

Development of a Nutation Damper Reservoir Mechanism

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ABSTRACT

A nutation damper was developed for the Radiation Belt Storm Probes (RBSP) program. The passive damper is fully filled with a custom-blended silicone and comprised of 1-inch titanium tubing formed into a racetrack shape measuring approximately 35 inches wide and 27 inches high. One of the critical components of this nutation damper is the reservoir, which provides sufficient space for the fluid to expand and contract throughout the damper's operational temperature range, while maintaining positive fluid volume in the damper. The reservoir contains a piston-and-spring mechanism that applies pressure generated by the spring compression to maintain a fully-filled state, and to prevent bubbles from forming. The reservoir's piston-and-bore interface is sealed with two inline O-rings that are compatible with the silicone oil and with temperature conditions ranging from -30 to 60 °C. The design of the reservoir involves the complex relationships of fluid volume, thermal expansion, O-ring compression, friction, spring stiffness, and spring stroke. This paper will address the development of the reservoir and its mechanism, the engineering model tests at various developmental stages, the leak failure mitigation efforts, and the successful mechanical qualification tests of the flight units.

EXECUTIVE SUMMARY

The RBSP nutation dampers were designed to function over a -30 to 60 °C temperature range. A reservoir was designed to provide sufficient space for the fluid to expand and contract throughout the damper's operational temperature range while maintaining positive fluid volume in the damper. The reservoir uses a piston-and-spring mechanism to achieve a fully-filled state at all times.

The piston-to-bore interface was initially designed with two in-line ethylene propylene O-rings. Extensive qualification testing was performed to ensure that the mechanism performed as designed, which ultimately led to a lengthy failure investigation, as a leak was found at the piston-to-bore interface early in the testing process.

The initial investigation determined the cause of the leak. Factors that were investigated included:

- Damage to the O-rings, piston, and cylinder
- Alignment of the piston within the cylinder
- CTE effects on the O-rings
- Rolling or slipping of the O-rings during rapid movement of the piston
- O-ring performance at cold temperatures
- Rate of change of the fluid temperature affecting the piston motion
- Pressure differential across the O-rings affecting their sealing capabilities

The root cause was determined to be a combination of a number of factors. Cold temperatures result in the O-rings shrinking, which results in less compression on the cylinder walls. As the temperature of the fluid increases, the difference in the pressure between either side of the O-ring seal becomes greater than that of the O-ring compression on the bore, resulting in a leak.

The piston was redesigned to increase the compression of the O-rings and the damper was retested for leaks. Another leak occurred, resulting in further failure investigations. The surface finish of the sealing surfaces was added to the possible factors causing the leak; however, this was

determined to not be the cause of the leak. It was determined that additional compression was needed on the piston O-rings.

The damper was again re-tested for leaks after a second piston redesign and another leak occurred. It was determined that simply increasing the compression on the O-rings could not solve the failure. The final iteration in the design was to change the O-ring material itself. The material of the O-rings was changed from ethylene propylene to low temperature nitrile. The damper was again re-tested for leaks and the seal was shown to be leak tight. Nitrile O-rings are currently used in the flight nutation dampers. The development process of the reservoir mechanism is illustrated in [Figure 1](#).

