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SIMULATION, DESIGN, AND TESTING OF A HIGH-PERFORMANCE MULT-AXIS HEXAPOD FOR VIBRATION ISOLATION

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ABSTRACT

This paper describes the simulation, design, and testing of a high-performance six degree-of-freedom hexapod for the purpose of isolating sensitive payloads from low-frequency vibrations. Design criteria required the hexapod to support a generic payload up to 500 lb with an isolation plunge frequency of approximately 1 Hz. Simulations were performed using Matlab in order to determine the optimum geometry of the base and platform structures in order to provide the best combination of translation-rotation uncoupling, frequency spread, plunge frequency, and jitter. Based on these simulation results, hexapod base and platform structures were designed and fabricated based on a 50 inch-diameter platform size. All of the accumulators and pneumatic hardware were embedded into the base structure to allow for a totally contained system. Modal testing of the hexapod was performed in order to verify the modes predicted by the model.

INTRODUCTION

Parallel positioning systems such as hexapods are often required for isolation of sensitive payloads in harsh environments. The advantages of parallel positioning systems over serial systems include built-in redundancy, high load-carrying capacities, as well as absence of error summation issues. Hexapods can be traced back to the so-called Stewart platform, designed by D. Stewart (1966). In this application, Stewart used a parallel arrangement of six isolators connected to base and platform structures to simulate flight conditions. Since then, parallel positioning systems have found their way into applications that can be grouped into two categories: 1) Precise positioning and 2) Vibration isolation. Precise positioning systems include precision machining, robots, as well as beam-pointing optical systems. Vibration isolation involves either isolating a sensitive payload from base excitation or isolating the base from a noisy payload, e.g., a rotating machine. Within the vibration isolation group, hexapods may be distinguished as being either active or passive, depending on the type of isolators used. Active isolators may use magnetic or piezoelectric technologies while passive isolators are often based on pneumatics or viscoelastic material.

Along with the isolators themselves, the design of a hexapod is not a simple task. The hexapod design can be based on several performance criteria that are commonly payload-specific. These criteria may include: payload weight, payload jitter (assuming beam-pointing device), and suspension frequencies. One important criterion that is difficult to design for and often gets ignored is uncoupling of payload translations and rotations. This criterion becomes very important when isolating a beam-pointing payload. Rotations of the payload due to base translations are highly undesirable and can lead to large jitter errors. These rotations are often difficult to counteract with passive systems however active systems can be used to cancel any moments caused by translational displacements. If a passive system is desirable, the geometry of the hexapod is a major factor in reducing the coupling effects. In particular, isolator connection locations on the payload and base, isolator azimuth and elevation angles, as well as payload platform geometry are important considerations for minimizing the coupling effects.

NOMENCLATURE

- a upper attachment (payload platform) radius
- b lower attachment (base platform) radius
- α angle between upper isolator attachment locations (on payload platform)
- β angle between lower isolator attachment locations (on base platform)

DESIGN SPECIFICATIONS AND OBJECTIVES

The overall objective was to design a hexapod based on the following specifications:

- Payload platform size to allow for payload of approximately 30 x 30 inches (76 x 76 cm).
- Supporting a payload of 200 to 500 lb (90 to 227 kg).
- Utilizing an isolator air pressure between 40 and 120 psig (0.28 and 0.83 MPa). This specification was desired in order to use COTS pneumatic components to keep the overall system cost relatively low.
- Uncoupling of the payload rotations and translations. The isolators were to be arranged so that the resulting mode shapes contained either a dominant translation or a dominant rotation. This meant that rocking and swinging motions of the payload were to be minimized.
- Grouping of natural frequencies.
- Plunge frequency of approximately 1.0 Hz.
- Minimize jitter to allow operation of optical instrument. Jitter was calculated as the powerwise sum of the absolute rotation about the x and z axes, which are perpendicular to the line of site, to each of the three base acceleration PSD functions. This assumed that the x, y, and z base accelerations were uncorrelated, which they were not. However, this procedure was used in order to compare different designs. The coordinate system used is defined in the Methodology section.

