Novel Guided Head Expander Design Uses Close Coupled Inertial Masses and Hydrostatic Bearings to Minimize Cross-Axis Motion

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Abstract

When vertical vibration testing of large test articles is required, it is common to install a head expander on the armature of a shaker. Larger test articles often have a center of gravity relatively far above the mounting surface. When combined with the armature and head expander, these test articles may exhibit multiple structural resonances within the desired test band that do not exist in the intended application. These test configuration-driven characteristics are likely to create unwanted cross-axis excitation during a vibration test.

The difficulty in controlling unwanted cross-axis motion usually increases when testing large items. Excessive cross-axis motion can "over-test" the test item, creating the risk of damaging the test item, or can limit the input in the test axis, thus jeopardizing a successful test.

Orbital Sciences commissioned the design of a guided head expander system that greatly reduces the cross-axis motion at the test article mounting surface of the head expander. The design submitted by Team Corporation couples large inertial masses to the head expander through high-stiffness, hydrostatic, self-aligning bearings. Together, the guided head expander and inertial mass structures have a first resonance higher than the test band of interest and provide high dynamic stiffness. The head expander and inertial masses are supported by a suspension system with a low first resonance, below the test band of interest. It is noteworthy that this design approach exhibits high "dynamic" stiffness and low static stiffness.

Conventional designs for this type of equipment may have relatively high cross-axis load ratings, which might suggest that such designs would provide good cross-axis motion control, but these designs often suffer from structural resonances within the test band of interest that produce unwanted cross-axis motion.

KEYWORDS

Lateral acceleration, cross axis acceleration control, vertical vibration test, guided head expander, hydrostatic bearings, spacecraft vibration test

INTRODUCTION

Launch vehicle (LV) providers require vibration tests (usually sine vibration) to demonstrate compatibility with LV launch loads. The typical LV users' manual presents a conservative enveloping environment derived by the LV supplier as a general load case for use in design and analysis and as the nominal input environment for testing. Mission-specific load requirements are based on a coupled LV-to-spacecraft finite element analysis performed by the LV supplier using the spacecraft developer's finite element model and an LV model. From the coupled analysis and the LV supplier's flight experience, responses at specific locations throughout the spacecraft are recovered and minimum input excitation requirements are established. The minimum input requirements are usually significantly less than the users' manual level. The spacecraft provider must demonstrate through the mechanical test program that these requirements have been met by stimulating the spacecraft with a sine environment such that the predicted loads are developed throughout the spacecraft.

Input notching (narrow frequency band reductions in input level) is not only allowed but common in spacecraft testing. Since the test boundary conditions produce modal responses unique to the mechanical test setup, there may be load cases in test that do not exist in the flight environment. Notching is performed on the input levels such that the local responses throughout the spacecraft achieve the predicted levels multiplied by the protoflight test factor, while keeping those responses below the design or qualification envelope of the components. Typically, low-level sine surveys are performed on the spacecraft to understand the response differences between analysis and test configurations. From this low-level survey data, notching of the users' manual input profile is determined.

In late 2002, the first vibration test at Orbital's Dulles Satellite Manufacturing Facility was conducted on the second Star2 GeoComm spacecraft. Although the test was completed successfully, meeting all LV requirements, the test had been particularly difficult in the vertical axis. Loads analysis performed in preparation for the test focused on LV-induced environments and on analyzing the test configuration with fixed-base boundary conditions. The analysis did not include the shaker characteristics and therefore would not predict the test configuration-specific loads. The low-level sine survey test data identified several strong, complex bending modes produced by the vertical axis test configuration. If the input levels were not reduced at these modes, excessive loads would be developed in the lateral direction for various components. The extent of the notching was greater than expected, bringing the input levels down to near the minimums defined by the LV supplier. The vertical axis test-configuration equipment was becoming a limitation on Orbital's ability to meet LV requirements.

Test Configuration Effects

It is not uncommon for the test-configuration boundary conditions to produce unique load cases. In this case, the unusual finding was that the magnitude of the lateral response transfer function was as high as 200% relative to the spacecraft base input. The lateral response was a complex stack-bending mode that occurred in a frequency band from 60 to 80 Hz of a sine sweep test that ranged from 5 to 100 Hz.

The second spacecraft in a series production was tested. The magnitude of the responses decreased slightly and shifted in frequency for the nearly identical design. Prior experience with shaker systems had not shown coupling with the test article in the manner experienced here. Spacecraft in test are instrumented with accelerometers to capture the responses throughout the spacecraft. The accelerometer signals are used to limit the test input to levels that do not exceed analytical predictions, and to prevent over-test in cases such as we were experiencing. From head expander in-axis accelerometers, rotational accelerations could be calculated that correlate with the increase in lateral acceleration. The tests met LV requirements but were difficult to conduct. Notches had to be carefully defined to enable testing without placing the hardware at risk or failing to meet LV requirements.

Testing of the next spacecraft to be evaluated commenced with analysis of the anticipated rocking modes. This spacecraft had a slightly higher center of gravity and increased mass. The results of this test showed that with increased mass, for a structure with approximately the same stiffness, the response decreased in frequency and became more pronounced in magnitude. Despite the complication introduced by the strong first rocking mode, the test met LV requirements and was fully successful. The rocking produced by the shaker and head expander guidance hardware might have been tolerated had there not been programs in production that were larger than this third spacecraft and built on the same primary structure design. The trend would put the next larger spacecraft at risk of not meeting the LV requirements in test.

