Mars Science Laboratory Differential Restraint: The Devil is in the Details

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Abstract

The Differential Restraint, a mechanism used on the Mars Science Laboratory (MSL) rover to maintain symmetry of the mobility system during the launch, cruise, and entry descent and landing phases of the MSL mission, completed nearly three full design cycles before a finalized successful design was achieved. This paper address the lessons learned through these design cycles, including three major design elements that can easily be overlooked during the design process, including, tolerance stack contribution to load path, the possibility of Martian dirt as a failure mode, and the effects of material properties at temperature extremes.

Introduction

The Differential Mechanism, a series of linked, passive pivots, is a component of the Mars Science Laboratory's rocker-bogie mobility design. The differential acts as a motion reverser for the suspension system. The Differential Restraint was designed to prevent the rotation of the Center Differential Pivot (see Figure 1) of the Mars Science Laboratory mobility system during the launch, cruise, and Entry Descent and Landing (EDL) phases of the MSL mission.

The Mars Science Laboratory is JPL's next generation Mars Rover, known to the public as "Curiosity". The mission launched on November 26, 2011 on an ATLAS V rocket and is scheduled to land on Mars on August 5th, 2012. The rover mobility system incorporates the heritage rocker-bogie suspension design which was invented at JPL for the first Mars rover missions. The large size of the rover, approximately nine feet in length, requires the mobility system to be folded during the cruise stage of the mission. During the Skycrane phase, in which the rover is lowered to the ground via bridles and a descent stage (see Figure 2) the rocker arms of the mobility system are released from their stowed positions and allowed to rotate about passive pivot joints until the four rockers latch in their ready-for-touchdown positions. The Differential Restraint is the last of the mobility restraints to be released, via a pyro, approximately 2 seconds before rover touchdown.

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Proceedings of the 41st Aerospace Mechanisms Symposium, Jet Propulsion Laboratory, May 16-18, 2012