The two conditions that are extremely important and not always easy to meet are the uncoupling and jitter specifications. If a high degree of coupling between translation and rotation exists, any translational base motion could result in extreme rocking or swinging motions of the payload leading to large jitter errors.

The operating environment of the hexapod was such that the above requirements were to be met with the platform subjected to a random input consisting of a PSD function recorded inside a Boeing 747 in straight-and-level flight. Based on these design specifications, the specific objectives were:

1. To determine the hexapod geometry including:
   a) Isolator connection locations on the payload end.
   b) Isolator azimuth angles. These angles are defined in the following section.
   c) Isolator elevation angles (also defined in the next section).
2. To characterize the performance of the simulated system based on selected geometries in response to a PSD function of a 747 in straight-and-level flight. The performance criteria are described in the following section.
3. To design a pneumatic isolator based on frictionless air bearings and an air piston to allow for 3.2 cm of stroke.
4. To perform modal testing on the actual hexapod.

SIMULATION INPUT PARAMETERS

A program created in Matlab was used to perform the simulations of this six degree-of-freedom passive isolation system. Input parameters included:

- Payload weight. For the results presented, all the simulations are based on a payload weight of 200 lbs.
- Payload inertia values. For these simulations, the payload was assumed to be a symmetrical right circular cylinder with a height of 61 cm and a diameter of 91 cm.
- Stiffness and damping values for each isolator.
- x, y, and z coordinates for both ends of the isolators.

Isolator coordinates were defined by two angles: α and β. The following figure is used to show what α and β measured. As noted on the diagram, the smaller inner circle represents the platform while the outer circle represents the base. The bold lines are the isolators. The coordinate system consisted of the origin at the center of the circles, the +x axis horizontal to the right, +y axis vertical and up, and +z axis through the center of the circles up out of the page. Each circle was divided into three regions by three lines spaced 120 degrees apart. The 120 degree-dividing lines on the platform (located at 90, 210, and 330 degrees) were rotated 60 degrees relative to the 120 degree-dividing lines on the base (located at 30, 150, and 270 degrees).

Besides the x and y coordinates, the following variables were also used to describe the orientation of the struts:

- Isolator azimuth angles, defined as the angle between the +x axis and the projection of the isolator on the xy plane with the isolator end being on the payload end.

- Isolator elevation angles, defined as the angles between the isolator axes and their projections on the xy plane.
- Isolator length.
- Isolator stiffness and damping. Based on results obtained from previous studies, a piston diameter of 3.5 cm and a chamber volume of 13109 cm^3 were selected for each isolator.

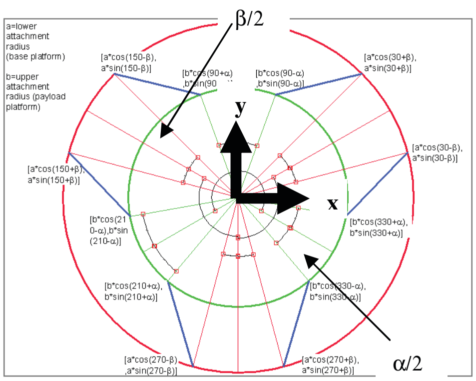

Figure 1: Top view of diagram used to define isolator locations. The variable α is the angle between upper isolator attachment points (payload platform) and the variable β is the angle between lower isolator attachment points (base platform).

The coordinate system used in the Matlab simulation program was set up similarly to that shown in Fig. 1. The origin was placed at the payload center of gravity. For all the analyses unless noted, the payload CG was recessed 1 inch below the upper strut connection plane.

SIMULATION PROCEDURE

The simulation experiment was set up as a factorial design with the factors being α, β, and platform diameters. This type of experiment meant that all possible combinations or levels of α and β were considered. The angles α and β were varied from 0 to 50 degrees with 5 degree intervals. Due to these angles as well as selected base and platform diameters, the isolator elevation angle varied from 28 degrees to 45 degrees. An angle less than 28 degrees would require an air pressure more than the specified maximum of 120 psig and an angle more than approximately 45 degrees results in poor isolation performance.

