Mars Science Laboratory Rover Integrated Pump Assembly Bellows Jamming Failure

Michael R. Johnson[1*], Joel Johnson*, Gajanana Birur*, Pradeep Bhandari* and Paul Karlmann*

Abstract

The Mars Science Laboratory rover and spacecraft utilize two mechanically pumped fluid loops for heat transfer to and from the internal electronics assemblies and the Radioisotope Thermo-Electric Generator (RTG). The heat transfer fluid is Freon R-11 (CFC-11) which has a large coefficient of thermal expansion. The Freon within the heat transfer system must have a volume for safe expansion of the fluid as the system temperature rises. The device used for this function is a gas-over-liquid accumulator. The accumulator uses a metal bellows to separate the fluid and gas sections. During expansion and contraction of the fluid in the system, the bellows extends and retracts to provide the needed volume change.

During final testing of a spare unit, the bellows would not extend the full distance required to provide the needed expansion volume. Increasing the fluid pressure did not loosen the jammed bellows either. No amount of stroking the bellows back and forth would get it to pass the jamming point. This type of failure, if it occurred during flight, would result in significant overpressure of the heat transfer system leading to a burst failure at some point in the system piping. A loss of the Freon fluid would soon result in a loss of the mission. The determination of the source of the jamming of the bellows was quite elusive, leading to an extensive series of tests and analyses. The testing and analyses did indicate the root cause of the failure, qualitatively. The results did not provide a set of dimensional limits for the existing hardware design that would guarantee proper operation of the accumulator. In the end, a new design was developed that relied on good engineering judgment combined with the test results to select a reliable enough solution that still met other physical constraints of the hardware, the schedule, and the rover system.

Introduction

The Mars Science Laboratory Mission - Overview

The mission consists of four discrete sections or phases:

1. Launch Phase
2. Cruise Phase
3. Entry, Descent, and Landing Phase
4. Surface Operations Phase

An exploded view of the assemblies that make up the spacecraft as it heads to Mars is shown in Figure 1. The Cruise Stage is jettisoned just prior to Entry, Descent, and Landing (EDL). The Backshell and Heat Shield are necessary components during the Entry portion of EDL. The Descent Stage contains fuel and rocket engines to provide a powered descent to the surface with a deployed Rover for a soft touchdown. After the Rover is on the surface, the Descent Stage is cut loose and flies away from the Rover's location.

* Jet Propulsion Laboratory, California Institute of Technology, Pasadena, California

