Spider: Mechanical Design of a Resealable Window

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Abstract

The Spider balloon-borne telescope is a cosmology experiment designed to measure polarization in Cosmic Microwave Background radiation. Central to the success of the experiment is the engineering design of a retractable window that protects the detectors while on the ground and during ascent and descent. The design process was a balance between trying to make the simplest mechanism and ensuring robust performance. Design focused on four component groups: the linear guide rails, the rack and pinion system, the four-bar mechanism, and the o-rings. The rails consist of a custom aluminum frame that houses a pre-manufactured linear guide system from Pacific Bearings. MR12 rails were selected to withstand the front and back thick-window frame forces of 248 N and 154 N, respectively, in the lateral direction. The rack and pinion, with motor and gearbox, drive the window in and out to deploy and retract it. The rack and pinion selected are available from Atlanta Drive Systems and can easily withstand the 174 N force required of them. The four-bar mechanism includes the rails and provides the axial movement that creates the thick window’s seal. Movement of the four-bar mechanism is also driven by the rack and pinion gears and motor. The o-rings create the seal over the cryostat faceplate, protecting the astronomical instruments from large pressure differentials. Nitrile was selected as the o-ring material for its low temperature performance and good gas impermeability properties. The interaction of the four component groups was crucial in determining their respective properties, such as strength, motor torque, and dimensions.
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List of Notations

CMB – Cosmic Microwave Background

NSERC – Natural Sciences and Engineering Research Council

BLAST – Balloon-borne Large Aperture Sub-millimetre Telescope

FEA – Finite Element Analysis

mK – millikelvin

lbf – Pound force

N – Newton

C – Celsius

E – Young’s Modulus

I – Moment of Inertia

MPa – Megapascal

CAD – Canadian dollar

USD – American dollar
1. Introduction

The goal of this report was to complete the engineering design of the Spider window cover, a retractable, self-sealing plate to maintain pressure equilibrium in the cryostat. Spider is a precision cosmology experiment to measure the polarization of the cosmic microwave background radiation (CMB) using a purpose-built telescope on a platform (gondola) flown below a stratospheric balloon [1]. This project involved the design and analysis of four major component groups:

- the linear guide rails, which allows the window to be retracted during flight,
- the rack and pinion mechanism, which drives the window in and out along the rails;
- the four-bar mechanism, which allows a priming pressure to be applied to the o-ring,
- and the o-rings, which act as seals between the cryostat and the ambient atmosphere.

The scientific leadership for Spider comes from experimentalist Prof. Barth Netterfield, who recently won NSERC’s Steacie Fellowship on the basis of science from two pioneering stratospheric telescopes, Boomerang and BLAST, and theorist Prof. Richard Bond, recently award the NSERC Herzberg Gold Medal and the Killam Prize in recognition of his outstanding work on predictions and analysis of the cosmological content of CMB measurements. The scientific promise of Spider and the high standard expected by the team was certainly motivational [1].
Work focused on the design of the four-bar mechanism and the o-ring seals in particular, due to the custom nature of the application. In choosing the linear guide rails and rack and pinion systems it was a matter of matching design requirements to product specifications. Analyses of the design were performed using CosmosWorks, a finite element software package.

At the outset of this investigation, a functional prototype existed for the window, but a redesign was desired due to concerns about the seal quality and repeatability.

Chapter 2 presents some background on the Spider experiment and the window design requirements. The motivation for this project, as well as the specific objectives for the design as set out at the beginning of the project, are then outlined in Chapters 3 and 4. The main portion of this report is presented in the two ensuing Chapters (5 and 6), which discuss the overall design and the specifics of each component, respectively. The concluding chapter (7) is a short summary of the work conducted and a discussion of possible improvements and suggestions for further investigation in the few months remaining before these components are finalized and constructed (the launch goal is 2011).
2. Background

Spider’s science goals are to detect for the first time primordial gravitational waves through their effects on the mm-wavelength polarization of the CMB. These gravitational waves are theorized to have been created during the first $10^{-30}$ seconds after the Big Bang [1]. By mapping the mm-length radiation, scientists hope to better understand the very early structure of the universe. Spider is a balloon-borne telescope currently being designed at the University of Toronto and several other institutions across North America.