For the first simulation, β was set to 0 degrees indicating adjacent isolators were attached to the same point on the base platform. For this specified angle, α was varied between 0 and 50 degrees. The base diameter can be thought of as a nested factor within α in that for each value of α, the base diameter was also varied.

For the next simulation, β was incremented 5 degrees and the above steps were repeated. For each of the platform and base configurations, the performance was characterized in terms of:

- Plunge frequency (z direction).
- Frequency spread: ratio of maximum natural frequency to minimum natural frequency.
- Decoupling of translation and rotation. This was determined by examining each of the mode shapes for a dominant row and was more of a qualitative figure of merit rather than quantitative. The decoupling degree was qualitatively described as either moderate or slight based on the relative magnitude of the two translation and rotation values and the motion resulting.
- Jitter in units of microradians (μRad), as previously defined.

SIMULATION RESULTS

Selected geometries that provided the most promising results are presented here. The following figure shows a stick model subjected to increasing values of angle α, which resulted in increasing the distance between isolator connection locations on the payload platform. Numbers 1 through 6 correspond to the payload platform isolator connection locations and numbers 7 through 12 are the base platform isolator connection locations. Table 1 presents the results.

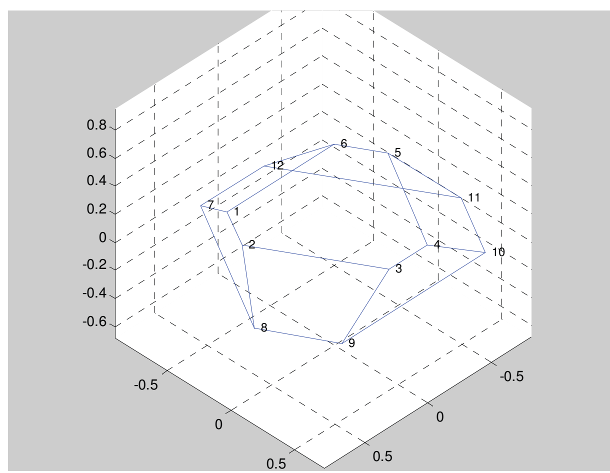

Figure 2: Model of hexapod with platform diameter=56 in., base diameter=66 in., α=30 degrees and β=35 degrees.

For this geometry the frequency spread reached a minimum around α=35 to 45 degrees. The degree of coupling increased with α, except for α=40 degrees where the rocking issue improved slightly from the previous design. This may represent an angle where the isolator ends could be located while minimizing rocking issues based on this design. Also, jitter decreased with increasing α.

Table 1: Effects of increasing angle α on hexapod shown in Fig. 2.

Angle α (degrees)	30	35	40	45
Plunge Frequency (Hz)	0.9	1.3	1.5	1.7
Ratio of max/min frequency	7.0	4.4	4.1	5.4
Decoupling degree	2 modes show moderate rocking	Worse than α=10	Slightly less rocking issue than α=10	Worse than α=40
Jitter (μRad)	306	241	178	128

Next, the effect of increasing the distance between isolator connection points on the base platform (increasing angle β) was investigated. Shown in Fig. 3 is a stick model of a design subjected to increasing values of β and the results are shown in Table 2.

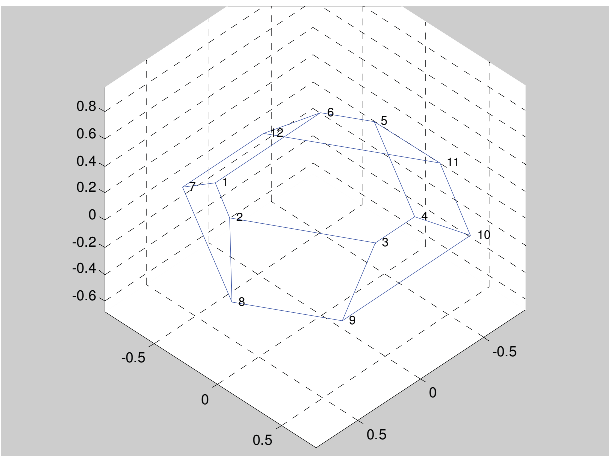

Figure 3: Model of hexapod with platform diameter=56 in., base diameter=66 in., α=30 degrees and β=45 degrees.