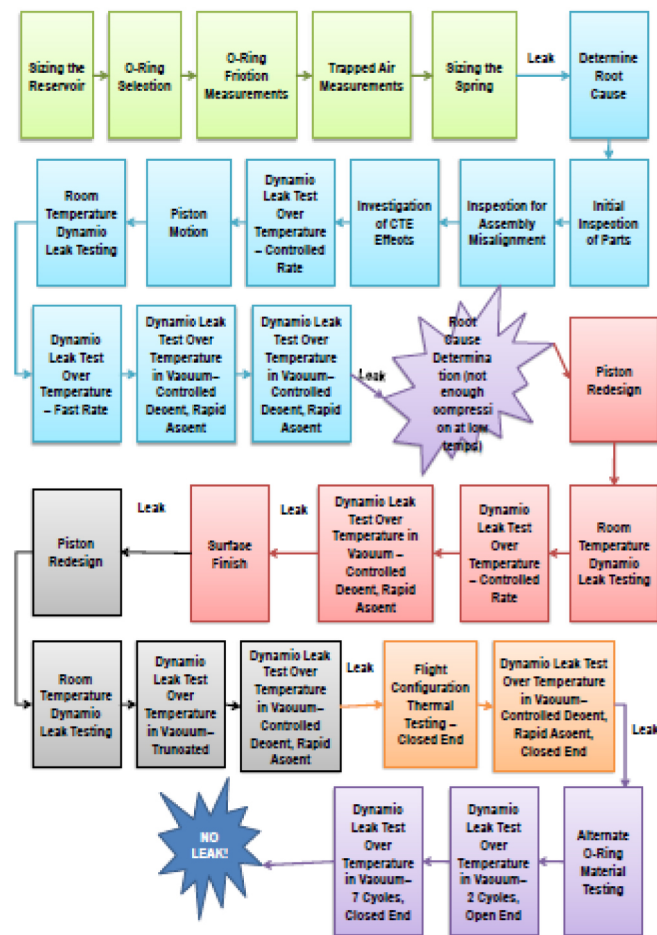


Figure 1. Flow Chart Of Development Process

INTRODUCTION

The RBSP Mission, part of NASA's Living With a Star Geospace program, will provide unprecedented insight into the physical dynamics of Earth's radiation belts and give scientists the data they need to make predictions of changes in this critical region of space. Beginning in 2012, two spacecraft will orbit the Earth, sampling the harsh radiation

belt environment where major space weather activity occurs and many spacecraft operate. The two spacecraft will measure the particles, as well as magnetic and electric fields that fill geospace. With two spacecraft taking identical measurements and following the same path, scientists can begin to understand how the belts change in both space and time.

The Johns Hopkins University Applied Physics Laboratory will build and operate the twin RBSP spacecraft. Each spacecraft will carry a number of identical instruments and instrument suites to support five experiments that will address the mission's science objectives. The RBSP spacecraft will operate entirely within the radiation belts throughout their mission. Each spacecraft is spin stabilized. Stabilization of each spinning spacecraft (SC) will be aided by two passive nutation dampers mounted orthogonally on the side of the main thrust cylinder ([Figure 2](#)).

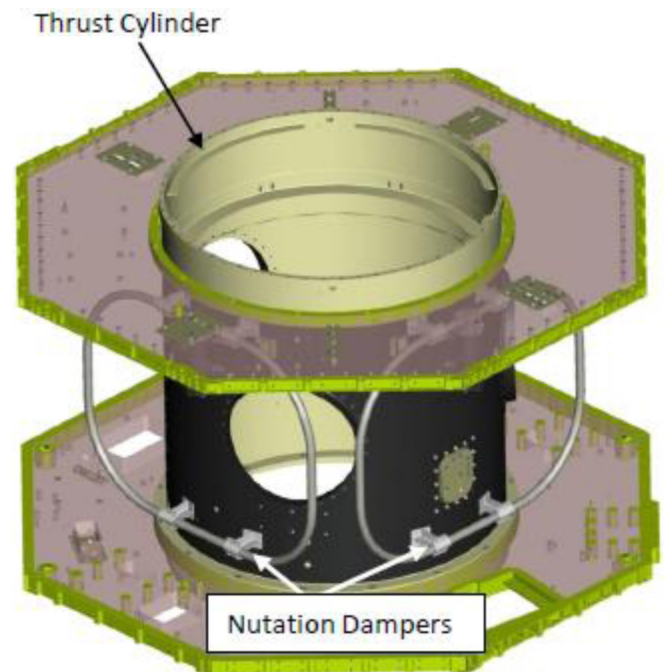


Figure 2. Nutation Damper Placement

The effectiveness of the nutation dampers depends on the moment-of-inertia (MOI) that the dampers generate to counter, over time, the forces that cause each spinning SC to wobble about its spin axis. Therefore, the nutation damper's size was based on the SC mass, and the size of the damper determines the volume of fluid used in the damping process

