Failure of Harmonic Gears During Verification of a Two-Axis Gimbal for the Mars Reconnaissance Orbiter Spacecraft

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Abstract

The Mars Reconnaissance Orbiter (MRO) spacecraft has three two-axis gimbal assemblies that support and move the High Gain Antenna and two solar array wings. The gimbal assemblies are required to move almost continuously throughout the mission's seven-year lifetime, requiring a large number of output revolutions for each actuator in the gimbal assemblies. The actuator for each of the six axes consists of a two-phase brushless dc motor with a direct drive to the wave generator of a size-32 cup-type harmonic gear. During life testing of an actuator assembly, the harmonic gear teeth failed completely, leaving the size-32 harmonic gear with a maximum output torque capability less than 10% of its design capability.

The investigation that followed the failure revealed limitations of the heritage material choices that were made for the harmonic gear components that had passed similar life requirements on several previous programs. Additionally, the methods used to increase the stiffness of a standard harmonic gear component set, while accepted practice for harmonic gears, is limited in its range. The stiffness of harmonic gear assemblies can be increased up to a maximum stiffness point that, if exceeded, compromises the reliability of the gear components for long life applications.

Introduction

The Mars Reconnaissance Orbiter Mission

During its two-year primary science mission, the Mars Reconnaissance Orbiter will conduct eight different science investigations at Mars. The investigations are functionally divided into three purposes: global mapping, regional surveying, and high-resolution targeting of specific spots on the surface. This detailed mapping of the surface of Mars will provide future landed missions with the high resolution data required to land safely in a desired area. The instruments on board the MRO spacecraft consist of five types: cameras, a spectrometer, a radiometer, a radar, and engineering. Refer to Figure 1 for an overall view of the MRO spacecraft's science deck.

Cameras

HiRISE (High Resolution Imaging Science Experiment)

This visible camera can reveal small-scale objects in the debris blankets of mysterious gullies and details of geologic structure of canyons, craters, and layered deposits.

CTX (Context Camera)

This camera will provide wide area views to help provide a context for high-resolution analysis of key spots on Mars provided by HiRISE and CRISM.

MARCI (Mars Color Imager)

This weather camera will monitor clouds and dust storms.

Spectrometer

CRISM (Compact Reconnaissance Imaging Spectrometer for Mars)

This instrument splits visible and near-infrared light of its images into hundreds of "colors" that identify minerals, especially those likely formed in the presence of water, in surface areas on Mars not much bigger than a football field.

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Radiometer
MCS (Mars Climate Sounder)
This atmospheric profiler will detect vertical variations of temperature, dust, and water vapor concentrations in the Martian atmosphere.

Radar
SHARAD (Shallow Radar)
This sounding radar will probe beneath the Martian surface to see if water ice is present at depths greater than one meter.

Engineering
The engineering instruments facilitate spacecraft navigation and communications.

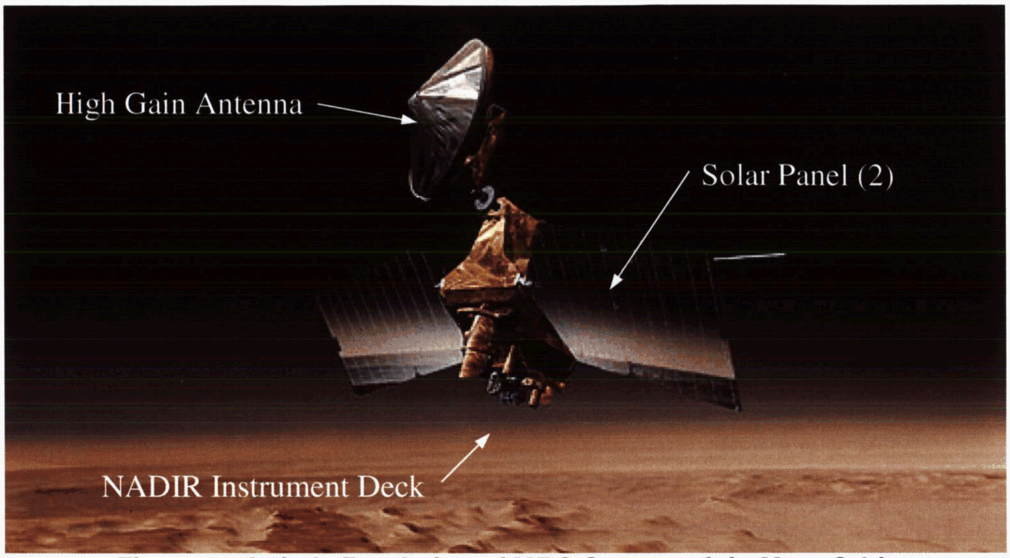


Figure 1. Artist's Rendering of MRO Spacecraft in Mars Orbit

Once the science phase is completed (two years after the mapping orbit is established), the MRO mission enters a second phase, communications relay. In this phase, the communication equipment on-board MRO will be used as a communications relay between the Earth and landed crafts on Mars that may not have sufficient radio power to communicate directly with Earth on their own. This capability allows landed crafts to use smaller antennas with reduced mass, improving the lander's science complement potential.

Due to the mapping nature of the mission, the instrument deck of the spacecraft must always be facing the surface of Mars. Additional pointing requirements include maintaining sun pointing of the solar panels and keeping the High Gain Antenna Earth pointed for communication purposes. The solution to this extreme panel and antenna pointing choreography was to put a two-axis gimbal at each of the appendages: two solar array wings and one High Gain Antenna.

The path from Earth to Mars orbit and mapping of the surface consists of launch, cruise, orbit insertion, aerobraking, and mapping phases. The spacecraft configuration of the solar array and the High Gain Antenna are different for each of these phases. During launch, the solar array wings are folded in half and the High Gain Antenna is positioned directly over the spacecraft bus to fit into the launch vehicle fairing (Figure 2). Once MRO is launched and in the cruise phase of the mission, the solar array and High Gain Antenna are pointing in roughly the same direction to capture sunlight and communicate with Earth (Figure 3). For Mars orbit Insertion and aerobraking, the appendages are moved slightly from the cruise configuration to produce an aerodynamically stable configuration. The High Gain Antenna and the solar array wings are the predominant source of atmospheric drag on the spacecraft and must be positioned to keep the spacecraft stable throughout the aerobraking maneuvers, which last for about 6 months (see Figure 4). For the mapping phase of the mission, the solar array wings and High Gain Antenna are almost continuously articulated so the wings remain sun pointed and the High Gain Antenna maintains a lock on Earth. This continuous motion must be performed while the spacecraft maintains precise pointing for high resolution imaging and high-speed data transmission to Earth (see Figure 1 for a mapping configuration). Because of the stringent pointing stability requirements, the gimbals were required to be exceedingly smooth and quiet.