Table 2: Effects of increasing angle β on hexapod shown in Fig. 3.

Angle β (degrees)	30	35	40	45
Plunge Frequency (Hz)	0.9	1.3	1.5	1.7
Ratio of max/min frequency	8.5	4.4	4.1	5.3
Decoupling degree	2 modes show moderate swinging	Slightly worse than β=20	Better than β=30	Worse than β=40
Jitter (μRad)	301	239	171	134

Based on the results in Table 2, jitter decreased with angle β. In general, the swinging modes became more severe with increasing β except for a slight improvement around β=40, suggesting a possible optimum value for β between 30 and 40 degrees. This was further investigated and the results are presented below.

Other simulations showed that the plunge frequency increased with elevation angle as one would expect since this caused the axial load on each isolator to increase. The frequency spread as well as the jitter decreased with increasing elevation angle. Jitter was dominated by y-direction (along the line of sight) ground motion.

It was found that an optimum angle of 37.7 degrees resulted in a minimal rocking motion. From Table 2, an apparent optimum value for α of 40 degrees was found that minimized rocking issues. Referring to Table 3, a value for β between 30 and 40 degrees results in a slight increase in decoupling degree compared to β=30 and 50 degrees.

Although the results aren't presented here, the effect of increasing the base diameter for a fixed payload platform diameter led to decreases in both jitter and frequency ratio. Based on these results and numerous other analyses involving different combinations of α, β, isolator elevation angle, and base diameter, it was determined that a base diameter of 66 inches and a payload platform diameter of 56 inches provided for the best combination of uncoupling, frequency spread, plunge frequency, and jitter. Table 3 shows the results of this optimum geometry.

Table 3: Optimum values of input parameters found for isolators

Parameter	Value
α (deg.)	40
β (deg.)	35
Elevation angle (deg.)	37.7
Base diameter (inches)	66
Platform diameter	56
Plunge Frequency (Hz)	1.6
Ratio of max/min frequency	3.9
Jitter (μRad)	151

The mode shapes and suspension frequencies for this optimum geometry determined from a 6 degree-of-freedom (DOF) Matlab model are shown in Table 4. DOF's X, Y, and Z are the respective x, y, and z translations of the payload CG. Alpha, Beta, and Gamma (not to be confused with strut angles α and β) are the rotations of the payload about the x, y, and z axes, respectively.

The swinging mode means that the payload is rotating about a point above its CG. There will be two swinging modes: one in the +x/-x direction and another in the +y/-y direction. The rocking mode means that the payload is rotating about a point below its CG. There will also be two rocking modes similar to the swinging mode directions. There will also be a plunge mode where the payload moves vertically up and down

Table 4: Calculated suspension mode properties

Undamped Freq., (Hz)	0.84	0.85	1.58	2.95	3.28	3.29
Modal Masses	0.60	0.58	0.52	4.04	79.0	5.19
X	1e0	-2e-2	-1e-5	1e0	2e-1	-2e-2
Y	-2e-2	-1e0	2e-2	-2e-2	-1e-1	-1e0
Z	-3e-4	-2e-2	-1e0	-6e-4	-4e-3	-4e-2
Alpha	-7e-4	-4e-2	-6e-4	5e-3	4e-2	4e-1
Beta	-4e-2	7e-4	-4e-7	3e-1	5e-2	-6e-3
Gamma	-2e-4	3e-6	-2e-9	7e-3	1e0	4e-3
Shape	swinging	swinging	plunge	rocking	yawing	rocking

as well as a yaw mode where the payload rotates about a vertical axis passing through its CG.