Figure 1 illustrates the cross-axis acceleration values of up to 2.5 times the base input value. These cross-axis responses, if uncontrolled, can exceed spacecraft allowable limits.

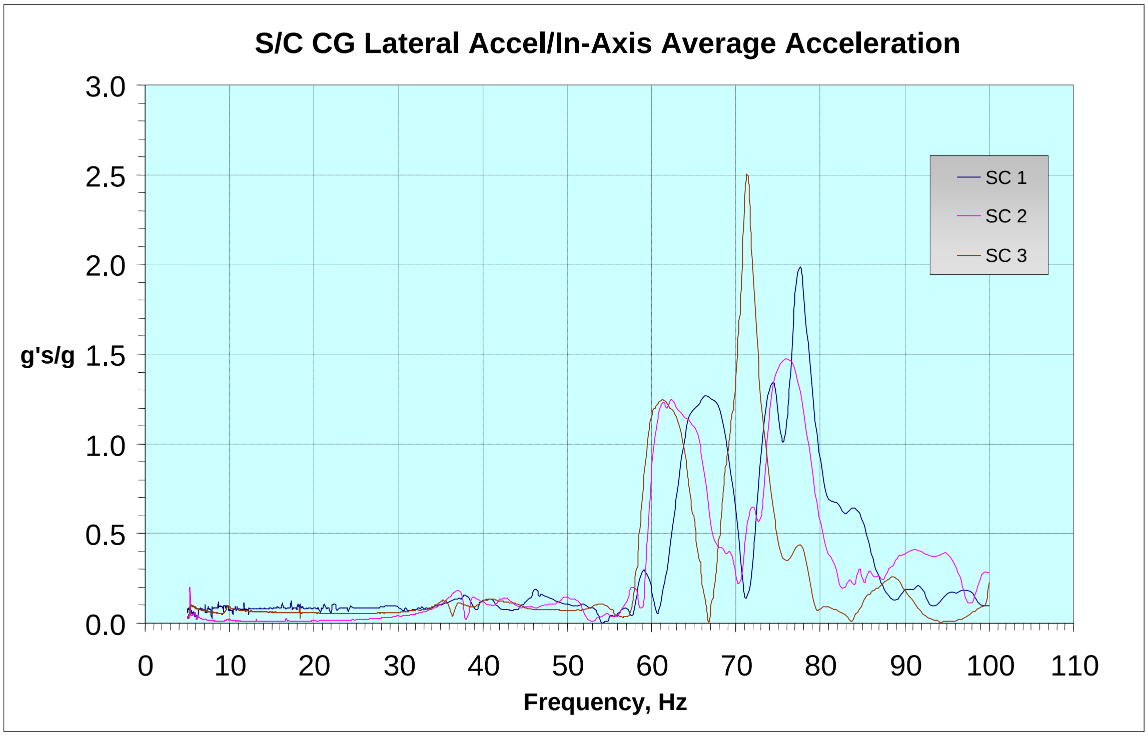

Figure 1. Spacecraft lateral acceleration/in-axis acceleration ratio (computed at spacecraft c.g. based on three accelerometers measuring head expander in-axis response) for several spacecraft. SC1 = spacecraft 1, SC2 = spacecraft 2, SC3 = spacecraft 3.

PROBLEM INVESTIGATION

The instrumentation demands from the spacecraft alone make it difficult to consider using additional instrumentation on the shaker hardware to determine the source of the problem. In early 2004, Orbital contracted with an outside company to conduct a modal test of the vibration equipment to better understand the source and magnitude of the problem and identify correction methods.

The modal test was conducted with an inert mass simulator (see Figure 2). Impact hammer excitation and operational modal surveys were conducted (see Figure 3). The operational testing provided a baseline that could be used for comparison after corrective modifications were installed.

Figure 2. Mass simulator shown mounted on guided head expander.

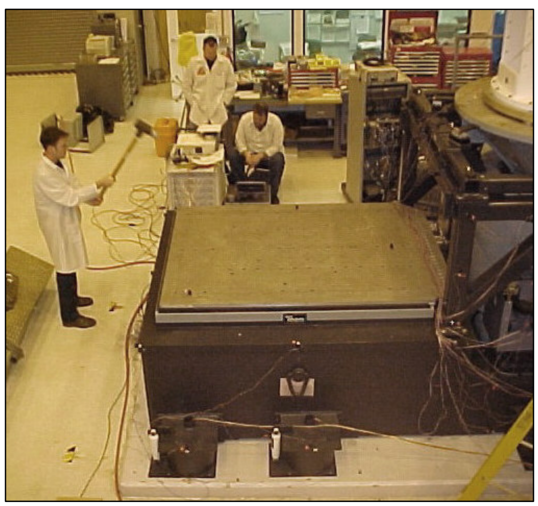

Figure 3. Instrumented hammer used in conjunction with modal survey. Slip table used for horizontal testing is visible, and part of the original guided head expander equipment is visible on the right.

The modal test was conducted in the vertical and horizontal test configurations, with and without a mass simulator. One hundred twenty-six locations were instrumented from the floor up, in 3 DOF. Eleven impact locations were used. The results of the modal test showed no fewer than

31 modes between 5 and 200 Hz for the vertical test configuration with mass simulator, 22 of which were below 100 Hz. The large number of modes below 100 Hz was the focus of the investigation. The major modes of concern could be identified easily from the modal test. Corrective action was not as easy to determine.