Balloon-borne astronomy addresses the need to make observations beyond the interfering (or debilitating) effects of the earth’s atmosphere, and is a growing field. Compared to a space-based telescope, a stratospheric telescope has several advantages: (i) can be implemented at a fraction of the cost, (ii) development cycle much shorter, (iii) can be tested and redeployed, and (iv) offers a fertile training ground for students.

Balloon-borne telescopes have some gross familial similarities from one experiment to another (Figures 1, 2, and 3), consisting of (from top to bottom) a large helium balloon, a lengthy cable or flight train (including the parachute for recovery), a swivel mechanism (termed the “pivot”), and finally the gondola bearing the telescope.

The science payload, or “telescope,” is contained within a large, cylindrical “cryostat” and is mounted to the gondola. The cryostat actually contains six distinct detectors, each with its own window. The cryostat is needed to cool the detectors, which operate at mK temperatures, and has
a large volume to enable a long-duration flight of up to 30 days. The centre of mass of the experiment (not including the flight train and balloon) is dominated by the cryostat, which has a mass of approximately 1250 kg when full of liquid helium.

Spider’s total payload mass is capped at 2500 kg by the maximum lift of the helium balloon used. The payload weight falls into three categories: experiment (telescope), supporting structure (gondola and pivot), and ballast. The ballast hopper is suspended from the bottom of the gondola. Ballast is dropped during flight to counteract slow leaks in the balloon. An experiment with a lighter gondola frame (structure) will be able to carry more ballast and will therefore be able to maintain a longer flight. Given the expense of designing and flying such an experiment, the risk that it may only fly once, and the minute signals being sought, long flights are a high priority. The weight of the science equipment is determined by the goals of the mission and cannot be reduced, and so it is up to the design engineer to minimize the structural mass of the remaining equipment, including the windows, in order to maximize ballast.

2.1 Terminology and Component Descriptions

There are several basics, including simply terminology, that should be understood and agreed upon before going into a detailed description of the components.

First, let us differentiate between the windows that seal the cryostat and the window mechanism that provides the means to seal. In this report the term window will refer to either of the two plastic discs that are used in to seal the cryostat. Namely, “thin window” refers to the permanent, inner window, while “thick window” refers to the retractable one. The thin window is permanently in place to provide protection of the cryostat once Spider is at float altitude. The
low pressure differential between the cryostat and the ambient environment at float (2 kPa) allows for the fixed window to be quite thin. Drawing on the work of Lepo, a Teflon (PTFE) film of about 0.0508 mm is sufficiently strong while having favourable radiative load properties [2]. That is, PTFE does not greatly dissipate the power of the signal which the experiment is trying to detect.

The thick window is in place during ground-level operations, ascent, and descent. This window must withstand the greater pressure differential experienced under these conditions (roughly 1 atm). The radiative properties of the thick, movable window are less critical than for the thin, fixed window. However, it is important that the thick window not distort the beam as it enters the cryostat for ground-testing purposes. Lepo’s results suggest a High-Density Polyethylene (HDPE) window of thickness 20 mm will withstand the greater pressure differential while not affecting the beam appreciably [2].

The term “window mechanism” will refer to the overall system that enables the cryostat to be sealed. While colloquially the entire system has been simply called “the window,” that is not the case in this report.

Second, it is beneficial to establish a cylindrical coordinate system with reference to the cryostat. As shown in Figure 4, let us define the z-axis to be along the central axis of the cryostat, with the positive direction being towards the telescope’s target of interest. The origin should exist at the surface of the cryostat top-plate. The positive r-axis radiates outward from the central axis of the cryostat. Finally, let theta, \( \theta \), be zero at the 12 o’clock position when looking at the cryostat from a position on the positive z axis, with theta increasing in the clockwise (CW) direction. A second axis of rotation is defined to pass laterally through the centre of mass of the cryostat, allowing
the elevation of the telescope’s line of sight above the horizon to be measured. Let gamma, $\gamma$, be zero at the horizontal and increase to 90° at the zenith.