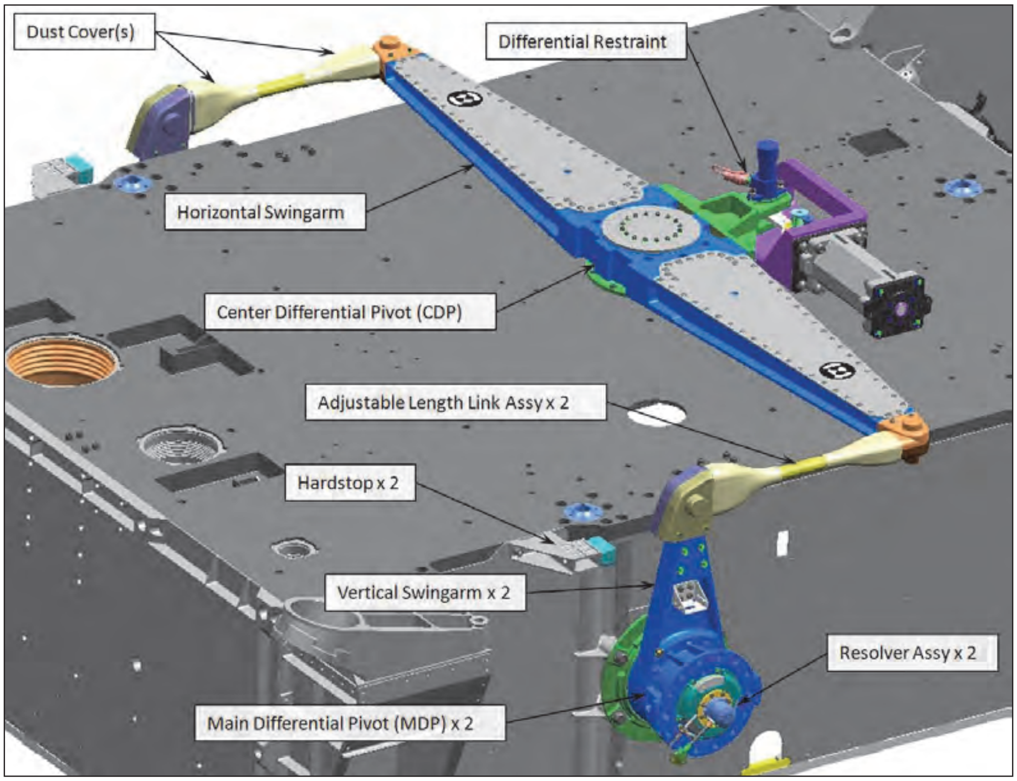

Figure 1: Key Components of the MSL Mobility Differential Suspension System

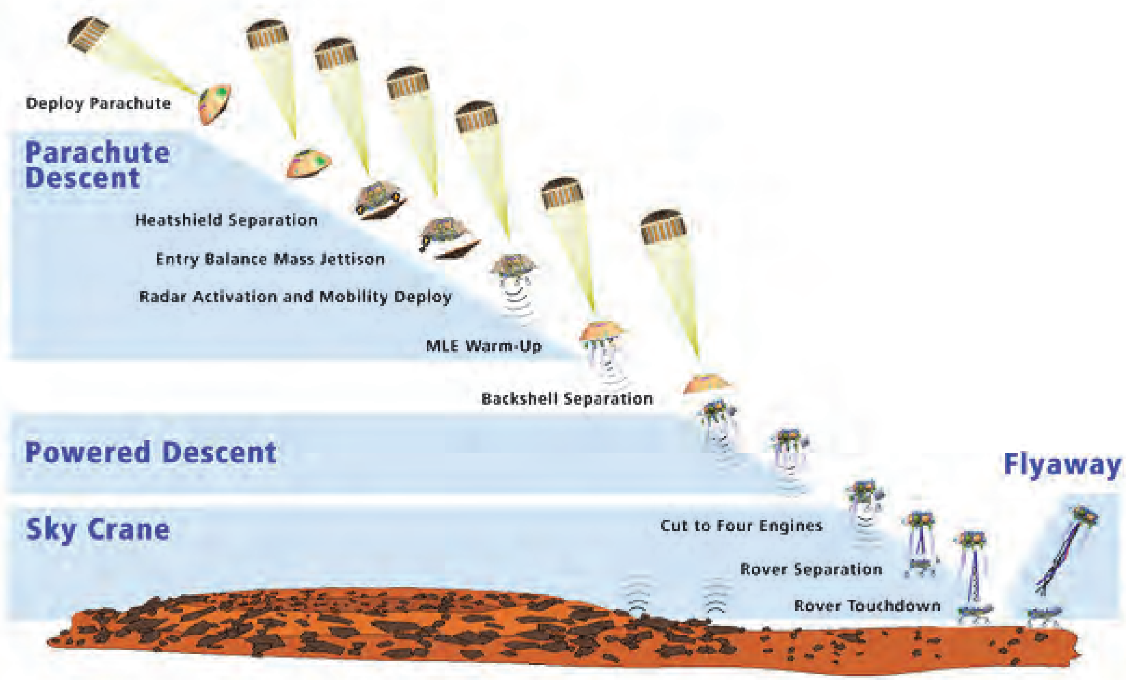

Figure 2: MSL Entry Descent and Landing Overview Diagram

Differential Restraint Design Development and Description

The Differential Restraint is a passive mechanism which relies on the compression of a series of Belleville Springs to limit the degree of rotational motion allowed at the Rover Mobility Center Differential Pivot during the Cruise and EDL phases, as well as to absorb and dampen the energy associated with the violent mobility deploy and latch events. Since the rover mobility system is also the vehicle's landing gear it is critical that the 6 wheels are in, what is termed, their "ready-for-touchdown" state, or as close to 6-wheels flat as possible when the vehicle impacts the Martian surface. The Differential Restraint enables this goal by limiting the rotation of the passive center pivot joint, known as the Center Differential Pivot (CDP), to a small angle, keeping the two sides of mobility, port and starboard, nearly balanced. Once the chassis dynamics, namely, chassis pitch and roll from the effects of the mobility deploy and latch events, have settled out the Differential Restraint is released, via a pin-puller pyro just prior to rover touchdown.

Design History

The driving load case for the Differential Restraint is derived from the near-simultaneous deployment (and end of travel latch events) of the mobility rockers during the Skycrane phase of EDL. In the event of actual simultaneous latch events of the port and starboard sides, large shear loads are expected at the CDP. Alternatively, if the deployment and latch events are staggered from port to starboard large moments are created about the CDP, which the Differential Restraint must absorb. The original Differential Restraint design was a rigidly pinned interface between the externally mounted mobility Differential and the top deck of the rover chassis which prevented nearly all motion, except for the small amount of slop between the pin puller pin and interfacing monoball (see Figure 3).

Approximately one year after the mobility Critical Design Review the increasing maturity of the ADAMS dynamic model produced a drastic increase in the predicted moment and shear loads at the CDP. This had the effect of creating significant negative margins in the original piece parts of the Differential Restraint; including the AerMet pin of the 3/8" Pyro, the shear pins and bolted joint which interfaced to the Rover top deck, as well as the Rover top deck itself. In late 2008 it was concluded that a completely new restraint design was required. The new design was to be analyzed, built, tested and integrated before the (then) planned 2009 launch. In addition to the tight timeline constraints, the re-design effort was further complicated by the fact that the hardware interfaces were already fixed, the available space was incredibly limited, and the 3/8" pin puller was the only feasible option for pyro devices.

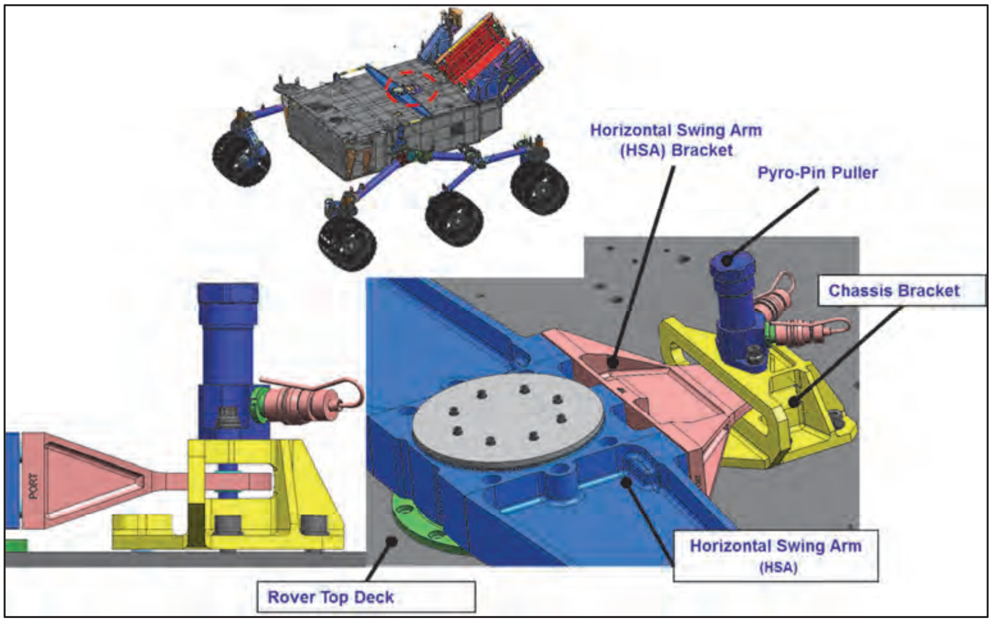

Figure 3: Original, Rigidly Pinned, Differential Restraint Design Concept

Since the rover top deck interfaces were already built, and the pyro device was already qualified and delivered the new design had to find a means by which the loads that the Center Differential Pivot would impart on these critical interfaces could be significantly decreased. The new design relied on the principle of applying a (theoretical) torsion spring at the center of the Differential Pivot which would absorb the large impact energy. After running idealized dynamics simulations, a spring rate of approximately 200,000 $^{N\text{-}m}/_{rad}$ was selected as the stiffness that would reduce the impact loads enough to prevent negative margins in mobility and chassis hardware but still restrain the mobility system from excessive motion during launch, cruise and the initial stages of EDL enough to be able to ensure that the mobility system was in its "ready-for-touchdown" state in time for landing. The new restraint was designed to accommodate the worst case combined loading event as predicted by a 2000-run Monte Carlo ADAMS dynamic model of the MSL EDL system.

Implementing a torsion spring at the Center Differential Pivot (CDP) was impossible due to packaging and size issues so it was decided that the Differential Restraint would use a 3-bar crank-slider approach to turn rotation into linear actuation (see Figure 4). Combinations of large Belleville washers in series and parallel were implemented to achieve the extremely high spring rate that was required in the limited space available. A rod, known as the Spring Plunger, was used to actuate the two stacks of Bellevilles. One stack is compressed from positive moments while the opposite stack is compressed by negative moment loads at the CDP (see Figure 5). The Spring Plunger, made from Titanium, was threaded at one end to allow it to be joined to a Clevis, which in-turn was connected to a Linkage with a Monoball in each end (see Figure 6 and Figure 7).

