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Evaluation of a Double Gimbal IPACS Design

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1.0 Introduction

Integrated Power/Attitude Control Systems, (IPACS), were investigated in the early 1970's to determine if the dual functions of electrical power storage and spacecraft attitude control could be efficiently integrated into a single package. An IPACS inner gimbal assembly (IGA) was designed and built by Rockwell International and described in NASA CR-172317 (reference 1). A gimbal sizing analysis was performed on this IPACS by Allied Bendix Aerospace and documented in NASA CR-172524 (reference 2). This effort reviewed the IPACS IGA design and produced a preliminary design for the gimballing assembly.

The scope of the task performed by the Allied Bendix Aerospace Corporation and described herein progresses from the results of reference 2. A parametric study analyzing the suitability of the IPACS rotor material was completed. This study investigated three materials: 6Al-4V titanium (current IPACS rotor material), B120 VCA titanium, and Custom 455 stainless steel. The preliminary linear vibration response analysis was updated to include the stiffnesses and the weights of the gimbals designed in Phase I (reference 2). Finally, a belleville washer spring preload mechanism was designed to replace the existing helical spring and interference fit preload mechanism.

2.0 Rotor Material Trade Study

2.1 Rotor Material Qualifications

A trade study has been performed on three different metals to investigate their suitability as the IPACS rotor material. Four main criteria were applied to initially restrict the material choices. Primarily, the candidate material had to have a very high strength to density ratio, excellent toughness, high resistance to stress corrosion, and possess an extensive history of use for life prediction security.

These criteria are of critical importance due to the unique demands placed upon the rotor of an IPACS system. A high strength to weight density ratio is crucial to the maximization of the power storage capability of the IPACS. However, strength without toughness is unacceptable for a high speed flywheel and even more so in an IPACS configuration. The rotor must live thru thousands of energy storage/discharge cycles each of which applies a stress whose magnitude is a considerable fraction of the yield strength of the material. Simultaneously, the rotor must endure the fatigue cycling that occurs when the rotor is torqued about a gimbal axis. The alternating stress in this case is rather small, but it is reacted at the rotor spin rate of up to 450 cycles per second (see reference 2 for the analytical rational for this maximum speed). Therefore, the material must have high toughness against both low cycle crack growth and high cycle fatigue failure. Stress corrosion is an incipient danger that has been identified in many high strength alloys being used in the aerospace industry. Because of the continuously high levels of stress present in energy storage and control moment gyro (CMG) rotors, a very high level of stress corrosion resistance must be maintained. Reference 3 is the standard document for stress corrosion suitability. Finally, the importance of an extensive database of mechanical and physical properties should not be underrated. The hardware development of any concept must be based upon a solid foundation of analysis and experience lest the construction of a prototype degrade into a very expensive form of destructive testing.

2.2 Candidate Materials

Most high strength metals are quite brittle and have low fracture and fatigue tolerances. The requirement for high toughness eliminated a considerable bulk of the ultra high strength alloys currently available. Lack of availability and extremely high cost eliminated the cobalt based superalloys. Maraging steels are unacceptable due to their tendency towards stress corrosion. Plastic composites, metal matrix alloys and ceramics were eliminated at this time due to the lack of substantial property databases which are necessary to insure the performance of this highly stressed component.

Throughout this process of elimination, three alloys stood out and were chosen as candidate materials for the trade study. These alloys were: 6Al-4V-Titanium alloy, B120 VCA Titanium alloy, and Custom 455 Stainless Steel alloy. All of these alloys have very good strength to density ratios while retaining excellent fracture toughness and fatigue strength. They are all very resistant to stress corrosion and have extensive backgrounds of testing data and actual hardware implementation. Bendix has been fabricating high speed rotors from Custom 455 Stainless Steel since 1979. Important mechanical and physical properties of the three alloys are presented in Table I (see references 4 and 5).

2.3 Parametric Rotor Study

The parametric rotor study that was performed on the three candidate materials attempted to determine which would yield the highest IPACS power to weight density. The basic configuration of the original IPACS rotor was retained and modification of the design was limited to scaling all of the coordinate axes simultaneously by the same amount. In this way, the basic rotor shape was preserved and, depending upon the material, the rotor either shrank or grew but kept the same proportions. References 1 and 2 contain more indepth information on the IPACS rotor configuration, and the system in general. It was necessary to modify the size of the wheel so that the highest possible rotor speed would develop the same amount of energy as the original IPACS rotor. By way of explanation, a Custom 455 stainless steel IPACS rotor can be made much smaller than a titanium wheel of the same energy storage capacity because of the steel's higher density.

The optimum rotor size and speed for each of the materials was found through an iterative procedure. An estimate was made as to the required size of the rotor, depending on the strength of the material and its density. The size of the rotor and the strength and density of the material determined the centrifugal stress in the rotor and allowed a

Table I. Mechanical and Physical Properties of the Three Candidate IPACS Rotor Materials

Material	Custom 455 Stainless Steel HT 1000	6Al-4V-Titanium	B120 VCA Titanium
Modulus of Elasticity	20.0×10^{4} MPa 29.0×10^{6} psi	11.0×10^{4} MPa 16.0×10^{6} psi	10.0×10^{4} MPa 14.5×10^{6} psi
Poisson's Ratio	0.30	0.31	0.31
Density	7860 kg/m^3 0.284 lbs/in^3	4430 kg/m^3 0.160 lbs/in^3	4820 kg/m^3 0.174 lbs/in^3
Elongation	10%	10%	8-10%
Thermal Expansion	10.6×10^{-6} cm/cm/^{o}K 5.9×10^{-6} in/in/^{o}F	8.8×10^{-6} cm/cm/^{o}K 4.9×10^{-6} in/in/^{o}F	9.2×10^{-6} cm/cm/^{o}K 5.1×10^{-6} in/in/^{o}F
Ultimate Strength (293^{o}K) (68^{o}F)	1380 MPa 200 ksi	895 MPa 130 ksi	860 MPa 125 ksi
(344^{o}K) (160^{o}F)	1325 MPa 192 ksi	835 MPa 121 ksi	785 MPa 114 ksi
Yield Strength (293^{o}K) (68^{o}F)	1310 MPa 190 ksi	825 MPa 120 ksi	825 Mpa 120 ksi
(344^{o}K) (160^{o}F)	1255 MPa 182 ksi	750 MPa 109 ksi	750 MPa 109 ksi
Fracture Toughness	110 $MPa*m^{0.5}$ 100 $ksi*in^{0.5}$	77 $MPa*m^{0.5}$ 70 $ksi*in^{0.5}$	66 $MPa*m^{0.5}$ 60 $ksi*in^{0.5}$

prediction of the maximum operating speed of the rotor. The maximum speed prediction and the moment of inertia of the rotor were used to calculate the energy storage capability of that particular design. Depending upon the outcome, the proposed rotor size was altered and the process repeated to iterate the design to accomplish the task required. In this way, it was guaranteed that each rotor would achieve the same energy storage capability at its maximum operational speed.