The gondola’s central function is to support the cryostat. The chosen structure is a space-frame construction (Figure 3). The cryostat will be able to rotate (tip in elevation) relative to the gondola by varying $\gamma$ from 0° to 58°. It is important that the window mechanism structure not interfere with the gondola as the cryostat is rotated in elevation (by varying $\gamma$).

Figure 5 shows the window mechanism with the significant components labelled. The thick-window frame carries the thick window in and out radially, driven by the rack and pinion gears. The rack gear is mounted to the rail of the four-bar mechanism. The thick-window frame slides along the rail on the linear guide. The thick-window frame makes contact with the thick-window base plate to protect the cryostat at ground level and during ascent and descent.

2.2 Thin and Thick Window Analysis

A series of finite element analyses (FEA) and theoretical studies were performed to verify the results of Lepo [2]. These only consider the strength and deflection requirements of the two windows, assuming her analysis of radiative properties to be sufficient. The analysis, presented in detail in Appendix A, suggests that Lepo’s analysis was sufficient.
3. Motivation

The success of Spider is dependent on the ability to detect minute signals. This requires, among other things, that there be as little power dissipation as possible. Thus, the retractable window design allows for two coverings to be used. At float altitude, where ambient pressure is low, a fixed, thin Teflon window is sufficient to maintain the internal vacuum of the cryostat. At ground level and during ascent and descent, the pressure differential between the cryostat and the ambient environment is large enough to require a thicker High-Density Polyethylene (HDPE) window. When the thick window is deployed there is significant power dissipation but minimal beam distortion, which allows for pre-flight pointing. The challenge, then, was to design a mechanism that could deploy and retract the thick window during flight.

The implications of failure of the window mechanism, whether failing to retract once at flight altitude or failure to deploy before descent, are large. Failure of the window to retract upon reaching float would mean that the detector in question would not function for the duration of the flight. Failure of the window to deploy before descent would cause catastrophic failure of all detectors. However, a deployment failure on descent would still allow for useful scientific observation.
4. Objectives

The serious implications of failure led to the creation of two goals set for the design. The first was that the window should consistently perform as designed. To that end, the design was continuously modified to become more robust. The second goal was, in the case of failure, to minimise the damage to the experiment. In balancing the two goals, the first took precedence. As an example, the final window design includes three two-detector window mechanisms as opposed to six single-detector mechanisms. It was judged that by combining neighbouring mechanisms the design could become more consistent, as well as reducing total mass.
5. Overall Mechanism Design

5.1 Mechanism Principles

The mechanism functions as follows, described from the retracted position as shown in Figure 6. When it comes time to deploy the thick window, the motor and gearbox provide torque to drive the pinion over the rack. The pinion, motor, and gearbox are mounted on the thick-window frame, which then moves radially inward. As the frame reaches the end of the rails, a peg on the frame engages the lever arm of the four-bar mechanism, shown in Figure 7. Up to this point the four-bar mechanism is in the open position, held up by a linear spring, as in Figure 8. The spring is necessary to overcome the weight of the thick-window frame, elevating the thick-window o-ring above the thick-window bottom plate, to which it will eventually seal (Figure 9). This elevation is needed so that the o-ring is not dragged along the bottom plate as the frame moves radially. As the frame pushes further inward, the four-bar mechanism closes, bringing the thick-window frame and bottom plate in contact with each other to create the seal, shown in Figure 9. To retract the thick window, the motor is driven backwards and the above steps are reversed.

The key to the window mechanism is the four-bar design. As described above, it is important to avoid shearing the thick-window o-ring along the base plate as the window is deployed and retracted. In a parallelogram-shaped four-bar mechanism, as is the case here, opposite sides always remain parallel. This means that the thick-window frame and base plate, which make up the two long edges of the mechanism, are always parallel. Thus, when the four-bar mechanism goes from open to closed the thick-window frame falls into place flush with the base plate,
creating an even seal across the o-ring. Chapter 6.3 describes the four-bar geometry in more detail.

