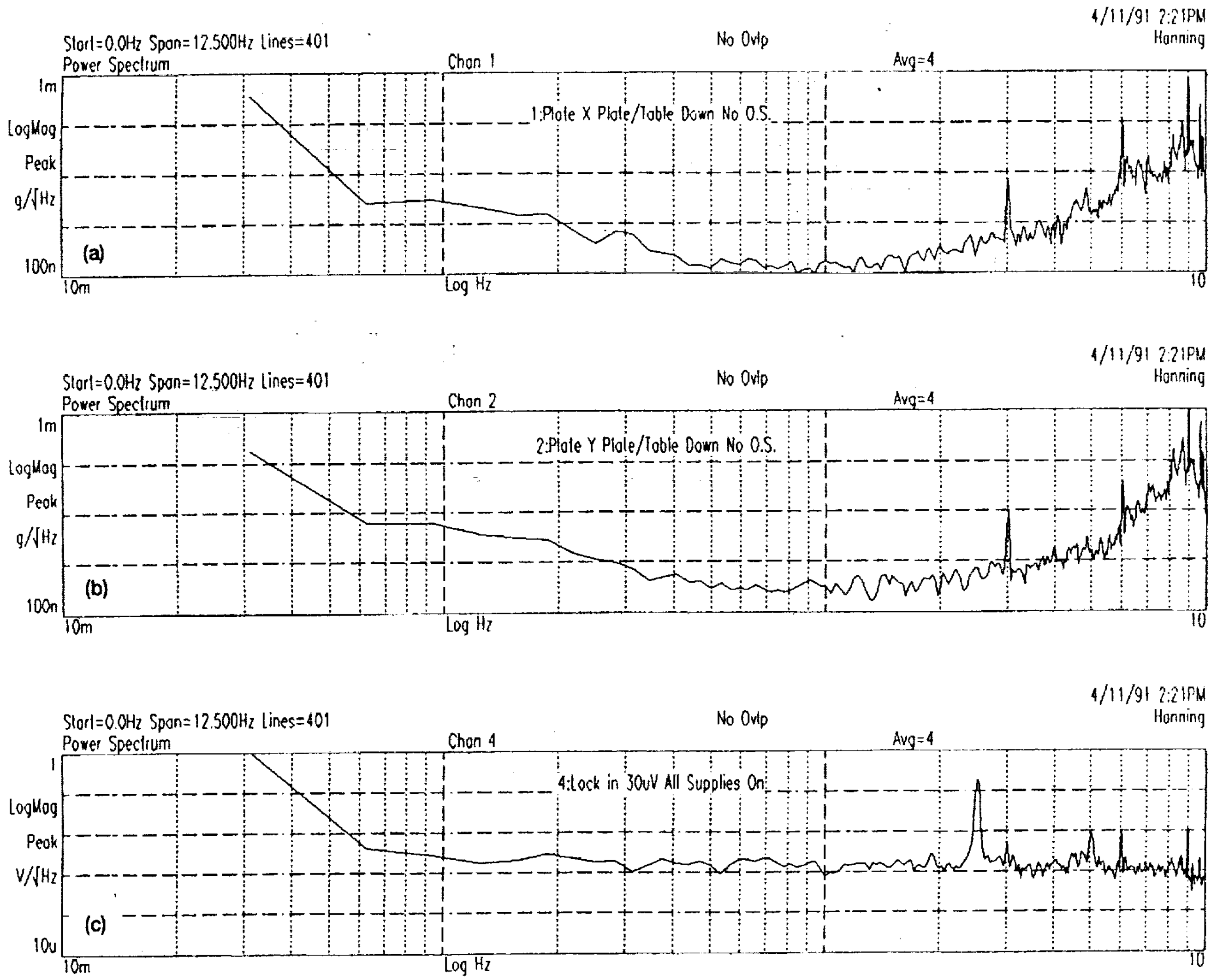


Figure 3.—(a) Accelerations present on unlevitated isolation platform in the "X" horizontal direction. Note that noise floor is near 1×10^{-7} $g/\sqrt{Hz}$ near 1 Hz.
(b) Accelerations in the orthogonal horizontal "Y" direction.
(c) The spectrum of the output of lock-in amplifier used to measure position of torsion oscillator. Background of 2×10^{-3} $V/\sqrt{Hz}$ is due to preamp noise of 5 $nV/\sqrt{Hz}$. Peak at 2.52 Hz is viscometer's torsion resonance excited by vibration levels.