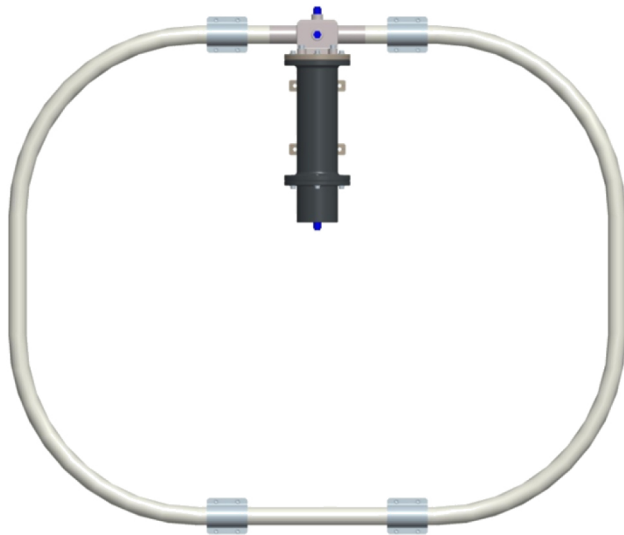


Figure 3. RBSP Nutation Damper

This MOI is supplied by a 1-inch diameter tube fashioned into a racetrack shape approximately 33 inches wide and 27 inches high (Figure 3), fully filled with a custom blend of silicone oil. Because the damper's structure and fluid will contract and expand throughout the RBSP on-orbit survival temperature range of -30 to 60 °C, it requires a reservoir in order to maintain its fully-filled state. A piston-and-bore mechanism which includes O-rings and a spring (Figure 4) is used to accommodate the thermal expansion

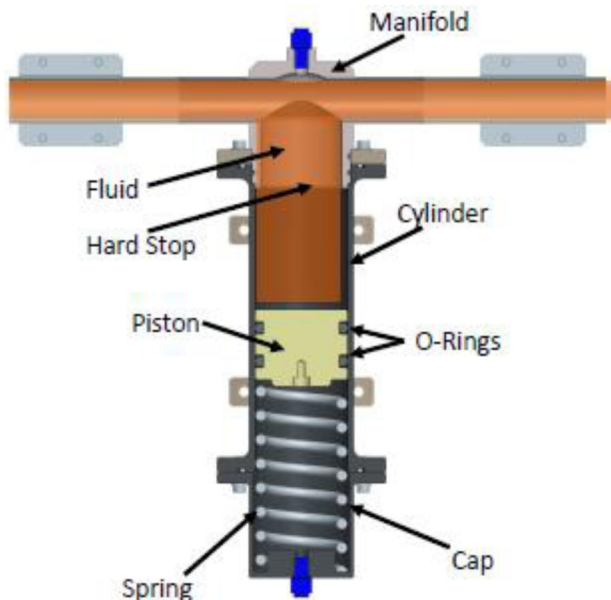


Figure 4. Reservoir Assembly

The piston stroke is limited by the length of the cylinder and the compression of the spring. As the fluid contracts in the cold environment, the spring extends, and the piston is

pushed up, and it will eventually hit a hard stop (see Figure 4) in the design of the reservoir assembly.

The nutation damper is composed primarily of titanium parts in order to meet project magnetics requirements. The tubing is 1-inch diameter 3Al-2.5V titanium which is welded to the reservoir assembly. The cylinder, cap, manifold and piston are machined 6Al-4V titanium. The spring is a custom manufactured Inconel spring. Ethylene propylene O-rings are used for static seals and nitrile O-rings are used for dynamic sealing on the piston which are indicated in Figure 4.