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

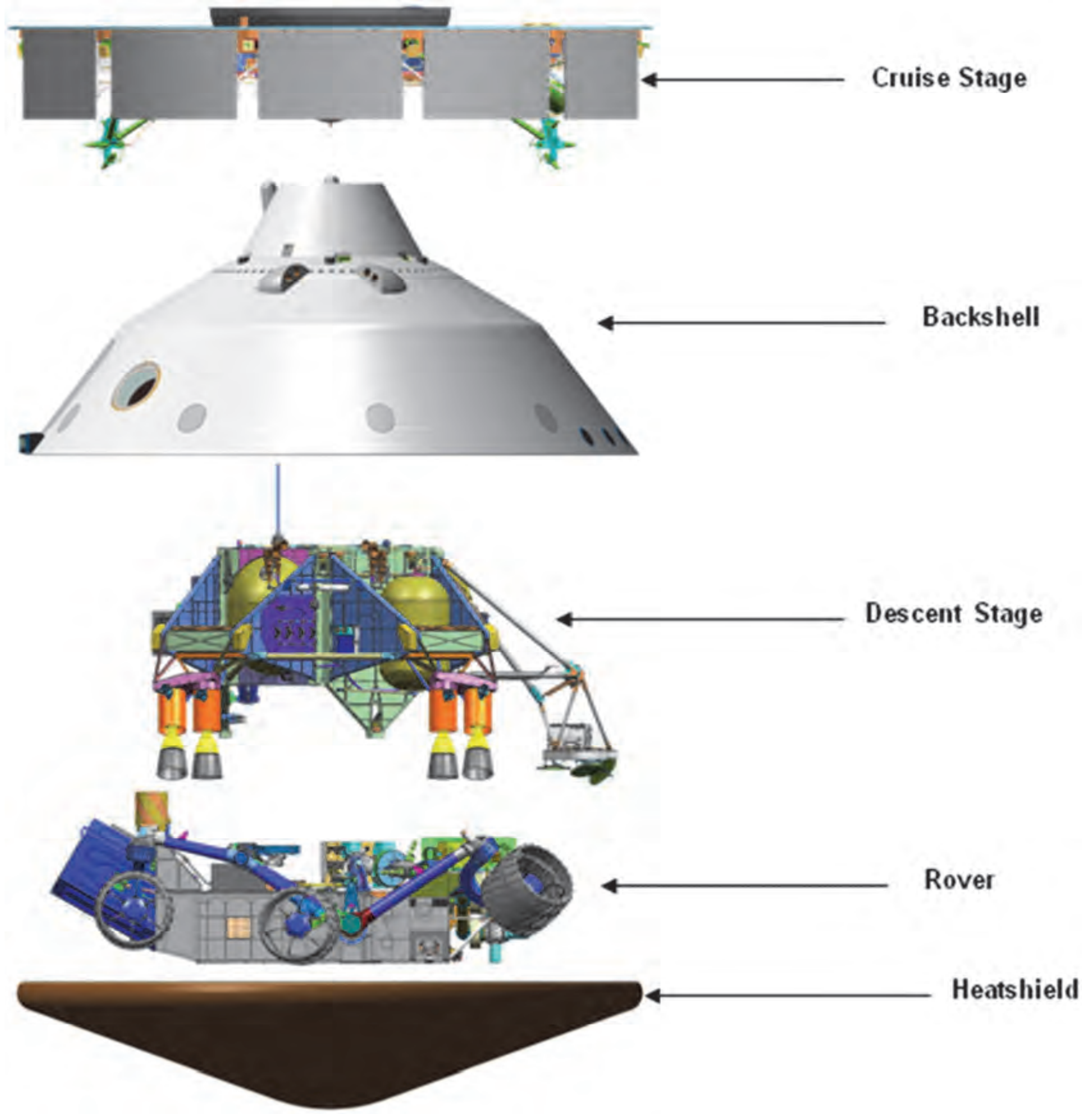

Figure 1. Mars Science Laboratory Spacecraft Major Assemblies

The Mars Science Laboratory Mission – Thermally

The cruise portion of the mission consists of the rover and cruise stage traveling from Earth to Mars over a period of 8½ months. During this time, power is provided to the cruise stage and the rover by solar panels located on the cruise stage and an RTG attached to the rover chassis. The RTG is located inside the spacecraft entry body. Heat generated by the RTG, as well as waste heat from the electronics, is transferred to radiators on the cruise stage using a system of pipes containing Freon R-11. This heat management system is called the Heat Rejection System (HRS). There are two separate fluid systems, one that is part of the cruise stage (CHRS), and one that is part of the rover assembly (RHRS). More detail on these two systems is provided in references 1 and 2. The CHRS is no longer needed after the several month cruise period to Mars so it is vented and then jettisoned from the entry body with the Cruise Stage just prior to the Entry, Descent, and Landing phase. The separate fluid system within the rover must continue to operate throughout landing and the 690 Earth-day primary surface mission, as well as any extended mission time. The main function of the heat transfer system is to distribute heat around the rover assembly for the purpose of maintaining a smaller operating temperature range for the rover electronics than the environment would otherwise require. Heat is transferred from the RTG to the internal Rover Avionics Mounting Plate (RAMP) during cold periods. The fluid is directed to radiators when excess heat must be exhausted. By utilizing the heat from the RTG and the fluid Heat Rejection System (HRS), the temperature range for the internal electronics assemblies is substantially reduced from -128°C to +55°C to a smaller range of -40°C to +55°C for the length of the mission.

Thermal Hardware Description

The thermal system for the rover and cruise stage are very similar. The hardware consists of an Integrated Pump Assembly (IPA), a gas-over-fluid accumulator, a significant length of fluid tubing, and heat transfer surfaces. The IPA consists of a motor, centrifugal pump, motor drive electronics, and directional valves for maintaining the coolant temperature. The valves direct more or less Freon to the radiators depending on the temperature of the coolant. A representation of the internal piping of the rover is shown in Figure 2 and the piping on the cruise stage is shown in Figure 3.