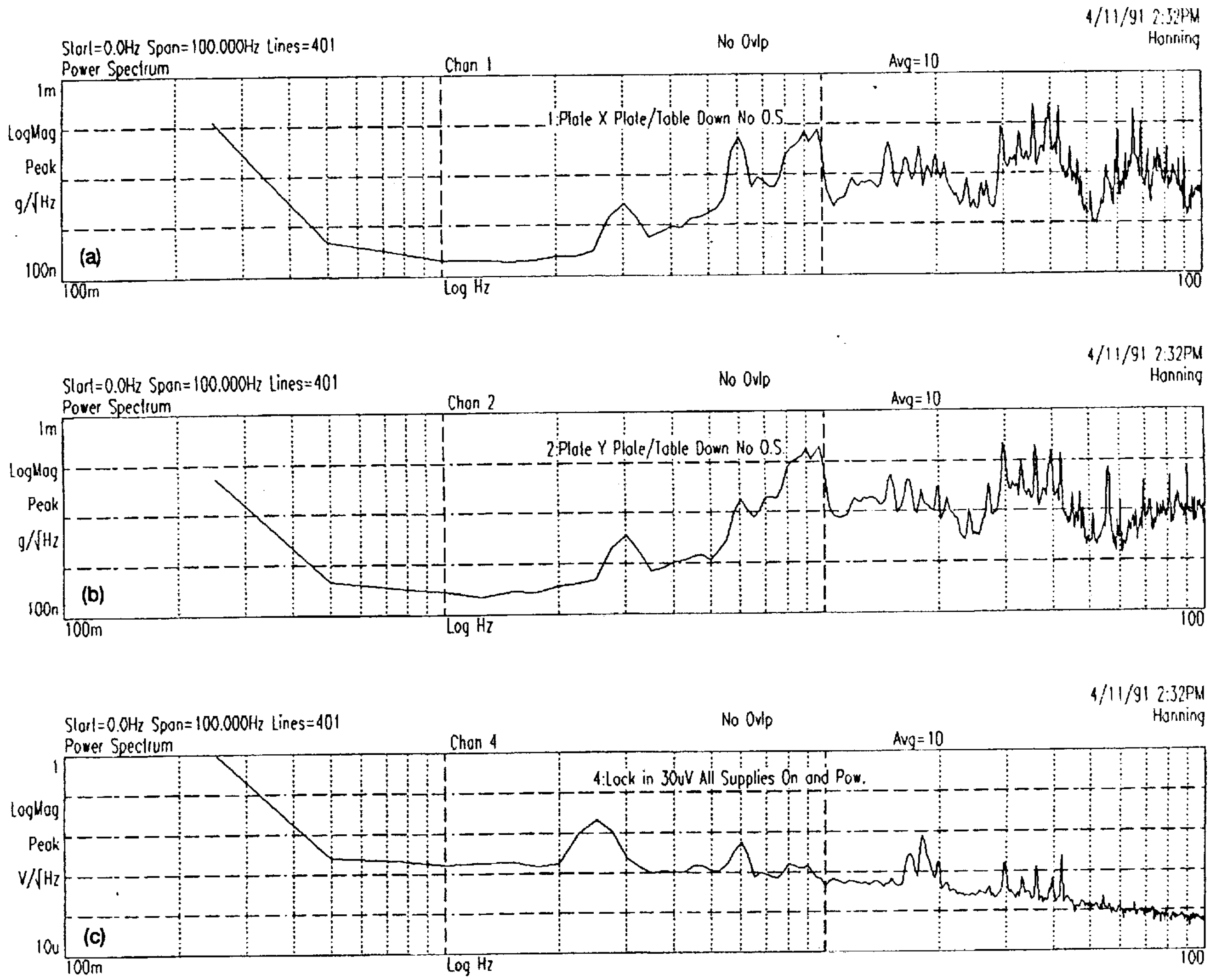


Figure 4.—Plots of Fig. 3 extended to 100 Hz. Lock-in amplifier's output rolls off due to its 1.25 ms integration time constant.