HEXAPOD DESIGN AND FABRICATION

A hexapod was designed based on the optimum geometry found from the Matlab simulations. The base and payload platforms were constructed using a rib and stringer technique (often called former and longeron, respectively) similar to that found in light aircraft fuselages. When combined with skins and fastened using rivets, this type of construction is known for its high stiffness/weight ratio. The total weight of the base platform including all the pneumatic fittings and pipes is approximately 120 lb (54 kg) and the total weight of the payload platform is 68 lb (31 kg). The entire hexapod is shown in Fig. 4.

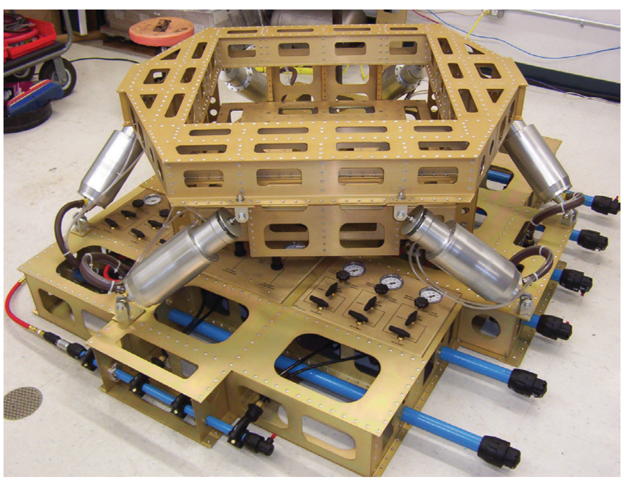

Figure 4: Actual hexapod designed and constructed.

The sheet metal parts (ribs, stringers, and skins) were fabricated using 12 gauge aluminum sheet metal with a thickness of 0.081 in. (0.21 cm). Also, the parts were covered with a gold chemical conversion coating. The ribs, stringers, and skins for the hexapod platforms were assembled using pop rivets with aluminum bodies and mandrels. The sizing and spacing of the pop rivets depends on parameters such as: clamping thickness, distances from edges, distance from rivet center to rivet center, and joint strength. The pop rivets selected for the entire assembly were 3/16 in.-diameter (0.48 cm) and spaced 1.25 in. (3.2 cm) rivet center-to-center. Note that temporary Cleco-style fasteners were used to ensure the pieces fit together correctly before pop riveting them. The platforms were designed in order to place rib-stringer intersections at the isolator connection locations, as shown in Fig. 5.

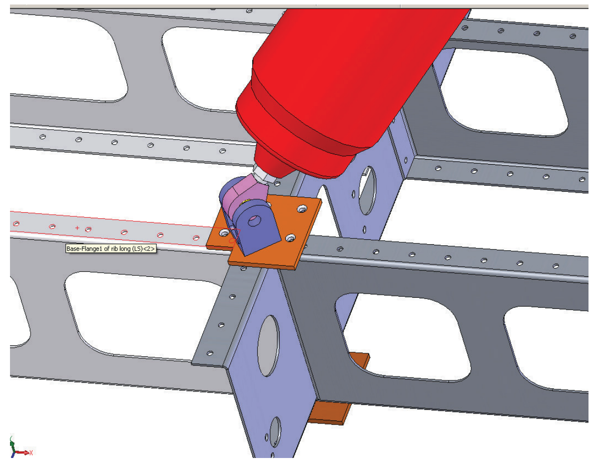

Figure 5: CAD model showing intersection of ribs and stringer where the strut attached to the platform. Note that the skin was removed in this figure in order to show the intersection.

Aluminum stiffener plates (4.6 mm thick) were used at these strut connection locations between the clevis and rib-stringer intersections. The stiffener plates were riveted to the stringers and ribs. The payload plate of the platform was designed in order to allow a wide variety of payloads to be secured. The plate was constructed out of 6.4 mm thick aluminum and contained several cutouts to save weight as well as to allow for wires and hoses from the payload to pass through. Tapped holes of ¼"-28 threads were spaced two inches center-to-center to allow for securing a number of different payloads with various sizes and shapes. The payload plate was designed to accommodate any payload with cross section dimensions up to 69 x 74 cm. Note the height dimension was unrestricted. Figure 6 shows a picture of the payload plate.