The different phases for the MRO appendages lead to a large range of requirements for the two-axis gimbal assemblies. The gimbals must be capable of carrying launch loads through their output bearings for the launch phase. The cruise phase is very benign with minimal load on the gimbals. The orbit insertion and aerobraking phases put a significant load on the output bearings and gears, since the loads have a significant component in the backdriving direction for the gimbal actuators. Once the spacecraft is in the mapping phase, the high resolution capability of the instruments on board require that the gimbal assemblies do not produce any significant disturbance to the spacecraft platform while they are continuously scanning to maintain the required pointing of the attached appendage. The gimbals must withstand all of these load combinations and still maintain extreme pointing accuracy and smooth operation once at Mars. In addition to the smooth motion, the lowest structural frequency in the mapping configuration is determined by the natural frequency of the deployed appendages. A major contributor to the frequency of the deployed appendage is the gimbal actuator output stiffness for each axis.

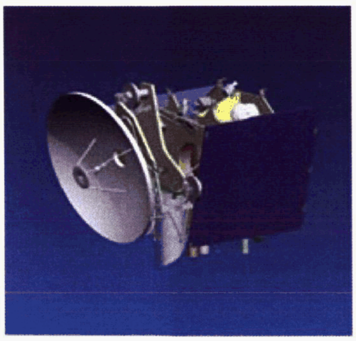


Figure 2. MRO Launch Configuration

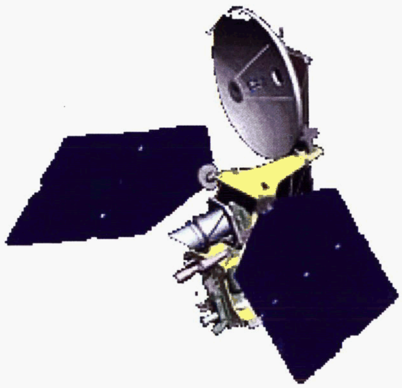


Figure 3. MRO Cruise Configuration

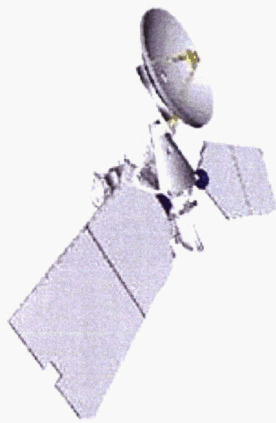


Figure 4. MRO Orbit Insertion & Aerobraking Configuration

<u>Gimbal Actuator Configuration</u>

Each two-axis gimbal consists of two identical gimbal actuators, structurally connected with application specific components. The core of the actuator is a 130-mm diameter two-phase brushless dc motor with a large number of poles in order to maintain smooth rotor velocity. The brushless motor is commutated using a resolver with the same number of poles as the motor to simplify the commutation logic. The motor directly drives the wave generator of the output harmonic gear component set through a bellows coupling. The bellows coupling was used to minimize speed ripple that would cause disturbances while operating. The harmonic gear is a size-32 HDC, standard-cup-type unit. The flexspline is mounted to the actuator housing and provides the torque reaction mount. The circular spline is mounted in a pair of preloaded angular-contact ball bearings. A multi-speed output resolver is installed between the angular-contact bearings for a compact assembly that measures the output position to the accuracy required for the MRO mission pointing. This arrangement of drive components provides a zero-backlash actuator with minimal mechanically generated disturbance sources and applies all of the externally generated loads directly to the harmonic gear teeth. A photo of a completed flight gimbal actuator assembly is shown in Figure 5.

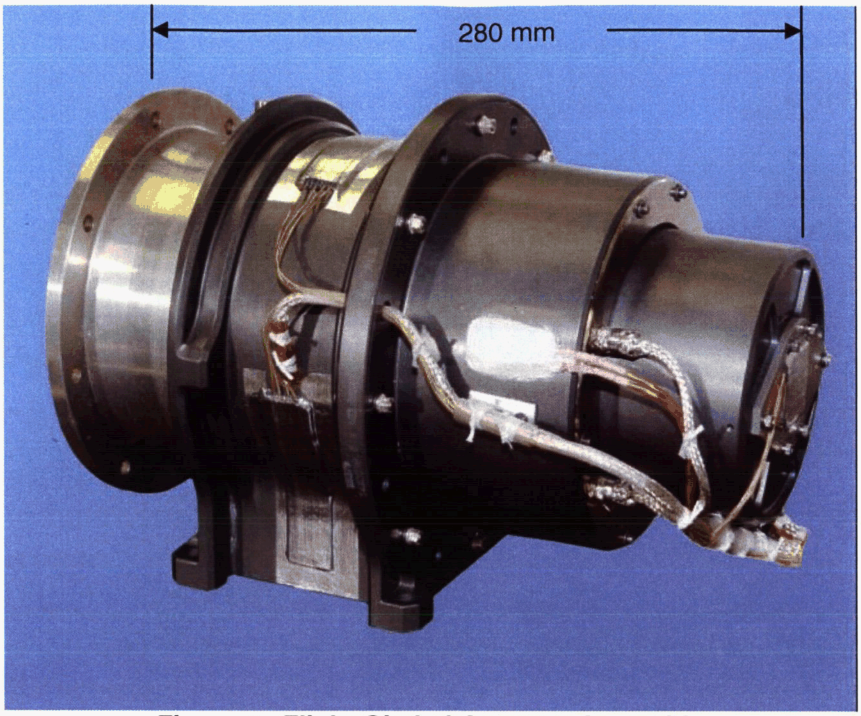


Figure 5. Flight Gimbal Actuator Assembly

The need for the harmonic gear to take the external loads led to the selection of the size of the harmonic gear in concert with the required output stiffness. The available volume and mass allocation for the actuators was minimal on the MRO spacecraft, requiring use of the smallest gears possible to achieve the required load and stiffness capability. The loads from the Mars orbit insertion and aerobraking phases needed to be taken into account along with all other applied loads during the mission. The mass of the solar array wings and the antenna assembly loading the output gear teeth in the acceleration environment of orbit insertion was one source of applied load. The force from aerodynamic loading on the large area array and antenna during aerobraking was another source of loading. These conditions together defined the magnitude of the loads that would be applied to the harmonic gear output teeth in flight. It was determined that the applied loads could be carried with appropriate margins by a size-32, cup-type harmonic gear component set.

