N88-18955

Spacecraft Redesign to Reduce Microphonic Response of a VCO Component

J. C. Strain
S. Mittal

ABSTRACT

Reaction wheel vibration was found to induce out-of-specification sidebands on the carrier frequencies of some spacecraft components containing mechanical Voltage Control Oscillators (VCOs). Concurrent investigations were performed to redesign the VCOs to reduce their response to the wheel vibration, and to design a reaction wheel isolation system to reduce the vibration input to the affected components. Component level tests have indicated that both efforts produced viable solutions. The redesigned VCO will be incorporated into future spacecraft of the series, while affected spacecraft already in production will be retrofitted with the reaction wheel isolation system.

INTRODUCTION

This paper describes redesign efforts made to resolve a problem with a vibration sensitive Voltage Control Oscillator (VCO) assembly on the Defense Satellite Communications System III (DSCS III) Spacecraft. The mission of the DSCS III Satellite is to provide an enhanced military communications capability through the 1980s and 1990s. At anytime, four DSCS III satellites, each with a ten year mission life, operate in geosynchronous orbit to provide worldwide protected services to ground, sea, and airborne forces.

The DSCS III satellite is designed to be placed into a low inclination geosynchronous orbit on either the Titan 34D/IUS, Titan 34D/Transtage or Shuttle/IUS launch systems. The satellite can be configured for a dual launch with either a DSCS III satellite or another DSCS III in a tandem launch mode, as shown in Figure 1. Also shown in this figure is the satellite in its orbital configuration. On orbit stabilization is about all three axes with four Attitude Control System reaction wheels.

The DSCS spacecraft has several components which contain microphonic VCOs. During the production cycle of the program, it was found that in some vehicles the VCOs in two components on the North Equipment Panel may be sensitive to the vibration induced by the reaction wheels. Out of specification sidebands were noted on the carrier of affected components at multiples of the 400 Hz pulse which runs the wheels. Sidebands were greatest and out of specification at 1600 Hz. Significant sidebands were also noted at 3200 Hz.

The problem was to redesign the spacecraft to reduce the amplitude of sidebands caused by reaction wheel excitation with minimal impact on the cost and the production schedule. A two pronged approach was used for the solution. Structural changes were sought to reduce sensitivity to the 1600 Hz and 3200 Hz excitation by VCO component evaluations. Simultaneously, investigation was begun to reduce the excitation through reaction wheel isolation. This paper discusses the two solution approaches, results of the investigations and recommended solutions along with test results. The next section discusses VCO component evaluations and is followed by a section on the design for vibration isolation.

VCO COMPONENT EVALUATION

The VCO assembly is shown in Figure 2. The outer box consists of machined aluminum sides and a separator wall running through the middle of the box. The box has an aluminum top cover attached by means of five screws. The figure shows the box with the top cover partially removed to reveal details. The resonator rod is located in the lower right cavity. It is a hollow rod supported on two L-shaped rexolite brackets. The left bracket is glued to the bottom of the box whereas the middle bracket is attached to the outer side with two screws. The right end of the rod is screwed to the outer wall. The left end of the rod is attached to a strap spring. A probe extends from the middle wall in between the two brackets.

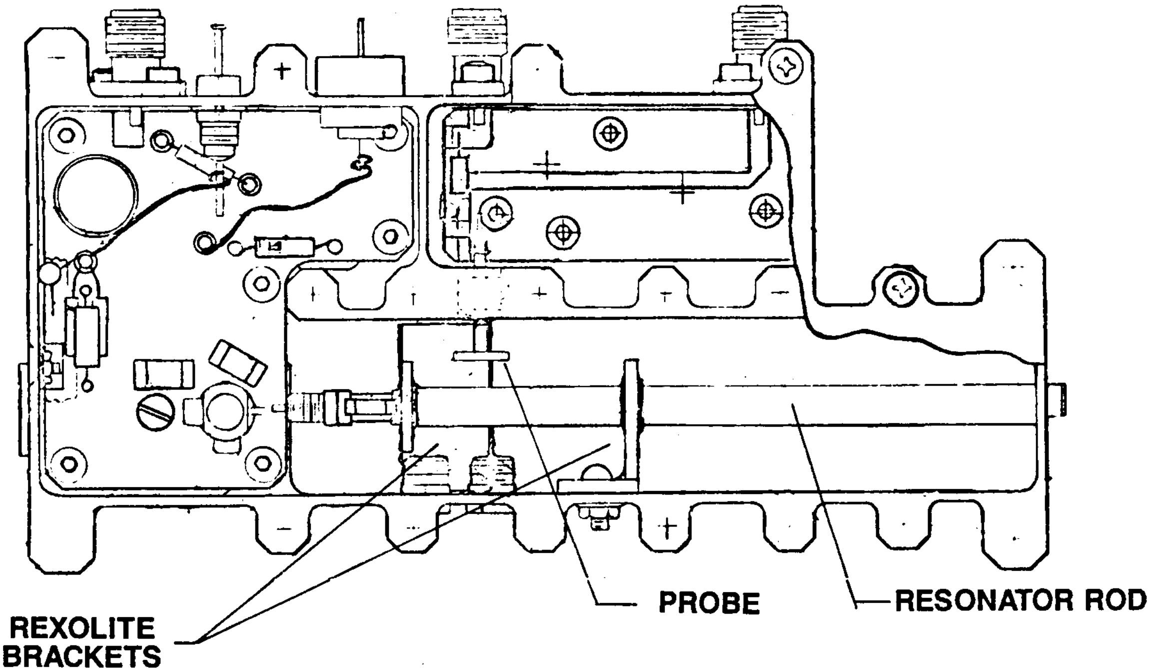

Figure 2. VCO Assembly (Top View)

The component evaluation was performed in three phases. In Phase I, dynamic analysis was performed on the VCO assembly and components to determine vibrational characteristics and recommend any solutions. The Phase II consisted of performing impedance hammer tap tests on the component hardware to validate the analytical model. Shaker tests of the component were conducted in Phase III of the evaluation process. This involved verification of analytical predictions and recommended design changes.

Dynamic Analysis of VCO Components

A detailed NASTRAN finite element model of the VCO assembly was made with the SUPERTAB/IDEAS system. The analysis was performed on the IBM 3090 system with the MSC/NASTRAN finite element program. The model is shown in Figure 3. Special care was taken to model the resonator rod stiffness and inertia distribution and its attachment to the VCO outer box. Rexolite brackets were modelled in detail to provide for variable fixity, to and offsets from the box center lines. The model consisted of 254 structural elements, 153 nodes and 175 dynamic degrees-of-freedom. A number of simpler component models were analyzed to trace the source of the 1600 Hz amplification. These included the probe model, the strap spring model, plate models to represent VCO outer box sides, and the resonator rod on its bracket model extracted from the overall assembly model. Of these component models, only the resonator rod model provided a clue to the sideband problem. It revealed that the rod resonant frequencies are very sensitive to the bracket fixity. The natural frequency varied from 1458 Hz to 2291 Hz for the first bending mode in the lateral plane and from 2036 Hz to 3289 Hz for the first bending mode in the vertical plane depending on the bracket attachment conditions. The rexolite brackets themselves, as originally designed, appeared to be very flexible.