SIZING THE RESERVOIR

As mentioned above, the purposes of the reservoir are twofold. One is to store sufficient fluid such that when the fluid reduces in volume due to a cold temperature exposure, the damper tubes will continue to be fully filled. The other is to provide sufficient storage for the expansion of the fluid in a hot temperature environment. Therefore, the size of the reservoir depends primarily on the thermal expansion and contraction of the silicone oil in the system. Due to the silicone oil having a higher coefficient of thermal expansion than the titanium damper structure, in the cold case the fluid will shrink and leave the reservoir, and the piston will move towards the inlet of the reservoir. It can be seen, then, that the cold stroke defines the initial position of the piston within the reservoir bore. A 0.25-in end-of-stroke margin (which corresponds to a 10 % margin on overall piston stroke) is incorporated so that the piston will not hit the hard stop at the reservoir inlet. The stroke can be determined by calculating the initial volume of the fluid and deriving the thermal shrinkage and expansion of both the fluid and the damper, as shown below in Table 1.

Table 1. Piston Stroke in Reservoir

	Tube	Manifold	Reservoir
Length (in)	102.306	1.925	1.537
Diameter (in)	0.938	1.755	2.018
Volume (in ³)	70.696	4.657	4.916
System Volume (in ³)	80.269		
Room Temp (°C)	20		
	Cold		Hot
Temp (°C)	-30		60
Linear CTE Ti (1/°C)	8.64E-06		8.91E-06
Volumetric CTE Fluid (1/°C)			
Silicone 200-5cSt	1.05E-03		1.05E-03
Fluid Volume (in ³)	76.055		83.640
Tube-Manifold Volume (in ³)	75.255		75.434
Reservoir Volume (in ³)	0.799		8.207
Reservoir Length Change (in)	-1.288		1.028
Displacement of Piston (in)	2.316		

The reservoir will need to accommodate a total change in volume related to a piston stroke of 2.316 in. The bore diameter of the reservoir is dependent firstly on the geometric constraints in the spacecraft for the damper assembly to fit, and secondly on the sizes of the off-the-shelf O-rings. The O-ring interaction with the reservoir bore involves O-ring radial compressive forces for sealing. These compressive forces and the size of the O-ring would then affect the amount of friction at that interface that would need to be overcome.

SELECTING THE O-RING AND SIZING THE GLAND

The critical factor in the selection of the O-ring for this application is the amount of squeeze necessary to seal the silicone oil in front of the piston. The reservoir piston mechanism is thought to operate in a dynamic environment where the piston cycles fore and aft within the cylinder during the in-orbit exposure to low and high temperatures. The recommended amount of squeeze for dynamic conditions is a max of 16 % of O-ring cross-section (1). The thermal effects of the environment and the dimensional tolerances of the gland components influence the degree of compression applied to the O-ring.

From the Design Table 5-2 in the Parker O-ring Handbook (1), an O-ring with an internal diameter of 1.600 inches and a cross-section of 0.210 inches fits the nominal bore diameter of 2 inches. An ethylene propylene (EP) O-ring material was selected as recommended by Parker for compatibility with silicone oils and the operating temperature range. Table 2 details the computation of the bore and groove dimensions and the generated compression of the O-ring. The goal in the computation was to obtain a max compression of 10 % in the O-ring at the hot temperature.

It can be observed that the compression variable is extremely sensitive to the tolerances of each component of the mechanism and to the temperature variation. This is of concern because the compression affects the friction magnitude at the O-ring and bore interface, and the higher it is, the more the force required to slide the piston along the reservoir bore. The friction force is the same at any position in the bore, but the situation becomes critical when the piston is at end-of-travel in the cold position, close to the hard stop of the reservoir at the inlet, because at this position the spring extends close to its unsprung length and loses most of its potential energy. Without other influences, such as pressure differential across the piston, the spring force needs to be higher than the friction force that resists the motion of the piston. It is necessary to determine the friction force at the O-ring and bore interface so that a suitable spring of sufficient stiffness and deflection can be incorporated into the design of the mechanism.