With the modal test as a beginning, Orbital developed an analytical model of the vibration machine to help evaluate possible solutions and predict the degree of difficulty when testing various spacecraft. Developing an analytical model of a complicated machine required manufacturing drawings of the components. The manufacturers of the existing shaker, head expander, and slip table/shaker support base provided valuable details useful in the development of the model. Inquiring about the attributes of the equipment resulted in frequent contact with the engineering staff of the three suppliers. Where component properties were not available, components were removed, measured, and tested to determine the required information to aid in modeling. The high degree of non-linearity within the shaker system made the model difficult to correlate to modal test results. Even so, the model proved useful in analyzing potential solutions and helped to avoid costly modifications that would not fix the problem. Some inexpensive modifications were identified to stiffen the head expander support frame, which had modal responses coincident with the frequencies identified as problems. However, those modifications produced only modest improvements. The weak flexural links between the head and support frame were a significant limitation. The primary modes of concern remained.

Off-the-shelf proven systems were investigated prior to developing a unique solution, which entails developmental risk. Through contacts made at the annual AIAA Working Group on Dynamics Space Simulation, Orbital's environmental test engineers discussed the problem with other system operators involved in similar testing. It was found that most vibration test personnel were either resigned to the problem of uncontrolled cross-axis acceleration, actively working to rid their systems of the problem, or were unaware of the problem as a result of the methods used to monitor test performance. Those attempting to correct the problem reported a litany of issues. One company had a system that used stiff pylon support structures to control the guided head expander cross axis loads. In this case, the head guidance design had six journal bearings above the shaker drive motor and was created with the intention of having no significant modes below 200 Hz. The system did not meet specification and had a major rocking mode at 112 Hz with no test article. With a 1588-kg (3500-lb) satellite, the system has a significant rocking response at about 45 Hz. The purchase requirement had been to limit cross-axis response from 5 to 200 Hz to a level of 1/10 the in-axis input. In addition to not satisfying the response requirements, the guided head suffered from binding at the bearings due to thermal expansion of the head during operation. Evaluations of other large-capacity systems showed similar modal characteristics.

Orbital had exhausted the possibility of buying a system that would solve the problem and started looking into a retrofit solution. A stiff support structure to guide the head expander with a hydrostatic oil bearing interface and a means of compensating for the thermal expansion was the baseline concept. Team Corporation had experience with guided head expanders, produced spherical self-aligning bearings, and produced different slip tables for various vibration test applications. Initial discussions expanded on the design concept and showed promise of a solution. Study funding was provided to cover direct costs and goals were established for a more capable guided head.

PROBLEM SOLUTION

Team Corporation reviewed the requirements and began an engineering study of possible solutions.

The Orbital vibration lab is busy and space is at a premium. The new guided vertical head expander (1.5-m (4.921-ft) diameter) would preserve the existing horizontal slip table and shaker and use the existing shaker. The existing shaker is used to drive the slip table, and when rotated into the vertical position is also used to drive the vertical head expander.

The first concept for a new guided head expander began with the idea of using hydrostatic bearings around the perimeter of the head expander to control the line of motion. These bearings were to be mounted on pylon structures anchored to individual concrete footings. In this case, the allowable size and weight of these footings were limited by lab schedules and space allocation. Figure 4 shows a traditional arrangement of hydrostatic bearings and pylons.