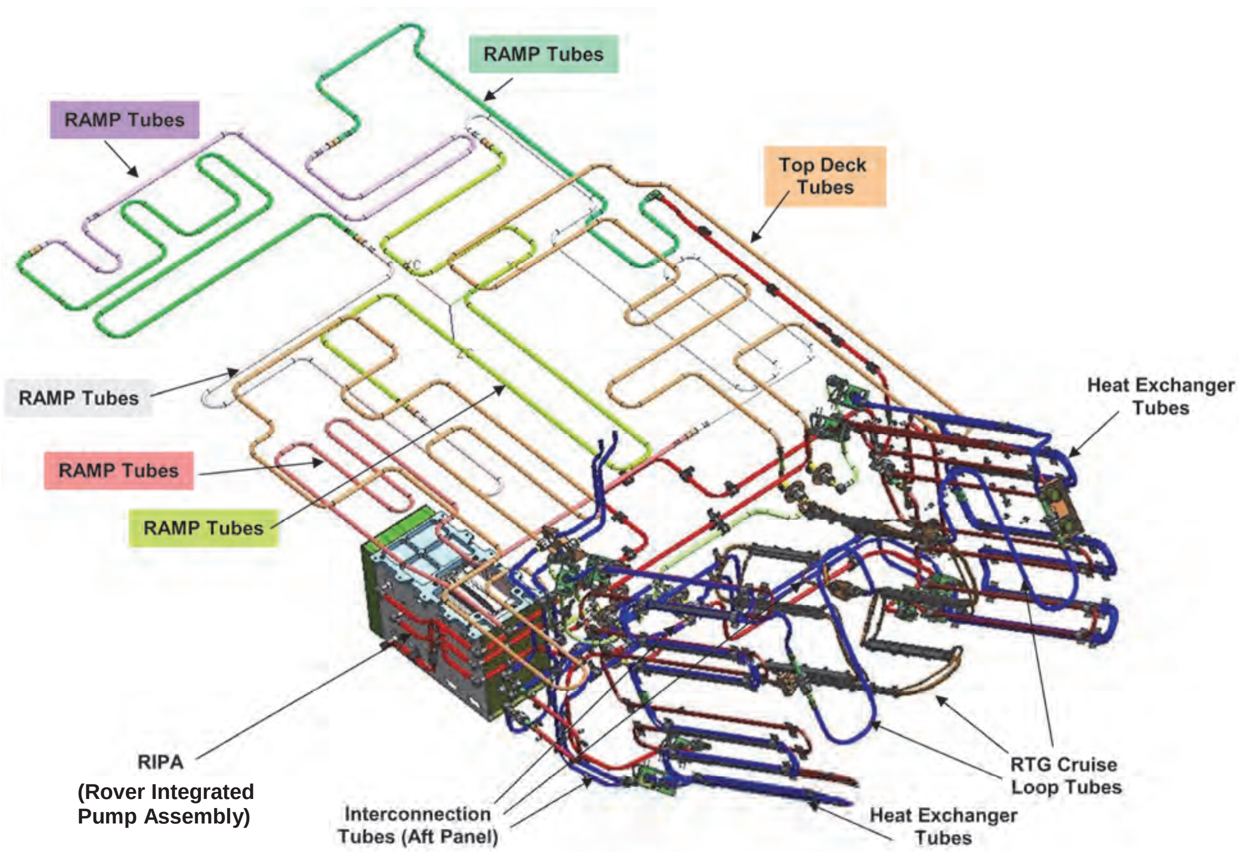

Figure 2. Heat Transfer System of Piping in the Rover and Around the RTG

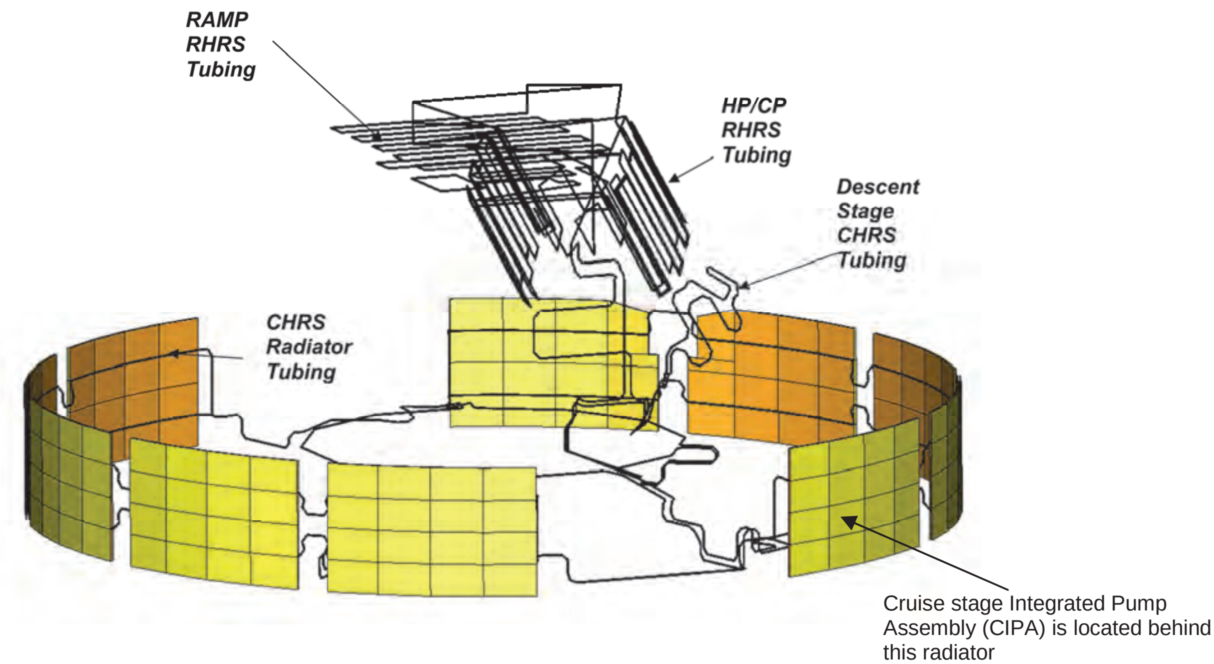

Figure 3. Heat Transfer System of Piping in the Cruise Stage and Relationship to the Rover System of Piping

Components of the Integrated Pump Assembly (IPA)

There are two separate IPA units, one on the Cruise Stage and the other on the Rover. The Cruise Stage IPA is called the CIPA and the Rover IPA is called the RIPA. A model of the RIPA is shown in Figure 4. The pumps provide the circulation power for the Freon R-11, the pressure transducers provide telemetry back to the system indicating the health of the fluid system, the directional valves divide the flow between the radiators and the RTG, and the accumulator provides expansion volume for the fluid in the closed system. The accumulator expansion volume is absolutely necessary because of the large thermal volumetric coefficient of expansion of the Freon of 20% per 100 degree Celsius change in temperature. Providing a volume for the Freon to expand into prevents the following failures:

- Mechanical failure of the system tubing due to over-pressurization
- Bubble generation from low pressure

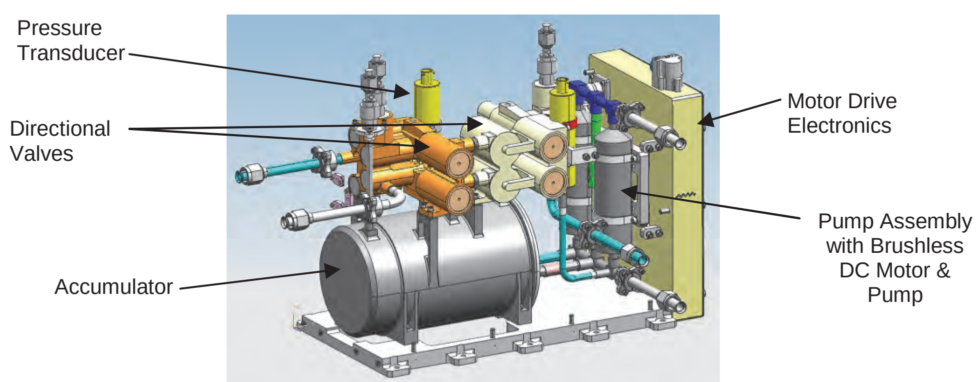

Figure 4. Rover Integrated Pump Assembly (RIPA)

The accumulator is a gas-over-fluid design that uses a metal bellows for the barrier between the fluid and gas. The gas is dry Nitrogen and the fluid is the system coolant of Freon R-11. A cross-section of the accumulator is shown in Figure 5.

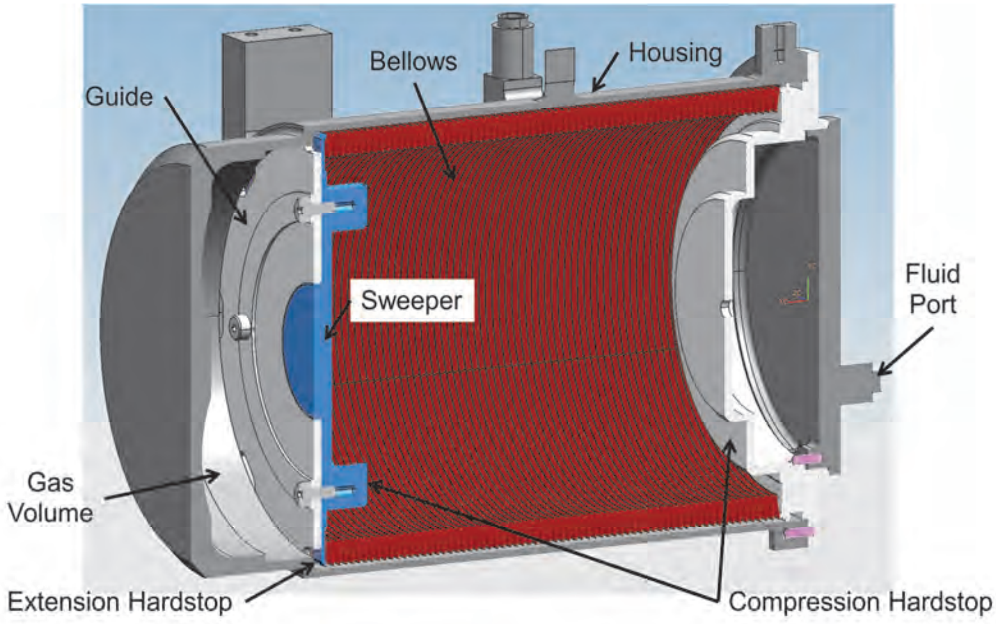

Figure 5. Accumulator Assembly showing bellows in fully extended position (maximum fluid volume, maximum pressure of gas)

The free end of the bellows assembly consists of a metallic sweeper to which a Teflon® guide disk with a larger outer diameter is mounted to provide low friction sliding on the bore of the outer accumulator housing. The bellows is prevented from stroking too far by a step in the housing bore that corresponds to the maximum design fluid volume increase along with the highest gas pressure. As fluid is removed from the accumulator, the gas pressure compresses the bellows until other features of the assembly act as a compression hardstop. This is the state of minimum fluid volume and minimum gas pressure within the accumulator assembly.