The output torsional stiffness of the actuator axes affects the spacecraft dynamics. The spacecraft sensitivity to jitter disturbances and the attitude control system authority dictated a minimum natural frequency for the deployed solar array wings and High Gain Antenna. Since the gimbal actuator was a significant contributor to the overall appendage stiffness and the harmonic gear teeth were the load reacting devices, an output stiffness was required of the chosen harmonic gear that exceeded its standard specification significantly. The magnitude of this stiffness increase for the selected size of harmonic gear was within the range of experience for this type of application on other programs. Since there was no significant difference in the stiffness requirement when compared to other heritage programs, this was not considered a significant risk to the program.

The selection of the harmonic gear materials from the available set involved comparing the MRO requirements with those of previous flight programs in order to maintain as much heritage as possible. A fundamental tenet for this and many programs was to use only corrosion resistant materials in all space mechanisms as is commonly done in the medical, semiconductor, and food processing industries. The materials of the various pieces of the MRO harmonic gear component set are listed in Table 1. This combination of materials had been used successfully in several programs with stiffness and total lifetime revolutions requirements that were similar to MRO. Table 2 lists the other heritage applications of the same material combination with similar functional requirements. To maximize the life cycle capability of all of the actuator components, Penzane 2001-3PbNp Oil was utilized in the motor bearings and Rheolube 2004 grease was chosen for the lubricant throughout the harmonic gear. Other flight applications with the same material combinations for the harmonic gear components had also used Braycote® because of a much lower operating temperature requirement than MRO. Since the required operating temperature range did not necessitate the use of bray oil or grease, the Penzane lubricant family was chosen since it tends to demonstrate more life capability over the Bray series when the operating temperatures are moderate.

Table 1. Commercially Available Materials for MRO Gimbal Actuator Harmonic Drive Components

Component	Material	Processing
Circular Spline	15-5 PH	H1075
Flexspline	15-5 PH	H1075, Melonite case harden
Wave Generator	15-5 PH	H1075
Bearing	440C	

Note: The Melonite process is a Nitrocarburizing case hardening per SAE-AMS-2753B

<u>Hardware Verification</u>

The motor and gearbox assembly was the same for all of the six axis applications on MRO, allowing one life verification to be performed on the worst case loaded design that would encompass all of the other five axis applications on MRO. The life verification program for these actuators consisted of operating a harmonic drive assembly alone in ambient environment with standard lubrication, followed by operation of an assembled flight-like actuator assembly in a vacuum with thermal cycling and a representative wiring harness for loading of the output gear.

The lone harmonic drive gear assembly was operated in a standard gear test fixture used at the harmonic drive vendor. This was performed at ambient pressure and temperature in a bath of low viscosity commercial oil at an input speed of 1750 RPM and with an applied load on the output that matched the cable loading from harnesses. The wave generator was driven for 32 million revolutions with no sign of unusual wear or failure of the bearing or the gear teeth. This was done without incident to a rotational life of five times the flight requirement of 6.2 million input revolutions, indicating harmonic gear rotational lifetime was a low risk.

Next an Engineering Development Unit (EDU) actuator was operated in a flight-like configuration. The EDU actuator used the flight housings with output hardware that supported the High Gain Antenna cable management system. The cable management system was incorporated to cycle the cabling as well as provide flight like output loading to the harmonic gear. The High Gain Antenna application was chosen because it has the largest number of cables across any of the gimbal axes. The EDU actuator was operated over a total output angle of 340 degrees. The EDU actuator was run in a thermal/vacuum environment, with the temperature slowly cycled from -25°C to +40°C at the rate of one thermal cycle every 18 hours. The actuator was driven at a motor rate of 125 RPM for approximately 3.8 million input revolutions, and then run at 65 RPM for the remainder of the time. The EDU motor was driven with an industrial stepper motor driver, severely limiting visibility into the performance of the actuator. The stepper motor driver was used for this operation for several reasons: flight actuator drivers were not available for the start of the running, the motor is a two-phase brushless dc (not three phase), and the rotary life was believed to be low risk so the limited visibility was not considered to be significant. The EDU life setup is shown in Figure 6.

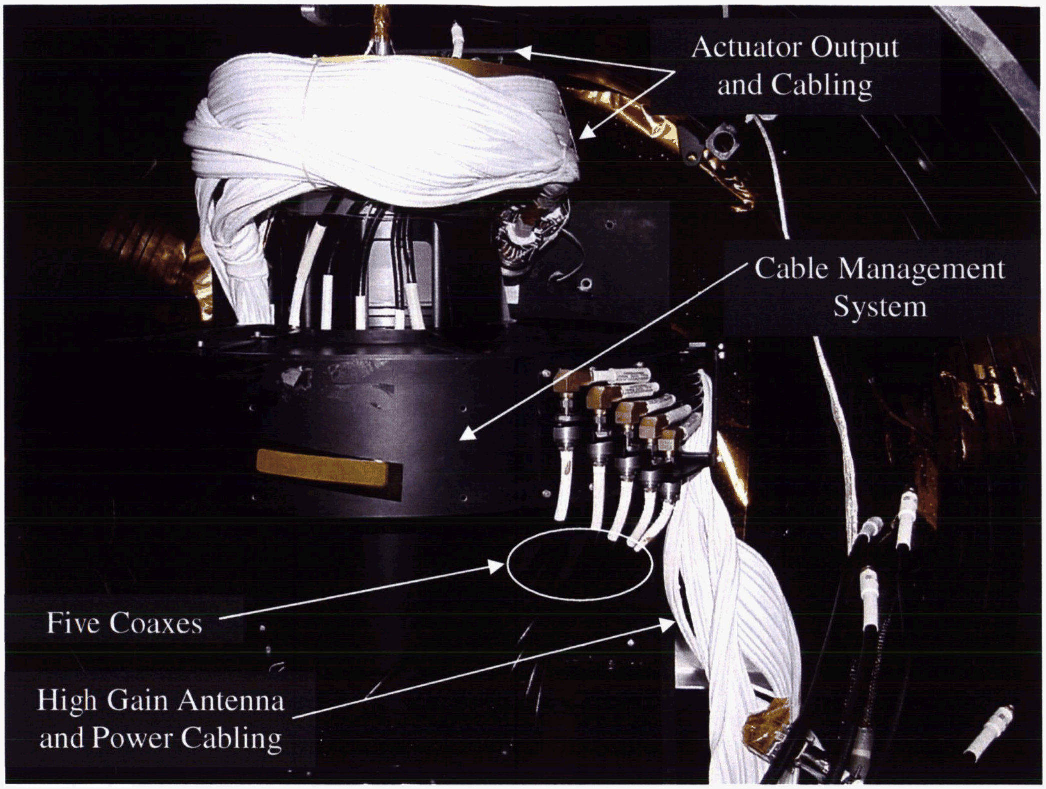


Figure 6. Engineering Development Unit Configuration in Vacuum Chamber

Table 2. Materials and Surface Finishes of Heritage Hardware from Previous Programs with Similar Stiffness and Life requirements as the MRO Applications