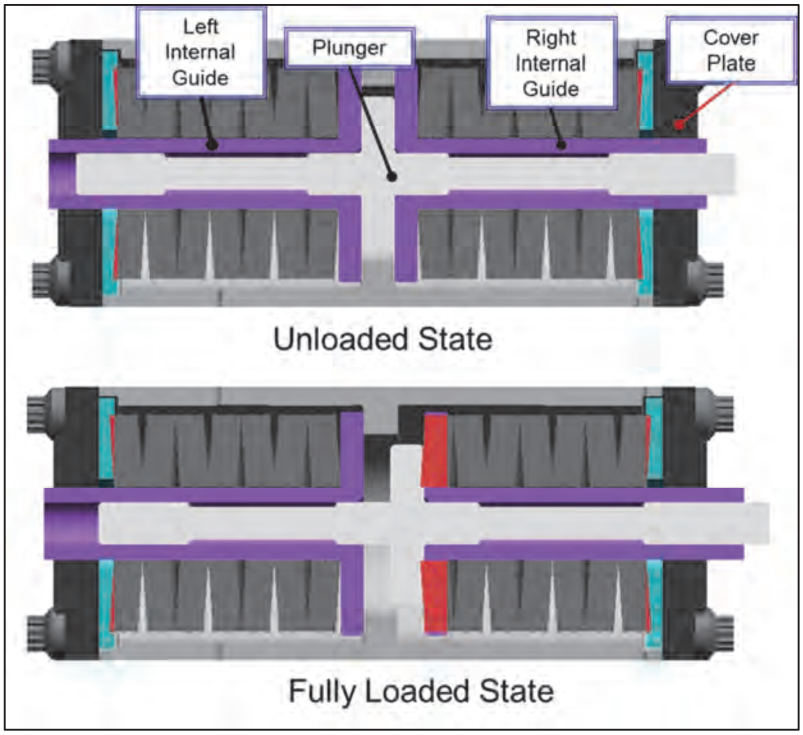

Figure 4: Crank-Slider Diagram of Differential Restraint

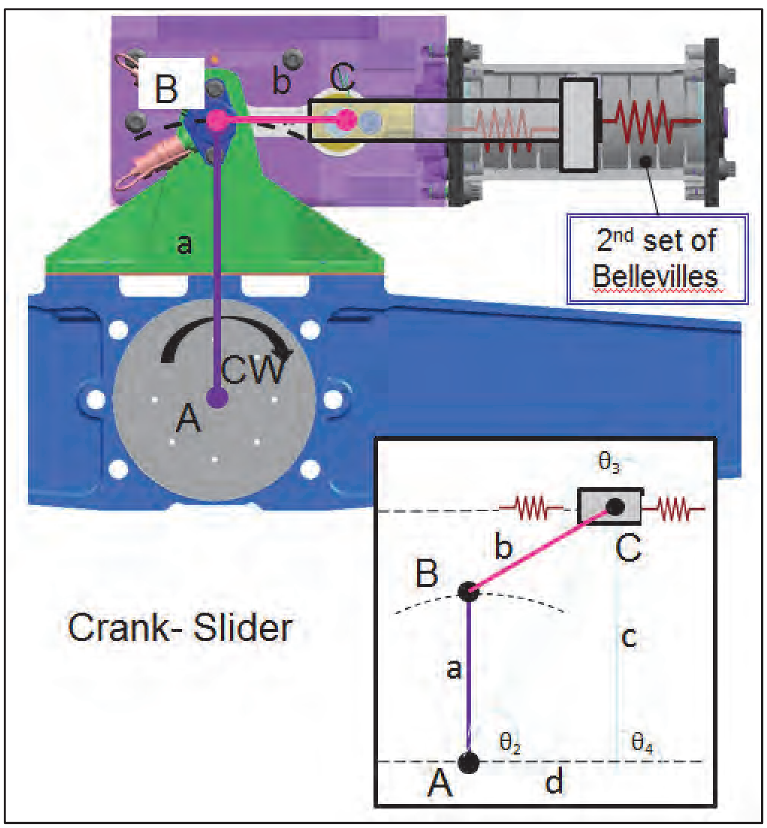

Figure 5: Belleville Spring Cross-Section Loaded and Unloaded States

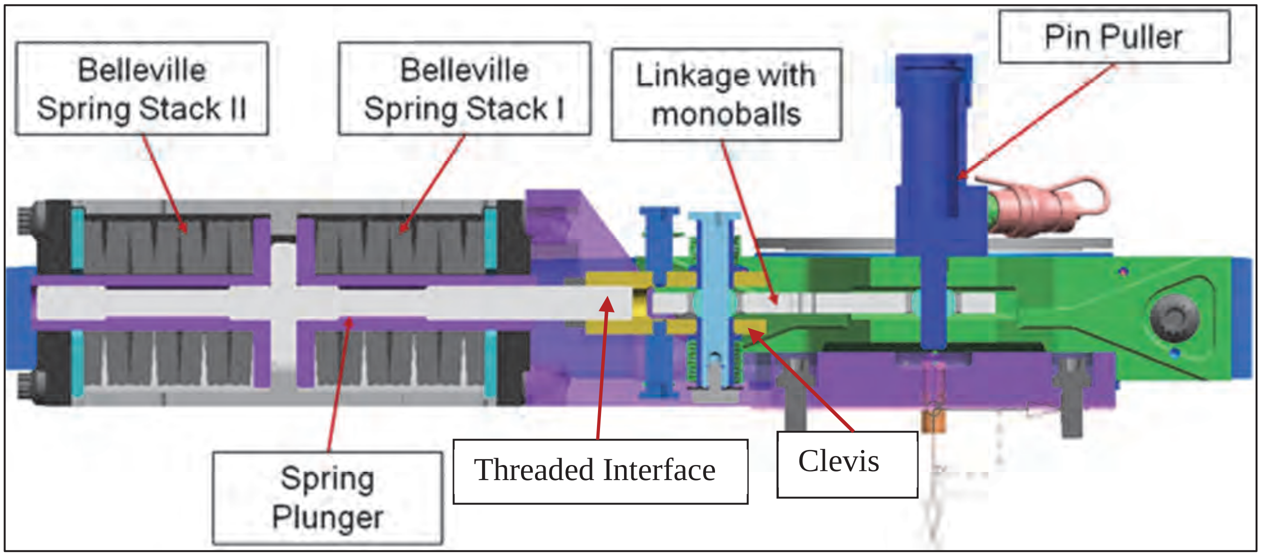

Figure 6: **Cross Section of Differential Restraint Mechanism**

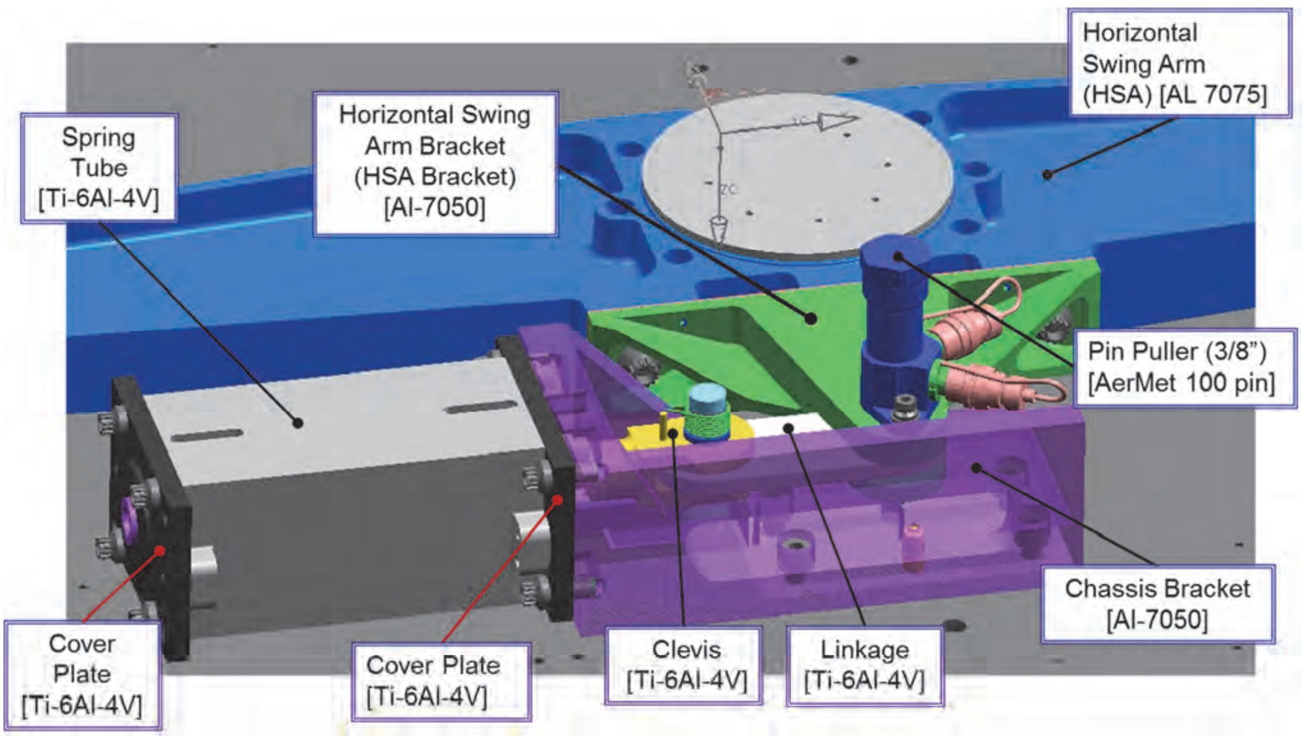

Figure 7: Differential Restraint Mechanism Nomenclature Overview

Tolerance Stacks and Stress Analysis

<u>Test Failure</u>

With the delay of the MSL launch the new Differential Restraint hardware was built but the planned component level testing of the hardware was delayed in an effort to conserve funds. In July of 2009, the Differential Restraint was integrated to the mobility differential system (see Figure 1) on the Dynamic Test Model of the MSL rover, in preparation for the differential system static test. With the restraint in place, the test was designed to load the entire differential system and generate a large moment load about the