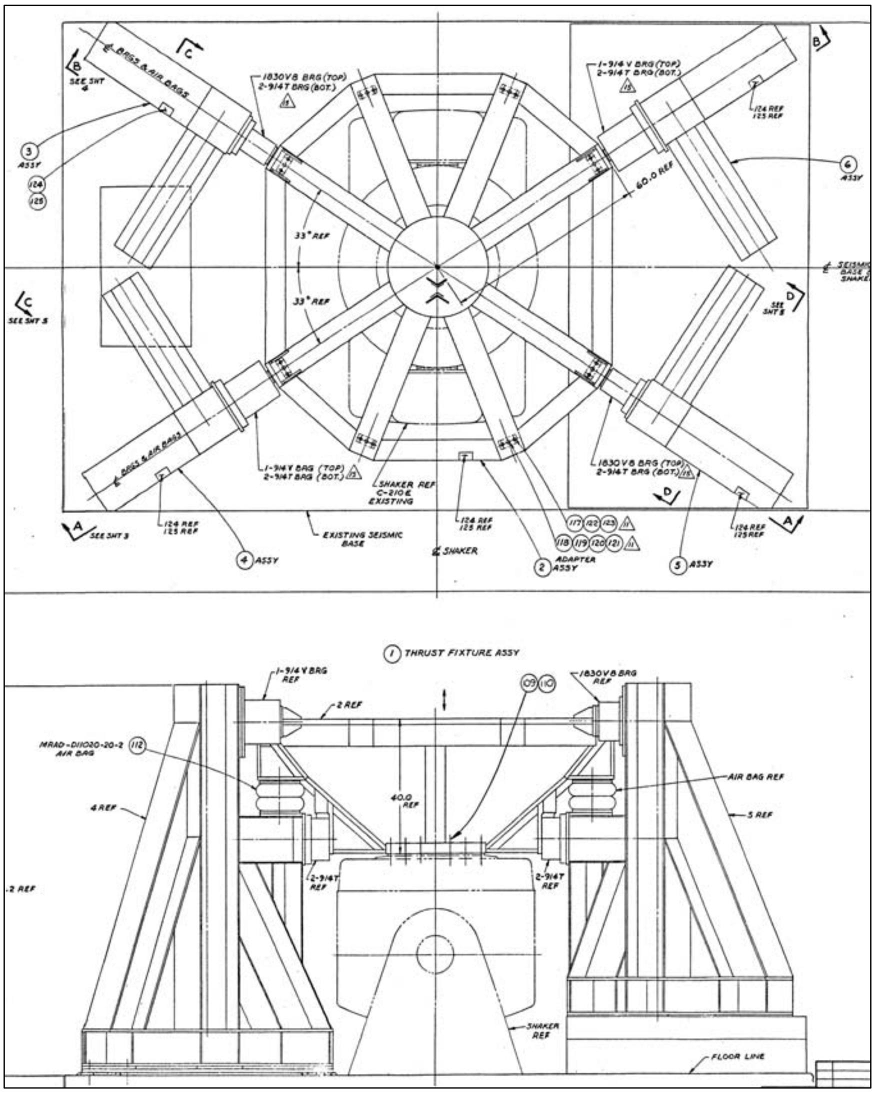

Figure 4. Example of traditional pylon structures and bearing arrangement.

Finite Element Modeling

Finite element modeling of the structural system comprising the head expander, hydrostatic guide bearings, pylon structures, and mass simulator test article was used to evaluate dynamic behavior. A number of structural designs were checked, but all exhibited undesirable frequency response characteristics within the desired test band of 5 to 200 Hz. The structural elements (pylons) that connected the hydrostatic guide bearings to the in-ground concrete footings could not be made stiff enough (within the allowable physical space limits) to deliver the desired dynamic behavior. The physical distance from the planned concrete footings (reaction masses) to the guide bearings and the limitations of the pylon structures would not produce the desired dynamic behavior. It became apparent that the location of the potential reaction mass footings was too far from the guide bearings.

New Concept

A new concept emerged. If the reaction mass could be coupled directly to the guide bearings, then the dynamically "soft" intermediate structure (pylons) that caused the undesired dynamic behavior could be eliminated. Figure 5 shows schematically a section view of the design that proved to be successful. Figure 6 shows an overall view of the final solution.

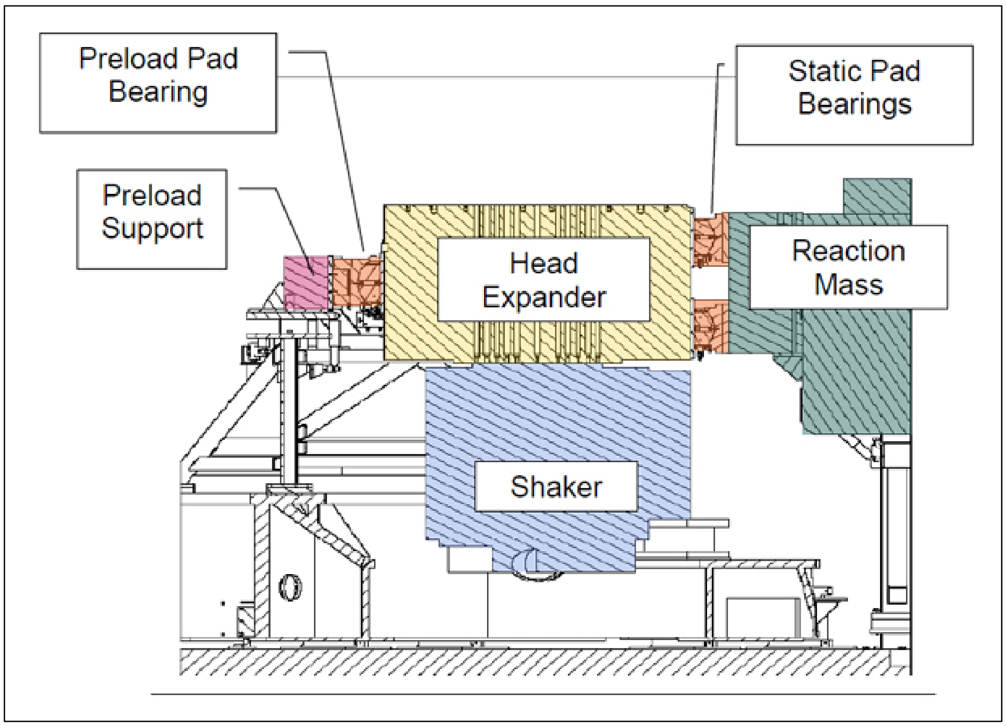

Figure 5. Section view of the successful design. All major components (shaker, reaction mass, and shaker) are supported on air isolators.