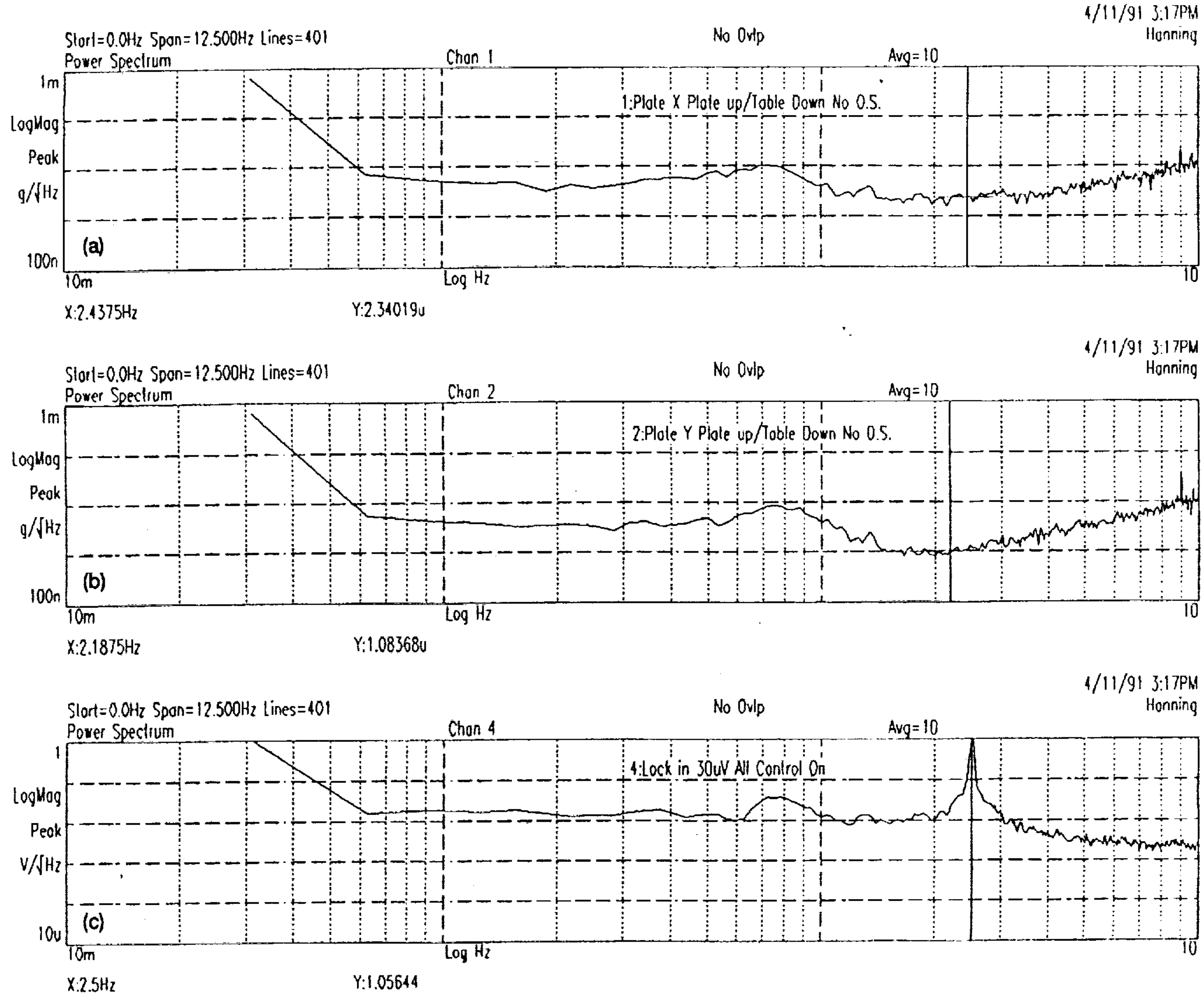


Figure 5.—Same accelerometer (traces a and b) and lock-in amplifier (trace c) plots as in Fig. 3 but with isolation platform levitated. Note the following differences from Fig. 3:
(1) Lowest acceleration level is raised to 1x10–6 g/√Hz
(2) New low-Q peak near 0.8 Hz
(3) Suppression of accelerations above 6 Hz
(4) Increase in low frequency noise from lock-in amplifier output
(5) Increase in vibration-excited viscometer at 2.52 Hz corresponds approximately to increase in vibration levels