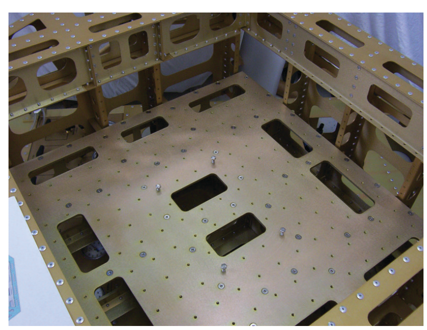

Figure 6: Payload plate of platform.

DETAILS OF PNEUMATIC COMPONENTS

The control of air pressure for each strut involved five pneumatic components:

1) Regulated air pressure gauge.
2) Pressure regulator.
3) Two-way valve for air pressure blowdown (relieving strut pressure).
4) Three-way valve to select the source for the strut as either the regulated air or vent.

All of the pneumatic components of the hexapod were contained in the base platform. This resulted in a relatively compact all-in-one system with no external components needed, except for an air supply. As previously mentioned, there are three control panels. Two of the control panels are used for controlling the pneumatic strut pressures while the center control panel is used to control the strut air bearing pressures. Each strut has its own reservoir of approximately 1640 cm^3. For the modal testing, external accumulator tanks were used to expand the strut reservoir value to 13110 cm^3. Shown in Fig. 7 is one of the two strut control panels and in Fig. 8 the air bearing control panel.

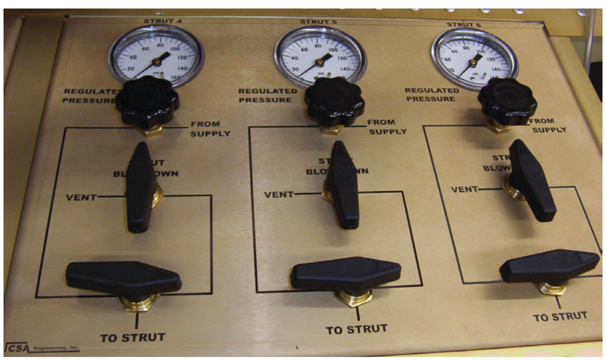

Figure 7: Strut control panel.

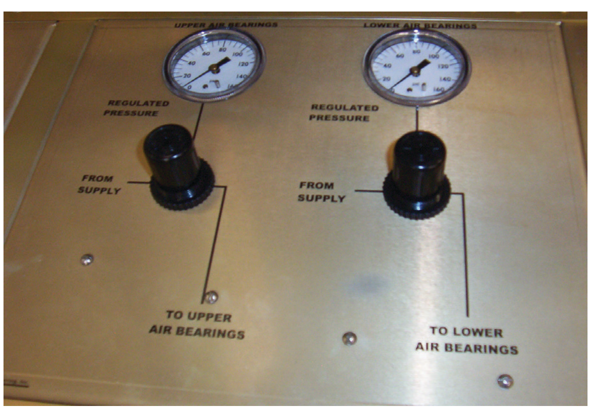

Figure 8: Air bearing control panel.

STRUT DESIGN

Friction effects can render isolators useless against low-level base motion. The heart of the hexapod is a system of six pneumatic isolators (struts) with each incorporating custom-designed air bearings and an air piston. Each pneumatic isolator was designed to incorporate two custom air bearings and an air piston. The air bearings and piston allowed for essentially zero friction resulting in lower attainable isolation frequencies leading to increased isolator performance. The isolators were designed to have a length at mid-stroke of 46.2 cm, overall diameter of 12.5 cm, and a weight of 5.9 kg. The area of the piston in each isolator was 9.7 cm^2. The total stroke was 3.2 cm.

MODAL TESTING

Modal testing was performed on the hexapod shown in Fig. 4 in order to determine the six suspension frequencies. The total weight of the payload for the modal tests including the structure was 200 lbs. A single electromagnetic shaker was used as the excitation device (Fig. 10). The shaker was connected to the payload platform by a stinger as well as a load cell. Matlab simulations were performed in order to determine shaker locations (Fig. 9) such that each of the six suspension modes shown in Table 4 could be excitable and measurable.