The primary concern of the window mechanism design is to provide enough compressive force to create a full seal in the thick-window frame o-ring. Chapter 5.4 describes the overall analysis undergone to determine the forces needed.

5.2 Detailed Window Seal Description

This section describes the different components that make up the intricate seals of the thin and thick windows. Figure 10 shows a cross-sectional view of the two windows in the deployed position. Visible in this image are six separate o-rings, labelled A – F. Seal A represents the interface between the window mechanism and the cryostat faceplate. O-rings B and C are used to sandwich the thin window (not shown), creating the first barrier between the cryostat and the ambient pressure. Ring D seals the bottom of the thick-window bottom plate, which also contains a venting hole (not shown). Ring E is the thick-window frame seal, which is discussed in detail in Chapter 6.1.4. Finally, o-ring F serves to seal the bottom of the thick window to the thick-window frame.

5.3 Double Window Design

The original window mechanism was designed to seal only one detector at a time. Such a design required six separate window mechanisms, as seen in Figure 11. This design had the advantage of minimize damage incurred upon mechanism failure. That is, if one mechanism failed to open, five of six detectors would still function. However, it was thought that mechanism reliability
could be improved by creating window mechanisms that sealed two detectors at a time, as in Figure 12. The rationale behind this decision was that the weight and space saved by joining the neighbouring mechanisms could be used to make a stronger, stiffer, and more powerful design.

5.4 System-level Analysis

It was necessary to conduct a system-level analysis to determine the interactions between each of the four component groups. This information was needed to determine the specific design features of each component. Specifically, in order to specify the motor, gearbox, and rack and pinion components it was crucial to determine the forces required to satisfactorily compress the thick window’s o-ring when the deployed position. The connectivity of the components, described previously, called first for a step-wise analysis of the o-ring and thick-window frame, followed by the interaction between the frame and the linear guides. The information from these analyses allowed the forces in the four-bar mechanism to be determined. Finally, the four-bar forces were converted to forces on the rack and pinion and the torque generated by the motor and gearbox.

The specifics of each analysis are presented in Chapter 6. The results are presented here to illustrate the interactions between components and how the overall mechanism will perform. It was found from FEA that to satisfactorily compress the thick-window o-ring (0.5 mm compression) the thick-window frame must experience 154 N and 248 N downward forces at the back and front carriage points, respectively. These forces were used to specify the Mini-Rail guide system to be the MR12 model. Next, using the force results and further FEA, the four-bar mechanism forces were determined. It was found that a driving force of 174 N must be applied to the lever-arm in order to generate the forces necessary for o-ring compression. The driving force
was then used to specify the models of the motor, gearbox, and rack and pinion to be used. It was found that 1.3 Nm of torque was required. Model specifications are given in Chapter 6.
6. Component Specifics

6.1 O-ring System

The Spider window system requires six separate o-rings (Figure 10). Five of these (A – D, F) are static face seal applications, meaning that the seal is permanent. The last (E) is the thick-window o-ring, which is compressed at ground level, becomes decompressed at float, and is recompressed before descent. This is also a face seal, but has a dynamic aspect.

6.1.1 O-ring Principles

An o-ring seal is generated when the ring is compressed between two faces. The contact pressure between the ring and the faces determines the pressure differential that can be maintained across the seal. There are several factors to consider when designing a seal, the first of which is ring geometry. O-ring’s are specified by the inner diameter (ID) and cross-sectional width (W). Next, the retaining groove, known as the gland, must be designed. Figure 13 shows a typical gland design for the geometries used in Spider. Gland depth determines the amount of compression that the o-ring will experience, measured in percentage of W. Percent compression should range between 10 and 40 % [3]. When designing the groove it is important to seat the ring on the low-pressure side. In Spider’s case, all of the rings should be seated against the inner wall. A certain amount of stretch is acceptable when designing the groove. O-ring stretch should be no more that 5 % of the circumference [3].
6.1.2 Material Considerations