Table 2. O-Ring Compression

	Room		Cold		Hot	
Temp (°C)	20		-30		60	
		tol (± in)	min	max	min	max
O-ring ID (in)	1.600	0.015	1.572	1.602	1.595	1.625
O-ring Width (in)	0.210	0.005	0.203	0.213	0.206	0.216
Bore Diameter (in)	2.017	0.001	2.015	2.017	2.017	2.019
Groove Diameter (in)	1.631	0.001	1.629	1.631	1.631	1.633
CTE O-ring (1/°C)	1.602E-04					
Ethylene Propylene						
CTE Ti (1/°C)			8.640E-06		8.910E-06	
O-ring Stretch (%)			1.700	3.752	0.322	2.346
O-ring Width Reduction (%)			1.540	2.709	0.358	1.996
$r := \begin{cases} 0.56 + 0.59S_{og} - 0.0046S_{og}^2 & \text{if } S_{og} > 3 \\ -0.005 + 1.19S_{og} - 0.19S_{og}^2 + 0.001S_{og}^3 + 0.008S_{og}^4 & \text{otherwise} \end{cases}$						
Ref: Parker O-ring Handbook						
O-ring Width after stretch (in)			0.198	0.210	0.202	0.216
Gland Depth (in)			0.192	0.194	0.192	0.194
O-ring Compression (%)			2.0	8.6	4.0	10.9

MEASURING THE O-RING FRICTION

A test was conducted to determine the friction force between the O-ring and bore. More specifically, through this test, the coefficient of friction was established for the coupling of ethylene propylene and titanium that has been surface treated with titanium anodized coating and smoothed to 32 microns, and lubricated with silicone oil.

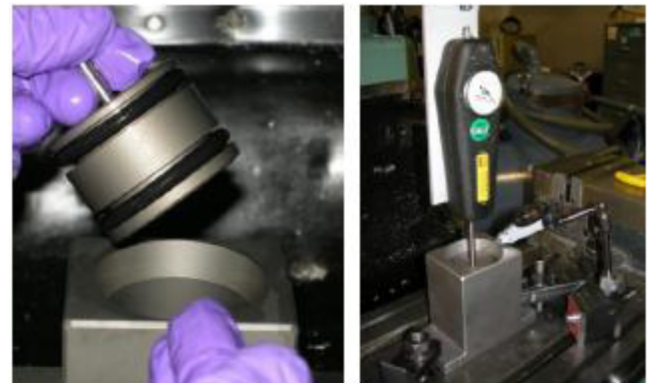


Figure 5. O-ring Friction Test Setup

An engineering unit of the piston and bore was built and assembled with the O-rings (Figure 5). The selected O-rings were ethylene propylene, of 2.0-in nominal diameter, had a 0.210-in cross-section, and were rated at 70-durometer hardness. The piston and bore were made of titanium (Ti-6Al-4V), and had their surfaces finished to 32 microns and anodized (Titanium Anodize Type 2 AMS- 2488). Three dimensional variations of the piston groove diameter were made by fabricating three different pistons that resulted in three O-ring compressions of 6.5 %, 5.9 % and 5.3 %. These

compression levels were the outcome of the dimensional tolerances in the design of the piston and bore.

The O-rings and bore were well lubricated with silicone oil (Dow Corning Silicone 200 10 cSt) prior to assembly. A force scale attached to a milling machine head was used to measure the static friction force when pulling on the piston. With the unit assembled and the test fixtures set up, the test was conducted 10 times for each compression level. The O-rings were replaced with new ones at each compression setting. The results of the test are presented below in [Table 3](#).

Table 3. Results of O-ring Friction Test

Compression	6.5%	5.9%	5.3%
Test	Force (lb)	Force (lb)	Force (lb)
1	14.00	12.00	10.25
2	14.00	13.50	10.00
3	15.00	12.50	9.50
4	14.00	12.50	10.50
5	14.25	12.75	10.25
6	14.25	12.75	10.25
7	14.25	11.75	11.00
8	15.00	12.50	11.00
9	14.25	12.25	10.75
10	14.75	11.75	11.25
Average	14.38	12.43	10.48

The friction force at 6.5 % O-ring compression was found to be 14.4 lb, and it can be seen from the results that when the O-ring compression was reduced, the friction force also reduced. The O-ring compression translates to a normal force that is applied by the O-ring onto the wall of the bore. This force can be estimated by utilizing the compression test chart ([Figure 6](#)) found in Parker's design guide.