The eigenvalue analysis performed on the VCO assembly showed that there are no natural frequencies below 1500 Hz. The first two modes calculated are shown in Figures 4 and 5, where the dotted lines represent the deflected shape superimposed on the undeformed configuration shown by the solid lines. The natural frequencies were found to be 1949 Hz and 2618 Hz for the two beam bending modes. This analysis was based on the assumption that the left bracket is glued

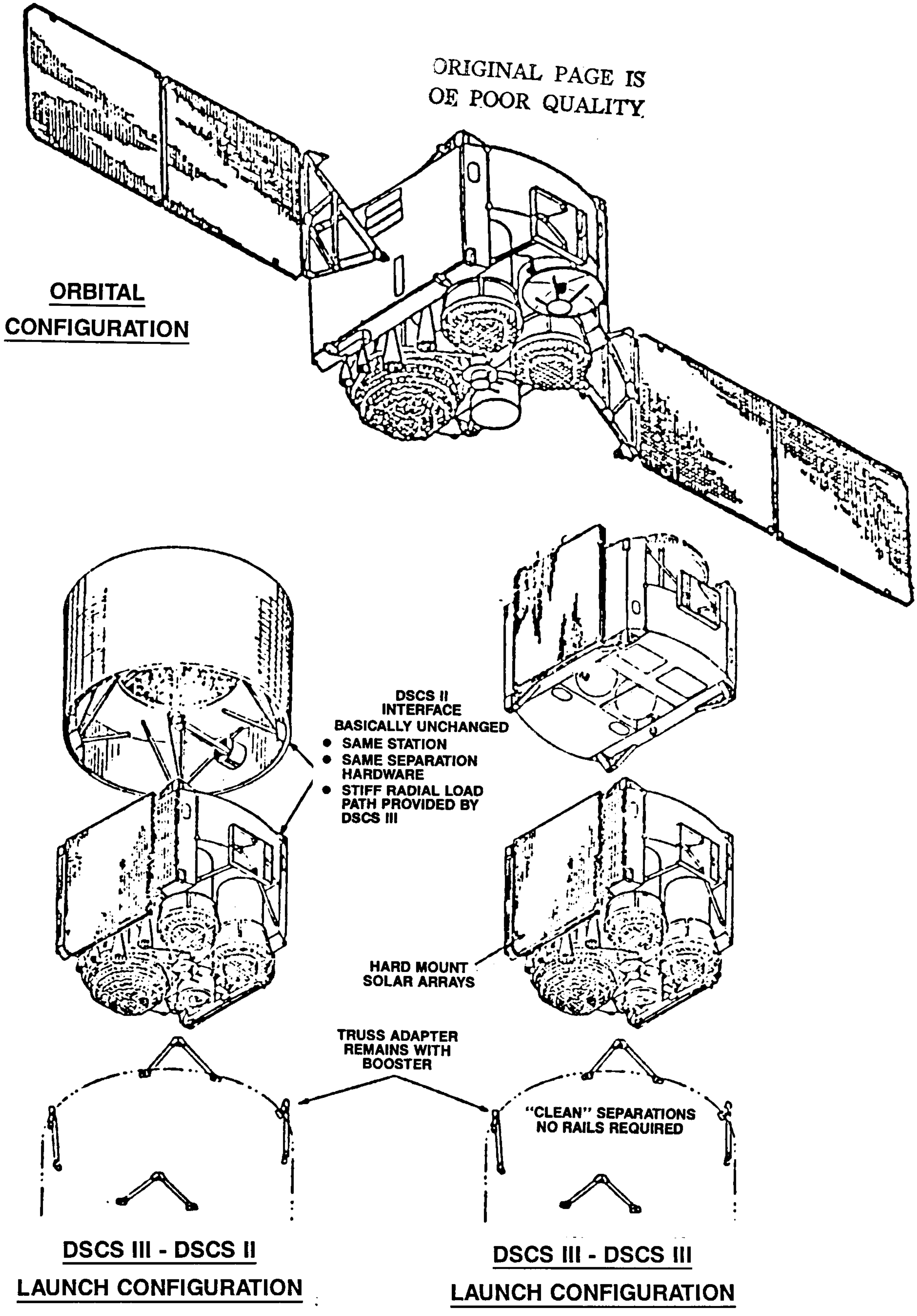

Figure 1. DSCS III Spacecraft Configurations

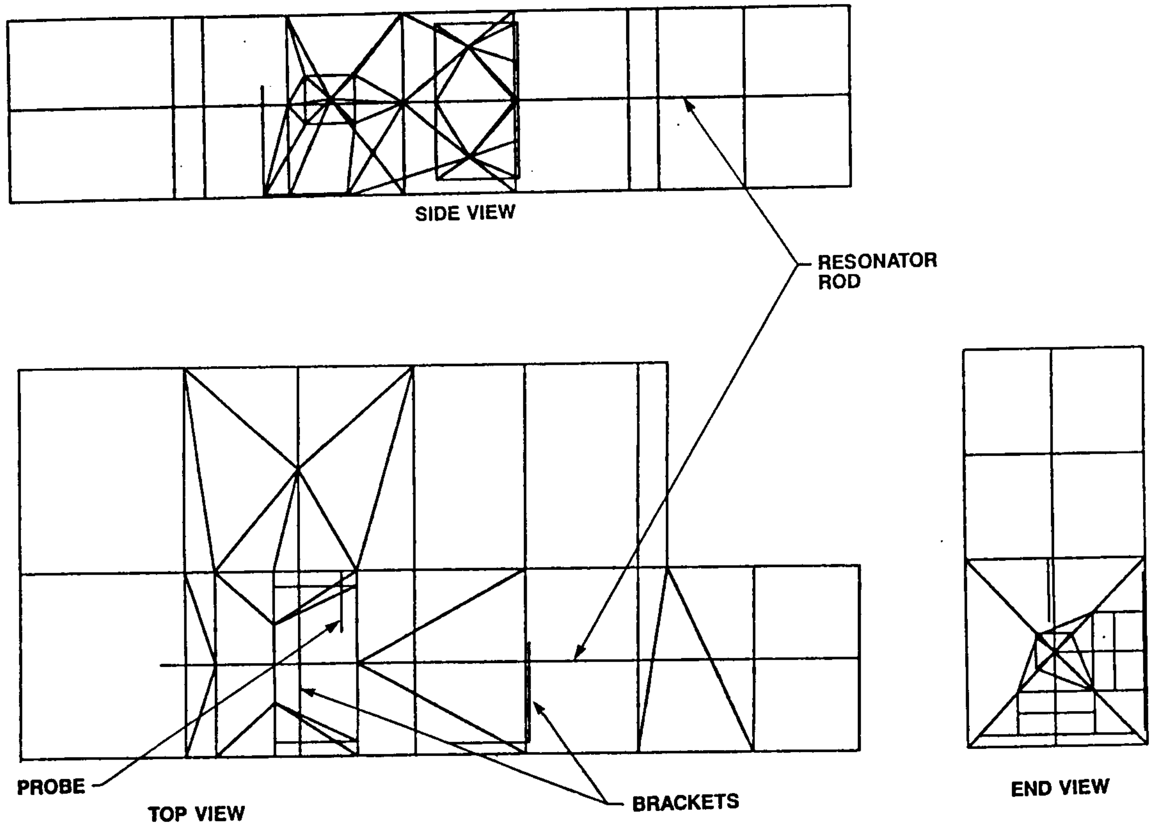

Figure 3. VCO Assembly Finite Element Model

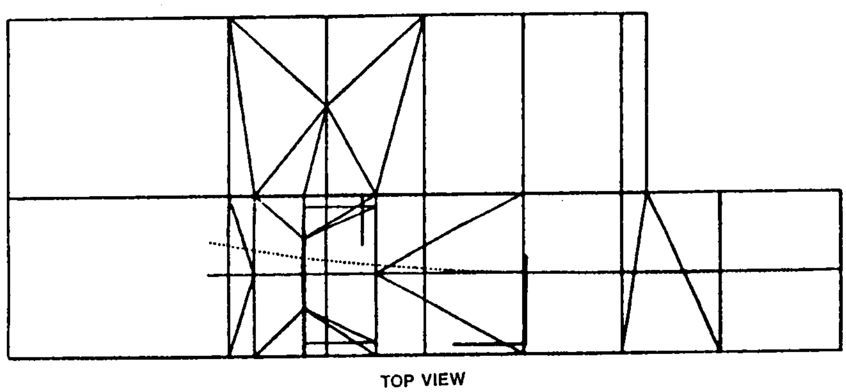

Figure 4. VCO First Mode (Beam-Lateral at 1949 Hz)