<u>Failure Event</u>

The initial functional testing of the accumulator assembly involved the following steps:

- Closing the accumulator assembly without welding (using a mechanical clamp) for a functional check
- Filling the bellows with fluid and removing any air bubbles (input tube upright)
- Pressurizing the gas side of the accumulator until the bellows is fully compressed
- Pumping fluid into the accumulator to extend the bellows, until the pressure difference across the bellows reaches a limiting value (this was the point where the bellows was calculated to be at full stroke with the guide seated against the step in the housing bore)
- Removing the pump connection
- Allowing the bellows to drain into a graduated cylinder to measure the amount of fluid the accumulator could contain
- Verifying that the fluid capacity of the accumulator is within specified tolerances
- If the assembly passes the test, weld the bellow end assembly onto the housing for a final flight seal
- Repeat the functional test above to verify no change occurred during welding

During final testing of one of the welded accumulator assemblies, the accumulator's fluid capacity appeared to be less than half the required value. Upon repeating the test several times, the capacity of the accumulator did not change. No test setup anomalies were observed and the accumulator assembly did not have any discrepancies that could account for the reduced capacity.

The accumulator assembly internals were then imaged using X-rays with the fluid in the accumulator at the limiting pressure difference. The picture in Figure 6 shows what was observed.

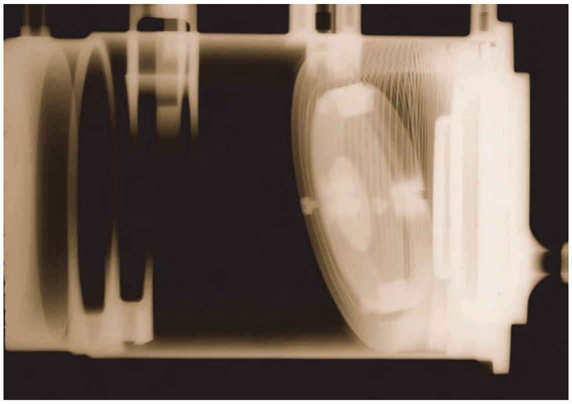

Figure 6. X-ray of Accumulator Assembly showing bellows shape with the maximum allowable pressure difference across the bellows. The accumulator fluid capacity is significantly reduced from the required value.

The end of the bellows assembly with the sweeper and Teflon® guide had managed to tip to an angle where the piston would no longer move down the bore and had jammed in place. The force on the piston in the direction of extending the bellows with the maximum allowable pressure difference is 370 N, indicating the jamming force was quite high. When the fluid was removed, the bellows compressed flat against the compression hardstop (the right end in Figure 6) as designed. Repeating the test and measuring the fluid capacity indicated that the failure point was consistent and very stiff – meaning the location along the bore where the piston jammed did not move detectably when cycled between no fluid and the maximum possible fluid, limited by the allowable pressure difference across the bellows.

The X-ray was used to make measurements of the geometry of the failure location so it could be identified when the assembly was opened for inspection. Figure 7 shows some of the measurements made to determine the location and the possible source of the jamming.

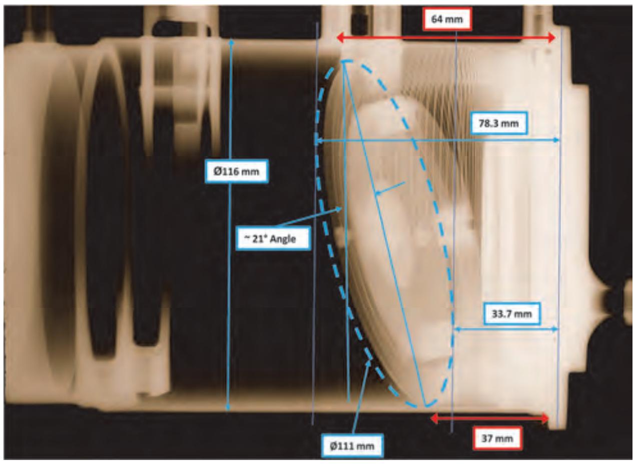

Figure 7. Measurements made using the X-ray of the Accumulator Assembly

It was clear there was some obstruction to the motion of the bellows guide that would hopefully be identified when the assembly was opened up and inspected. The next step was to carefully remove the bellows assembly from the housing to find the source of the jamming by cutting the weld off the end of the housing. Figure 8 shows what was found inside the housing assembly. The location of the residue seemed to indicate it was the source of the jamming, as indicated in Figure 9.

Figure 8. Tape adhesive residue on the housing wall, left behind from previous rework operation

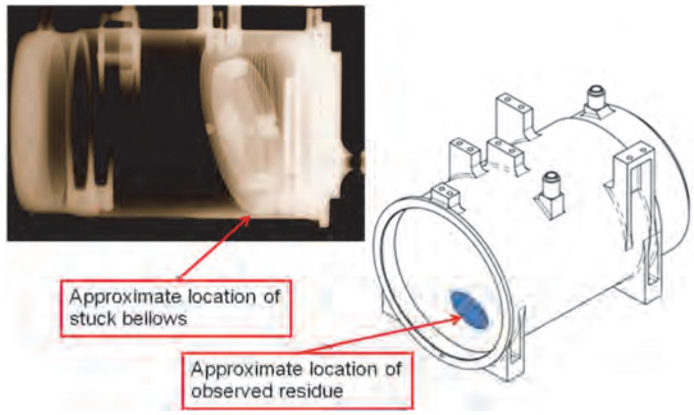

Figure 9. Location of tape adhesive residue correlates well to the tip-initiation point

The tape residue was cleaned from the housing bore and the accumulator reassembled. The test was repeated with the full expectation that the problem had been solved. The result of the test indicated **<u>no change</u>** in the behavior of the assembly, including the measured fluid volumes. The following analysis on the tipping phenomenon was performed (see Figure 10) to understand the sensitivity of the design to the variables involved. A description of the mechanical behavior model is presented.