Center Differential Pivot by pushing on an additional lever arm connected at the Main Differential Pivot Interface. At 90% of the full test load the measured load began to drop rapidly and an audible snap was heard. Upon investigation it became apparent that the Spring Plunger had catastrophically failed at the point of thread termination at the interface between the Spring Plunger and the Clevis (see Figure 8).

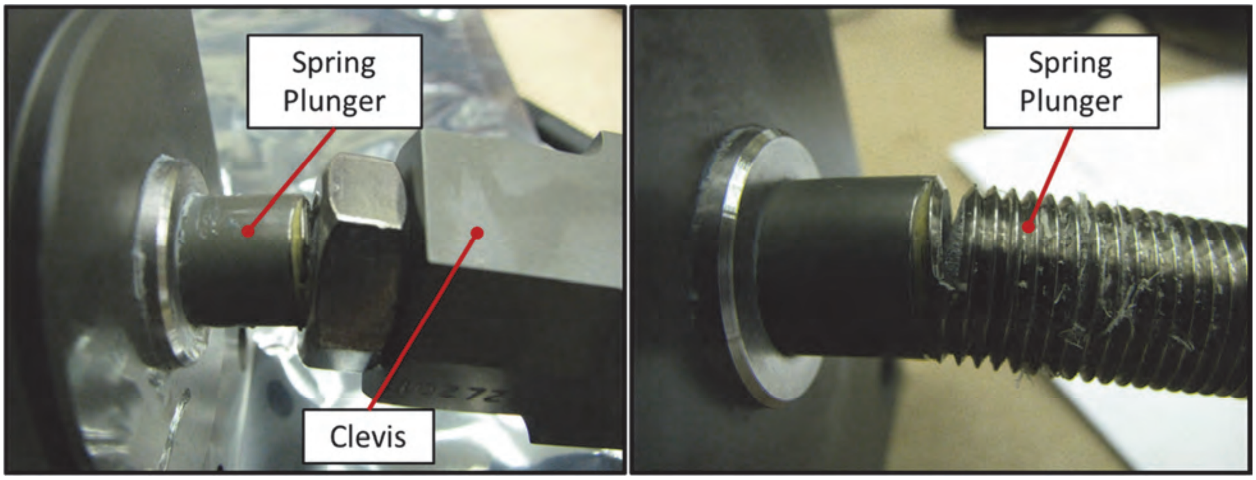

Figure 8: Failed Differential Restraint (Left) Disassembled from Clevis (Right)

<u>Determining the Cause of Failure</u>

Since the component level tests of the Differential Restraint had been skipped a wide array of failure causes existed that needed to be examined. Linear sliding mechanisms are often avoided at JPL on principle. This fact made the theory that one of the sliding surfaces between the Spring Plunger, Internal Guides, and Bellevilles had jammed, thereby preventing actuation of the mechanism, a prime suspect. Another theory was that the Spring Plunger, which was made from Titanium, had been made by a fraudulent vendor, and that imperfections in the material caused the crack initiation. The lack of a thread relief callout on the Spring Plunger drawing was also considered as a possible cause of failure, as was the large bending load on the Spring Plunger.