O-rings are available in a wide range of materials and grades. The temperature at float altitude was determined to be in the range of -50 to -40\degree C, requiring a material that maintains its elastic properties at low temperatures. Further, the vacuum application requires a material with high gas impermeability. Table 1 shows the properties of a number of materials. Nitrile was selected for its excellent low temperature and vacuum properties. Further, nitrile is resistant to lubricating oils, which rings may be exposed to in the operating environment. Environmental attack is not a consideration for Spider because of the short duration of the experiment. However, it is recommended that new o-rings be used for each flight.

Softer materials, measured in Shore A durometer, require less clamping pressure to create a good seal. This is desirable in the case of the thick-window seal, where lower clamping forces translate into lower motor requirements. Figure 14 shows the clamping pressure, measured in pounds per linear inch of seal, for various materials and cross-sections [3]. Low-temperature nitrile has a Shore A hardness of 40, which falls below the range given in Figure 14. However, a relation between Young’s Modulus and durometer is given by (1) and can be used to estimate clamping pressure [4].

\[ \log E = 0.0235 \times S - 0.6403 \]  

where E is in MPa and S is Shore A hardness.

A simple FEA test revealed that a 20 inch ID, 0.21 inch cross-section o-ring made of 40 Shore A nitrile would require 2.38 lbf/inch for 10\% compression, or roughly 670 N in total. This estimate corresponds well with the values one might expect when examining Figure 14. Given these
results, subsequent FEA tests were conducted with Young’s Modulus set at 1.349 MPa for nitrile.

6.1.3 Static Seal Specifications

The specifications of seals A – F are provided in Table 2. The ring and gland geometries are determined by examining the space available for the seal and then looking up the specifications in a table [3]. In choosing the ID it is important to consider the faces that make up the gland and the faceplate. Widths were chosen with consideration to plate thickness and the percentage of stretch that the ring would experience. The compression forces for these seals are provided by the permanent screws that secure the metal plates to the cryostat faceplate.

6.1.4 Thick-Window O-ring Specifications

The considerations for seal E are similar to those of the static seals. A notable difference, however, is the desire to minimize seal compression pressure, which is due to the power requirements of the motor. Materials with low durometer require less pressure to create a satisfactory seal. This is fortuitous for Spider because low-temperature nitrile has a Shore A hardness of 40, which is considered quite low. Further, lower clamping pressures are required for rings of smaller widths [3]. However, it was felt that a larger width would allow for greater disparity in faceplate displacement. Such disparity is caused by the fact that the thick-window frame is compressed only on one side of each seal, as in Figure 15. To balance these two considerations, a cross sectional width of 0.21 inches was selected, as suggested by the engineers at Apple Rubber Products [3].

Using the material property results of (1), a FEA model was established to determine the forces necessary for ring compression by the thick-window frame. A compression of 10 – 20 % was
recommended by Apple Rubber Products, corresponding to a faceplate displacement of 0.53 – 1.06 mm for the 0.21 inch (W) ring [3]. Figure 16 shows the results of the model, demonstrating a minimum compression of 0.97 mm when the frame is displaced 1.33 mm. This displacement corresponds with creating a flush contact between the bottom face of the frame with the top face of the plate below.

The results suggest that sufficient compression is attainable with the given design of the thick-window frame. Initial designs were found to be unsatisfactorily stiff, leading to an uneven compression of the o-ring, as in Figure 17. To create a stiffer design, while maintaining a low mass, two aluminum reinforcements were applied across the top of the thick-window retainer, as in Figure 18. These serve to increase the moment of inertia of the frame, just as in an I-beam.

6.2 Linear Guide Rails

A wide variety of linear guide rails are available from a number of manufacturers. Pacific Bearings makes a product called Mini-Rails (MR), which are designed for light-duty applications such as Spider [5]. The Mini-Rails are designed for carriages that do not require bearings. Instead, MR carriages are made of Frelon, a plastic, which has low-friction properties as it glides over the 6061-T6 aluminum rails. The advantage of the Frelon carriage over traditional bearing designs is that bearing are affected negatively by low temperatures. Frelon carriages are also lighter weight.