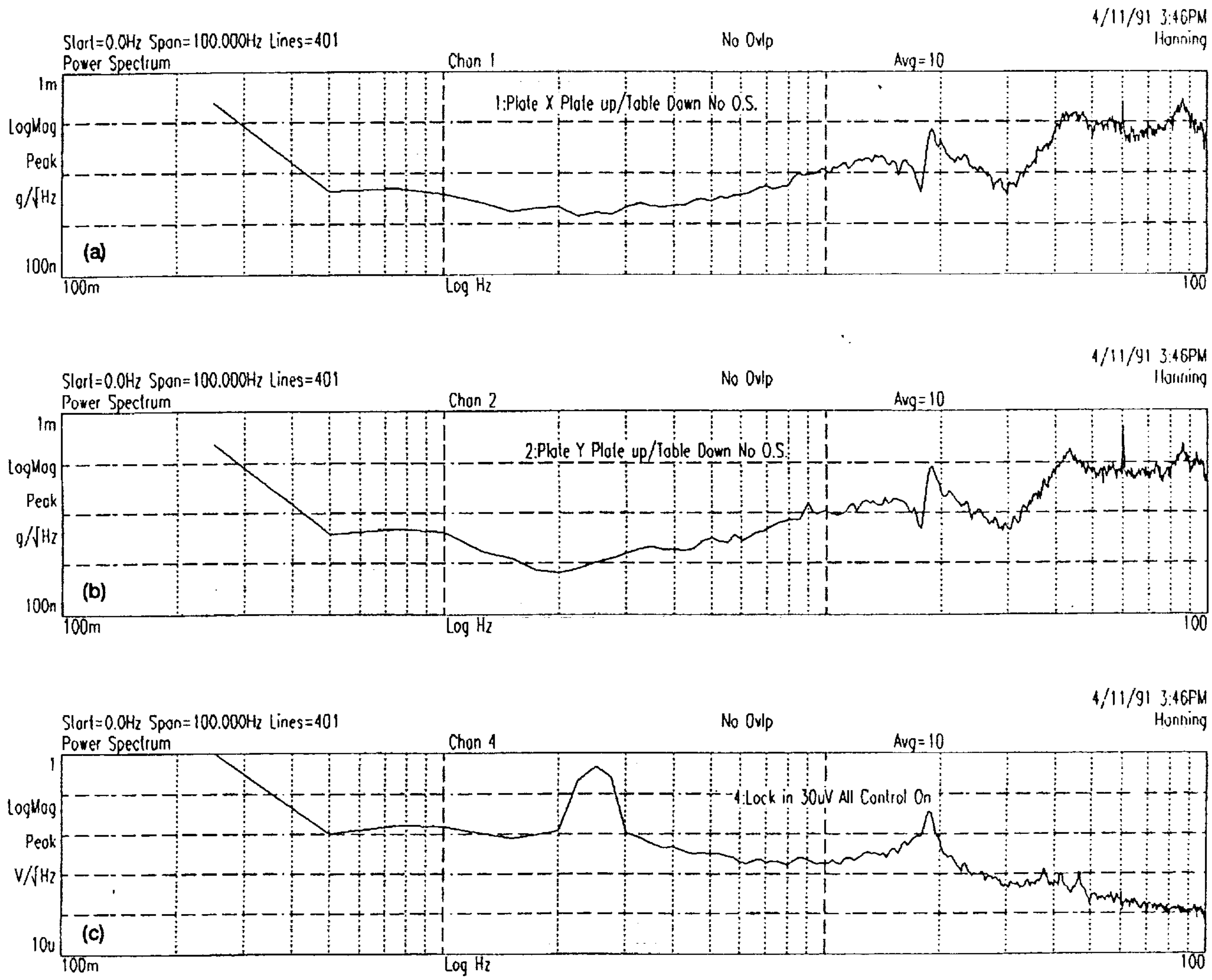


Figure 6.—Plots of Fig. 5 extended to 100 Hz.