Failure Theories:

- Jammed sliding surfaces
- Fraudulent Titanium
- Improper thread termination on Spring Plunger
- Excessive bending load on Spring Plunger

The leading failure theory, that the sliding surfaces had jammed and prevent motion, was disproven in two ways. First, a line of Braycote, the lubricant used inside the Belleville Spring Assembly, was left on the Spring Plunger from where it had entered the Belleville Spring Assembly. By measuring the distance between the grease marks left on the Spring Plunger, it was possible to approximate the distance traveled between the start of test and the time of failure to be 5.46 mm (0.215 inch). At the load applied at the time of failure the total compression of the Belleville Spring stack was predicted to be 5.59 mm (0.220 inch). The close proximity of these values was strong evidence that the Spring Plunger did in fact slide along the Internal Guides and compress the spring stack as intended. Furthermore, the shape of the load deflection curve measured during the test exactly matched the prediction (see Figure 9), including the inflection point where the stiffness of the Belleville Stacks drops by one-half due to only one stack being compressed after the initial motion during which both stacks are in the load path. If the mechanism had jammed the applied load versus displacement would have spiked significantly rather than following the predicted linear line before dropping off when the Spring Plunger yielded and eventually snapped.

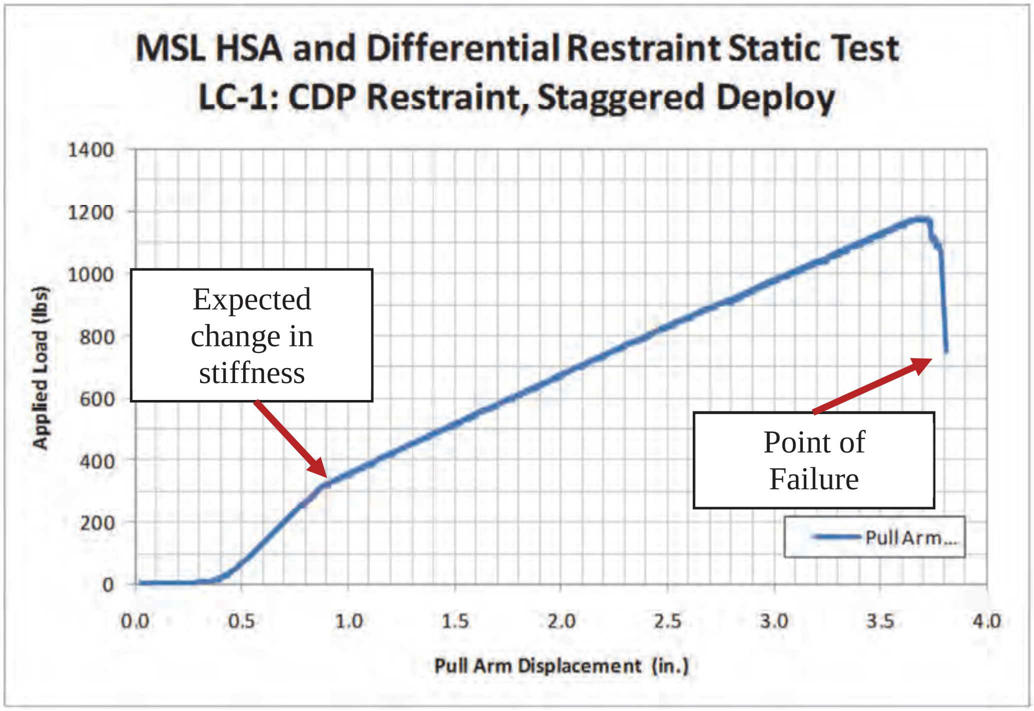

Figure 9: Applied Load vs. Pull-arm Displacement measured during failed Static Test

Upon inspection of the material certs for the Spring Plunger it was confirmed that the material was from Western Titanium. However, upon SEM inspection of the failed threads (see Figure 10) it was determined that the failure was ductile and there was no evidence that a material imperfection initiated the crack.

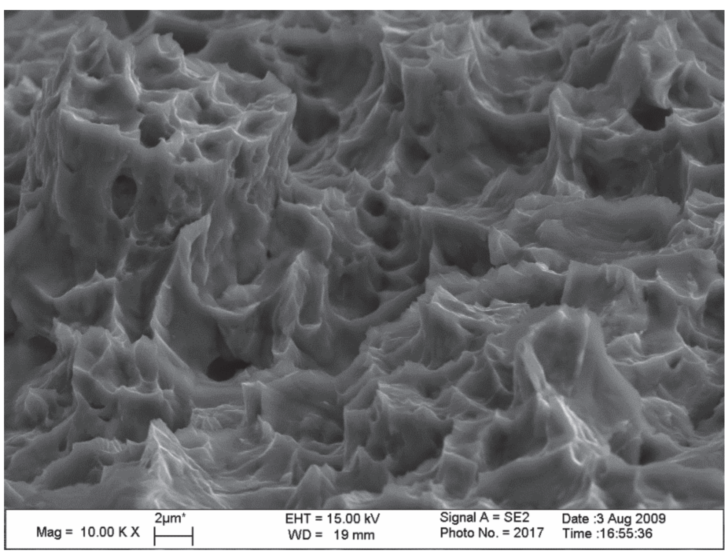

Figure 10: 10kX magnification of Spring Plunger Fractured Surface

<u>Bending Loads</u>

The Spring Plunger was originally analyzed for a bending load which would be applied as the Center Differential Pivot rotated. Since the Differential Restraint mechanism was designed to limit the rotation of the Center Differential Pivot to ±2°, the total misalignment between the line of action of the Linkage and the Spring Plunger was calculated by assuming a worst case rotation of the CDP to be 5°. If the CDP were allowed to rotate 5 degrees the misalignment between the line of action of the Linkage and the

Spring Plunger axis caused by this rotation would be 0.5°. With these assumptions the Spring Plunger was shown to have very small, but positive, margins of safety at the root of the threads. What was thought to be a very conservative analysis, (doubling the expected rotation of the CDP) was in fact not nearly conservative enough.