The design of the window mechanism is such that the rails are loaded laterally, as in Figure 19. The results of the thick-window frame and o-ring FEA model, discussed previously, demonstrated that forces of 154 N and 248 N at the back and front carriage mounts, respectively,
were needed to compress the thick-window o-ring, E. The Pacific Bearings line of Mini-Rails includes models MR7 to MR20, with the number designation corresponding to the rail width. Increasing sizes leads to increased height, which is not desirable given the tight space constraints on the rails and four-bar mechanism. Thus, the MR12 model was chosen for its suitable strength and minimal profile. Specifically, the MR12 can withstand lateral forces of 400 N per carriage and has an overall height of just 13 mm [5]. In order to spread out the load and provide a more even compression of the o-ring, it was decided to use two carriages rather than just one.

The cost of the MR12 guide rails, in lengths appropriate for the Spider application, is approximately $100 CAD per rail. Carriages are sold individually for approximately $30 CAD each. This corresponds to an overall cost of $960 CAD for the guide rail system for all six detectors. The Canadian distributor is Motion Industries.

6.3 Four-bar Mechanism

As described in Chapter 5, the purpose of the four-bar mechanism is to provide clearance for the thick-window o-ring, E, while the window is being deployed or retracted. Once again, the key to this mechanism is the parallelogram arrangement, shown in Figure 20. This design ensures that the thick-window frame and base plate remain parallel at all times.

Another key aspect of the four-bar design is that the motion of the frame is completely vertical at the time of contact of the o-ring with the base plate. As shown in Figure 21, this is achieved by setting the short arms of the mechanism to be horizontal at the point of contact. This is a desirable function because it further minimizes the lateral drag on o-ring. The range of motion of the four-bar mechanism is limited by a stopping block, so that the angle between the short and
long arms never exceeds $30^\circ$. This corresponds to a vertical translation of 15 mm between the open and closed position. According to the geometry of the thick-window frame, the o-ring will initially contact the base plate when the frame to base separation is 1.33 mm.

The natural tendency of the design is for the four-bar mechanism to be in the open position. This is because of the linear compression spring placed between the base plate and the frame. The spring has an outer diameter of 0.3 inches and a spring constant, $k$, of 7439 N/m. The spring is mounted such that it is compressed 7.92 mm at the point of closure, corresponding to a force of 59 N. While this force adds to the total torque that the drive motor must provide in closing the mechanism, it is necessary to keep the base plate and frame separated during retraction and deployment.

A FEA simulation was run to determine the bearing forces within the mechanism. The forces found for the linear guide rails were applied to the model to simulate the loads exerted by the o-ring seal. These forces were modeled as remote loads, as shown in Figure 22. The model accounted for the compression spring load as well. The rotary joints were represented as pins, which allow free rotation while limiting translation between the two faces. The end of the lever arm was limited in translation along the x-axis to simulate the load applied by the peg driven by the rack and pinion. This is the force that causes the four-bar mechanism to close. The results of the FEA study showed that force to be 174 N. The bearing shear forces (x and y) were shown to be on the order of 350 N each, leading to a resultant force of 495 N. The rotary joints will be made of 6061-T6 aluminum and commercially available acetal sleeve bearings. The acetal bearings are rated to 1000 N [6].
The design of the thick-window frame was previously described in Chapter 5.3, showing the stiffening strategy to ensure an even compression of the o-ring.

### 6.4 Drive System

The radial motion of the window mechanism is provided by the drive system. This system is made up of a rack and pinion gear, a motor, and a gearbox. The rack and pinion system was selected for its simplicity and compact design. Unlike linear actuators, the rack and pinion does not require extraneous radial space because the motor and gearbox are mounted perpendicular to the line of motion.