2.4 Rotor Material Trade Study Results

A tabular presentation of how the three rotors compare to each other can be found in Table II. The active constraint on the rotor designs was that the kinetic energy of each rotor at its maximum operating speed would equal 1140 watt*hours. This value was chosen because it is the amount of energy stored in the original IPACS rotor at its maximum operational speed. The maximum operational speed of a rotor is defined here as that speed at which the centrifugal stress in the rotor equals the yield strength of the material at 344° K (160° F).

The 6Al 4V titanium rotor has the greatest energy storage to weight density, achieving 20.3 watt*hours per kilogram (9.19 watt*hours per pound) of material. While the energy storage to weight densities of the titanium rotor is superior to the stainless steel rotor, the stainless steel rotor is significantly smaller in size. It is this size disparity that will allow the steel rotor to achieve higher overall system power to weight and power to volume densities.

The Custom 455 rotor is 16% smaller than the original IPACS 6Al 4V titanium rotor and is 5% heavier. But the reduction in rotor size brings with it a corresponding reduction in the size and weight of the inner gimbal, the outer gimbal and the mounting ring. The gimbal pivots and torquers are sized for stiffness and will not significantly change due to this package size reduction. A 15% reduction in required gimbal diameter will allow approximately a 25% reduction in gimbal volume while maintaining the same structural stiffnesses. When this factor is applied to the gimballing structure as described in Section 3.0 and in references 1 and 2, then the weight reduction for the gimbals is:

Table II. Results of a Parametric IPACS Rotor Sizing

Material	Custom 455 Stainless Steel HT 1000	6Al-4V-Titanium	B120 VCA Titanium
Rotor Radius	19.07 cm 7.51 in	22.71 cm 8.94 in	22.71 cm 8.94 in
Rotor Height	35.56 cm 14.00 in	42.29 cm 16.65 in	42.29 cm 16.65 in
Rotor Weight	59.0 kg 130 lbs	56.2 kg 124 lbs	61.2 kg 135 lbs
Rotor Inertia	0.586 N*m*sec^2 0.432 Ft*lbs*sec^2	0.789 N*m*sec^2 0.582 Ft*lbs*sec^2	0.858 N*m*sec^2 0.633 Ft*lbs*sec^2
Maximum Speed	35700 rpm	30800 rpm	29500 rpm
Kinetic Energy @ Maximum Speed	1140 Watt*hrs	1140 Watt*hrs	1140 Watt*hrs
Energy Storage to Weight Density	19.3 Watt*hrs/kg 8.77 Watt*hrs/lbs	20.3 Watt*hrs/kg 9.19 Watt*hrs/lbs	18.6 Watt*hrs/kg 8.44 Watt*hrs/lbs
Required Output[1] Torque	27 N*m 20 Ft*lbs	27 N*m 20 Ft*lbs	27 N*m 20 Ft*lbs
Momentum @ 1/2 Maximum Speed	1095 N*m*sec 808 Ft*lbs*sec	1270 N*m*sec 937 Ft*lbs*sec	1325 N*m*sec 977 Ft*lbs*sec
Required Gimballing[2] Rate	0.025 rad/sec	0.021 rad/sec	0.020 rad/sec

1. Original IPACS performance requirement.
2. Gimballing rate needed to produce the required output torque at 1/2 maximum speed.

Inner Gimbal...	8.39 kg	* 0.25	=	2.10 kg
Outer Gimbal...	10.4 kg	* 0.25	=	2.61 kg
Mounting Ring...	9.07 kg	* 0.25	=	2.27 kg

Total Reduction = 6.98 kg (15.4 lbs)

This reduction in gimbal weight would more than make up for the 2.7 kg (6 lbs) increase in rotor weight. In this way, an IPACS based upon a Custom 455 steel rotor would have an overall greater energy storage to weight density than either of the systems based on the titanium rotors. Additionally, there would be a reduction in system volume of nearly 40% below the titanium rotor based IPACS system. Table III compares the system and rotor weights and volumes.

Table III. Weights and Volumes for the Three Rotor Materials

Rotor Material	Rotor Weight kg	(lbs)	System Weight kg	(lbs)	System Volume m^3	(ft^3)
Custom 455	59.0	(130)	181	(400)	0.57	(20)
6Al-4V-Ti	56.2	(124)	186	(410)	0.91	(32)
B120 VCA Ti	61.2	(135)	191	(421)	0.91	(32)

There are several other advantages to using Custom 455 stainless steel over the titaniums for the IPACS rotor. Machining a titanium rotor would be a laborious and expensive process. When machining titanium, high temperatures develop at the machine tool's edge due to the low thermal conductivity of titanium. This localized heating effect causes a plastic smearing of the metallic surface instead of a cutting action and rapidly wears out machining tools. The steel selected is considerably easier to machine than titanium and would therefore significantly reduce the manufacturing costs of the rotor assembly. The reduction in gimbal size would also improve producibility thru the ease of fabrication of a casting or in reduced forging costs if hog-out fabrication is utilized.

3.0 System Response Analysis

The loading on any component of a structural system depends upon the interaction of all of the components in the system. Each component actively participates in the attenuation and/ or amplification of the loading environment as it propagates through the system. For this reason, a system response analysis was performed on the double gimballed IPACS package to determine the maximum loading conditions in a launch environment. This analysis and its results were discussed in detail in reference 2.