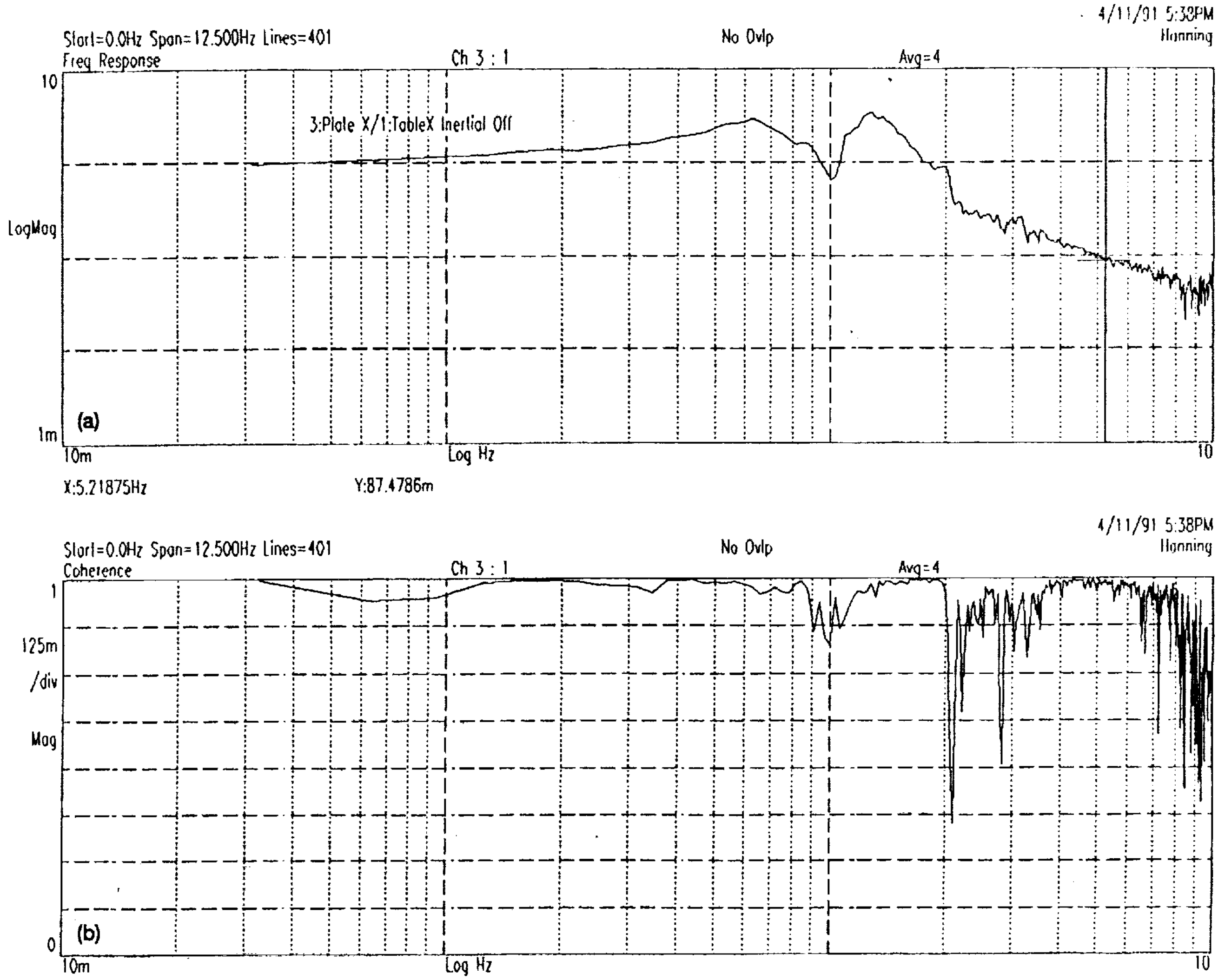


Figure 7.—Transmissibility plot for horizontal translation "X" accelerations. Inertial control is not used. Note the expected f^{-2} rolloff above 2 Hz. Lower trace plots coherence, a measure of signal-to-noise ratio for this measurement.