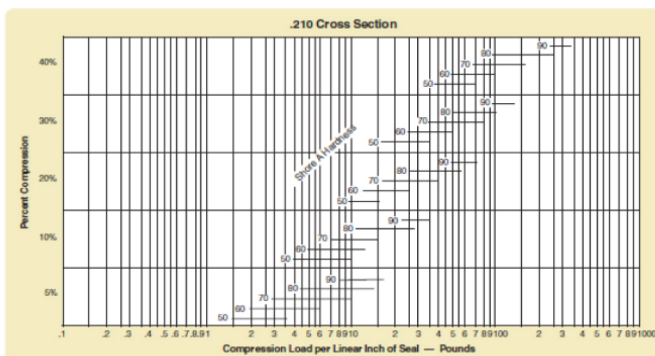


Figure 6. Compression Load per Length of Seal

From this chart, a trend can be derived for a 70-durometer hardness O-ring material, and an equation can be generated relating the percent compression to the compression load per linear inch of O-ring contact ([Figure 7](#)).

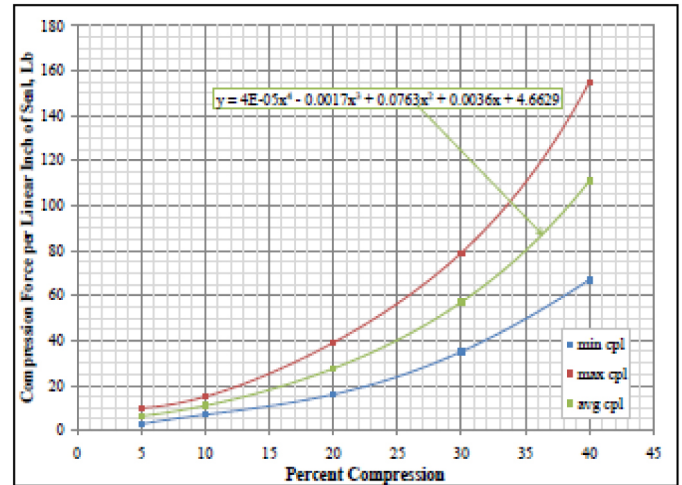


Figure 7. Radial Force on O-ring Interface

The friction coefficient for this particular combination of materials (EP-Ti Lubed) can then be derived for the range of compression levels tested using the equation in the chart above, and the averaged value is determined to be 0.138 ([Table 4](#)). This coefficient is useful in estimating the friction force when changes are made to the compression rates.

$$C_{\sigma_0} = 6.499$$

$$CPL := 4 \cdot 10^{-5} \cdot C_{\sigma_0}^4 - 0.0017 \cdot C_{\sigma_0}^3 + 0.0763 \cdot C_{\sigma_0}^2 + 0.0036 \cdot C_{\sigma_0} + 4.6629 = 7.514$$

$$F_{fr} := 14.375 \text{ lbf}$$

$$\mu := \frac{F_{fr}}{2 \cdot \left(CPL \frac{\text{lbf}}{\text{in}} \right) \cdot \pi \cdot D_b} = 0.151$$

Table 4. Average Friction Coefficient from Test

Compression	6.5%	5.9%	5.3%
Friction, lb	14.38	12.43	10.48
Friction Coeff	0.151	0.139	0.125
Avg Friction Coeff	0.138		

TRAPPED AIR FORCE

The cavity on the spring side of the reservoir bore is filled with air at atmospheric pressure ([Figure 8](#)). The design of the cavity is such that the space is fully sealed for possible silicone oil contamination purposes. With this condition, the trapped air will produce a tensile force on the piston when the piston moves towards the inlet of the reservoir. The trapped air tensile force was measured in a pull test described below.