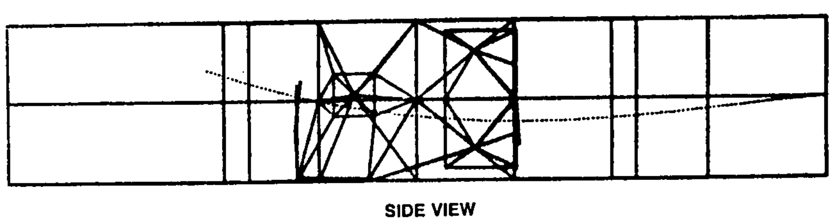

Figure 5. VCO Second Mode (Beam - Vertical at 2618 Hz)

to the bottom of the box. Inspection of the VCO hardware that had shown the sideband problem revealed that the bracket fixity was less than the perfectly glued condition. This fact coupled with the results of rod model analysis led to a redesign of the brackets shown in Figure 6. The reanalysis on the new configuration confirmed that the resonator rod modes in the frequency range of 1600 Hz and 3200 Hz have been eliminated. Figures 7 and 8 show the two lowest frequency modes for the redesigned configuration. These represent overall box modes which are not expected to cause any sideband problem.

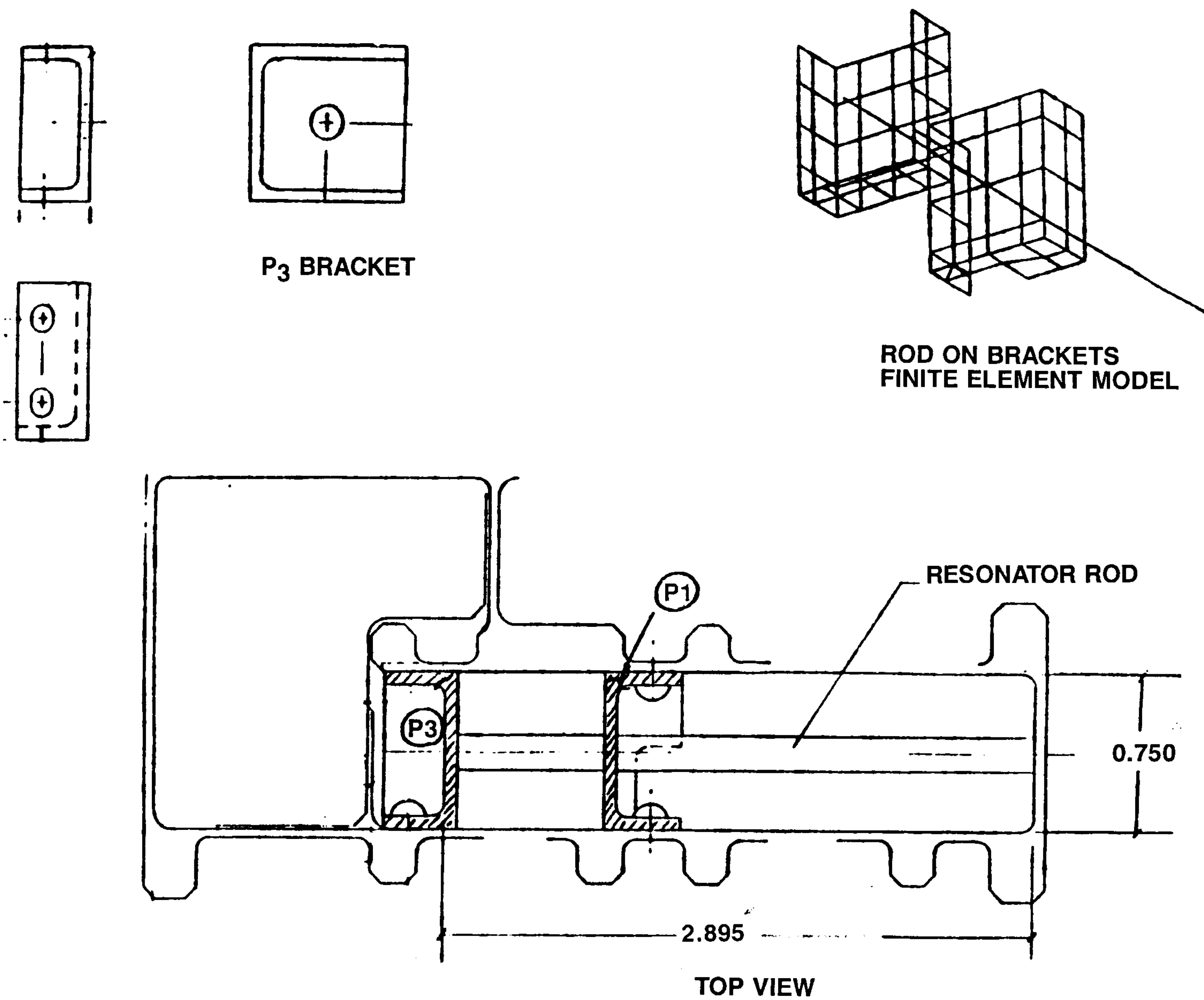

Figure 6. Redesigned VCO with New Brackets

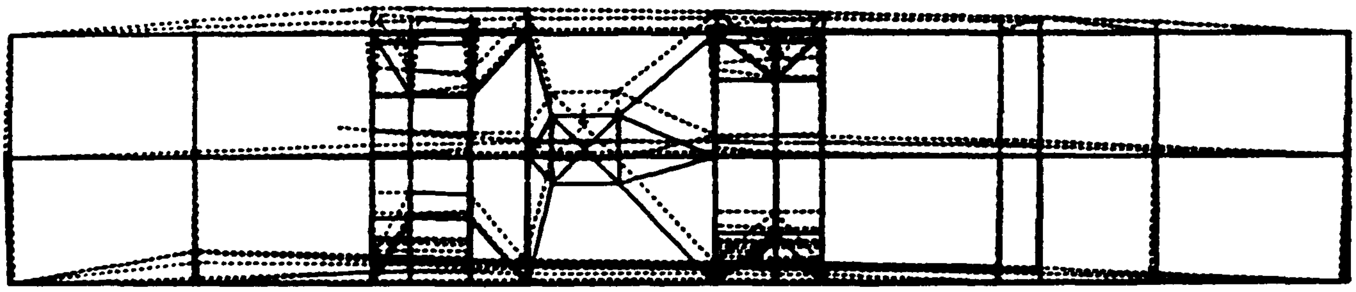

Figure 7. First Mode at 2606 Hz of Redesigned VCO

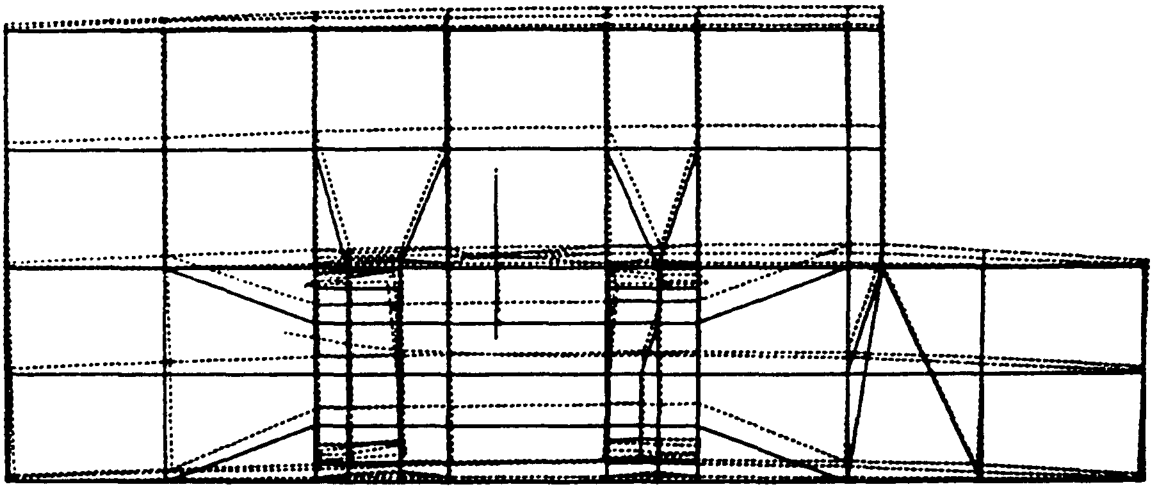

Figure 8. Second Mode at 3039 Hz of Redesigned VCO

VCO Tap Test Results and Correlation with Analysis

A series of tap tests were conducted on an engineering model unit of the VCO assembly to determine natural frequencies and to validate the analytical model. Two accelerometers were used on the resonator rod. Initial tests were conducted with the cover-off conditions to facilitate the accelerometer relocation and for ease of measurements. The analytical model was modified to match the test conditions and the accelerometer weights were included. The VCO assembly was suspended by means of three cords attached at screw mount locations to simulate free-free boundary conditions. The results of the tap tests are summarized in Table 1. Corresponding analytical results are shown in Table 2. Tests conducted with the cover-off conditions consistently showed natural frequencies at around 850 Hz and 1450 Hz which correspond well with the analytical results. Tests with the cover-on condition have measurement related problems and no conclusive data was recorded. However, good correlation with the analytical results for the first series of tap tests was deemed sufficient to validate the analytical model.

Table 1. Summary of Tap Test Results

Run #	Top Cover On/Off	Accelerometer Locations	Tap Location	Natural Frequency (Hz)
1	Off	A and B	rod	f1 = 850 f2 = 1475 f3 = 2300
2	Off	A and B	outside	f1 = 850 f2 = 1450 f3 = 2300
3	Off	A'and C	outside	f1 = 840 f2 = 1410 f3 = 2200
4	On	A and B	outside	f1 = 2650 f2 = 2850 f3 = N/A