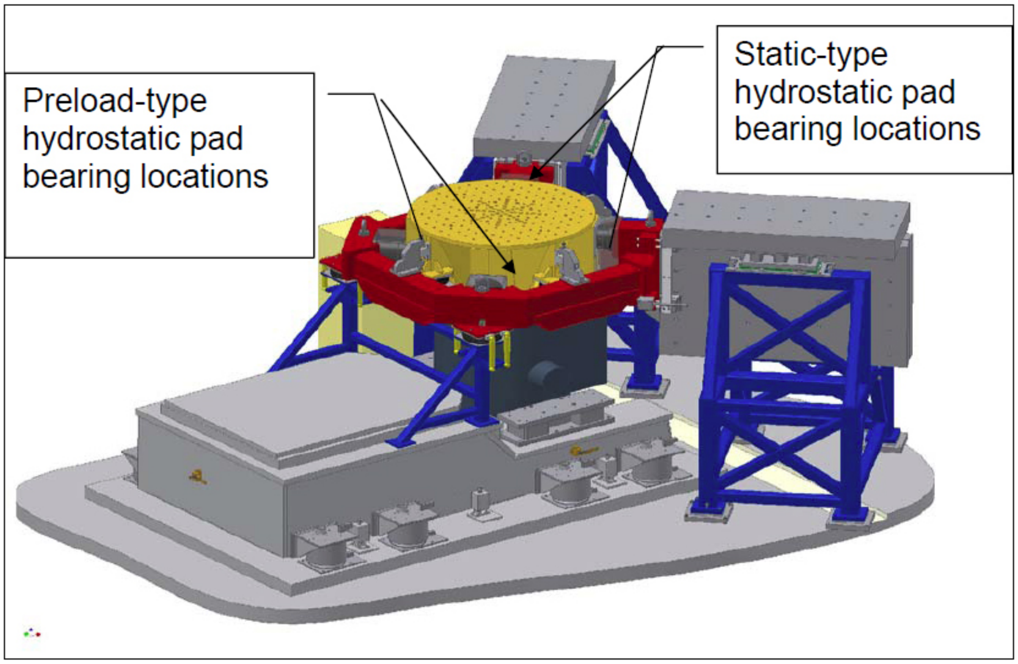

Figure 6. The design uses a 1.5-m (4.921-ft) diameter guided vertical head expander.

Final Design

The final solution consisted of two steel reaction mass structures supported on air spring isolators located directly behind two pairs of hydrostatic guide bearings. The load path from the guide bearings to the reaction mass structures is short, direct, and extremely stiff. The head expander and the reaction masses essentially become one continuous structure when the hydrostatic bearings are operating. High stiffness is one key to avoiding undesired mode frequencies within the test band.

The shaker armature is rigidly connected to the head expander. The shaker body is the reaction mass for the exciter. The compact symmetrical structure has no modes within the test band. The shaker body is not connected to the pad bearing reaction masses and is supported on air springs.

Each pair of hydrostatic guide bearings is positioned so that the bearings resist lateral (horizontal) translation motion of the head expander in one axis, and also resist rotation around one horizontal axis. The two sets of reaction masses and guide bearings together provide high dynamic stiffness that resists translation in any horizontal direction and resists rotations around any horizontal axis. The entire vertical vibration system—consisting of reaction masses, shaker, and guided head expander—is supported on air isolators. Supporting the functional items on isolators reduces the chance that vibration modes of adjacent structures will be excited and possibly interfere with the test.

The hydrostatic guide bearings used in this application are noteworthy. Each bearing is a combination of a spherical hydrostatic bearing and a flat hydrostatic bearing. The spherical surface provides a valuable self-aligning capability for each bearing. The flat surface provides the linear sliding interface between the bearing and the mating flat surface located on the guided head expander. The oil film that is established between the spherical surfaces and the flat surfaces is rather thin, approximately 0.025 mm (.001 inch) thick. This thin film provides high compression stiffness, both statically and dynamically. It is important to note that this bearing design can carry only compressive loads. In this application, both compression loads and tension loads must be transmitted through the bearings that couple the head expander to the reaction masses. The ability to carry dynamic tension loads is created by establishing an initial static compressive “preload” across the four bearings that connect the reaction masses to the head expander. Once the preload is established, any dynamic tension loads created during the vibration test produce a reduction in the initial compressive load across the bearings. As long as the static bearing preload forces are greater than the external dynamic tension loads that must be transmitted by the bearings, the stiff connection between the reaction masses and the head expander is preserved. The preload force is created by using self-aligning hydrostatic pad bearings that incorporate a hydraulic piston. This hydraulic piston maintains a constant preload force and absorbs small changes in head expander dimensions that might be the result of small changes in temperature, without the risk of guide bearing overload or binding that can occur in other arrangements of guide bearings.

INSTALLATION AND TEST OF NEW GUIDED HEAD EXPANDER

Head expander installation concluded on schedule. The planned acceptance test program included testing with Orbital’s proof test “mass simulator.” Tests performed on the original head expander during the initial problem characterization were to be repeated with the new guided head expander, followed by a test program with a qualification (qual) structure from a current program (see Figure 7).