Program (Harmonic Drive Size)	Circular Spline	Flexspline	Lubricant
MRO (size 32)	15-5 PH H1075	15-5 PH H1150 Melonite	Penzane 2001-3Pb Rheolube 2004
Program #1 (size 20)	15-5 PH H1075	15-5 PH H1150 Melonite	Braycote 602 Bray 815Z oil
Program #2 (size 25)	15-5 PH H1150	15-5 PH H1075 Melonite	Penzane 2001-3Pb Rheolube 2004
Program #3 (size 32)	15-5 PH H1150 Melonite	15-5 PH H1075	Penzane 2001-3Pb Rheolube 2004
Program #4 (size 40)	15-5 PH H1150 Melonite	15-5 PH H1075	Penzane 2001-3Pb Rheolube 2004
Program #5 (size 32)	15-5 PH H1075	15-5 PH H1150 Melonite	Rheolube 2000 with 3% lead Napthenate

The EDU actuator operated up to 6.1 million input revolutions, when the output telemetry indicated the actuator was not following the input signals properly. Later analysis revealed there were indications of improper operation as early as 4 million input revolutions that were not diagnosed due to limitations in the test setup with the stepper motor driver. The EDU actuator was removed from the chamber, disassembled, and inspected. Figures 7, 8, and 9 show the condition of the harmonic gear teeth at this inspection. Note that the tooth profile was completely obliterated and damaged across the entire width of the flexspline teeth and most of the width of the circular spline.