In this Phase II effort, the weights and stiffnesses have been updated and included in a reanalysis of the system response. In the following sections, the linear spring-mass model of the system will be described, the STS launch vibration environment will be defined, and the launch loads for each component along each axis will be presented.

3.1 IPACS Spring/Mass Model

A simple seven-degree-of-freedom model was used to describe the double gimballed IPACS. The model is a single path model in that there are no nodes that have more than two elements connected to it (1 mass, 1 spring). Figure 1 defines the masses and springs in the model and depicts their connectivity. Table IV presents the mass values for the model and the spring stiffneses along the inner gimbal, outer gimbal and spin axes. The frequency response of the rotor within the IPACS is given in figure 2.

The modifications made to this model which distinguish it from the original model developed in reference 2 have to do with the weights and stiffnesses of the gimballing structures. Analysis of the inner gimbal (reference 2) found that the stiffness of the structure was especially high along the outer gimbal axis (OA). The weight of the inner gimbal was known exactly since it exists as a piece of hardware (reference 1). The outer gimbal was generally softer than had been expected, notwithstanding an increase in weight from 9.07 to 10.4 kg (20 to 23 lbs). Substitution of an aluminum metal matrix composite would increase the structure's stiffness without altering any other properties if it becomes advantageous to stiffen the gimbal. Currently, the stiffnesses of the outer gimbal are sufficient for a moderately tight control loop bandwidth.

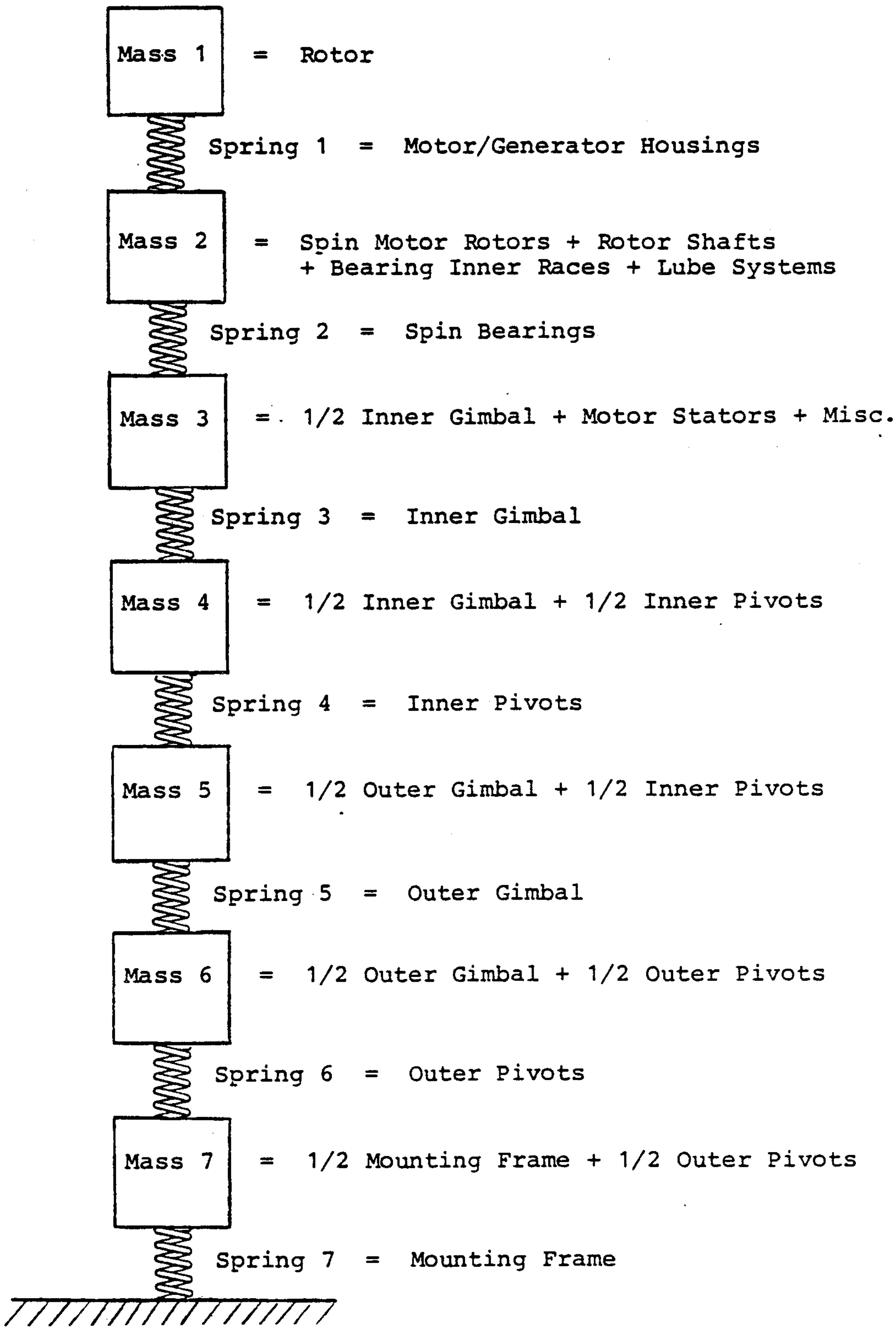
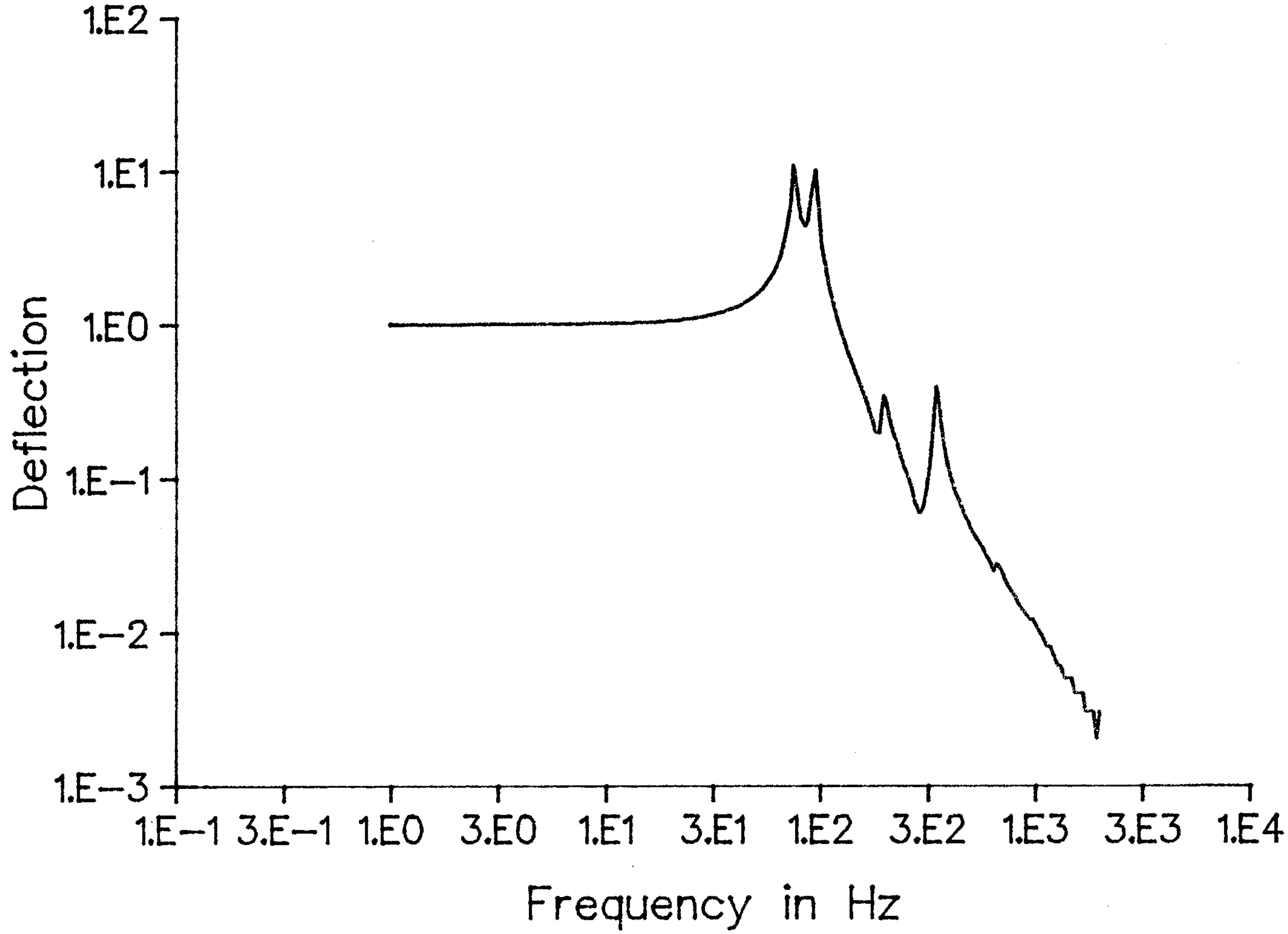