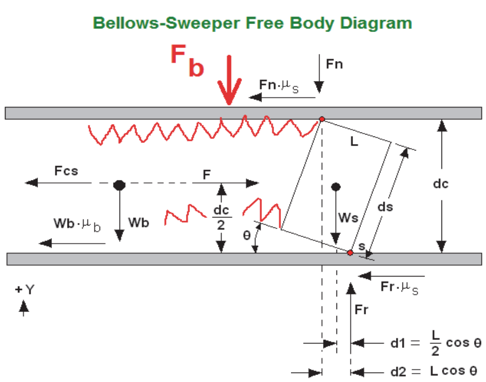

Figure 10. Free Body diagram of bellows assembly in housing

<u>Mechanical Behavior Model Description</u>

The free-body diagram variables include the weight of the sweeper (W_s), the weight of the fluid-filled bellows (W_b), the friction at the sweeper-to-housing interface (μ_s), and the normal force generated by a squirming bellows (F_b).

Prior to the tipping of the end of the bellows, F_b is zero. Once the end starts to tip, the bellows squirms upward and reacts against the housing wall. This additional normal force increases the frictional drag between the housing wall and the bellows surface, including the sweeper-to-housing interface.

An additional aspect of the tipping end of the bellows is how the contact between the Guide Disk and the housing behaves. The contact starts at a point and, as the tipping angle increases, this contact spot increases to an elliptical area. As the tipping angle is increased more, the contact patch splits into two small contact patches that travel around the housing surface. As the contact points get higher on the housing surface, the normal force necessary to balance the bellows squirm force, F_b ,increases substantially. As the normal force increases, so does the friction force, increasing the tipping moment on the end of the bellows. This runaway behavior demonstrates that the tipping of the end of the bellows will eventually jam, and added force will not help to release it. See Figure 11 for a pictorial description of this phenomenon.

Figure 11a. Bellows straight, normal force on bottom is from weight only (F_b = 0)

Figure 11b. Start of tipping, normal force on bottom is a small contact patch (F_b = 0)

Figure 11c. Tipping angle has increased and contact patch has split into two discrete areas

Figure 11d. As tip angle increases more, the contact areas move up the side of the housing. Radial force must increase to balance the bellows squirm force, F_b.

The most significant variable in this tipping and then jamming behavior is the friction force. The housing bores were cleaned very well to minimize the possibility for contamination. The Teflon® Guide Disk was relied upon for its low friction properties at the bellows-to-housing interface. No other Accumulator Assembly has exhibited this characteristic and it had not shown up in any of the environmental testing either.

Design History of the Accumulator Assembly

This accumulator design was not new. There was a Heat Rejection System on the Mars Pathfinder Lander in 1997, as well as one on each of the rovers, Spirit and Opportunity in the 2003 missions. The accumulator in the MSL mission was larger than any of the prior units because the volume of heat transfer fluid was much greater. The design was scaled up in size to accommodate the volumetric increase.

The mission life requirement for MSL is eight times longer than the Spirit and Opportunity rovers. The larger size of the accumulator and bellows assemblies coupled with the longer operational life required a cycle life test be performed. The cycle life test resulted in a fatigue failure of the bellows assembly. The bellows was redesigned to accommodate the higher life requirement by reducing the thickness of the convolutions a small amount. The bellows displacement then resulted in a lower stress level in the material. This had the additional effect of reducing the bellows stiffness, both axially and laterally.

The Next Version of the Failure Mechanism Hypothesis

When the elimination of the adhesive residue on the housing wall did not affect the failure results, the additional factor of the reduced stiffness of the bellows was considered. This would make the design even more sensitive to bellows tipping under smaller frictional forces, since the bellows stiffness helps keep the end from tipping by resisting the applied moment from friction with the housing bore. The reduced stiffness of the MSL bellows combined with a surface finish in the housing bore that was at the coarse end of the tolerance and some tape adhesive residue all seemed to have pushed this unit over the edge and initiated the tipping of the end of the bellows. Once it had started, the bellows continued to tip and jam at the same location, even though the adhesive had been removed from the bore. While a stress analysis indicated that the tipped distortion of the bellows should not have resulted in any yielding, and there was no visual indication of damage or permanent deformation, it seemed that the bellows had taken a set that led to the repeatable failure. Additional stroke tests were performed with different orientations to gravity that indicated that the bellows was slightly biased to fail in the original location.

<u>Design Change Options – Highly Limited</u>

The opportunities for changing the design were explored at this time and included a new bellows assembly with a greater stiffness, reducing the stroke requirement, providing improved guidance of the end of the bellows, and increasing the length of the housing to accommodate more volume for some bellows guide options. The options that were possible in the schedule time available included reducing the stroke requirement and adding a guide to the end of the bellows. A new bellows could not be designed and fabricated in time and increasing the housing length was not possible since the rover had structure that would interfere with a longer housing. The option of reducing the stroke requirement did not actually solve the fundamental problem and would have required the temperature range to be less than the current system design provided, making this option unacceptable.

The remaining option of improving the design of the guide on the end of the bellows was pursued. The length of the outside of the accumulator housing could not be changed, but the inside of the housing had an option for moving the end-of-stroke hard stop 25 mm closer to the end of the housing. This did not affect the outer profile, making it acceptable in the tight volume the accumulator occupied within the rover chassis. While the housing could not be lengthened, the end-of-stroke hardstop inside of the housing could be moved closer to the end of the housing. This provided 25mm of additional interior length in which to implement the improved guide design.