5	On	A'and B	outside	f1 = 1950 f2 = 2500 f3 = 3000

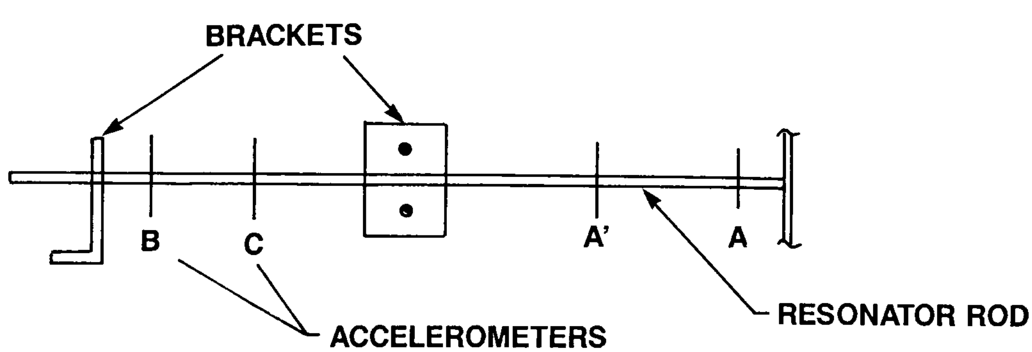

Table 2. Summary of Analytical Results

Run #	Top Cover On/Off	Bracket Design	Bracket Flexibility	Natural Frequency (Hz)	Mode Shape Description
1	Off	Original	Screws/ Screws	f1 = 866 f2 = 1422 f3 = 1760	Box Twisting Beam Lateral Box Side
2	On	Original	Screws/ Screws	f1 = 1412 f2 = 1964 f3 = 2852	Beam Lateral Beam Vertical Probe Bending
3	On	Original	Glue/ Screws	f1 = 1950 f2 = 2618 f3 = 2946	Beam Lateral Beam Vertical Probe Bending
4	On	New with Complete Bulkhead	Glue/ Screws	f1 = 2606 f2 = 3039 f3 = 3470	Box/Probe Box Twisting Box Bending

VCO Shaker Test Results and Analysis Verification

The VCO assembly was subjected to low level sine vibration tests to measure the sideband response. Tests were first performed with the original rexolite bracket configuration and then repeated with the resonator rod supported on the recommended brackets. Figures 9, 10 and 11 show the sideband amplitude to the carrier in the three axes as a function of frequency. The responses show significant reduction in sidebands when the recommended design change was implemented for the resonator rod support brackets. For example, in the Y-axis there was a 32 dB reduction at 1600 Hz and a 30 dB reduction at 3200 Hz. Similarly, reductions of 16 dB at 1600 Hz and 24 dB at 3200 Hz were measured in the Z-axis. Accounting for the variation in the fixity conditions between the analysis and the implemented hardware, this test provided a valuable verification of the analytical predictions and confirmed the validity of the recommended configuration.

ISOLATION EVALUATIONS

Tests were performed on a flight vehicle to determine which of the four reaction wheels were contributing significantly to the sideband response of the affected components. Each wheel was run separately and all four simultaneously while the sidebands were measured. One of the four wheels was found to be the major contributor. Exchanging this wheel with another determined that the wheel itself did not produce excessive vibration. Any wheel in this location would transmit enough excitation to the VCO to produce the out of specification sidebands. An isolation system was therefore designed to reduce the transmittance from this location at frequencies which could simulate a vibration response of the VCO resonator rod.

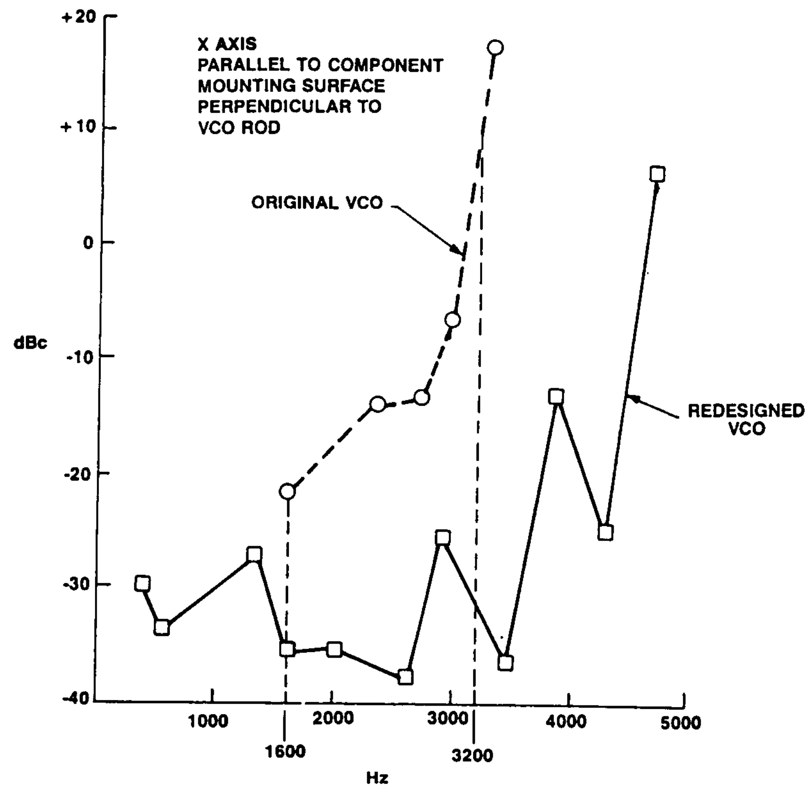

Figure 9. Component Vibration Response (Sideband to Carrier Ratio)