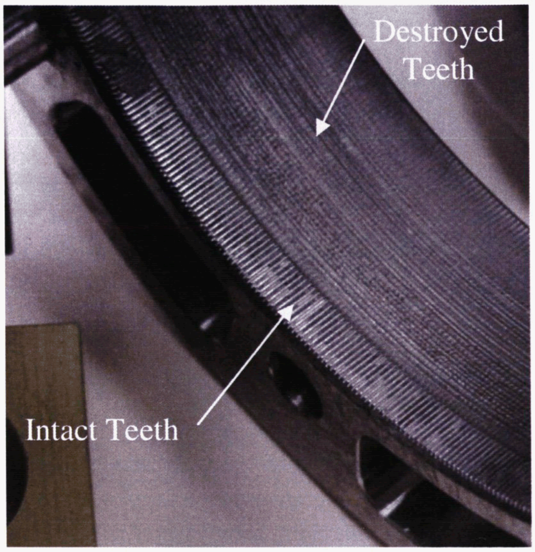


Figure 7. EDU Circular Spline Life Damage

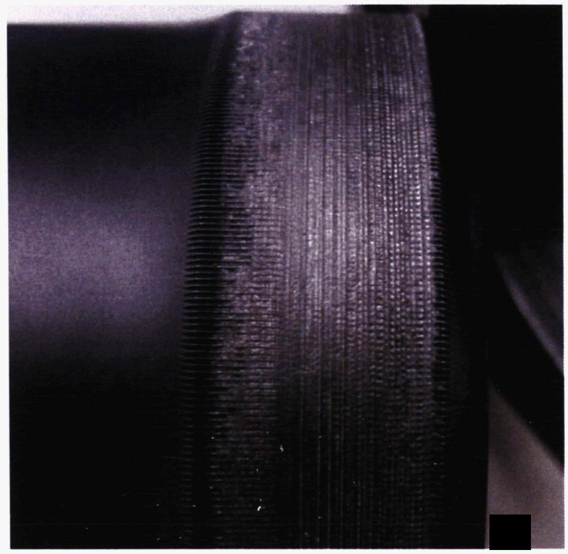


Figure 8. EDU Flexspline Life Damage

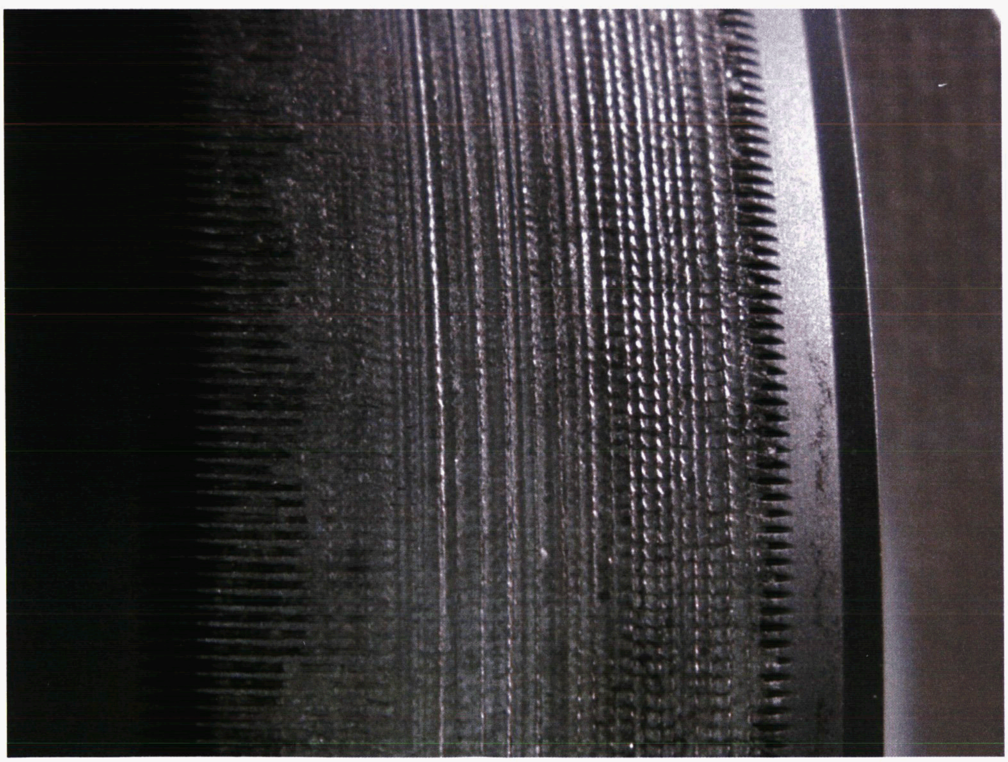


Figure 9. Magnification of EDU Flexspline Life Test Damage

Once the dramatic failure of the harmonic gear assembly was seen, a detailed review of the manufacturer's documents uncovered that there were some problems with the Melonite coating on the group of flexsplines that included the life unit, labeled Lot B. Two additional Melonite processing groups of flexsplines had been received at the time of the failure, labeled Lot C and Lot D. Operation of these units was started to determine if the Melonite processing on the EDU life test unit was the source of the failure. A harmonic assembly from each of the two remaining Melonite process groups was placed in a harmonic gear fixture (not in the gimbal actuator) and operated in vacuum with thermal cycling over a 12-hour period, an output load, and an input speed of 130 RPM. Additionally, one of the two units was tested using the same Penzane family of lubricants as the EDU and the other was tested using Braycote® 602. Both assemblies failed at approximately two million input revolutions. Figure 10 shows the failed splines that were tested with the Penzane family of lubricants. Figure 11 shows the failed splines that were tested with the Braycote® 602 lubricant. A significant result of these two tests was that the lubricant type made no difference at all, with both assemblies failing at nearly the same number of revolutions with the same type and level of damage.

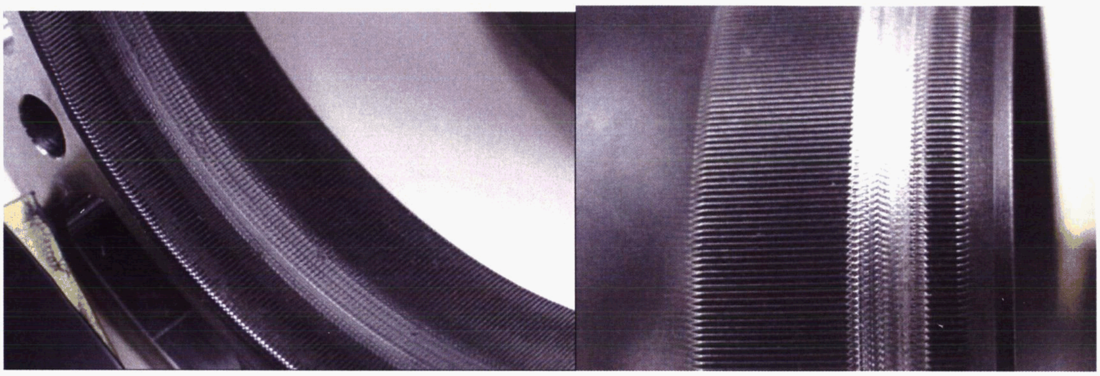


Figure 10. Circular Spline and Flexspline from Harmonic Only Test #1 Tested with Penzane

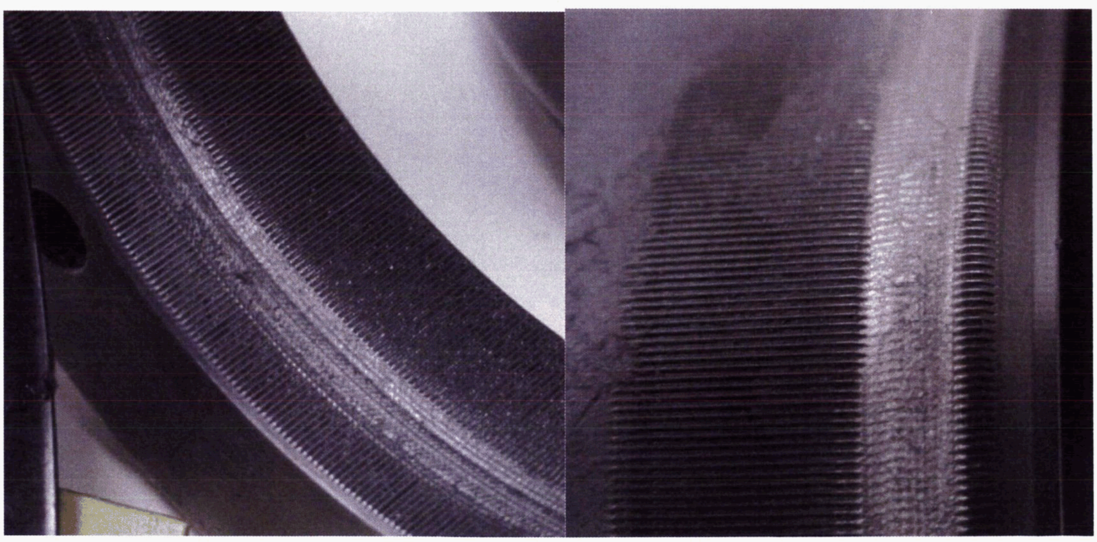


Figure 11. Circular Spline and Flexspline from Harmonic Only Test #2 Tested with Braycote® 602

Since the lubricant type made absolutely no difference in the life capability of the harmonic drive gear teeth, it was clear the problem was related directly to the material combination of the components and their internal stress level. The appearance of the failure surfaces gave the impression of a possible galling condition at work, but it was not clear if galling was the initiator of the failure or a consequence of the damage once the failure had been initiated. Rough mathematical analyses were performed to estimate the contact stress at the harmonic gear teeth from the preload and under the applied load in the operation. The constant external load in the harmonic fixture runs represented the worst-case load from the cable management system and was responsible for a roughly 40% increase in tooth contact stresses over the preload. The estimated contact stresses in the EDU life test were in the realm of 750 MPa. The galling threshold listed for a 15-5 PH stainless steel contact pair is around 14 MPa. This indicated that the contact stresses compared to the galling threshold for the selected materials was a strong candidate for the cause of the failures. To minimize the schedule time to a solution, the next group of tests used Nitronic 60 (another available harmonic gear material) with a listed galling threshold value greater than 345 MPa. Nitronic 60 was identified as a candidate in addition to other standard commercial materials for the harmonic gear assemblies, like nodular iron. Also, the output stiffness of the assembly was reduced so the internal harmonic gear preload (and with it the internal tooth stresses) could be reduced as well. At this point in the project schedule, the flight solar array panels had been fabricated and their stiffness was measured, allowing reduction of the stiffness margin for these panels in the MRO spacecraft stability analysis. This made it possible to reduce the gimbal actuator stiffness requirement significantly.

To investigate these issues within the remaining program schedule, three readily available harmonic drive assemblies were procured with different material combinations and tested on the harmonic gear fixture. The material combinations consisted of a unit with a nodular iron circular spline and an E4340 flexspline. The second unit had a Nitronic 60 circular spline and an E4340 flexspline. The third assembly was composed of a Nitronic 60 circular spline with an E4340 flexspline processed with a Melonite surface. Additionally, the internal preload of the harmonic gear (to obtain the new required output stiffness) was reduced in order to lower the gear tooth internal stresses. Table 3 lists all of the life test units and the material and lubrication configuration for each.