<u>What Guide Material and Length-to-Diameter Ratio is Acceptable?</u>

The additional guide length that could be obtained without changing the outside length of the housing was 25 mm and the diameter of the bore of the accumulator was 111 mm. For a piston type guide, the standard wisdom is to have a length to diameter ratio of 0.5 to 2.0, with most ratios in typical applications near 1.0. The best that could be obtained within the constraints of the MSL application was 0.225. Additionally, the best material choice would be one with the following characteristics:

- Low friction with the stainless steel housing
- Minimal to no particle generation over the life
- A life cycle capability of 10,000 cycles
- A coefficient of thermal expansion (CTE) that is close to the bellows and housing values

The best choices for low friction involved polymers and the best CTE choices were metals. The choices from a particle generation and cycle life perspective crossed between polymers and metals. Lubrication of the metals was considered, but no lubricants had been qualified with the nitrogen gas fill valve. If the gas fill valve leaked at all, the system would fail due to loss of pressure on the Freon R-11. The leak rate of the fill valve had been verified with everything clean, and there was not enough time available to repeat the valve qualification with a lubricant, liquid or dry. This eliminated the use of an applied lubricant as a possible solution to the failure mechanism.

The analysis of the self-energizing jamming phenomenon showed that friction and initial tip angle are critical to a successful design. A larger radial clearance between the guide and the housing bore would allow the guide to rotate more before contacting the housing wall. The larger the initial contact angle is, the smaller the required friction to create a self-energizing jam. This demonstrates that smaller radial clearances are preferred. In addition, the CTE difference between the guide and the housing had to be considered. These properties determined the smallest radial clearance that could be selected for a particular guide material choice.

In order to answer the question of minimum acceptable guide length and material choice, manual testing was performed using the test assembly shown in Figure 12. The plates at the end of the assembly determined the radial clearance to the housing bore and the plates in the middle region controlled the effective length of the test guide.

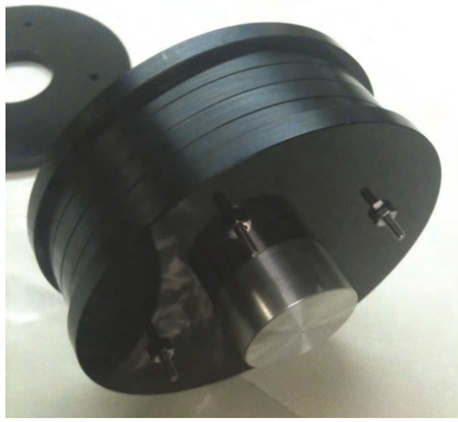

Figure 12. Test Guide Assembly - Used to qualitatively determine the best length and radial clearance combination using Delrin® disks

The test guide assembly was inserted into the test housing and the operator used the mandrel diameter to stroke the assembly within the bore. As the test guide was moving, the operator applied a moment to the assembly with the thumb and index finger. As moment was applied, some combinations of length and radial clearance would solidly jam, some would tip over freely, and others would get tight in the bore but not be self-energizing. The results of the testing are listed in Table 1.

Table 1. Results of Qualitative Manual Testing of Test Guide Assembly as a function of radial clearance and guide length using Delrin® disks

	Clearance, Ø [mm]					
Length [mm]	**0.025**	**0.127**	**0.254**	**0.381**	**0.508**	**0.635**
6.35	FAIL -	FLIP	FLIP	FLIP	FLIP	FLIP
12.7	FAIL	FAIL -	FAIL -	FAIL-	FAIL/FLIP	FLIP
19.1	FAIL	FAIL -	FAIL	FAIL	FAIL	FAIL-
25.4	FAIL	FAIL -	FAIL	FAIL+	FAIL+	FAIL
31.8	FAIL	FAIL	FAIL+	FAIL +	FAIL+	FAIL+
38.1	PASS -	PASS -	PASS	PASS	PASS	PASS -
44.5	-	-	-	-	-	PASS

FLIP = Piston could freely flip through 360 degrees, i.e. provided no tip restraint
FAIL - = Jammed badly, i.e. became stuck with little or no effort
FAIL = Would jam, but only when a moment was applied or was pushed at edge
FAIL + = Would jam, but took a lot of effort. Would not recover on its own
PASS - = Judged not to jam, but considered to be closer to jamming than others
PASS = Could not be made to jam

The manual testing indicated that a 38.1 mm minimum guide length was required; no length/clearance combination less than 38.1 mm was successful. However, the maximum possible length of the guide that could be implemented with the housing and structural constraints was 25 mm. The test results demonstrated that a Delrin® solution was not acceptable.

Since the longest possible guide length was 25 mm, a new set of tests were implemented with different guide materials and two different guide designs. The first test guide design was a solid cylindrical guide that was 25 mm long on a stainless steel mandrel. The second design was the same cylindrical guide with a mandrel and a slotted guide, with the slot in the circumference of the guide, similar to automobile engine piston ring designs. The slot allowed the CTE mismatch between the guide and housing to be accommodated in the expansion/contraction of the slot, rather than in the radial clearance between the guide and housing. This was crucial for high CTE materials, since a solid guide design would have required the guide to actually be smaller in diameter than the bellows themselves. The slotted guide was also placed on a stainless steel mandrel to provide support for the bore of the guide. Figure 13 shows the virgin Teflon® version of the guide.

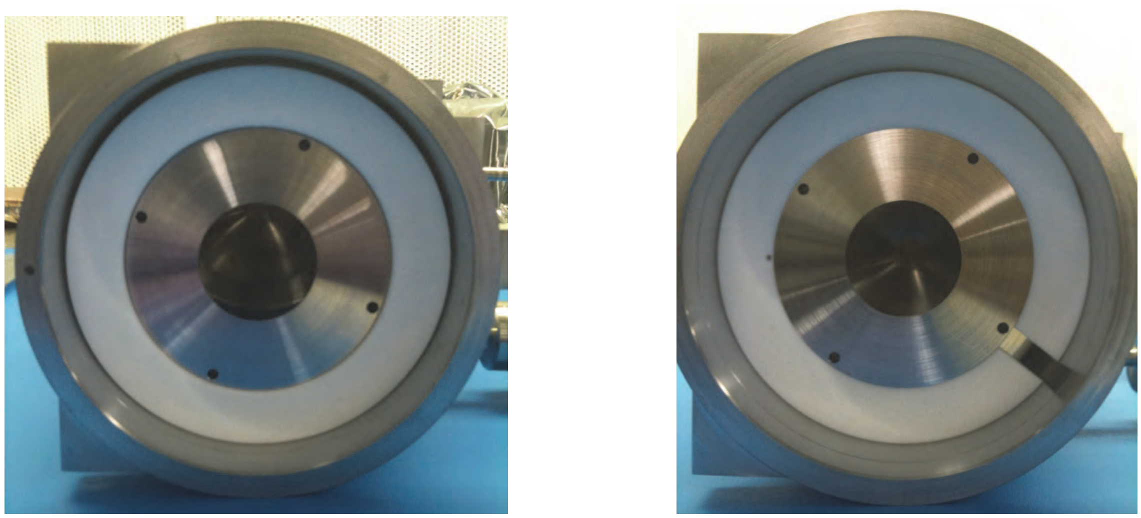

Figure 13. Solid and Slotted designs of the guide on Test Mandrel and inside the Test Housing with the largest possible radial clearance. Note that the slotted design springs outward, closing the radial clearance under all conditions.
(the terms Solid and Slotted refer to the circumference of the guide)