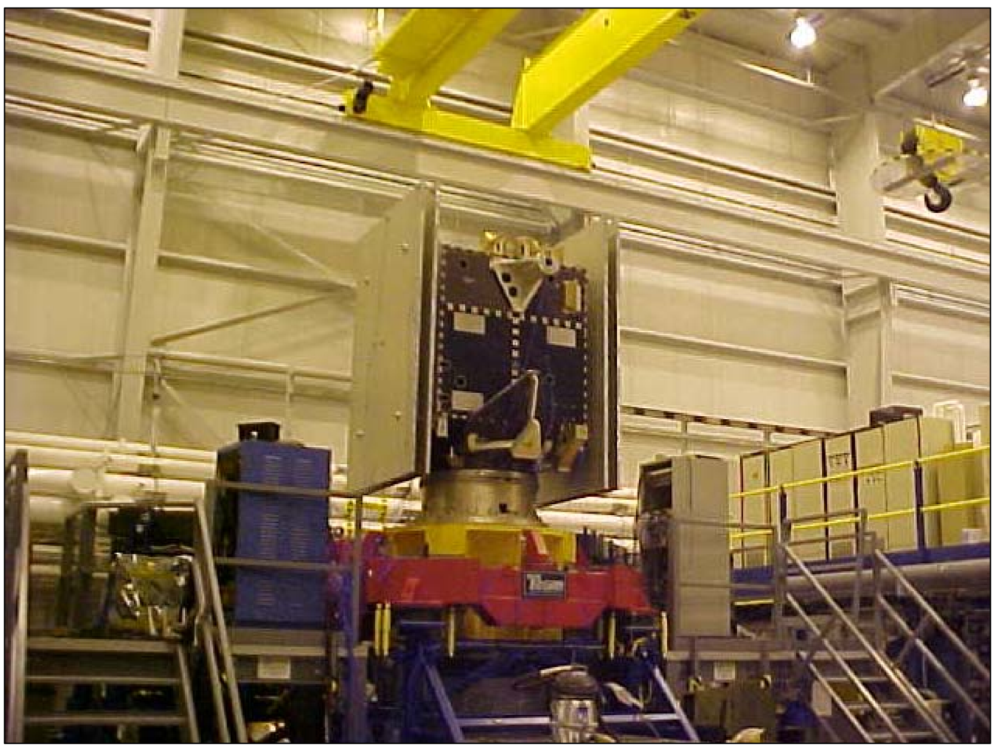

Figure 7. Science mission spacecraft qualification primary structure with mass simulators undergoing vertical vibration test on new guided head expander.

The requirements imposed on Team Corporation were the basis of the acceptance test program and formed the basis of the formal acceptance of the head expander. In addition, tests with a qualification structure and mass models were added. The test program would provide the confidence that the vibration facility was ready to test spacecraft beginning in January 2006 (on schedule).

Acceptance testing of the new equipment included installation, operation, and removal of the guided head. Orbital technicians were trained to perform the activities unassisted within the time required for the previous guided head. The requirement was for two technicians to perform installation or removal within 6 hours. Installation was accomplished initially within 4 hours and removal completed in 3 hours. Subsequently, those times were reduced to 3 hours for installation and 2 hours for removal.

Performance testing was against requirements of <10% cross-axis response for a bare head from 5 to 200 Hz and <15% with a 909-kg (2000 pound-mass (lbm)) load with a center of gravity at 864 mm (34 in.) high and a lateral offset of 38 mm (1.5 in.) (similar to the modal testing performed during the initial problem characterization). Excitation levels were from 0.1 *g* to 2.0 *g*.

The following charts show measured results comparing 1-*g* test inputs for two orientations of the proof mass with a lateral center of gravity offset. The lateral response was calculated just as it was for the spacecraft testing, using the three equally spaced accelerometers around the perimeter of the head measuring in-axis (vertical) acceleration. Significant reductions in cross-axis acceleration were achieved across a range of frequencies.

Offset CG along Shaker X axis		
Old Head	New Head	% reduction
56 Hz / 0.21g/g	55 Hz /0.12g/g	43%
72 / 0.22	75 / 0.03	86%
104 / 0.42	104 / 0.06	86%
125 / 0.15	158 / 0.15	
173 / 1.6	165 / 0.25	84%
183 / 2.4	184 / 0.50	79%
202 / 2.8	194 / 0.69	75%

Offset CG along Shaker Y axis		
Old Head	New Head	% reduction
57 Hz / 0.15g/g	58 Hz /0.05g/g	66%
73 / 0.25	75 / 0.06	76%
105 / 0.19	105 / 0.05	74%
135 / 0.15	147 / 0.15	
149 / 0.26		
	165 / 0.74	
175 / 1.2	178 / 1.1	8%
184 / 1.5		
197 / 1.6	195 / 1.0	37%

Requirements were met with the exception of a narrow frequency range centered at 160 Hz. An analysis identified the cause of the non-compliance as a manufacturing issue that was easily corrected on Orbital's guided head.

Acceptance testing was continued with a science mission spacecraft qualification (qual) structure. A high-fidelity modal test correlated finite element model produced test response predictions for the qual structure. The environments and instrumentation consistent with the original qual-structure test program performed at NASA Goddard Space Flight Center were used, as were inputs consistent with those used in the prior test program. The qual-structure had strong 60 to 90 Hz modes and weighed 1227 kg (2700 lbs). This was 270 kg (600 lbs) more than any prior Geo spacecraft test weight. The qual-structure was built upon the same structural core, and had a center of gravity consistent with the geosynchronous satellites. The test results showed a significant improvement over the prior test results.

Figure 8 shows the cross-axis motion of various spacecraft vertical vibration tests. In one case, the cross-axis acceleration is 2.5 times greater than the input vibration. The heavy line shows the results for a more demanding load being tested on the new guided head expander. The maximum cross-axis accelerations are significantly reduced.