The results of the four-bar mechanism FEA suggest that a 174 N force is needed to seal the thick window. This is a relatively low load for typical rack and pinion systems. The lightest-duty racks available from Atlanta Drive Systems are suitable for loads up to 4448 N [7]. As such, a module 1 rack was selected with the smallest available pitch diameter of 15 mm [7]. The corresponding pinion gear has 15 teeth and a pitch diameter of 15 mm [7]. The respective model numbers for the Atlanta Drive Systems rack and pinion are 36 00 050 and 21 10 15 [7].

To achieve the 174 N force the motor and gearbox must deliver 1.3 Nm of torque to the rack and pinion. First, a gearbox was selected that can provide 1.8 Nm of continuous torque, with a maximum intermittent torque of 2.8 Nm [8]. The gearbox was selected to be able to provide greater torque than necessary to ensure complete compression of the thick-window o-ring. The specified gearbox has a ratio of 231:1 and is model number 143991 from Maxon Motor [8]. The motor was also selected from the Maxon line. The 283838 model bushless stepper motor has a continuous torque of 10.3 mNm [8]. The combination of the motor and gear ratio provides a
maximum of 2.379 Nm, which is within the tolerable torque of the gearbox. The combined price of the motor and gearbox is $310 USD [8]. Six sets are required, totalling $1830 USD.
7. Conclusion

The success of the Spider balloon-borne telescope is contingent on the window mechanism design for several reasons. Failure of the system can occur in two ways. First, the mechanism can fail to open. This would result in the loss of two detectors during flight. Second, the mechanism can fail to seal before or during descent. This would lead to pressurization of the detectors, resulting in significant damage. Thus, the goal of this project has been to minimize the chance of either failure method. To that end, the prototype design was modified to cover two detectors at once. This allowed for a stronger, stiffer, and more powerful mechanism to be constructed in the allotted space.

The window mechanism is designed around a four-bar mechanism, which creates the thick-window seal by bringing the frame and base plate in contact. The frame is moved radially along linear guide rails, powered by a drive system made up of a rack and pinion, a motor, and a gearbox. The physical seal is created when the o-ring is evenly compressed. Nitrile was selected as the o-ring material because of its low-temperature performance, its gas impermeability, and its resistance to damage by lubricants.

The majority of the mechanism is made up of custom-machined components, all made of 6061-T6 aluminum. However, several off-the-shelf parts were specified. The guide rails were chosen because of their simple design and high degree of accuracy. The rack and pinion system was selected because of its simplicity and compact package. Polymer sleeve bearings were selected because of their easy of installation and excellent performance under the given loads.
While the design of the mechanism is complete, there remains the task of designing a venting system. This system must be able to equalize pressure between the inter-window space and the ambient environment. In addition, shop drawings must be completed so that a prototype can be made. Stringent testing is recommended given the high cost of failure for the window mechanism. The remaining tasks are estimated to be complete by December 2010, which puts Spider in a good position to meet its target launch date.
References

Figure 1: The BLAST telescope on the launch pad. Of interest is the pivot, which connects the gondola to the flight train via the suspension cables. The Spider geometry will be similar.
Figure 2: The BLAST telescope during ascent showing the flight train and balloon.

Figure 3: An early rendering of the Spider experiment. The gondola is shown as white tubing while the cryostat is shown in blue.
Figure 4: The cryostat showing the reference coordinate system.

Figure 5: The double window mechanism, labelled. A – thick window, B – rails, C – rack and pinion, D - four-bar mechanism, E – thick-window frame, F – thick-window base plate.
Figure 6: Axial view of the window mechanism in the open position.

Figure 7: The thick-window frame’s peg in contact with the four-bar lever arm.
Figure 8: Compression spring keeps the four-bar mechanism in the open position.

Figure 9: Thick-window frame and base plate in contact to create seal.
Figure 10: Cross-sectional view of thin and thick windows.

Figure 11: Original six-window design.
Figure 12: Double window design.

Figure 13: A typical gland profile for the o-rings.
Figure 14: Compression force per linear inch of seal (after [3]).
Figure 15: Each thick window is only compressed from one side.