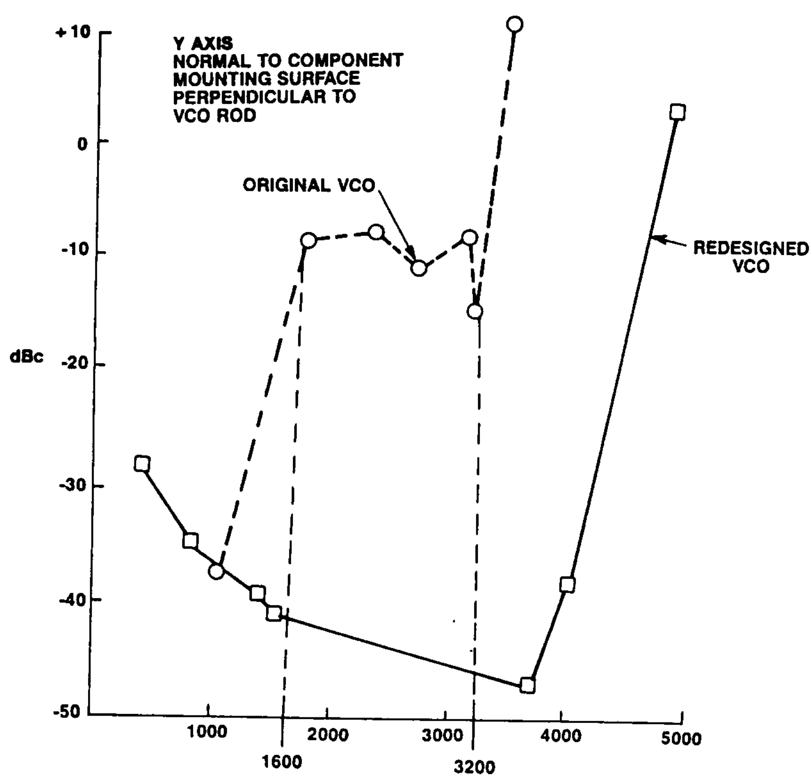

Figure 10. Component Vibration Response (Sideband to Carrier Ratio)

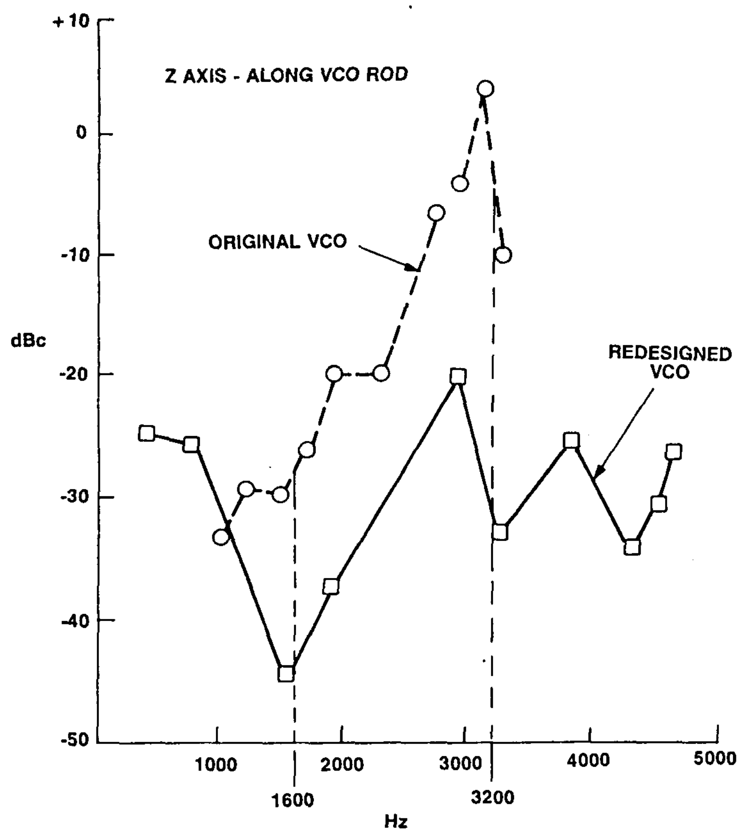

Figure 11. Component Vibration Response (Sideband to Carrier Ratio)

Isolation System Design

Isolation concepts were evaluated based on used of four isolators supporting a wheel to provide six degrees of freedom wheel isolation. Due to space limitations, no suitable commercial isolator could be found which could be put on the reaction wheel without some vehicle modification. It was therefore decided to design a damped stainless steel spring isolator which would fit into the available space, picking up existing bolt holes and requiring no modification to either the mounting surface or the reaction wheel harnessing. To use a commercial silicone isolator, it was determined to be necessary to invert the wheel, to change its direction of rotation, to reroute harness, and to use an interface adaptor plate. The steel spring mounts were preferred since they would cause a smaller impact to the interface and be lighter weight. Use of the silicone mounts which have an elastometer load path also required investigation to ensure adequate material properties for the space environment.

The preliminary requirement from spacecraft control system considerations was for an isolator frequency of approximately 100 Hz so as to provide no appreciable magnification below 1 Hz. The final design of a steel spring isolator meeting this requirement is shown in Figure 12. The flexible portion of the spring is .055 inch thick stainless, .5 inches wide. The fatigue requirement of design for four lifetimes is also met by this design. The damping treatment inside the spring was .25 inch thick GE SMRD 100F90B with a .030 inch thick stainless steel