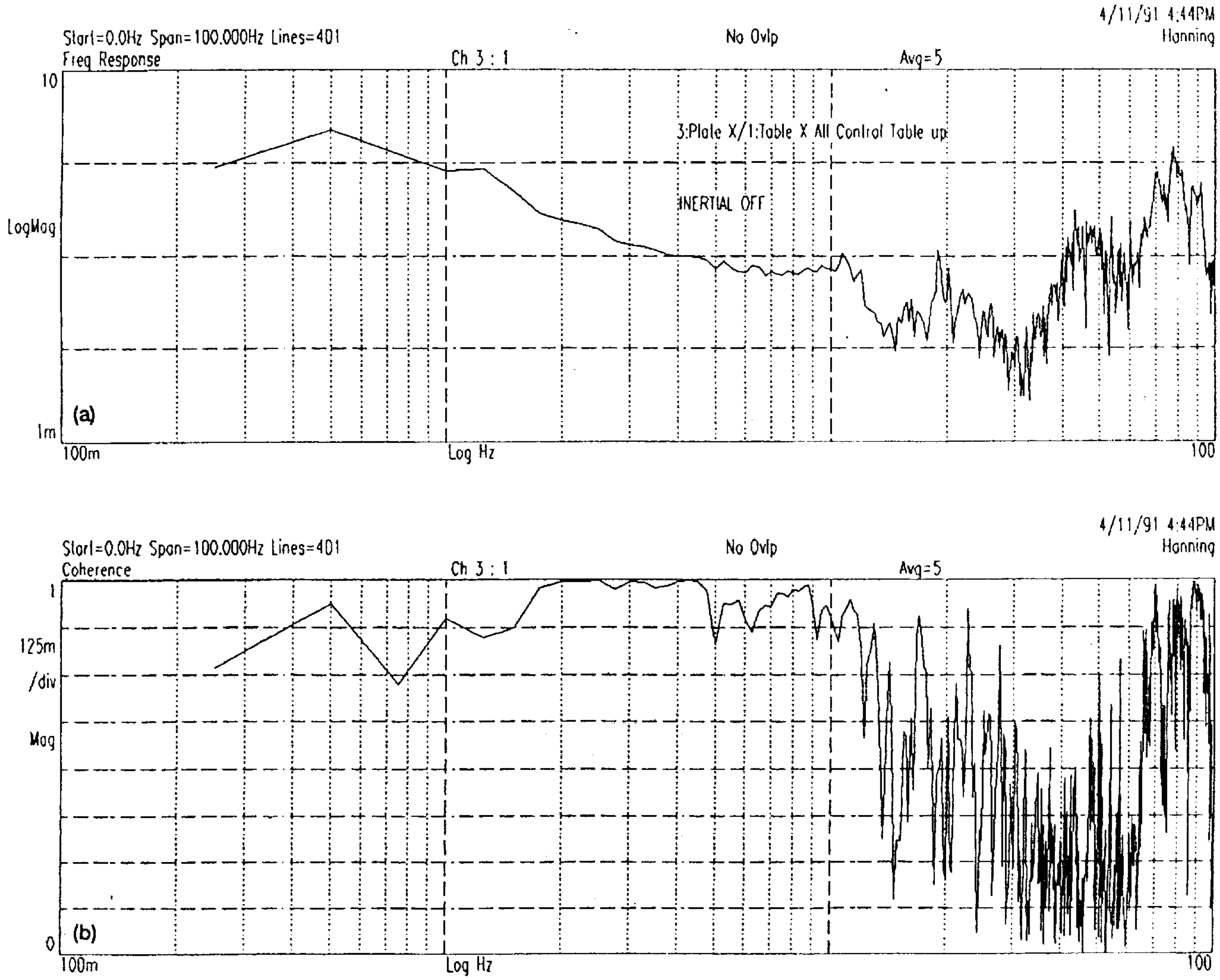
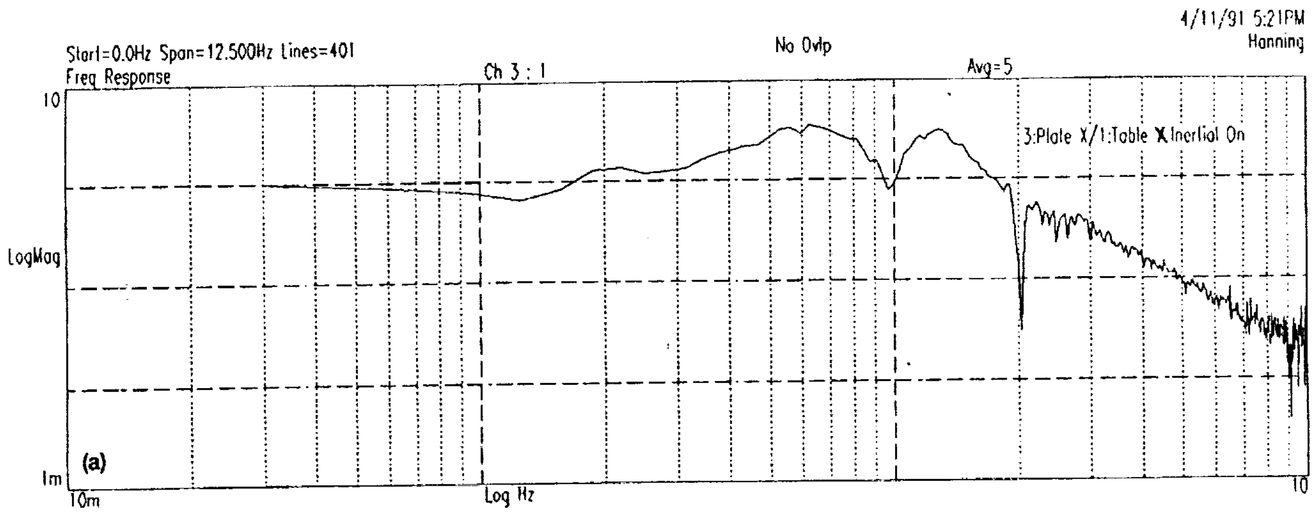


Figure 8.—Fig. 7 extended to 100 Hz. Signal-to-noise ratio is poor above 15 Hz.