Figure 1. IPACS Model for the System Response Analysis

Figure 2. IPACS Rotor Frequency Response

Normalized Deflection vs Frequency



The mounting ring structure was very stiff primarily because the bolt down points were positioned quite close to the pivots. Redesign and analysis of the mounting ring cut the weight from 11.3 to 9.07 kg (25 to 20 lbs) while reducing the stiffnesses only slightly. These high stiffnesses do not pose any problems to the IPACS. In fact, they simplify the development of a tight gimbal control loop. The penalty assessed for this gain is usually an increase in weight for the gimbals and an increase in loading throughout the system during launch vibration. In this case, the position of the mounting bolts creates a very stiff structure. Furthermore, the loads throughout the IPACS during launch vibration are still comfortably within the capabilities of the double gimballed IPACS design as will be shown in the System Response Launch Loads section.

Table IV. Spring and Mass Values for the System Response Model

Body	Mass	Stiffness Along IA	Stiffness Along OA	Stiffness Along SA
	kg ($lbs*sec^2/in$)	10^8 N/m (10^6 lbs/in)	10^8 N/m (10^6 lbs/in)	10^8 N/m (10^6 lbs/in)
1	54.0 (0.308)	57.4 (32.8)	57.4 (32.8)	6.46 (3.69)
2	4.54 (0.0259)	1.60 (0.92)	1.60 (0.92)	0.27 (0.15)
3	14.33 (0.0818)	3.24 (1.85)	4.52 (2.58)	2.07 (1.18)
4	20.0 (0.114)	1.75 (1.0)	5.25 (3.0)	5.25 (3.0)
5	21.0 (0.120)	0.66 (0.38)	0.70 (0.40)	0.58 (0.33)
6	21.0 (0.120)	5.25 (3.0)	1.75 (1.0)	5.25 (3.0)
7	20.3 (0.116)	75.3 (43.0)	43.8 (25.0)	158.0 (90.0)

3.2 Launch Environment

Three launch environments were applied to the IPACS spring-mass model: 1) the original Skylab launch at 4.6 g's rms; 2) the final Skylab launch at 5.2 g's rms; and 3) an STS launch environment of 9.6 g's rms random vibration. This final environment was that which caused the highest component loads throughout the IPACS.

On Earth testing and in space operation impose relatively benign loading conditions on the outer gimbal, mounting ring and pivot assemblies. Qualification level vibration testing on Earth and the actual vehicle launch vibration are the most severe environments to which these components will be subjected. The spectral density plot of the STS qualification vibration environment is presented in figure 3. The precessional loads generated at 0.02 radians per second are insignificant relative to the launch or qualification loads.