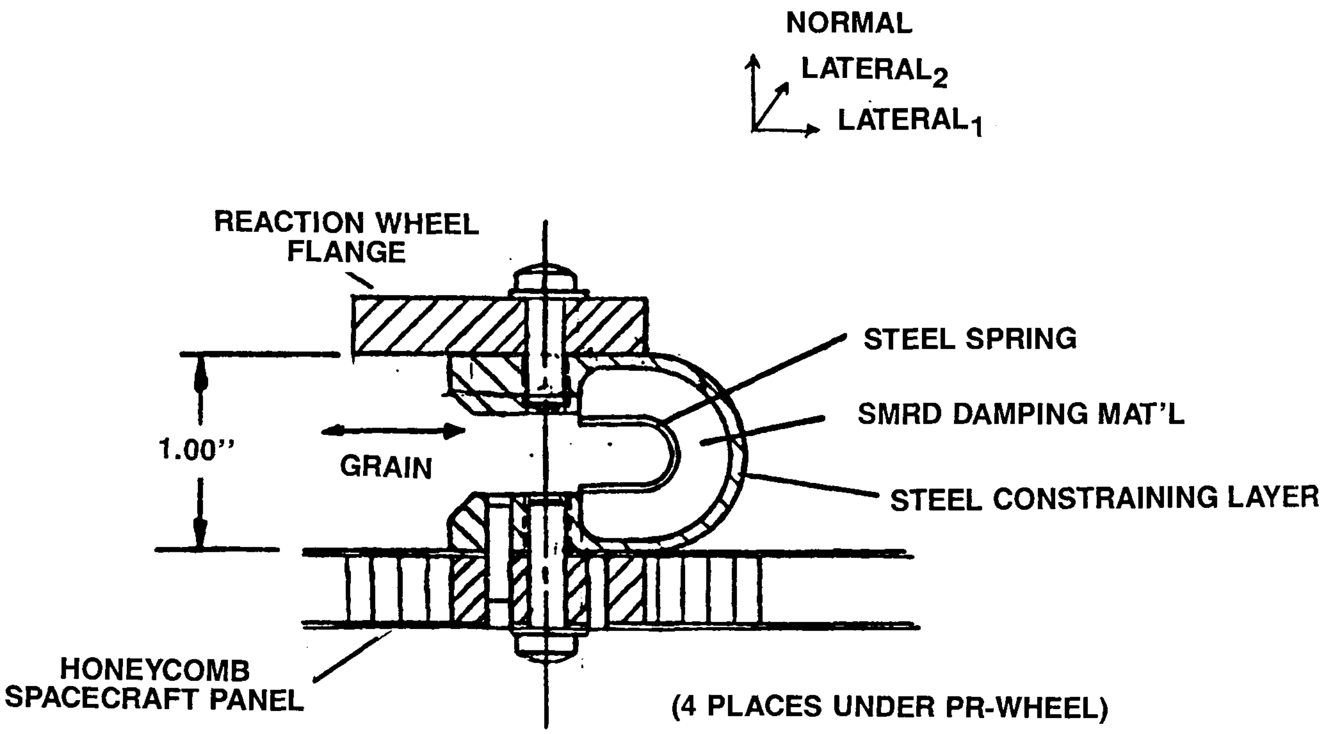

Figure 12. Isolation Mount Assembly Detail

constraining layer, providing a loss factor of approximately .3 (ζ = .15). This high damping will minimize dynamic loads at the isolator resonance while still providing good high frequency isolation. A base shake test performed on these isolators with an Engineering Model reaction wheel showed their fundamental frequencies to be 120 Hz laterally and 130 Hz axially with a measured Q of approximately 3 in both directions.

The chosen commercial silicone mounts have two attachment holes each. To avoid modification of the honeycomb structure to which the reaction wheels attach, a milled aluminum adaptor ring (Figure 13) was designed to provide an interface which would pick up the four existing mounting holes. A base shake test on these mounts showed the fundamental frequencies to be 105 Hz and 115 Hz in the lateral and axial directions respectively. Damping at resonance was adequate providing a Q of about 7.

Test Results

Both the steel spring isolators and the commercial mounts were tested mounted to a simple plate fixture which was readily available (Figure 14). With an Engineering Model Reaction Wheel providing the excitation, vibration was measured on the fixture and on the wheel mounting flange. These tests showed that the steel spring isolators did not work well on excitation normal to the plate at both 1600 Hz and 3200 Hz. Isolation was very good in the lateral directions. The silicone isolators were found to work well on 3200 Hz vibration in all three directions, but they did not provide good isolation of the 1600 Hz excitation.

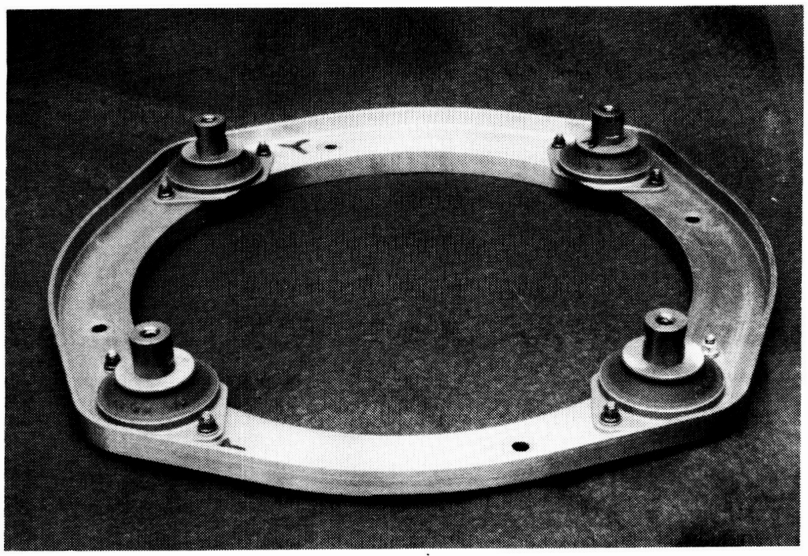

Figure 13. Silicone Isolators and Adaptor Ring

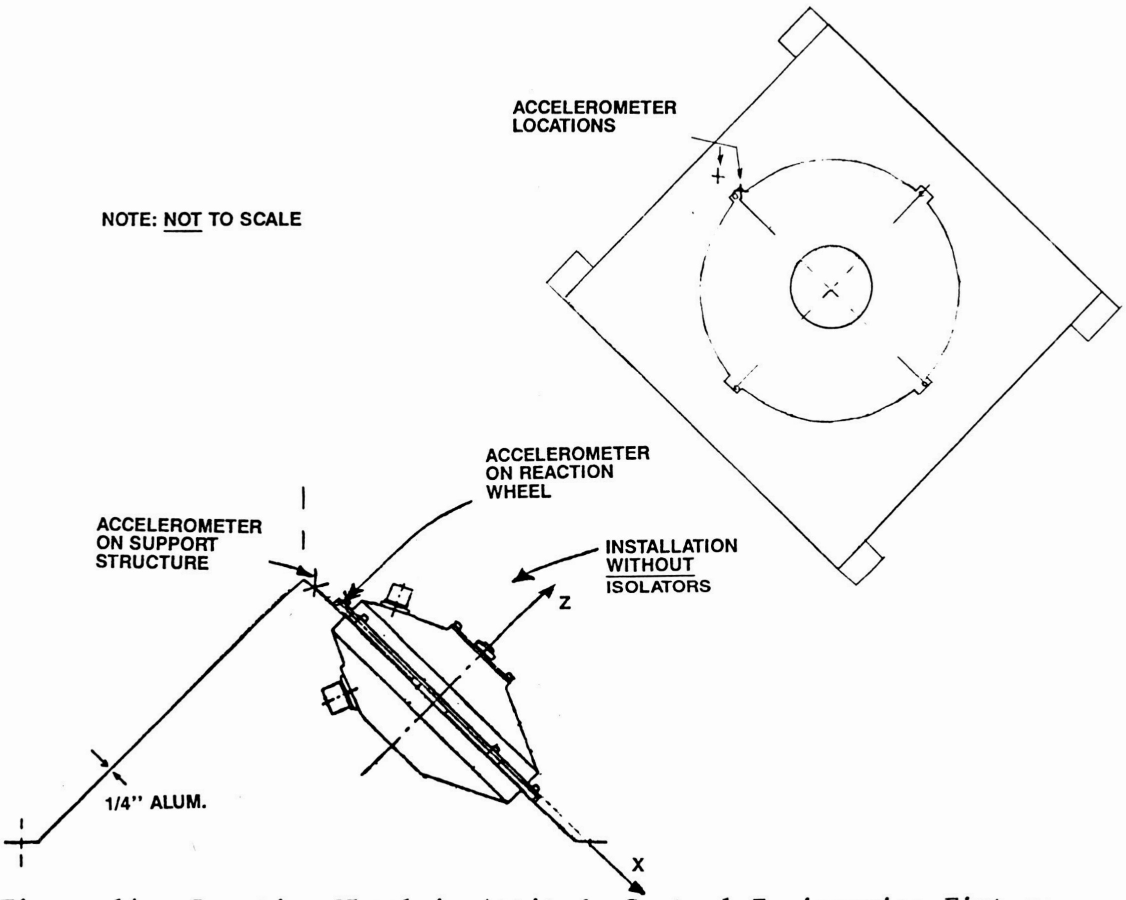

Figure 14. Reaction Wheel in Attitude Control Engineering Fixture

Since neither set of mounts was shown to be clearly superior to the other, and the extent to which the mounting structure of the spacecraft would change the results was unknown, further tests were performed on a prime spacecraft structure. A prime centerbody structure, without North or South panels attached, was made available for the further evaluations. Figure 15 shows this centerbody structure. To reduce the possibility of damage to this structure when the North and South equipment panels are not attached, honeycomb panels are attached in