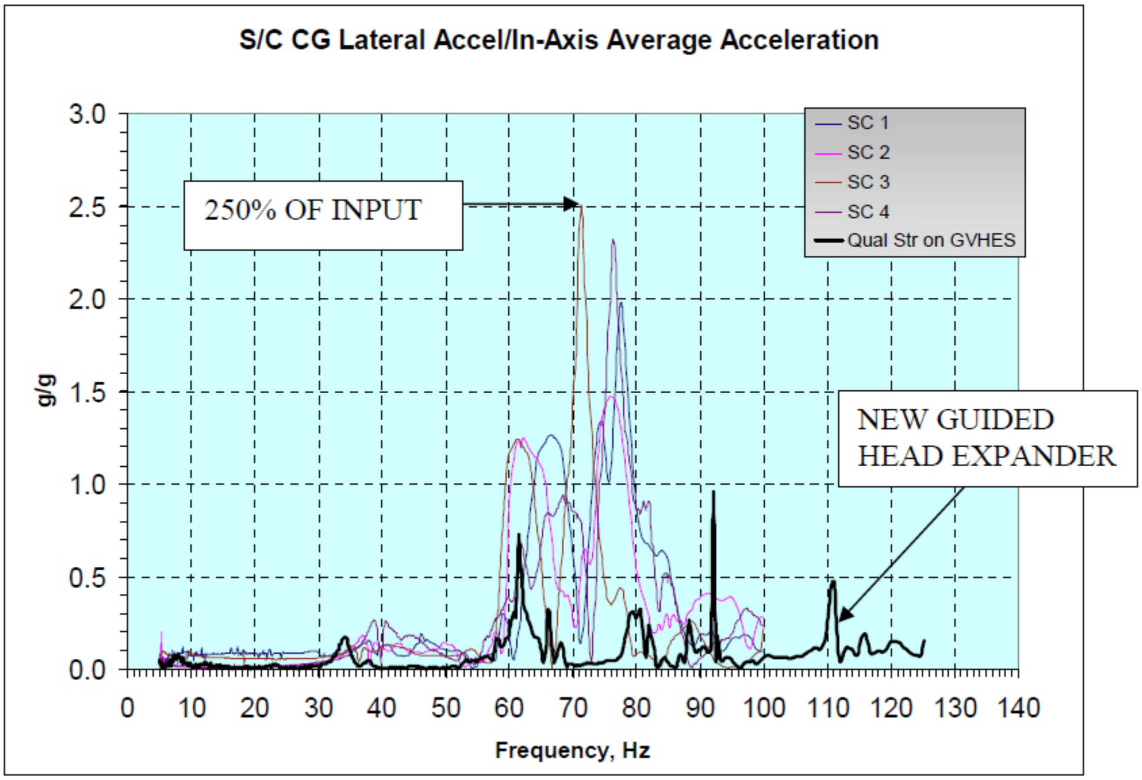

Figure 8. Cross-axis acceleration response/in-axis acceleration ratio for various spacecraft tested on the original head expander versus the cross-axis response of the heavier qualification structure (Qual Str) tested on the new guided vertical head expander system (GVHES).

The head expander was placed into service in November 2005. Figure 9 shows the first flight spacecraft test that was performed in January 2006 with results that surpassed expectations. The reduction in cross-axis rotations and the resulting cross-axis translations at various locations on the spacecraft were dramatic. Longitudinal in-axis test inputs now required far less notching during the test and easily met the coupled-loads predictions. The test inputs were nearly equal to the full (LV users' manual) specification. Such positive test results are not always achieved so easily.

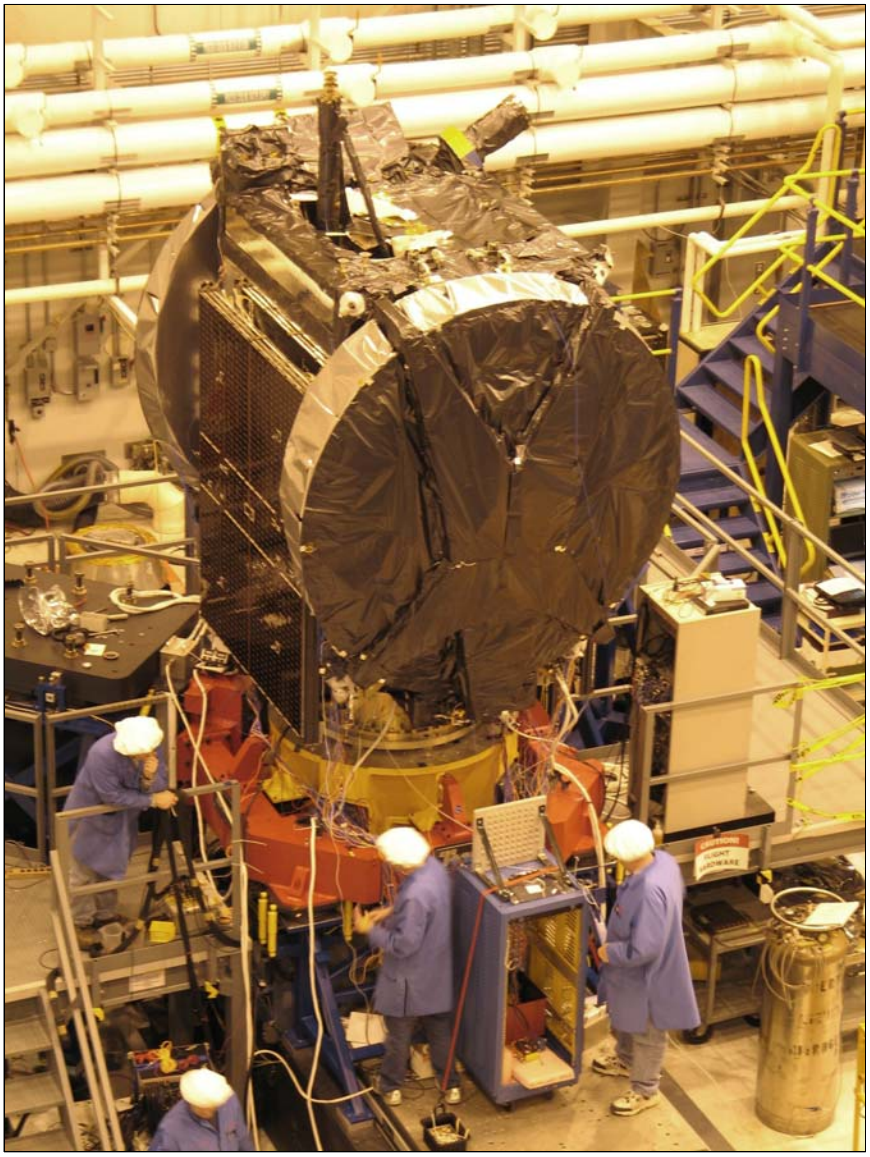

Figure 9. Orbital Sciences GeoComm spacecraft undergoing vertical-axis vibration test on new guided vertical head expander.