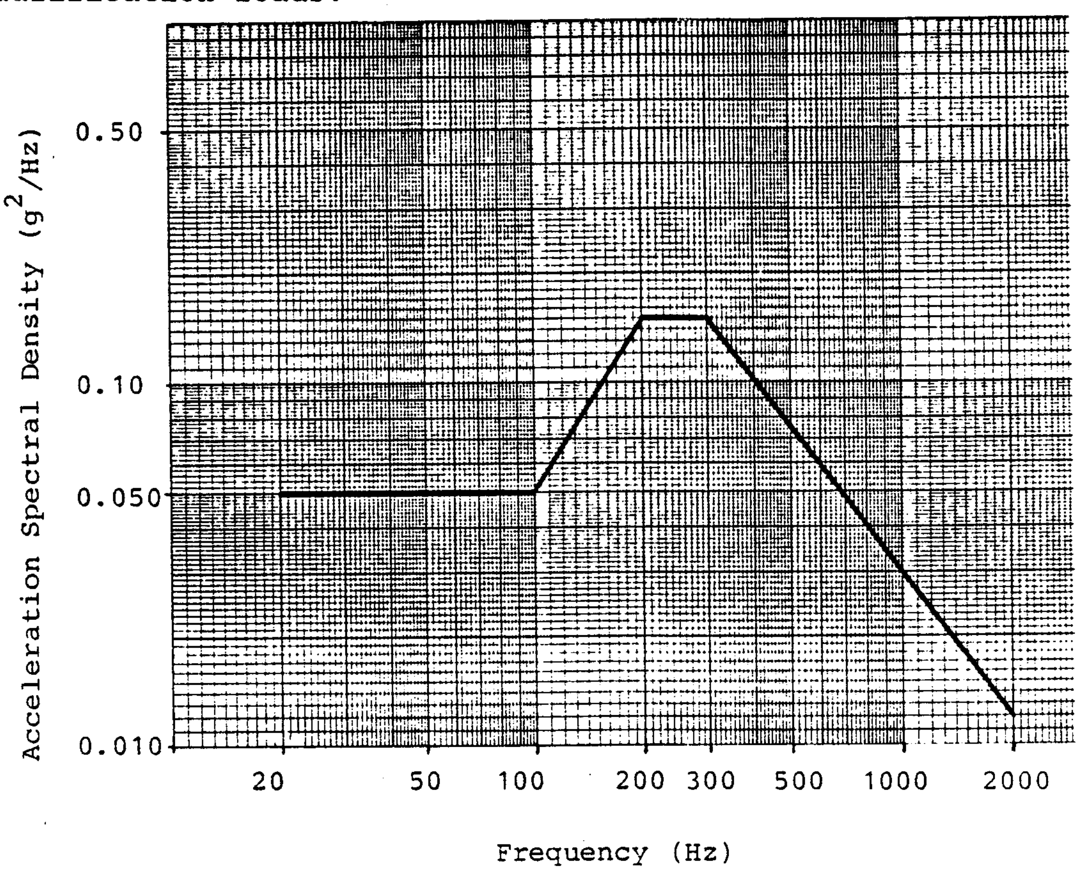


Figure 3. Spectral Density Plot of the Qualification Level Random Vibration Environment based on STS Launch

3.3 Launch Loads

The system component loads due to qualification random input as predicted by the analysis are presented in Table V. These loads are slightly higher than those presented in the original analysis (reference 2) but are still comfortably within the capabilities of the current design.

Table V. System Response Analysis Results

Maximum Design Loads (1σ)

Body	Along IA N (lbs)	Along OA N (lbs)	Along SA N (lbs)
1	6180 (1390)	6225 (1400)	5735 (1290)
2	6670 (1500)	6760 (1520)	6625 (1400)
3	7780 (1750)	7825 (1760)	6445 (1450)
4	9245 (2080)	9290 (2090)	7290 (1640)
5	10450 (2350)	10890 (2450)	8670 (1950)
6	10980 (2470)	11335 (2550)	9510 (2140)
7	11110 (2500)	11425 (2570)	9910 (2230)

On the average, loads along the inner gimbal axis (IA) decreased by 2%, loads along the outer gimbal axis (OA) increased by 14% and the loads along the spin axis (SA) increased by 2%. The magnification of the loads along the OA axis is largely the result of the high stiffnesses of the inner gimbal and mounting ring structures. The gimbals are designed to be stiff in order to enhance the control loop bandwidth of the gimballing system. These high stiffnesses also have the effect of increasing the loads throughout the system. However, these loads are still small compared to the capability of the structures within the system and no problem is foreseen with the stresses arising from the launch vibration environment.

Table VI presents the margins of safety that the 1σ launch loads have against the components' endurance strength. The lowest margin of safety for fatigue is 1.0 which indicates that the high cycle fatigue loading is about one-half of the endurance strength of the component.

Table VI. Margins of Safety Against Fatigue Strength

Margin of Safety = (Endurance limit/1σ Launch Load)-1.0

Body	Along IA	Along OA	Along SA
Rotor	185.	185.	13.
Inner Gimbal	2.6	2.0	2.2
Inner Pivots	1.4	1.9	2.6
Outer Gimbal	6.3	8.8	4.7
Outer Pivots	1.4	1.0	1.8
Mounting Ring	66.	30.	93.

Table VII presents the margins of safety that the 3σ launch loads have against the components' yield strength. The lowest margin of safety is 0.3 which indicates that the greatest single load that can occur during vibration is still 30% below the capability of the component.

Table VII. Margins of Safety Against Yield Strength

Margin of Safety = (Yield Strength/3σ Launch Load)-1.0

Body	Along IA	Along OA	Along SA
Rotor	111.	111.	7.3
Inner Gimbal	1.7	1.2	1.3
Inner Pivots	0.6	1.2	1.8
Outer Gimbal	4.0	5.7	3.0
Outer Pivots	0.8	0.3	1.1
Mounting Ring	46.	20.	64.

4.0 The Belleville Washer Preload System

Currently, the IPACS rotating assembly's bearings are preloaded with a combination of interference fits at the bearings and a large helical spring. The interference fits on the bearings are on both the inner and outer races. At operating temperatures these interference fits are reduced to nearly zero, and consequently so is the preload from this reaction (see reference 2). The large helical spring is therefore the sole remaining preloading mechanism during high speed operation. In order to keep the bearing preloaded during gimballing operation of the IPACS, the force produced by the helical spring would have to be 245 N (55 lbs), (see reference 2). This is a fairly high preload for a high speed energy storage device, resulting in a considerable power loss. Furthermore, an increased preload generates higher bearing temperatures and reduces the fatigue life of the bearing.