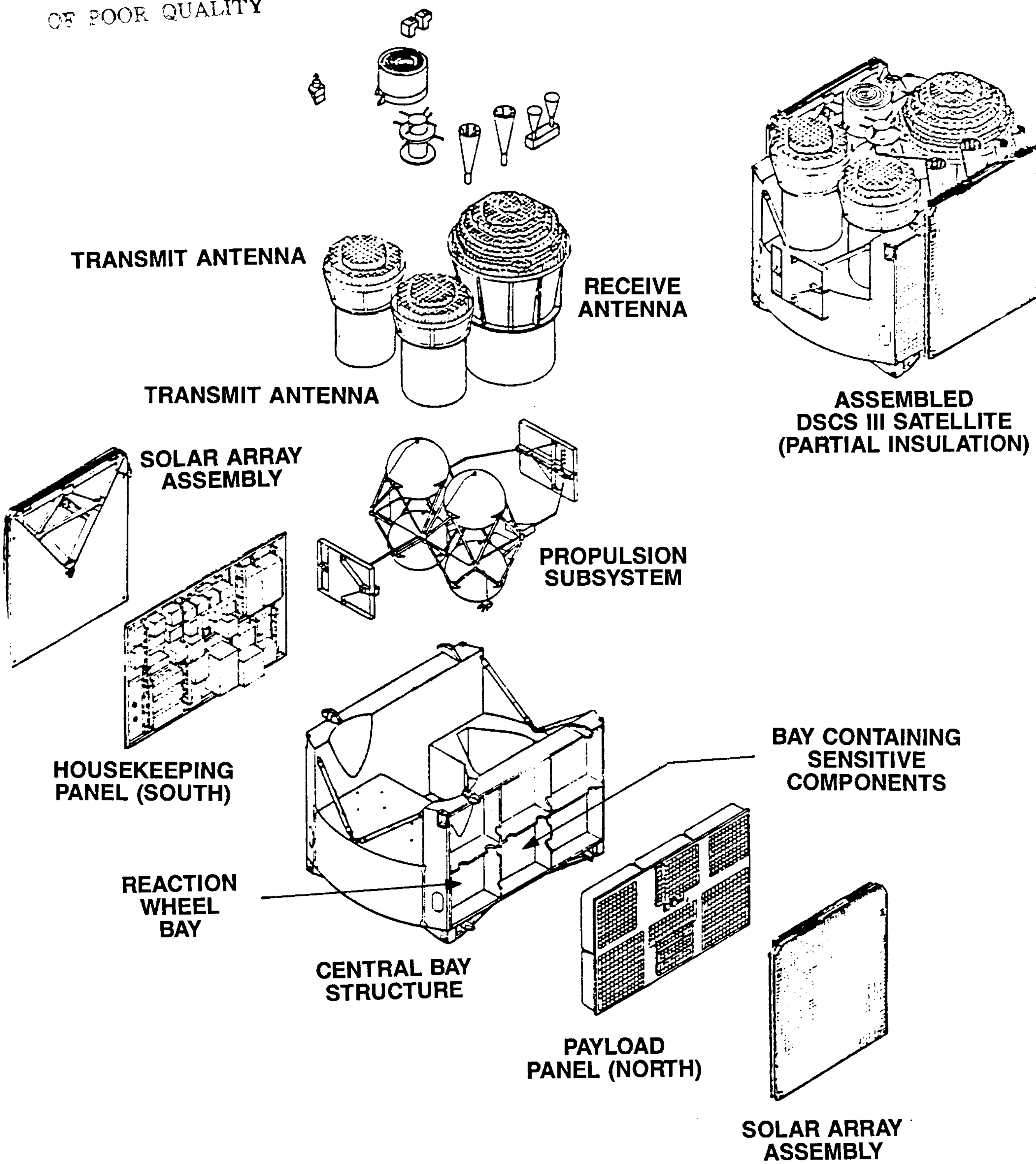

Figure 15. DSCS III Spacecraft Major Assemblies

their place to add stiffness to the structure and were in place during these tests. Four tests were run on the centerbody with an engineering reaction wheel providing the excitation in the location which showed the highest transmissibility to the components with sidebands. The three other wheel locations were also filled with engineering wheels to have their weight in place, but these other wheels were not run. The response was measured on a bulkhead adjacent to the mounting bolt for the running wheel. This was done for the running reaction wheel hard mounted to the spacecraft and for this wheel mounted through three sets of isolators: the steep spring mounts, the previously tested commercial silicone mounts, and a set of softer silicone mounts as it was determined from the control system analysis that these isolators, at about a 55 Hz frequency would be acceptable.

A sample from these tests is shown in Figure 16 which shows Z axis accelerometer response due to reaction wheel excitation for all four mounting conditions. Figure 17 summarizes the results of these tests in a bar graph for all three axes. The steel isolators are seen to have worked as well as the silicon mounts in two axes (X and Z) at 1600 Hz. For the Y axis the silicone mounts were clearly superior. Rather than spend additional time in cut and try to improve the Y axis performance of the steel mounts it was decided to proceed using the soft silicone isolators.

The silicon mounts were tested on a shaker to qualification levels in random vibration and inspected for damage. They passed the qualification test and questions as to their suitability for use in space were resolved satisfactorily.

CONCLUSIONS

The resonator rod supported on the existing brackets was found to be a potential cause for the sideband problem observed on some of the spacecraft assemblies. The two pronged approach of modifying the VCO rod to reduce its response to wheel excitation and isolation of the reaction wheel to reduced the level of the disturbance proved to be very successful.

The beam resonant frequencies were found to be very sensitive to the bracket stiffness and fixity. A timely solution was recommended with a revised bracket design to eliminate the resonance problem from mechanical vibration.

The reaction wheel isolation design was found to be practical from both structural dynamic and control system analyses. Retrofit of this design change onto already built spacecraft with out-of-specification sidebands has been planned. The implementation will be via an adaptor plate between the isolators and the spacecraft so that no rework will be required on the spacecraft except for a harness modification.

Both solutions must still be tested on a fully assembled spacecraft (the real North and South panels may change the degree of isolation) and measurements must be performed on the affected components to determine the extent to which the sidebands have been reduced. Sideband reduction on a fully assembled spacecraft will be the true indication of the success of this investigation.