Table 3. Selected Harmonic Gear Material and Lubricant Configurations

Test Unit	Circular Spline Material	Flexspline Material	Flexspline Surface Treatment	Lubricant
Harmonic drive vendor Unit #1	15-5 PH, H1075	15-5 PH, H1075	Melonite, Lot C	Rheolube 2004 grease Nye 2001-3PbNp oil
EDU Assembly #1 (in actuator)	15-5 PH, H1075	15-5 PH, H1075	Melonite, Lot B	Rheolube 2004 grease Nye 2001-3PbNp oil
LM Harmonic Assembly #1 (same unit as Harmonic drive vendor Unit #1)	15-5 PH, H1075	15-5 PH, H1075	Melonite, Lot C	Rheolube 2004 grease Nye 2001-3PbNp oil
LM Harmonic Assembly #2	15-5 PH, H1075	15-5 PH, H1075	Melonite, Lot D	Braycote® 602
LM Harmonic Assembly #3	Nodular Iron	E4340 Steel	None	Rheolube 2004 grease Nye 2001-3PbNp oil
LM Harmonic Assembly #4	Nitronic 60	E4340 Steel	None	Rheolube 2004 grease Nye 2001-3PbNp oil
LM Harmonic Assembly #5	Nitronic 60	E4340 Steel	Melonite, Lot E	Rheolube 2004 grease Nye 2001-3PbNp oil
EDU Assembly #2	Nitronic 60	E4340 Steel	None	Rheolube 2004 grease Nye 2001-3PbNp oil

Note: LM stands for Lockheed Martin

The test units for LM Harmonic Assemblies #3, #4, and #5 were tested in the harmonic drive test fixture, in the environment, with the applied load and input speed listed in Table 4. The stiffness of the units was measured at the start of the testing as shown in column 2 of Table 4. The life tests already discussed are listed in Table 4 for completeness.

Table 4. Life Unit Stiffness, Environments, Conditions, and Revolutions at Failure

Test Unit	Initial Stiffness with 45/140 N·m Applied Torque (N·m/rad)	Environment	Applied Load on Output	Input Speed	Approx. Revolutions to failure
Harmonic drive vendor Unit #1	Approx. 68,000	Ambient	11.3 N·m	1750 RPM	No Failure
EDU Assembly #1 (in actuator)	> 56,000	Vacuum, -25°C/+40°C 18 hour cycles	HGA Cable Harness	125 RPM & 65 RPM	4M
LM Harmonic Assembly #1 (same unit as Harmonic drive vendor Unit #1)	Approx. 68,000	Vacuum, -15°C/+40°C 12 hour cycles	11.3 N·m	130 RPM	2M
LM Harmonic Assembly #2	69,000/75,000	Vacuum, -15°C/+40°C 12 hour cycles	11.3 N·m	140 RPM	2M
LM Harmonic Assembly #3	56,000	Vacuum, -15°C/+40°C 12 hour cycles	11.3 N·m	140 RPM	4M
LM Harmonic Assembly #4	55,600/60300 56,500 (28 N·m) 43,800 (28 N·m)	First 3M revs Vacuum, -15°C/+40°C 12 & 24 hour cycles Next 3M revs Vacuum, +23°C constant Next 8M revs Vacuum, -10°C constant	11.3 N·m	140 RPM	No Failure @ 14M
LM Harmonic Assembly #5	54,600/61,000 43,800 (28 N·m)	Vacuum, -10°C/0°C constant	11.3 N·m	140 RPM	No Failure @ 9.1M
EDU Assembly #2	37,300 to 46,300 (28 N·m)	Vacuum, -10°C/+10°C constant	HGA Cable Harness	60 RPM	No Failure @ 13M

Key results of the above are as follows:

- The high input speed of the harmonic gear only operation performed at the harmonic drive vendor allows the lubricant to support high contact stresses that would otherwise result in complete failure of the gear teeth.
- The difference from the EDU Assembly #1 unit and the LM Harmonic Assembly #1 was the Melonite lot and coating details. The results of the LM Harmonic Assembly Unit #1 showed that the Melonite coating was not involved in the failure.
- The applied output load, while small when compared to the maximum torque capability of the harmonic gear, made a difference in the revolution life to failure. Note that the EDU life unit exhibited failure around 4 million revolutions, while assemblies #1 and #2, with a constant applied load, failed at 2 million revolutions. This difference could be due to the EDU assembly #1 using a cable wrap harness, with a variable load depending on output position, compared to a constant load.

- LM Harmonic Assemblies #1 and #2, with failure at a similar number of revolutions using very different lubricants, indicated the lubricant was not a significant player in the failure mechanism. This eliminated the lubricant as a variable in further failure investigation.
- The LM Harmonic Assembly #3, with nodular iron and a reduced preload, had a longer life to failure than the stainless steel, supporting the galling hypothesis. Note the regions of damage in Figure 12.
- The LM Harmonic Assemblies #4 and #5 used Nitronic 60 for the circular spline, the highest galling threshold material that could be obtained in a harmonic gear assembly. The late date of the testing permitted a reduction of the output stiffness to two-thirds of the initial values used for the gimbal actuators. The stiffness of the unit in Harmonic Assembly #4 was reduced as the operation progressed and a method of setting the harmonic gear preload was established. The result of the run was no failure at all with some minor wear of the harmonic gear teeth, as shown in Figure 13. The schedule dictated that the first successful combination be used, so the final material selection was a Nitronic 60 circular spline and a flexspline of E4340 with no additional surface processing. Note that schedule dictated changing more than one variable at a time.
- The setting of the internal preload of the harmonic gear assembly during assembly at the harmonic drive vendor was critical in achieving the required output stiffness without compromising the reliability of the harmonic gear assembly.