An improvement over this type of spin bearing preload system would be the use of belleville washer constant force springs. These small conical springs can achieve high spring rates in extremely small packages, allowing the use of lower preload forces than the helical spring. In the following sections the advantages of using belleville washers to preload spin bearings will be enumerated; a preload system utilizing belleville washers and designed to be compatible with the existing IPACS rotating assembly will be presented; and the resulting improvements upon bearing reliability, assembly power losses and system torque bandwidth will be discussed.

4.1 Belleville Washer Advantages

A high performance rotating assembly like an IPACS or a CMG is a very interactive device in that the alteration of any component can produce a multitude of reactions throughout the system. The spin bearing preload mechanism is one of the most sensitive components in this regard. For this reason, optimization of the preload mechanism will often result in significant improvements in the performance of many of the other components in the assembly.

The use of belleville washers in a preload mechanism allows the utilization of very low preload forces on the spin bearings. The impact of this preload reduction throughout the IPACS system is far reaching. Reducing the spin bearing preload directly produces a reduction in spin bearing drag torque power losses, and thereby, a reduction in the spin bearing operating temperature. Decreasing the spin bearing drag effectively increases the efficiency of the charge/discharge cycle of the IPACS, the spin bearings being a significant source of power dissipation. This reduction of drag torque would then allow a reduction in motor power to achieve the same run up and run down performance characteristics. This would in turn cause a reduction in the operating temperature for the motors and coupled with the reduced bearing temperatures would result in a lower flywheel temperature. Because of the inverse relationship between temperature and the flywheel material's strength, lowering the flywheel's temperature increases its strength and would allow a greater top speed for the rotor and consequently greater power storage.

4.2 Preload Mechanism for the IPACS

4.2.1 Spring Parameters

Parameters of the belleville washer spring are given in Table VIII. A load versus deflection graph for the spring is shown in figure 4. Inspection of this graph shows that the spring bottoms out (runs out of travel) before the spring can snap thru. Snap thru is the action that occurs in washer type springs when the differential stiffness of the spring (dK/dx) changes from positive to negative. Physically, the top of the washer deflects past the base and turns inside out.

Table VIII. Belleville Spring Parameters

Outside Diameter:	5.08 cm	(2.00 inch)
Inside Diameter:	3.81 cm	(1.50 inch)
Free Height:	0.508 mm	(0.020 inch)
Thickness:	0.508 mm	(0.020 inch)
Stress at Flat Position:	335 MPa	(48 ksi)
Spring Load at Flat Position:	52.9 N	(11.9 lbs)

4.2.2 Spring Material

Type 302 Stainless Steel was chosen as the spring material. It would be cold worked to about 40% which greatly increases the strength of the material. The steel has a high resistance to stress corrosion, can operate at high temperatures and has excellent fatigue strength. Important physical and mechanical properties for this stainless steel are presented in Table IX.

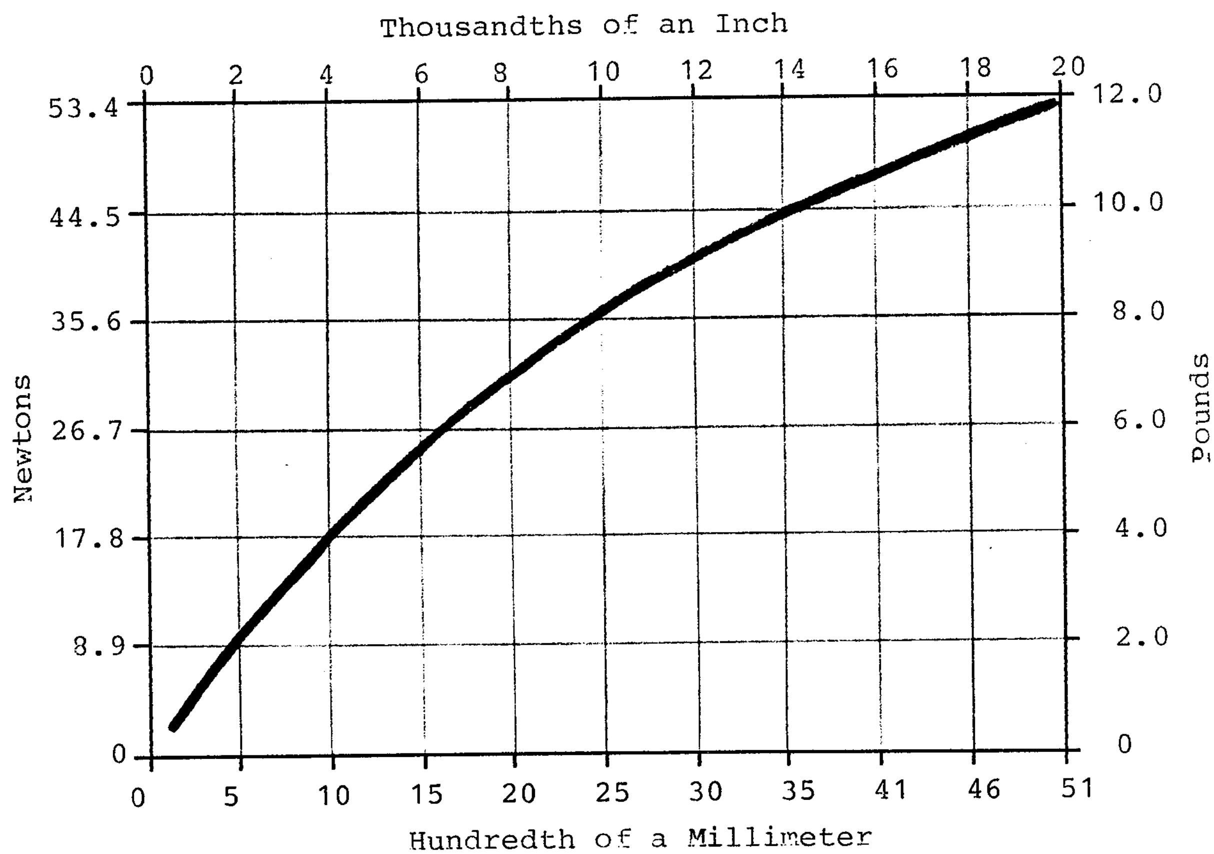


Figure 4. Load versus Deflection for the IPACS Belleville Spring

Table IX. Physical and Mechanical Properties of the Belleville Washer Material

Material: Type 302 Stainless Steel, MIL-5-5059
40% Cold Rolled

Ultimate Strength:
$293^{\circ}K$ = 1190 MPa ($60^{\circ}F$ = 170 ksi)
$366^{\circ}K$ = 1070 MPa ($200^{\circ}F$ = 153 ksi)