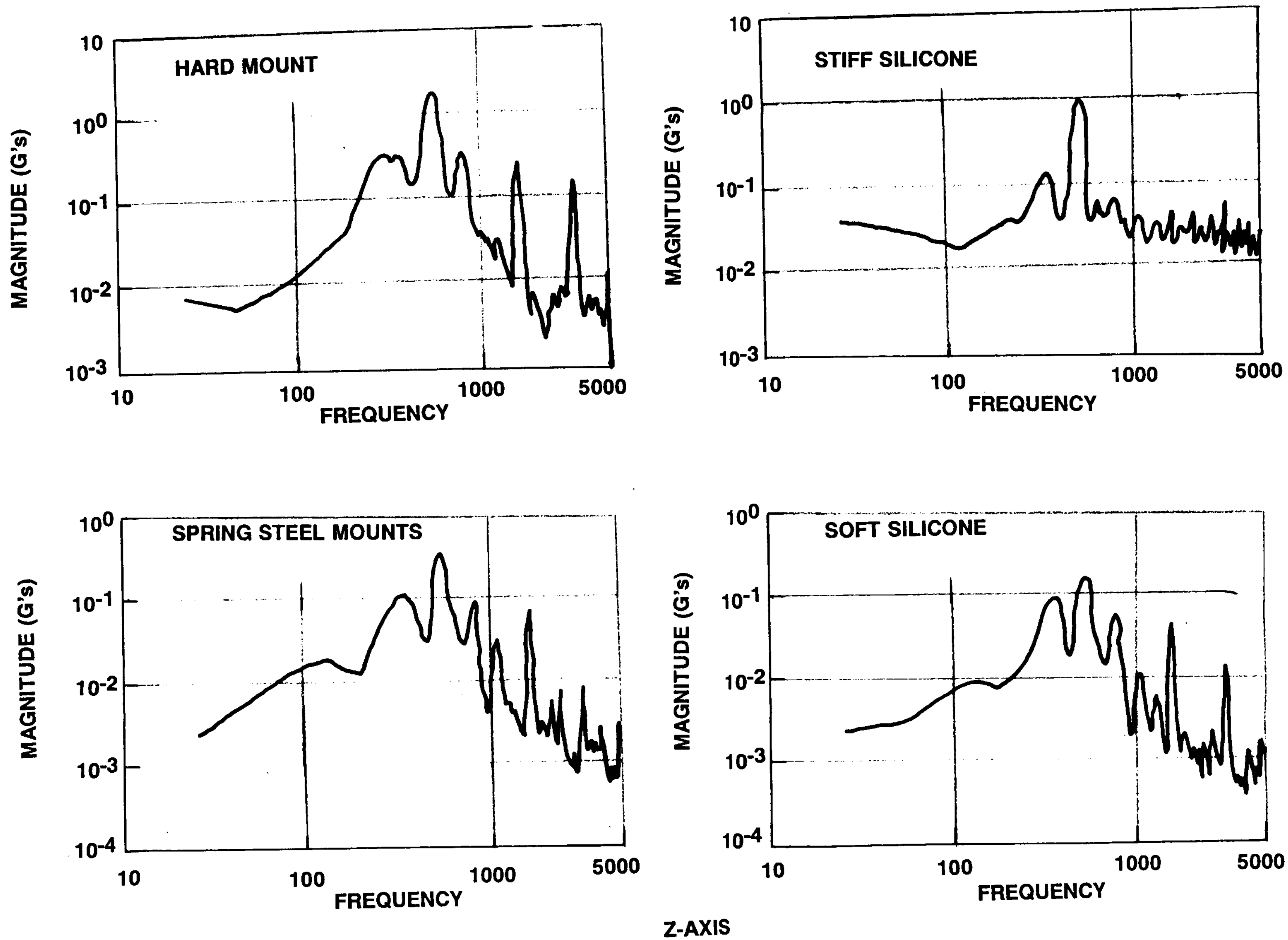

Figure 16. Centerbody Panel Response to PR - Wheel Spacecraft Axes (Fig. 3-6)

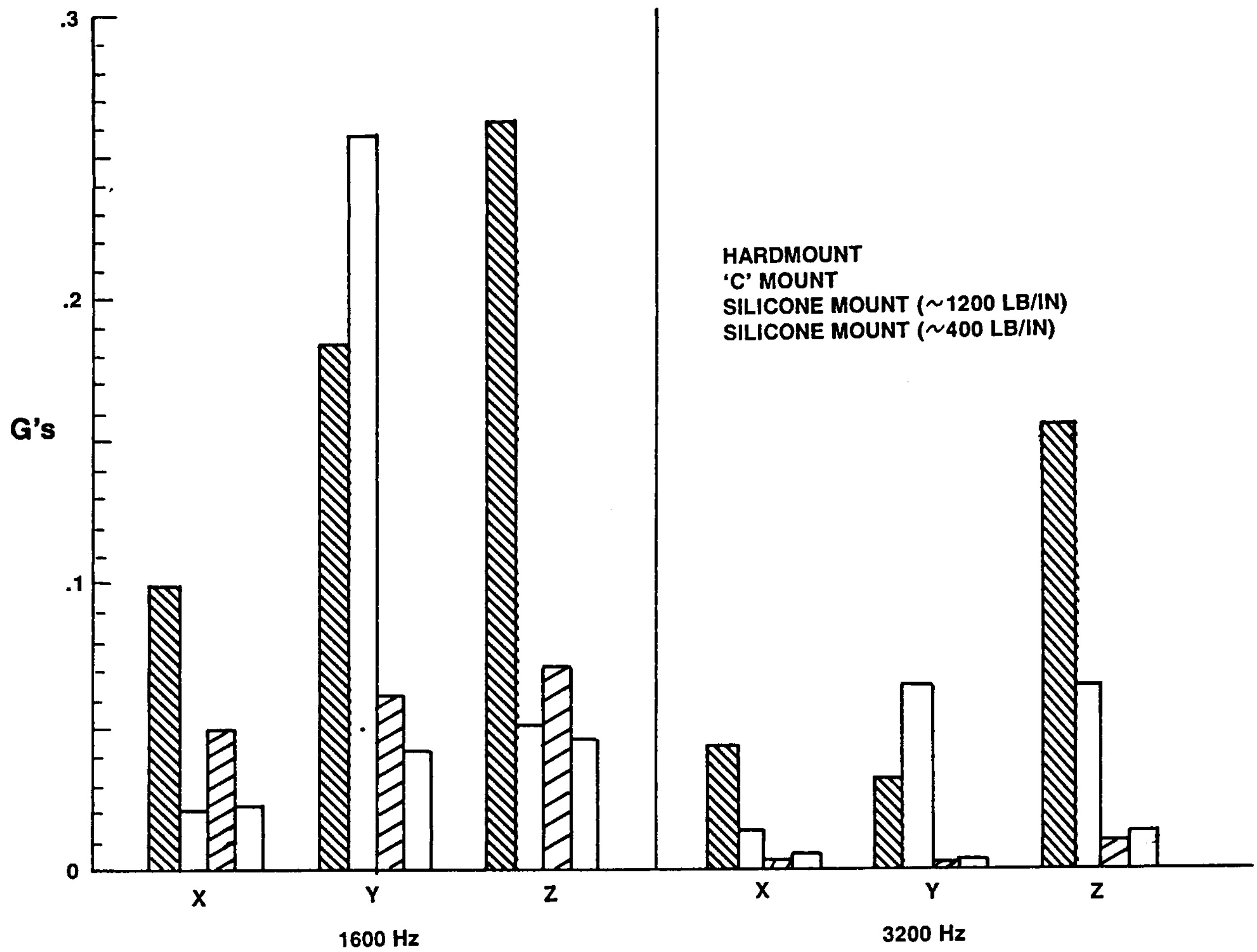

Figure 17. Test Results - Isolators on Centerbody Structure