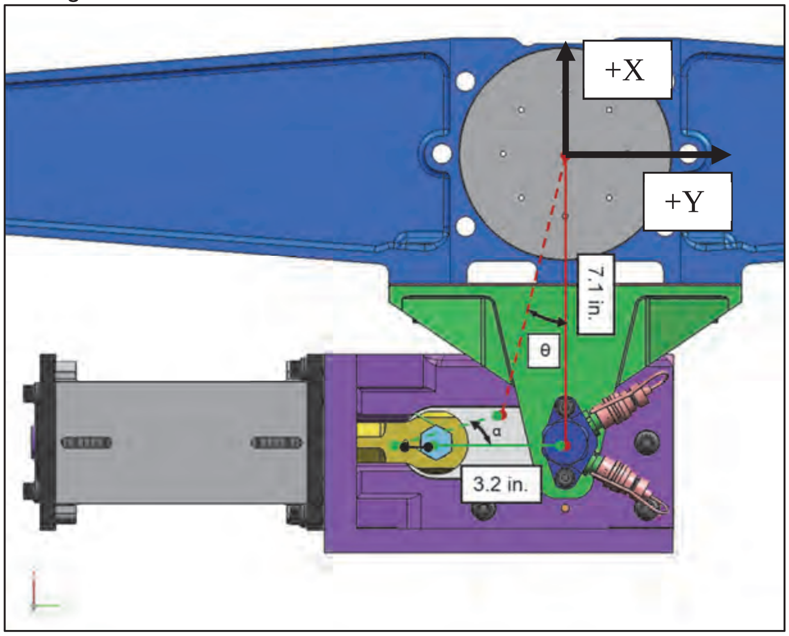

Figure 11: Line of action misalignment due to rotation of Center Diff. Pivot

In the post failure analysis the total angular misalignment between the Linkage force vector and the Spring Plunger was further scrutinized. It quickly became apparent that the misalignment caused by the rotation of the CDP was actually one of the smallest contributors to the overall misalignment. More significant misalignments were due to piece part misalignments due to imperfect shimming of the mechanism, the vertical slop of the Linkage in the mouth of the Clevis and Horizontal Swingarm Bracket, the slop of the Center Differential Pivot in its own bushings, and the elastic deformations of the Chassis Bracket and the Rover Chassis top deck. The maximum possible misalignment was calculated to be 2.2 degrees (see Figure 12), which was over four times what was initially accounted for in the Spring Plunger stress analysis.

Angular Misalignment

(Include RSS where applicable)

Misalignment Source		ΔX [in.]	ΔZ [in]	ΔRX [°]	ΔRZ [°]
1.0	1.1 Sprng Stack Deflection (.215")	0.00458			
	1.2 HAS Shim	0.02			
	1.3 Chassis Bracket Shim		0.02		
	1.4 Monoball to HSA		0.0233		
	1.5 Monoball to Clevis		0.0094		
	RSS	0.020518	0.03211		
	1.0 RSS_X_Z	0.03811			
2.0	2.1 CDP Translation	0.022	0.013		
	2.1 RSS X_Z	0.02555			
	2.2 CDP Rotation about X/Y		0.0448		
3.0	3.1 Chasis Bracket Elastic Deformation (8443 lbf)	0.0031	0.0053	0.629	0.0822
	3.1 Sum_X_Z	0.0084			
	3.2 Top Deck Elastic Deformation (8443 lbf)	0.0045	0.0290		
	3.2 Sum_X_Z	0.0335			
	3.3 Clevis Elastic Deformation		0.0188		
	3.4 Spring Plunger Elastic Deformation		0.0442		

RSS (1.0,2.1,3.0)	0.0749	0.6290	0.0822
Angular Misalignment (1.0,2.1,3.0)	2.0614	0.7112	

RSS (1.0,2.2,3.0)	0.0834	0.6290	0.0822
Angular Misalignment (1.0,2.2,3.0)	2.2154	0.7112	

Figure 12: Differential Restraint Angular Misalignment Calculations

Recovery From Failure

Immediately following the failure of the Differential Restraint many senior engineers began taking a closer look at the restraint design. They highlighted the fact that the restraint poorly implemented the concept of a crank-slider because the point at which the crank grabbed the slider was not at the center of the slider but rather it was offset- the link between the Clevis and the Spring Plunger was at the end of the Spring Plunger rather than the middle of the Spring Plunger (see Figure 13). Many engineers advocated redesigning the restraint to properly implement the theory of the crank slider; however limited schedule and funds made a complete redesign impractical. Instead, the design concept and most of the original hardware was kept intact, but key components were changed from Ti-6Al-4V material to significantly higher strength steel. By remaking two components and modifying two additional ones it was possible to quickly and relatively cheaply make the restraint design work.

Using the interaction formula:

$$R_C + \sqrt[3]{R_S^3 + R_b^3} = 1$$

$$R_c = ratio\ of\ compressive\ stress\ to\ yield\ /ultimate\ stress$$
$$R_s = ratio\ of\ shear\ stress\ to\ yield/ultime\ stress$$
$$R_b = ratio\ of\ bending\ stress\ to\ yield\ /ultimate\ stress$$

it was possible to predict the allowable side load, and hence the misalignment angle between the Spring Plunger and Clevis that a steel Spring Plunger could withstand, if the Flight Limit Load was applied axially. After calculating the misalignment angles that would produce yield and failure for a Spring Plunger made from steel it was decided that the values calculated were too close to the worst case possibilities predicted by the tolerance analysis to make simply changing the material a comfortable approach. Therefore, in addition to changing the material, the Spring Plunger diameter and thread diameter was increased. With the larger Spring Plunger diameter, and the material swap from Ti-6al-4v to steel, the new Spring Plunger would be able to withstand 4.24° and 4.29° misalignments before yielding and ultimately failing respectively. Additionally, a thread relief was added to the Spring Plunger thread termination (see Figure 14) to prevent excessive stress concentrations.

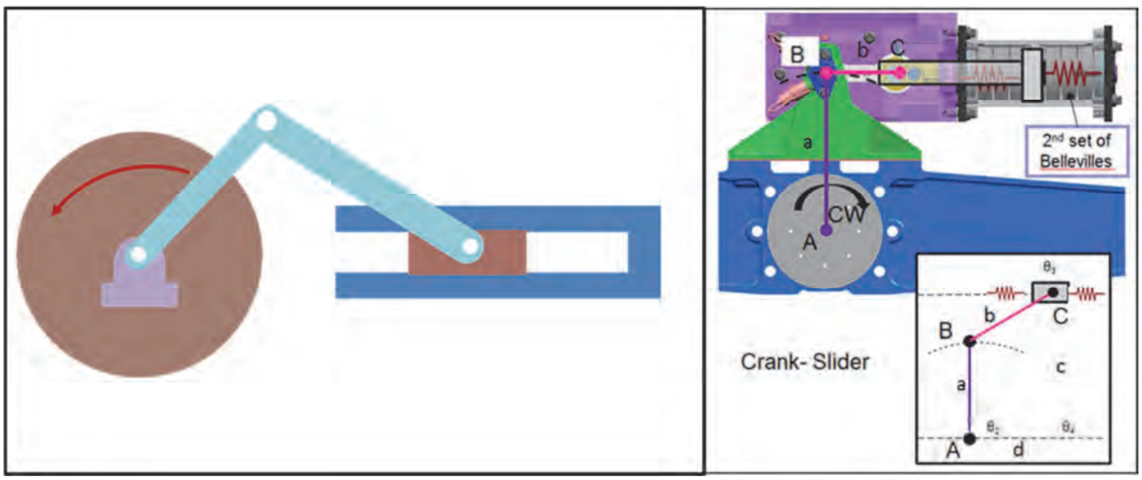

Figure 13: Comparison of “proper” crank-slider design methodology vs. Differential Restraint Design