The different materials tested using circumferentially solid and slotted guides on a mandrel were

- Torlon® 5030
- Vespel® SP-3
- Delrin® 100 AF
- Teflon® 25% glass filled
- Virgin Teflon®

The testing was done manually with the same pass/fail criteria as was used in the previous Delrin® guide testing (see Table 1). The test guides were fabricated to diameters representing a nominal radial clearance at 25°C and the maximum radial clearance associated with the lowest temperature condition and the smallest manufactured outside diameter (Least Material Condition, or LMC). The results of this testing are listed in Table 2. Note that a solid Delrin® version was not tested. This was the version tested previously and represented in Table 1. For the Torlon® and Vespel®, the slotted versions were not tested because the CTE of these materials did not require them to be slotted and the solid versions failed. Both Teflon® versions were successful, except that the solid version with virgin Teflon® had too high a CTE and lost all guiding function at cold temperature and least material condition.

Table 2. Results of Qualitative Manual Testing of Test Guides using circumferentially slotted and solid test items as a function of radial clearance [Nominal and Least Material Condition Cold (LMC Cold)], and material

Material		CTE	Solid		Slotted	
		[ppm/C]	nominal	LMC Cold	nominal	LMC Cold
Torlon 5030	glass fill	16	FAIL			
Vespel SP-3	MoS2	52	FAIL			
Delrin 100AF	Teflon fill	100			FAIL	
Teflon 25% GF	glass fill	100	PASS	PASS	PASS	PASS
Teflon, virgin		151	PASS	FLIP	PASS	PASS

The test results demonstrated that both glass filled and virgin Teflon® were acceptable candidates, though only a slotted design was acceptable with the virgin Teflon®. Additionally, the glass filled Teflon® showed signs of abrading the bore of the housing by capturing fine particles of housing material within and on the wear surfaces of the guide. This abrasion was deemed undesirable and the virgin Teflon® slotted design was chosen to proceed forward. The final design of the guide and hard stop for the flight assembly is shown in Figure 14 along with a representation of the original design, emphasizing the changes made to mitigate the demonstrated failure mode.

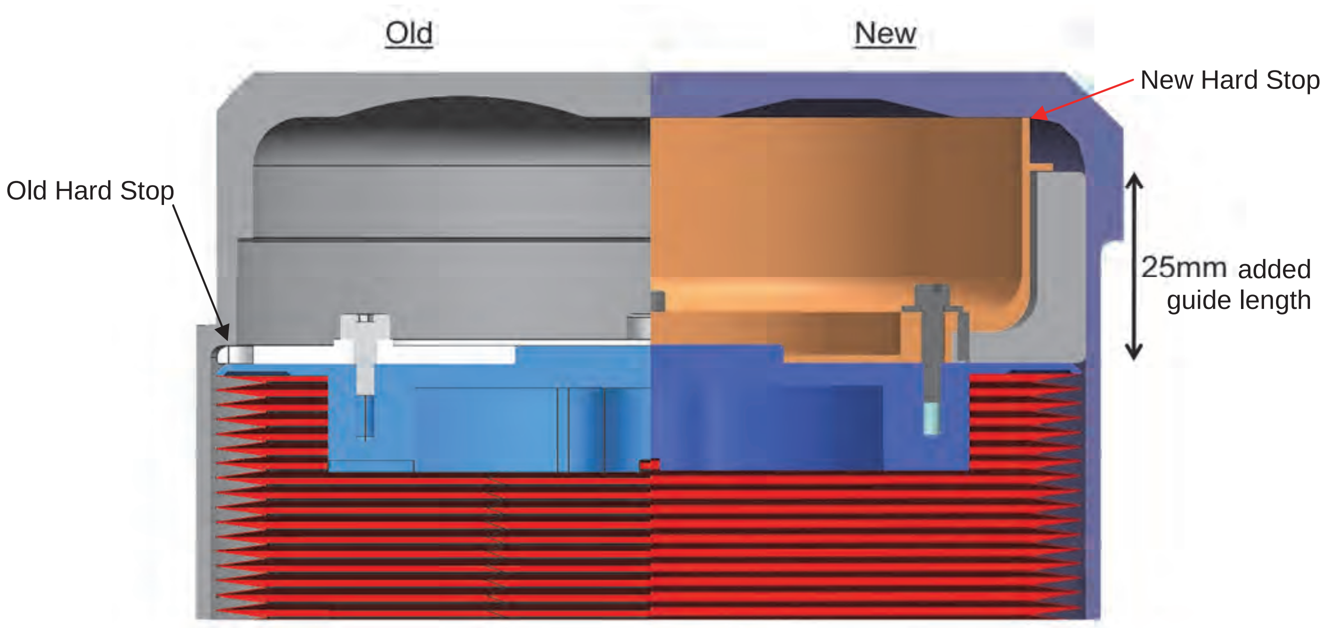

Figure 14. Original and Final designs of the end of the bellows assembly. The figure shows the bellows stroked to the maximum fluid condition and against the end-of-stroke hard stop for both design versions

Acceptable to Fly?

The final design of the bellows end guide was selected based on intuitive “feel” and engineering judgment. These are not exactly verifiable quantities that make an assembly ready to fly on a mission. The next challenge was to demonstrate, using quantifiable methods under worst case conditions, that the new design would not exhibit the previous functional limitation or any new failures. This was particularly difficult since the landing and launch environments present significant loading to the assembly, but they were not representative of the long life mission loads while the bellows is actually stroking within the bore. However, if the large loading of launch or landing were to initiate a self-energizing condition, then they were appropriate and representative to use for testing.