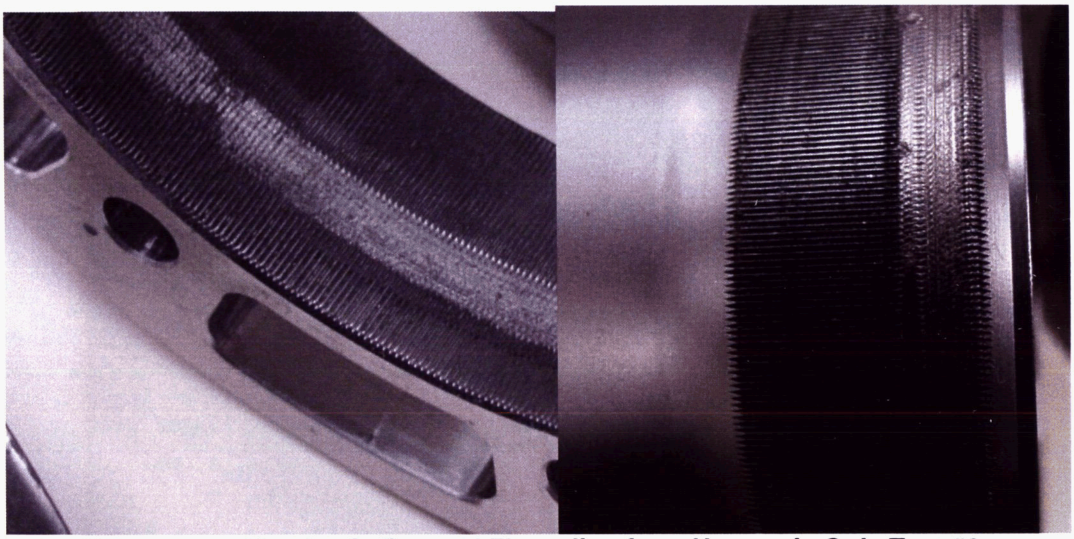


Figure 12. Circular Spline and Flexspline from Harmonic Only Test #3
Note the damaged regions show where the flexspline was engaged with the circular spline.

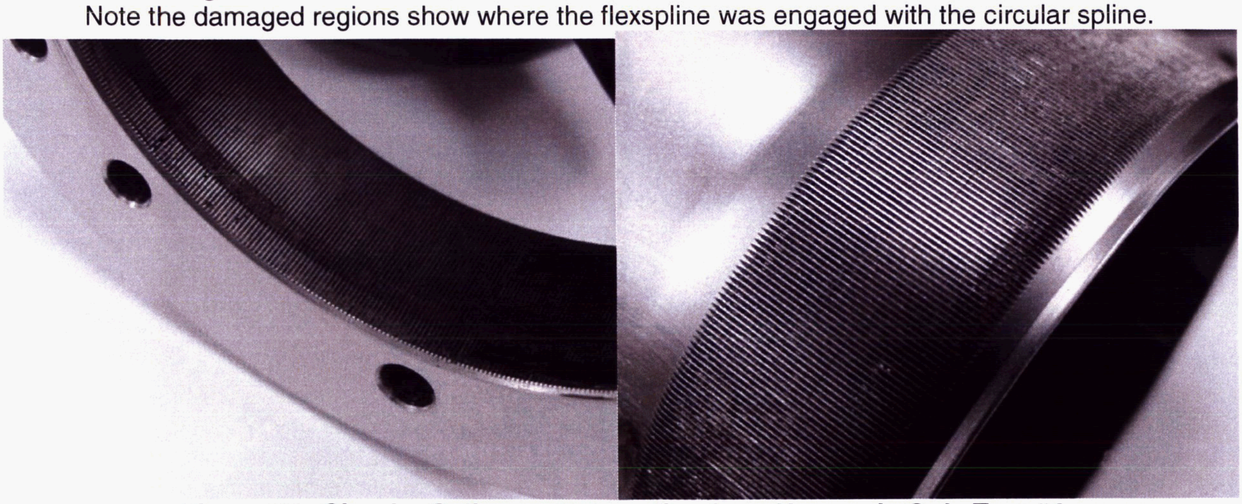


Figure 13. Circular Spline and Flexspline from Harmonic Only Test #4
Note the slight wear region on the circular spline showing where the flexspline was engaged with it.

Harmonic Gear Assembly Internal Preload Setting

The cup type harmonic gear assembly has zero backlash due to the angle of approach, along the rotation axis, of the flexspline teeth relative to the circular spline teeth (Figure 14). This arrangement leads to a two-sloped stiffness curve of the output of a harmonic gear. As torque is applied to the gear, the cup flexes to allow more of the width of the teeth to engage with the circular spline. As more tooth area is engaged, more load sharing occurs and the stiffness increases. Once the angle between the teeth of the splines is reduced to near zero, the stiffness has reached its maximum value. As the torque is increased from this point, the stiffness is related to deflection of the individual teeth and the change in roundness of the housing and circular spline. Figure 15 shows a stiffness curve for a typical harmonic gear assembly prior to increasing the internal preload for stiffness improvement.

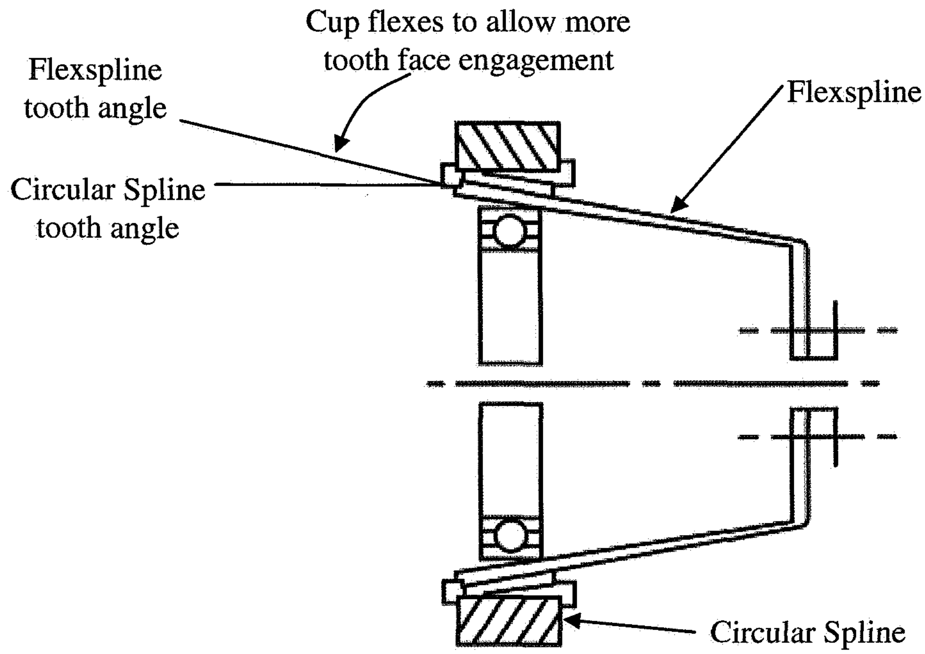


Figure 14. Harmonic Drive Cross Section Showing Significant Source of Variable Stiffness
Note the largest stresses on flexspline teeth occur at the point of initial contact, the open end of the cup.
Drawing courtesy of Harmonic Drive, LLC