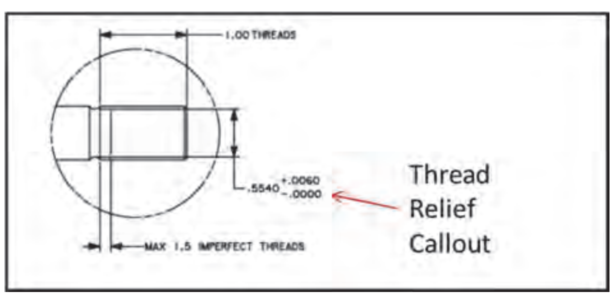

Figure 14: Spring Plunger Thread Termination Callout

<u>Key Findings:</u>

- Tolerance stacks and shimming procedures affect load paths- Make sure to account for this in stress analysis
- The location of pivot joints in kinematic mechanisms is extremely important- Failing to follow best practices early-on can cause unexpected difficulties down the road
- Proper thread relief callouts on drawings is essential for limiting the effect of stress concentrations

- Although not always the most elegant design approach- making a design "good enough" rather than ideal can be a more effective approach for saving time and money

Lubrication and Vacuum Bake

Since it was a concern that the properties of the lubricant used in the Differential Restraint, Braycote 601EF, could be altered by vacuum bake, the vacuum bake out, required for all MSL hardware to meet contamination and planetary protection requirements was performed prior to performing life and component level testing on the Belleville Spring stacks. Upon removing the Differential Restraint components from a 50 hour, 110°C vacuum bakeout an oily substance was noticed on the bottom surface of the mechanism. Upon chemical analysis of the substance it was determined that it was the lubricant Braycote 601EF. According to the Braycote 601EF product data sheet, the temperature range for Braycote 601EF is -80°C to 204°C, however the viscosity at these temperatures had been overlooked. At a temperature of 99°C Braycote 601EF has a viscosity of 45 cST, which is approximately midway between the room temperature viscosities of Honey (73.6 cST) and Olive Oil (24.1 cST). When viewed in this context it is not surprising that the Braycote flowed and leaked out of the assembly when baked at 110°C.

The Differential Restraint, Belleville Spring Assemblies were weighed before and after entering vacuum bake. With the knowledge of the decrease in weight due to vacuum bake and the knowledge of the mass of Braycote originally in the assembly it was determined that the Engineering Model and Flight units lost between 10 and 12% of their original Braycote lubricant due to the leak. Despite the loss of lubricant the stiffnesses of the Belleville Spring stacks measured during thermal characterization testing were well within the range of predicted and acceptable values and it was deemed unnecessary to add replacement lubricant to the assemblies.

By exposing the potential for Braycote to leak from the assembly the unfortunate leak that occurred during vacuum bake turned out to be a blessing in disguise, in that it focused attention on a failure mode that would have otherwise been missed. Due to the hardware's proximity to critical lenses for imagers on the Rover, focus was shifted away from the concern over the Belleville spring rate which was vetted through testing, to how to prevent the Braycote from leaking during flight and possibly contaminating other sensitive hardware. The actuation of the Spring Plunger through the Belleville Spring Assembly and its ability to squeeze Braycote out of the Belleville Spring Assembly like toothpaste became a concern. This problem was solved by adding a cover over the end of the Belleville Spring Assembly to catch any Braycote squeeze out (see Figure 15). Kapton shields were epoxied over the witness holes in the Belleville Spring Assembly (see Figure 15) to close out those leak paths as well. With these new covers in place the venting ratios of the Belleville Spring assembly came under scrutiny. A balance was struck between closing vent paths and preventing the Kapton tape from being blown off during depressurization by poking pin holes in the Kapton covers.

<u>Key Findings:</u>

- Just because a material is advertised to work over a specified temperature range doesn't mean it will work the same over the specified temperature range-make sure to understand all key material parameters over the expected temperature range to avoid surprises
- Consider "Band-Aid" solutions carefully while fixing one problem to avoid creating another

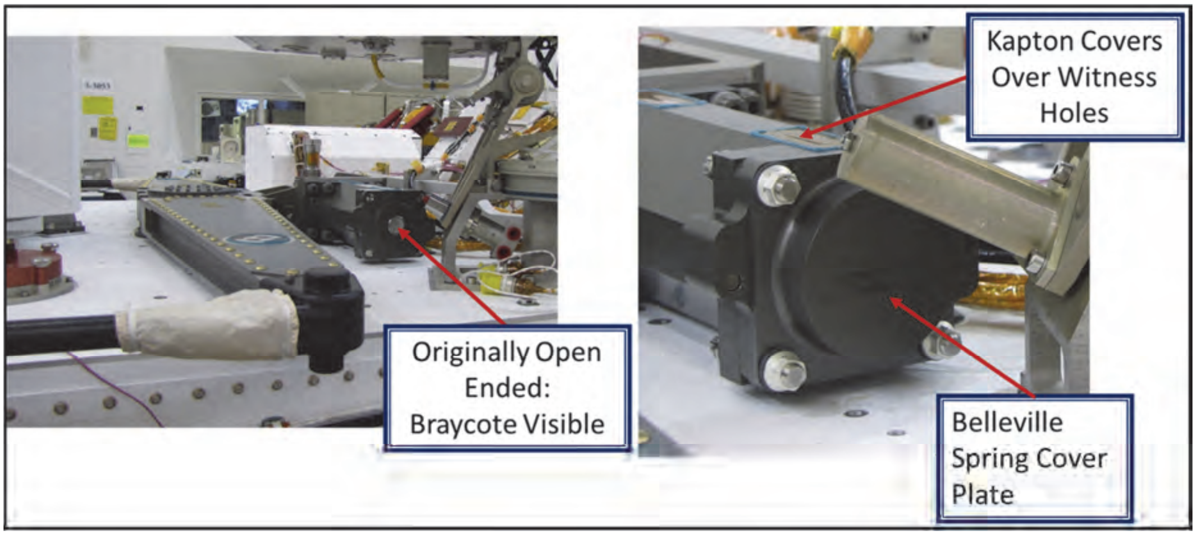

Figure 15: Before (Left) and after addition of Braycote leak path closures (Right)