Upon studying the types of loading the unit would be subjected to during all of the phases of the mission, the worst case conditions consisted of combined radial and moment loading at the end of the bellows. The magnitude of the moment loading would be dependent on the stroke position of the bellows, which is a function of the temperature around the internals of the rover. In order to cover the conditions on the surface of Mars for the majority of the mission as well as the shorter term higher loading conditions, a combination of moment and radial loading conditions were formulated. It turned out that the maximum radial load could not be achieved along with the proper moment load due to the dimensional constraints of applying the loads to the bellows inside of the housing. Since the moment loading was determined to be the most detrimental to the operation of the accumulator, the moment loading was matched and the radial loading was allowed to be undersized.

The test loads were generated by attaching steel plates to the end of the bellows assembly and cantilevering them far enough to produce the required moment loading and best possible radial load. This was not possible to do with a closed and sealed accumulator, so it was performed with the nitrogen gas end cap removed from the end of the accumulator housing. Figure 15 shows the test setup.

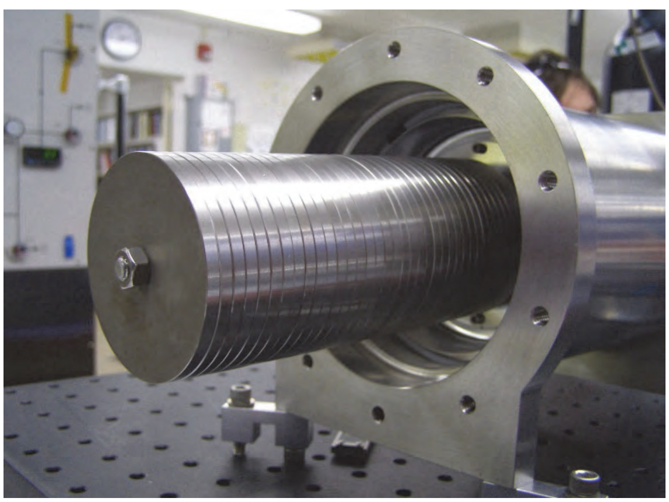

Figure 15. Stroke Testing of the final guide design under high loading conditions (load is equivalent to 3.2-g radial and 5.4 times the maximum flight moment)

The test loads were applied as shown above and the bellows stroked using internal pressure. The maximum allowable safe pressure difference across the bellows assembly was 38 kPa. The pressure difference across the convolutions that moved the guided end of the bellows in the housing was measured and recorded. There was no nitrogen gas pressure applied to bellows during the testing. This meant that the only force available to compress the bellows was its own internal spring force, which was very low. The results of this testing are shown in Table 3. Note that for the higher moment loads, the bellows did not retract on its own due to the friction force. This would not occur in a sealed and pressurized accumulator due to the presence of the nitrogen gas on the end of the bellows, forcing it to retract.

Conclusions and Lessons Learned

A bellows assembly without end guiding that must carry moment loads is a marginal design at best. A design without adequate end guiding relies on the internal moment stiffness of the bellows itself to prevent a significant rotation of the free end. If the radial clearance and friction conditions are beyond certain limits, then the rotation of the end of the bellows can quickly result in a self-energizing jamming condition that cannot be overcome.

With a piston guide of reasonable length, friction and the allowable radial clearance are extremely important design parameters because this mechanical configuration is very sensitive to these variables. The best option, where possible, is to use a lubricant to control the sliding friction. Otherwise, the lubrication function must be provided by careful material selections of the moving components. Polymers like Teflon® are great, but their CTE is large compared to metals and the applied contact pressure between the sliding surfaces must be limited. The slotted guide presented here worked exceptionally well to accommodate the large CTE difference.

Table 3. Results from the load testing of the final guide design (all bellows pressure values were well below the 38 kPa limit)

Number of test plates	0	14	22	29	37
Added mass [kg]	0.000	1.540	2.368	3.093	3.921
Total mass [kg]	0.680	2.220	3.048	3.773	4.601
Total applied load [N]	6.671	21.778	29.901	37.013	45.136
Cantilever length [m]	N/A	0.069	0.065	0.072	0.096
Applied moment [Nm]	0.000	1.022	1.492	2.160	3.647
Equivalent Radial load [g's]	0.5	1.6	2.1	2.6	3.2
Equivalent Moment load (x Flight)	0.0	1.5	2.2	3.2	5.4
	Bellows Pressure, kPa gage				
Guide started to slide, extending	0.34	2.4	5.2	9.3	7.9
Guide is halfway, extending	1.4	4.5	8.3	10.7	13.4
Guide is almost at hardstop, extending	3.1	7.2	10.3	13.4	16.9
Guide started to slide, contracting	2.4	0.34	†	†	†
Guide is halfway, contracting	1.4	†	†	†	†

† the drag was high enough to prevent the bellows from contracting on its own

References

1. Bhandari, P., Birur, G., Pauken, M., Paris, A., Novak, K., Prina, M., Ramirez, B., and Bame, D., “Mars Science Laboratory Thermal Control Architecture,” SAE 2005-01-2828, 35th International Conference on Environmental Systems, Rome, Italy, July 2005.
2. Bhandari, P., Birur, G.C., et al, “Mechanically Pumped Fluid Loop Heat Rejection & Recovery Systems For Thermal Control on Martian Surface – Case Study of The Mars Science Laboratory,” 36th International Conference on Environmental Systems, Norfolk, Virginia, July 2006.
3. Flexial Corporation provided experienced advice on the details of bellows analysis methods within the confined space of the accumulator housing. Flexial Corporation, 1483 Gould Drive, Cookeville, TN, 38502-3105, USA, www.flexial.com

Acknowledgements

This work was performed at the Jet Propulsion Laboratory, California Institute of Technology, under a contract with the National Aeronautics and Space Administration. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not constitute or imply its endorsement by the United States Government or the Jet Propulsion Laboratory, Pasadena, California. Copyright 2012 California Institute of Technology. Government sponsorship acknowledged. The authors wish to acknowledge the many individuals working on the MSL project, of which the thermal subsystem is a part of the greater whole. Dave Bame and AJ Mastropietro from the MSL Thermal team provided excellent support for the testing of the bellows at JPL.

Pacific Design Technology (PDT) in Goleta, CA is acknowledged for their excellent work in developing the pump assemblies and providing significant support for this investigation.