Figure 10 shows the cross-axis transfer function from test measurements. The results showed cross-axis responses reduced from 250% of the input level on prior spacecraft (of lesser mass) to levels that never exceeded 13% of the input level. Cross-axis accelerations had been reduced by a factor of 19.

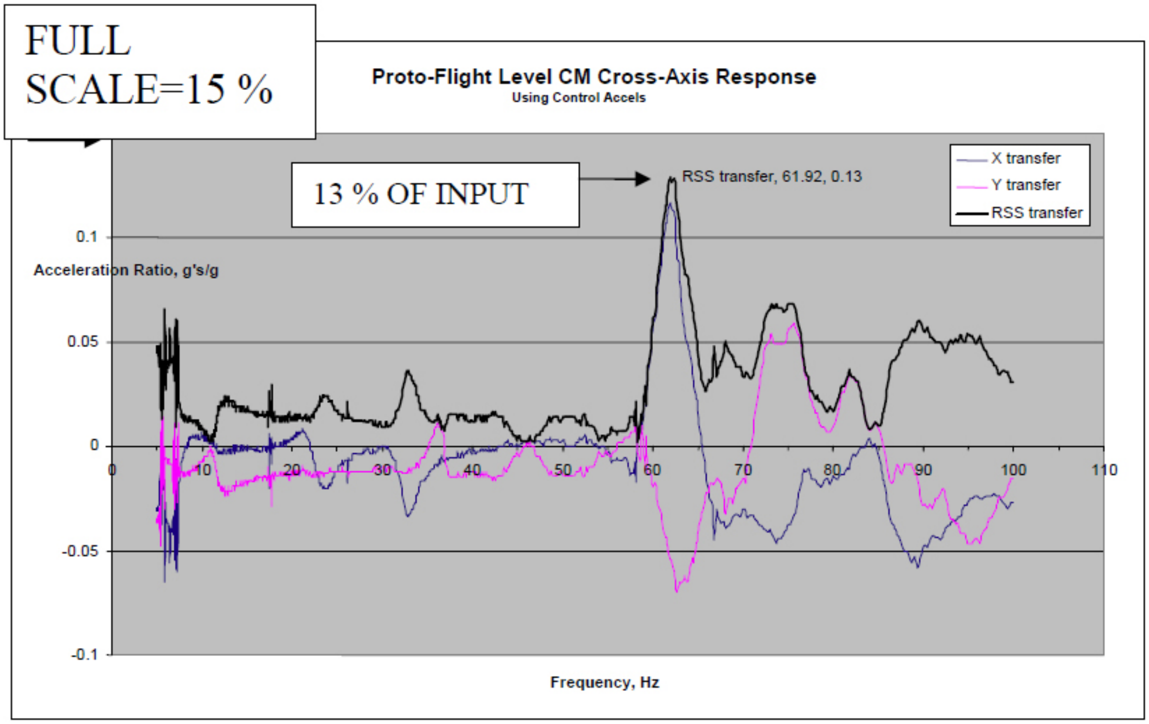

Figure 10. Root sum square (RSS) transfer function of cross-axis vibration vs. input vibration. Protoflight-level vibration inputs are 1.25 × the expected input levels generated by the LV.

CONCLUSION

If the existing test system had not been in place, alternate arrangements might have been possible. One alternate system might have been to construct a pit that would place the mounting surface of the head expander near floor level. This could have freed up some of the surrounding floor space if the isolated reaction masses were positioned below floor level with a floor cover-plate above. The pit could be constructed so as to support the air-isolated masses, eliminating the freestanding steel structures used in this installation.

Analysis of similar problems in other systems will probably follow the course of action used here. A finite element model of the system will usually be needed to understand the existing behavior and to evaluate possible improvements in the structure. If modifications to the existing system cannot achieve the desired results, then a new design will be necessary.

This new design for restraining cross-axis translations and rotations of vertical head expanders has proven to be a significant improvement over the previous equipment. The design simplifies the vibration test process and reduces the time required to complete the desired test program. This new design required considerable engineering effort to meet specific performance requirements and is a more complex assembly of hardware than the prior design. These factors add significantly to the procurement costs. However, the new system is comprised of not only a head expander and guide bearings, but also the reaction mass. Little facility work is required prior to installation, thereby minimizing laboratory downtime; no in-ground reaction mass is needed (eliminating facility construction costs); and the system can be moved in its entirety to a new facility as needed. These features will significantly influence any cost/benefit analysis beyond the operational advantages demonstrated by the new design.

ABOUT THE AUTHORS

Douglas Lund is the vice president of engineering at Team Corporation. He is a mechanical engineer with 38 years of experience. He earned a BSME from the University of Southern California in 1971 and an MBA from California State University, Fullerton in 1980. He has patented several concepts related to vibration testing applications.

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