Working in a Dirty World

As part of the final preparations for launch and closeout of the Verification and Validation matrices, the testing performed on all of the Rover pyros was summarized in order to ensure the Rover had been properly and completely tested. Although the pin puller used in the Differential Restraint had been qualified and testing had been completed at the sub-system and system level tests over temperature one concern was raised that the pin puller had never been fired in a dirty environment. None of the rover release pyros had been tested in dirt because they are fired shortly after the rover enters the Mars atmosphere while the rover is still connected to the Descent Stage and approximately one-hundred feet above the surface. The Differential Restraint pin puller, however, is fired much later in the deploy sequence, when the rover is approximately 2 seconds from final touchdown. Not only is there dust expected in the atmosphere at the level of the firing of the Differential Restraint pin puller, but the Descent Stage rocket engines are also expected to kick up significant quantities of Mars dirt into the atmosphere. The tight clearances between the Horizontal Swingarm Bracket bore and the pin puller pin (nominal 0.0015” diametral clearance) created a concern that small particulates of sand/dust could get stuck between the two surfaces and jam the joint, thereby preventing retraction of the pin. Additionally, there was a concern that if debris were to enter the pin puller device itself the retraction of the pin could fail.

Two tests were conducted to demonstrate the ability of the Differential Restraint Pin Puller to successfully retract in the presence of large quantities of dirt. Hardware which mimicked the Flight hardware clearances and tolerances was used to perform the test. This hardware had previously been used to qualify the pin puller for release while subjected to large lateral loads over temperature and varying percentages of NASA Standard Initiator charge. An acrylic chamber was built to sit around the Differential Restraint mock hardware and a GN2 line was setup with two inlets to the container to create a simulated dust storm. In total, 700 ml of dust simulant was added to the chamber, including 600 ml of cohesionless fine grain sand, which was used in traverse testing characterization for the MSL rover, 50 ml of JSC-1 lunar soil simulant and 50 ml of BP-1 dust (see Figure 16). GN2 was pumped into the chamber at 345 kPa (50 psi) for 30 seconds prior to firing the pin puller to create the simulated dust storm.

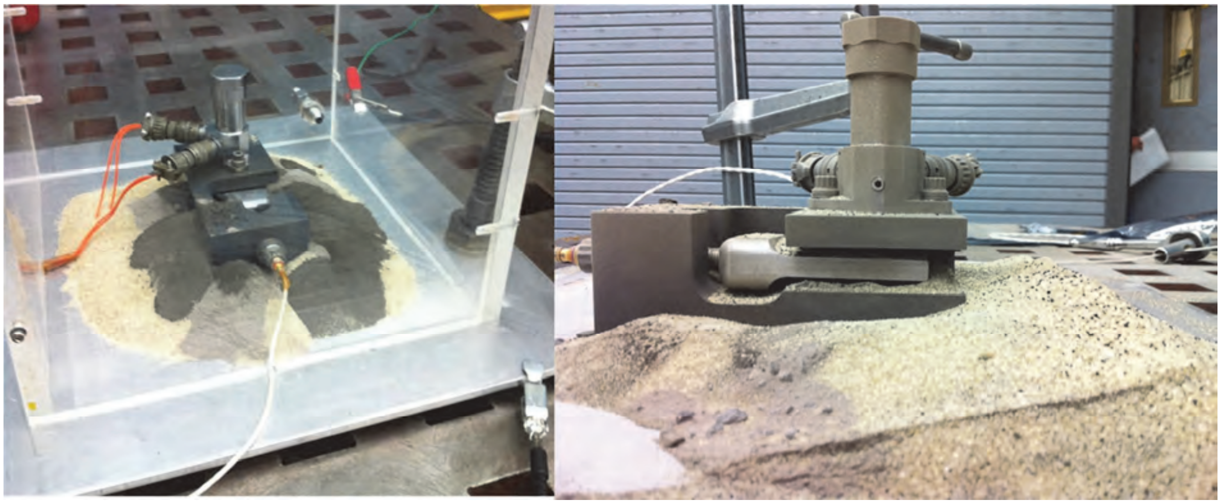

Figure 16: Dirty Pin Puller Test Setup: Pretest (Left), Posttest (Right)

The first test involved a dual firing of 100% charged NSI while the second test involved only a single NSI charged to 100%. Both tests included the application of the flight limit lateral load applied to the pin puller pin through the use of a strain gaged bolt connected to the Linkage hardware simulator. Both tests resulted in successful retraction of the pin puller. Upon inspection of the pin puller pin, Linkage monoball, and Horizontal Swingarm Bracket bore after the test no unusual wear marks or scoring were found, indicating that the fine grains of dust and sand had not jammed or impeded the retraction of the pin. The monoball in the Linkage, did however, have a significant increase in drag due to the presence of dirt in the lubricant on the monoball increasing the rolling friction between the ball and race. The increased drag in the monoball is not significant, however, since it did not provide enough resistance to jam the pin and prevent it from retracting.

<u>Key Findings:</u>

- Consider operational environments during the design of hardware (ex. how particulate contaminants affect hardware clearances and failure modes)
- Pin Pullers can successfully operate in the presence of fine grain particulates

Summary

The design of the Differential Restraint for the MSL mobility system involved striking a difficult balance between strict load and deflection requirements with the tight design space available. In total, three major design iterations were required to get the Differential Restraint from concept to the launch pad, and along the way some interesting lessons were learned. The first redesign effort was required when the loads model drastically increased the predicted maximum loads. The failure to consider the shimming procedure and misalignments allowed for by piece part clearances and tolerances was a costly mistake which led to the failure of the design in a major system level test causing the entire design to be subjected to major scrutiny and redesigned for a second time. The tolerances and clearances, when considered, allowed for over four times the assumed misalignment between the axis of the Spring Plunger and the Linkage which proportionally increased the bending stress in the Spring Plunger. The absence of a proper thread relief callout on the Spring Plunger drawing provided an easy target for scrutiny of the design, but was also easily corrected in the redesign efforts. The leaking of Braycote during vacuum bake should not have been a surprise, but the failure to assess the meaning of the viscosity of the lubricant at high temperatures led to the requirement of yet another design Band-Aid, the implementation of which would have neglected the effects of depressurization and rules of thumb for venting ratios except for a last minute catch of the newly created issue. The development of the Differential Restraint was not a linear or elegant process, but important lessons about tolerance stacks, lubricant leak paths, and even the use of pyrotechnic devices in dirty environments were learned; the knowledge of which will hopefully inform the design of future mechanisms for space applications.

Acknowledgements

This research was carried out at the Jet Propulsion Laboratory, California Institute of Technology, under a contract with the National Aeronautics and Space Administration.