Yield Strength:
$293^{\circ}K$ = 875 MPa ($60^{\circ}F$ = 125 ksi)
$366^{\circ}K$ = 810 MPa ($200^{\circ}F$ = 116 ksi)

Density: 7915 kg/m^3 (0.286 lbs/in^3)

Modulus of Elasticity: 2.0×10^5 MPa (29×10^6 psi)

Poisson's Ratio: 0.30

Fatigue Strength:
$293^{\circ}K$ = 510 MPa ($60^{\circ}F$ = 73 ksi)
$366^{\circ}K$ = 485 MPa ($200^{\circ}F$ = 69 ksi)

Stress Corrosion Resistance: High

4.2.3 Belleville Spring Analysis

Analysis of the belleville washer preload system involved the application of ten different possible loading conditions. The variables that were considered included vacuum shrinkage of the inner gimbal, thermal mismatches between the rotor and inner gimbal, operation with the spin axis both horizontal and vertical, and an axial pumping force generated by the rotor at operational speed. These loading cases are defined in Table X.

The short lengths of the rotor and inner gimbal, and the fact that there are no huge variations in these components' temperatures, produced thermal expansions that were extremely small. These thermal deflections between the titanium rotor and the aluminum inner gimbal were less than 5% of the other system deflections and were therefore ignored.

The axial pumping force is generated by the rotation of the rotor thru the spin bearing runout tolerances. This oscillation produces a pumping force at the wheel speed frequency. The natural frequency of the rotor/belleville

Table X. Belleville Washer Load Cases

Load Case	Description	Spin Axis	Vacuum[1]	Rotor Temp	IG[2] Temp	Rotor Pumping[3]
1	Assembly	Vertical	No	RT	RT	No
2	Assembly	Horizontal	No	RT	RT	No
3	Evacuation	Vertical	Yes	RT	RT	No
4	Evacuation	Horizontal	Yes	RT	RT	No
5	Run Up	Vertical	Yes	338°K 150°F	294°K 70°F	Yes
6	Run Up	Horizontal	Yes	327°K 130°F	294°K 70°F	Yes
7	Max Speed	Vertical	Yes	330°K 135°F	311°K 100°F	Yes
8	Max Speed	Horizontal	Yes	327°K 130°F	311°K 100°F	Yes
9	Run Down	Vertical	Yes	300°K 80°F	311°K 100°F	Yes
10	Run Down	Horizontal	Yes	294°K 70°F	311°K 100°F	Yes

Notes: 1. Vacuum of 2 microns pulled on inner gimbal assembly = 1 atmosphere external gage pressure.
2. IG stands for Inner Gimbal.
3. Pumping force generated by rotor at speed due to bearing runout.
4. Extreme temperatures taken for conservatism.

spring system is very low at approximately 8 to 10 hz, while the IPACS will operate at roughly 300 to 400 hz. Because of the great distance between the operating frequency and the natural frequency of the system, there is no significant dimensional movement of the rotor due to this axial pumping force anywhere near the operating speed range of the IPACS.

The greatest dimensional changes occurred between the vacuum/no vacuum conditions and the spin axis vertical/horizontal conditions. These four loading conditions primarily controlled the design of the belleville spring.

Installation of the belleville springs would initially specify a deflection of 0.305 mm (0.012 inch) for each spring in a spin axis horizontal configuration. Moving the IPACS to a vertical spin axis orientation would result in the lower belleville spring compressing to its maximum deflection of 0.508 mm (0.020 inch) while the top spring would expand 0.203 mm (0.008 inch) for a resulting preload deflection of 0.102 mm (0.004 inch).

When a vacuum is pulled on the inner gimbal (assuming there is 1 atmosphere of pressure outside), the entire structure shrinks. This shrinkage is approximately 0.203 mm (0.008 inch) and was computed from finite element analyses of the inner gimbal (see reference 2). If the IPACS was in a spin axis vertical configuration, this 0.203 mm (0.008 inch) shrinkage would be applied to the top spring, increasing its deflection to 0.305 mm (0.012 inch). Upon returning the unit to a spin axis horizontal orientation, the springs would equalize their deflections at 0.406 mm (0.016 inch) apiece.

It should be noted that the horizontal, no vacuum condition most simulates the situation that will exist in space. The tabulation of the four loading cases, the spring deflections for each, and the resulting spin bearing preloads for each case are presented in Table XI.

Table XI. Spin Bearing Preloads for the IPACS

Load Case 1:	Spin Axis Vertical No Vacuum Bottom Spring Flat (Snap thru not possible) Bottom Spring Deflection= 0.508 mm (0.020 in) Bottom Bearing Load= 578 N (130 lbs) Top Spring Deflection= 0.102 mm (0.004 in) Top Bearing Load= 18.2 N (4.1 lbs)
Load Case 2:	Spin Axis Horizontal No Vacuum Spring Deflections and Bearing Loads Equal Single Spring Deflection= 0.305 mm (0.012 in) Single Bearing Load= 40.5 N (9.1 lbs)
Load Case 3:	Spin Axis Vertical IG Evacuated to 2 Microns Pressure Bottom Spring Deflection= 0.508 mm (0.020 in) Bottom Bearing Load= 600 N (135 lbs) Top Spring Deflection= 0.305 mm (0.012 in) Top Bearing Load= 40.5 N (9.1 lbs)
Load Case 4:	Spin Axis Horizontal IG Evacuated to 2 Microns Pressure Spring Deflections and Bearing Loads Equal Single Spring Deflection= 0.406 mm (0.016 in) Single Bearing Load= 47.1 N (10.6 lbs)

4.3 Impact on System

The preload on the spin bearings peaks at 47.1 N (10.6 lbs) during spin axis horizontal, on Earth testing. This is a considerable improvement compared to the 245 N (55 lbs) that would be necessary to preload the system with the original helical spring preload mechanism. In space, the expansion of the inner gimbal due to the absence of a pressure differential across it allows the belleville washer preload system to expand slightly and drops the spin bearing preload to 40.5 N (9.1 lbs). The original preloading mechanism has no such reaction; the change in preload is insignificant due to the extremely small deflection combined with the low stiffness of the helical spring.