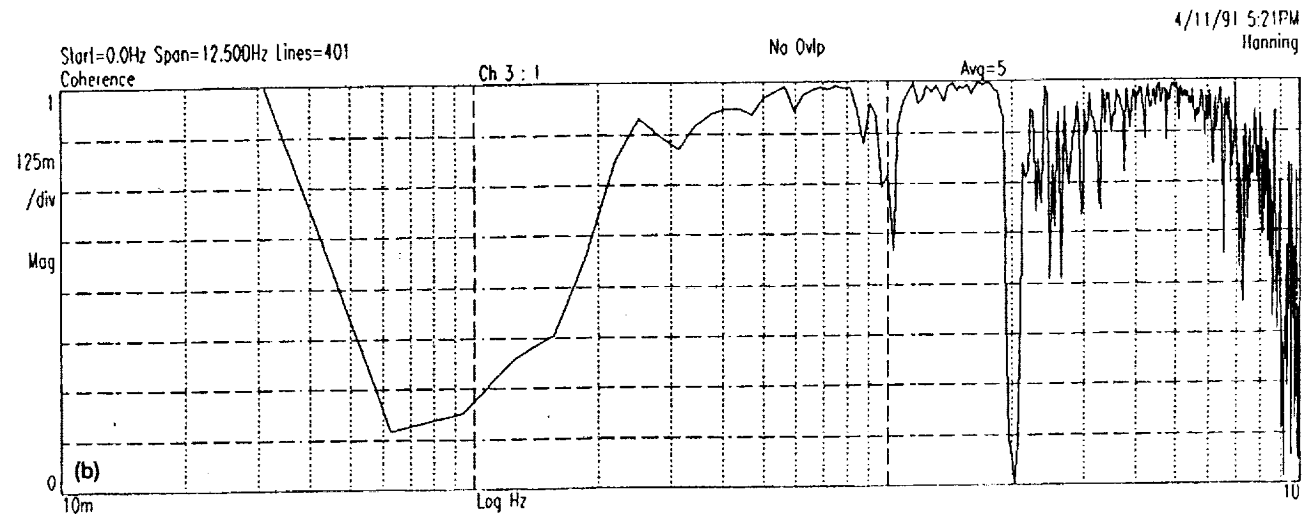


Figure 9.—Transmissibility measurement of Fig. 7 repeated with inertial control turned on. (See Tables II and III for the numerical parameters.) Isolator behavior is not significantly changed due to wrong inertial control parameter corresponding to a 180° phase shift of acceleration signals.