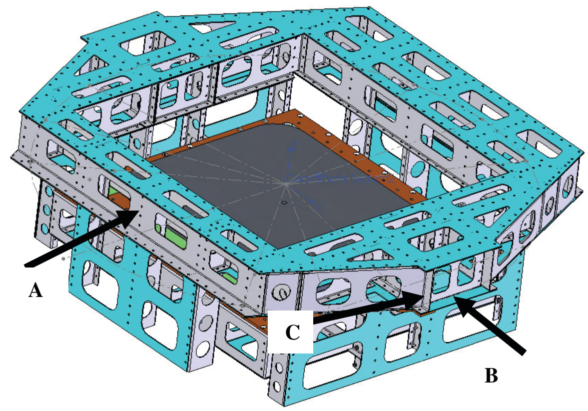

Figure 9: Shaker locations

From the simulations, it was determined that four shaker locations as shown in Fig. 9 will allow for the measurements of the six suspension modes. Location D (not shown) is with the shaker directly under the payload CG and oriented vertically.

- Location A for the swinging and rocking modes in the y direction.
- Location B for the swinging and rocking modes in the x direction.
- Location C for the yaw mode as well as swinging in the y direction.
- Location D for the vertical plunge mode, swinging in the y direction, and rocking in the y dirction

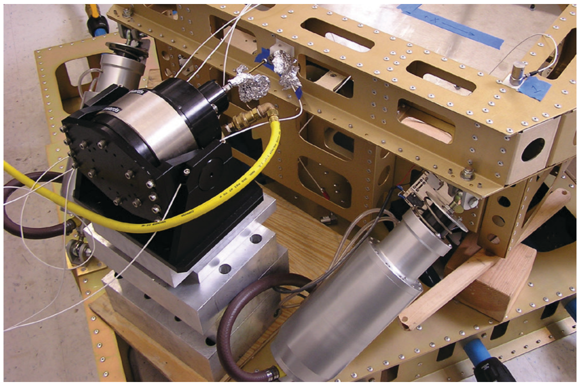

Figure 10: Shaker position A. Note that aluminum foil shown in the picture was used to shield the temperature-sensitive load cell.

The simulated frequency response functions of the acceleration/force at the driving points are shown in Fig. 11 through 14.

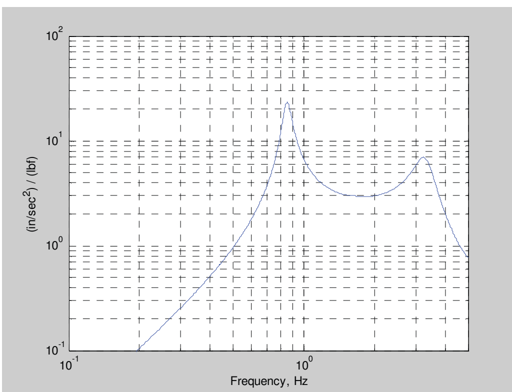

Figure 11: Shaker position A Drive-DOF Simulated Acceleration/Force frequency response.

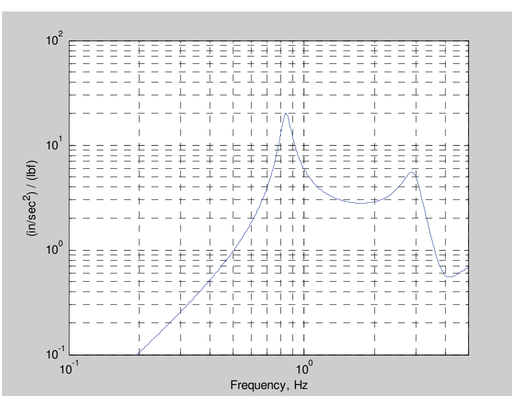

Figure 12: Shaker position B Drive-DOF Simulated Acceleration/Force frequency response.