A comparison between the spin bearing power levels can be made for an operational load. For comparison, from reference 2 the loading would be: the titanium rotor IPACS rotating assembly running at 24000 rpm in space with either the 245 N (55 lbs) preload from the original helical spring preload mechanism, or 40.5 N (9.1 lbs) from the belleville washer preload mechanism.

The drag torque power may be calculated from the component viscous, thrust, and radial drag torques as was done in reference 2. The original helical spring system would develop a drag torque power of 25.5 watts dissipation per bearing. The belleville washer system develops a drag torque power of only 8.9 watts dissipation, a reduction of 65%! As a consequence, there would be a considerable reduction in the motor power required to drive the IPACS flywheel. Combined with the reduction in heat generated at the bearings, there would be an overall lowering in the assembly's operating temperature.

Reliability of the spin bearings would benefit because of the reduced load and the cooler operating temperatures. The reduction of preload from 245 N (55 lbs) would increase the analytically derived survival rate from 99.84% to above 99.99%.

Associated with the reduced thrust preload on the spin bearings will be a reduced radial stiffness in the bearings. However, this softening of the spin bearings will not adversely affect the system torque output bandwidth. The electronic control loop that is responsible for determining the bandwidth would be designed to take into consideration the different stiffness of the spin bearings. The bearings would still be very stiff, and this would allow a wide bandwidth control loop to be established.

5.0 Conclusions

A parametric study investigating three rotor materials for the IPACS revealed the benefits of using Custom 455 stainless steel for the rotor as opposed to the high strength titaniums. Although this steel cannot match the titanium's strength to weight density, it is a better choice for the rotor material because of its high density. Because the steel is approximately 78% denser than the titanium, a much smaller and only slightly heavier steel rotor can be made having the same energy storage capability as the titanium rotor. The advantage of the steel rotor is that it is 16% smaller in every dimension than the titanium rotor and the weight savings due to the smaller gimbals surrounding the steel rotor more than makes up for the weight difference between the steel and titanium rotors. Not only would an IPACS with a steel rotor have a slightly higher energy storage to weight density than a titanium rotor IPACS, but the overall system volume would be 40% smaller.

The system response analysis updated the existing linear spring/mass model. Modifications to the model included changing the stiffnesses and weights of the inner gimbal, outer gimbal and mounting ring. These structures were analyzed and the results were included in this system analysis. A mild redesign effort was established for the outer gimbal and mounting ring in order to lower their weights without severely altering their stiffnesses. This was accomplished and an STS launch vibration environment was applied to the model. The loads on the components in the system increased slightly due to the gimbal stiffnesses being higher than were originally estimated. However, the peak loads are still comfortably within the capabilities of the system.

A belleville washer spring preload mechanism was designed to be compatible with the existing IPACS IGA. This mechanism was designed to replace the helical spring and interference fit preload mechanism currently in use in the IPACS. Preload reductions for the belleville washer system are significant. The existing preload system would require 245 N (55 lbs) for an on Earth, gimballing operation. The belleville washer spring preload system would require only 40.5 N (9.1 lbs). Drag torque power dissipation at the spin bearings would also decrease substantially. Operation of the IPACS in space with the existing preload design would dissipate 25.5 watts per bearing. The belleville washer preload system would develop a drag torque power of only 8.9

watts per bearing. Additional benefits include the reduction of the operating temperatures for the spin bearings and consequently an increase in reliability for these components. Motor power levels could also be reduced while maintaining the same run up and run down times. Because the spin bearings and motors would be dissipating much less power, the entire rotating assembly would be operating at a much reduced temperature. The reduction in operating temperature for the rotor would increase the allowable strength value for the material and subsequently increase the margins of safety for the IPACS.

Development of the IPACS concept has reached a point where hardware implementation is the next logical step. The inner gimbal assembly (IGA) can achieve 2.8 watt-hours per kilogram (6.3 watt-hours per pound) energy density. As was presented in this document significant improvements in the performance of the IGA may be achieved by using Custom 455 stainless steel for the rotor, and substituting a belleville washer constant force spring preload system to preload the spin bearings instead of the helical spring/interference fit method of preload currently being used.

Gimballing of the IPACS IGA is not difficult. The very small required output torque does not generate any significant loads in the system. For this reason, there is a moderate amount of gimballing capability that is not being used in this system. A precession rate approximately five times larger than the 0.02 radians per second currently required for the IPACS would be within the operating capability of the system. Although the advantage of the IPACS system over separate energy storage and attitude control devices becomes less pronounced as the capabilities of the system are utilized less and less, there is still a significant savings in weight and space in this application. However, with low output torque requirements, as developed in this IPACS, it may be useful to compare the double gimballed IPACS concept with a system of high speed energy storage flywheels configured as single degree of freedom reaction wheels. Analysis has shown the advantage of reaction wheel assemblies over control moment gyros in low output torque configurations and it may also be true for this application.

6.0 References

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15. Supplementary Notes

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16. Abstract

A parametric study analyzing the suitability of various IPACS rotor materials was conducted. This study investigated three materials: 6Al-4V-Titanium (the current IPACS rotor material), B120 VCA Titanium, and Custom 455 Stainless Steel. The preliminary linear vibration analysis was updated to include the weights and stiffnesses of the gimbals designed in NASA CR-172524. Finally, a belleville washer spring preload mechanism was designed to replace the existing helical spring and interference fit preload mechanism.

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