Inert. Ref. Platform vs. Inert. Ref. Off Transfer Functions

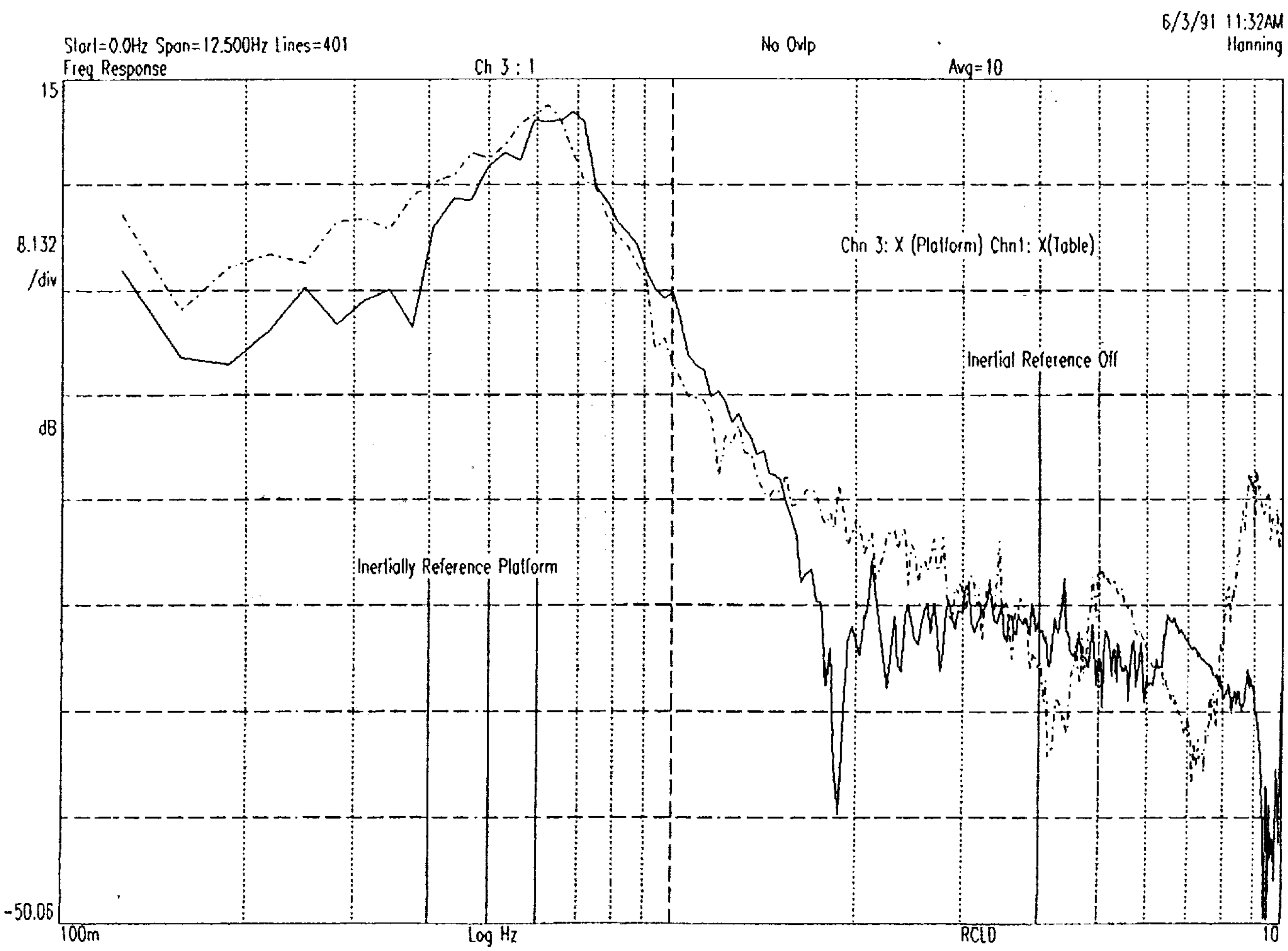


Figure 10.—Frequency response or transmissibility measurement without the torsion oscillator demonstrating inertial control vs. an active relative harmonic oscillator as in Fig. 7.

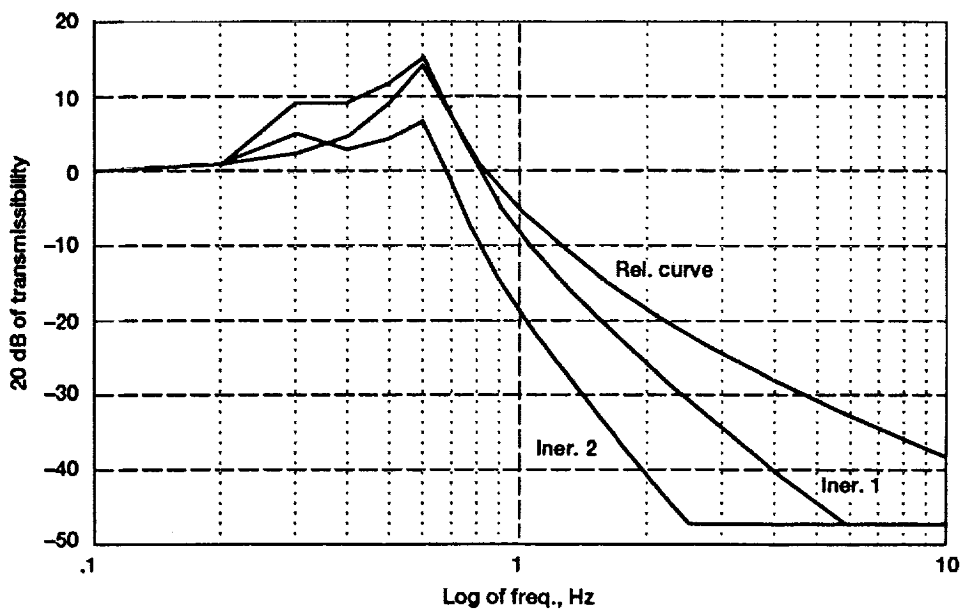


Figure 11.—Theoretical predictions of inertial vs. relative control.

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13. ABSTRACT ***(Maximum 200 words)***

Critical point viscosity measurements are limited to their reduced temperature approach to T_c in an Earth bound system, because of density gradients imposed by gravity. Therefore, these classes of experiments have been proposed as good candidates for "microgravity" science experiments where this limitation is not present. The nature of these viscosity measurements dictate hardware that is sensitive to low frequency excitations. Because of the vibratory acceleration sensitivity of a torsion oscillator viscometer, used to acquire such measurements, a vibration isolation sensitivity test was performed on candidate "microgravity" hardware to study the possibility of meeting the stringent oscillatory sensitivity requirements of a NIST torsion oscillator viscometer. A prototype six degree-of-freedom active magnetic isolation system, developed at the NASA Lewis Research Center, was used as the isolation system. The ambient acceleration levels of the platform were reduced to the noise floor levels of its control sensors, about 1 microgravity in the 0.1 to 10 Hz bandwidth.

14. SUBJECT TERMS

Vibration isolators; Vibration damping; Microgravity experiments; Active; Electromagnetics; Viscosity measurements; Shear mode viscometer

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