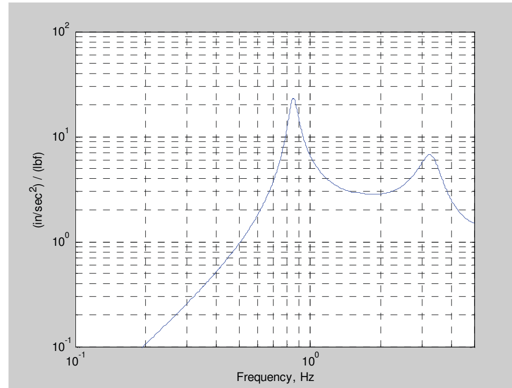

Figure 13: Shaker position C Drive-DOF Simulated Acceleration/Force frequency response.

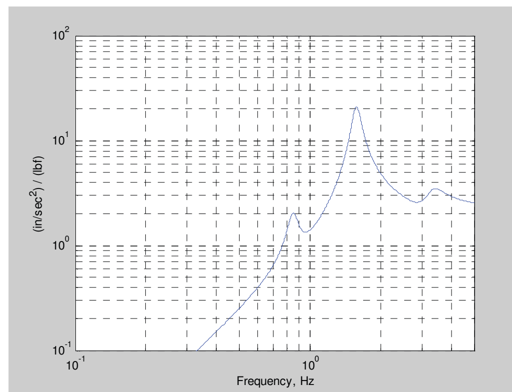

Figure 14: Shaker position D Drive-DOF Simulated Acceleration/Force frequency response.

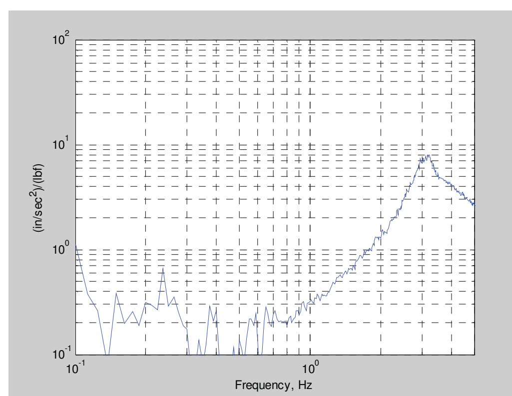

Figure 15: Shaker position A Drive-DOF Measured Acceleration/Force frequency response.

The shaker produced a random time waveform with a nominally flat spectrum band-limited to the range between 0.1 and 10 Hz. This resulted in the shaker imposing disturbances on the order of approximately 1 kg. The platform responses were measured using 5g accelerometers with a sensitivity of approximately 1 V/g. The accelerometers were mounted in a triaxial arrangement at four corners of the platform as well as at the shaker driving point location.

The measured frequency response plots verified those predicted by simulation shown in Fig. 11 to 14. However, due to limitations of the signal generator/shaker combination used, frequencies of less than 1 Hz were difficult to measure, as shown in the measured frequency response in Fig. 14.

From Fig. 15, the rocking mode in the y direction is easily discernable at 3.3 Hz. However, the swinging mode in the same direction at 0.85 Hz is not so obvious. Similar results were seen for the swinging mode in the x direction. Modal parameter estimation and mode shape displays will be performed in the near future.

CONCLUSIONS

The following are the conclusions from the simulations.

- Jitter was dominated by y-direction ground motion, which was along the line of sight.
- The frequency spread decreased as the strut elevation angle increased. In other words, increasing the strut elevation angle caused the natural frequencies to bunch closer together.
- There existed an optimum strut elevation angle of 37.7 degrees that resulted in minimal coupling of translation and rotation.
- There existed an optimum value for α of approximately 40 degrees that led to minimal coupling of translation and rotation. This was seen in the two mode shapes involving the rocking motions.
- There existed an optimum value for β of approximately 35 degrees that led to minimal coupling of translation and rotation. This was seen in the two mode shapes involving the swinging motions.
- The modal frequencies consisted of:
    - swinging modes at 0.84 and 0.85 Hz
    - rocking modes at 2.95 and 3.29 Hz
    - a plunge mode at 1.58 Hz
    - a yaw mode at 3.28 Hz

ACKNOWLEDGMENTS

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