Figure 16: Thick-window frame displacement when stiffening straps are included. Straps are shown and are included in the simulation.
Figure 17: Thick-window frame displacement when no stiffening straps are included. Straps are shown but not included in the simulation.
Figure 18: Stiffening straps (pink) are shown on top of the thick-window retainer (yellow).

Figure 19: Lateral loading of the guide rails, as seen from above the carriage.
Figure 20: The parallelogram arrangement of the four-bar mechanism can be seen from this side view.

Figure 21: Four-bar mechanism in the closed position. Note the horizontal position of the short arms.

Figure 22: Slide rail modeled with remote loads and pin connectors.
Figure 23: Thick window displacement under 101 kPa.
Figure 24: Thick window stress under 101 kPa.
## Tables

Table 1: Properties of select o-ring materials (After [3]).

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<td>Butyl</td>
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<td>Good</td>
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<td>Poor</td>
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<tr>
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<td>-65 to +177</td>
<td>Excellent</td>
<td>Fair</td>
<td>Poor</td>
</tr>
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<td>Fair</td>
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<td>Excellent</td>
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<tr>
<td>Silicone</td>
<td>-104 to +232</td>
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<td>Poor</td>
<td>Poor</td>
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Table 2: O-ring specifics.

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<td>B</td>
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<td>C</td>
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<td>Nitrile</td>
</tr>
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<td>D</td>
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<td>19.8</td>
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<td>F</td>
<td>388</td>
<td>19.21</td>
<td>1.3 %</td>
<td>Nitrile</td>
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</tbody>
</table>
Appendix A

Thin and Thick Window Analysis

A.1 Thin Window Analysis

The Lepo report did not provide specific material properties for the thin window [2]. A search of commercially available Teflon films suggests that 0.05 mm film has a tensile strength of 33 MPa [9]. Poisson’s ratio was taken as 0.46 Error! Reference source not found.. These properties were applied to a theoretical formulation of the window’s deflection [2]. The outer edge was constrained from moving while 2000 Pa was applied to the surface of the film, according to (2).

\[
\sigma_{\text{max}} = \frac{3}{4} \frac{pa^2}{t^2} 
\]  

Solving (2) for pressure, \( p \), and using the failure strength of 33 MPa it was found that the window would fail at 2200 Pa. This matches well with experimental data obtained by Lepo, which led her to select the 0.05 mm film in the first place [2].

Unfortunately, thin-plate mechanics are not applicable to this case due to the large displacement of the film [11]. (3) and (4) yield meaningless displacement data, assuming a Young’s Modulus of 412 MPa Error! Reference source not found..

\[
W_{\text{max}} = \frac{p}{64t^2} 
\]  

\[
D = \frac{Et^3}{12(1-v^2)}
\]  

(3)  

(4)
A.2 Thick Window Analysis

A FEA model was used to measure deflection and stress in the thick window. As described by Lepo, it was modeled as HDPE, with a modulus of 1.07 GPa and a tensile strength of 22.1 MPa [2]. Figure 23 shows the deflection of the thick window to be a maximum of 6.58 mm at the centre of the disc. This model better represents the actual boundary conditions of the thick window than does the Lepo model [2]. Rather than simply fixing the outer edge, the model clamped the top and bottom of the window around the edge, allowing for the edge of the disc to displace radially. This allows for the possible failure mechanism of disc pop-out to be tested. However, results showed that this is not an issue in this case, with the edges displacing only a fraction of a mm.

The stress distribution is also better represented in this model. Figure 24 shows a stress concentration around the lip of the thick-window frame. Peak stresses reach a value of 9.34 MPa, which is well below the failure stress for HDPE.

A.3 Conclusions

The analysis of the window materials suggests that Lepo’s analysis was sufficient. It would be beneficial to further test the deflection of the thin window once a prototype is constructed. It should be easy to change thin window films, in thickness or material, without altering the overall design. Any changes to the thickness of the thick window would require changes to the thick-window retainer but would be relatively easy to accommodate.