Increasing the output stiffness of a harmonic drive assembly involves increasing the diameter of the wave generator in the area where it forces contact between the teeth of the two splines. This is accomplished by using a different wave generator plug with a larger major diameter of the oval. As the diameter is increased, the flexspline cup is deflected in the direction of engaging more of the face of the teeth. This has the same effect as increasing the torque on a nominal unit in the low stiffness region. As the wave generator plug size is increased, the low stiffness region gets smaller. This trend continues until the stiffness curve is essentially straight. At this point, a further increase in the diameter of the wave generator plug will increase the overall gear assembly stiffness and significantly increase the internal tooth stresses. Figures 16 through 19 show the how the shape of the stiffness curve changes with different wave generator plugs in a harmonic drive assembly. The low stiffness region is very evident in Figure 16. A larger plug, after insertion into the wave generator bearing may nearly eliminate the low stiffness region, as seen in Figure 17. Figures 18 and 19 show the next two larger size wave generator plugs, without a significant change in the shape of the stiffness curve. The desired operating point for the flight plug is smallest wave generator plug that exhibits a fairly straight curve. If there is any uncertainty between units, the smaller one would always be installed to guarantee that the gear teeth were not being jammed together with high, and unknown, internal stresses. The wave generator plug used in this example would be Figure 17's.

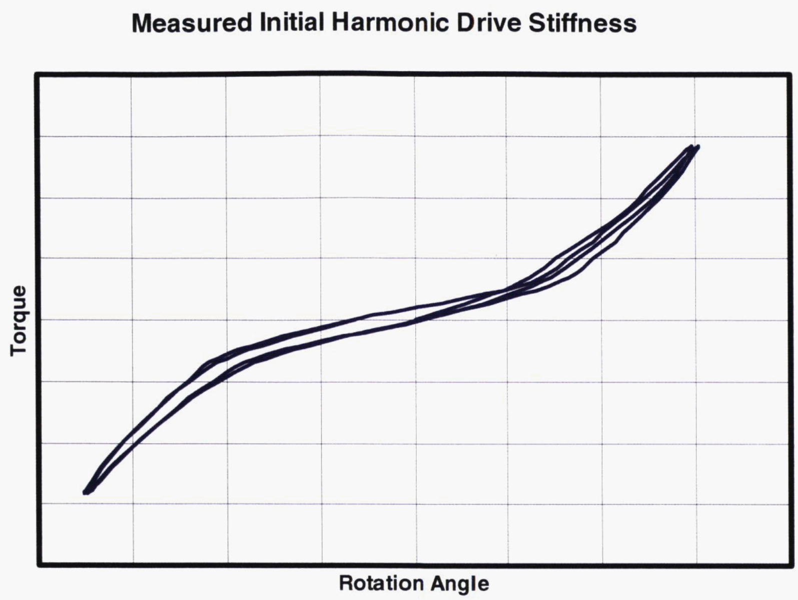


Figure 15. Typical Harmonic Gear Assembly Stiffness
Note two regions of stiffness: low near zero torque and larger at high torques.

<u>The following figures are from measured data on the flight harmonic gear assemblies:</u>
(Note: the following figures' axes are rotated relative to Figure 15)

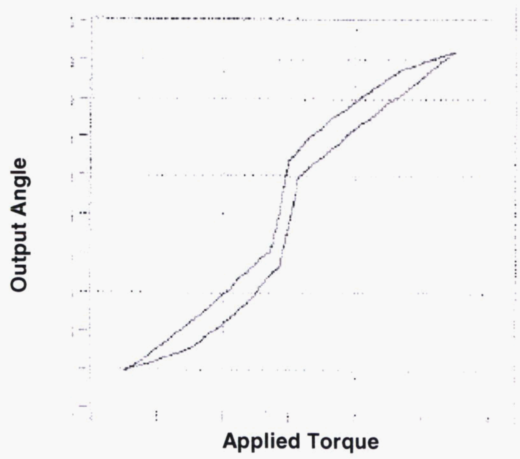


Figure 16. Stiffness Curve Showing Low Stiffness Region

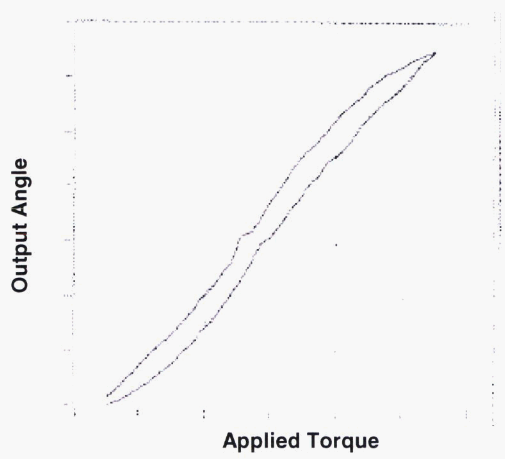
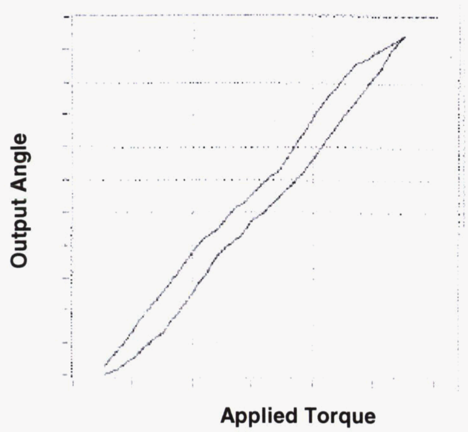


Figure 17. Stiffness Curve with Minimal Low Stiffness Region



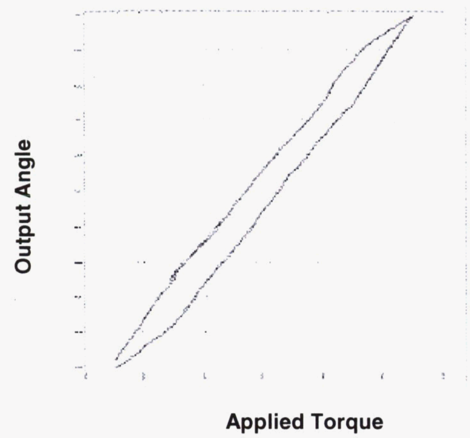


Figure 18. Next Larger Plug Size Over Figure 17 Showing Little Change

Figure 19. Next Larger Plug Size Showing Showing a Small Change in Shape

Conclusions and Lessons Learned

The most significant result of the failure investigation was determining that the internal stresses due to the preload and the cable harness loading caused the premature failure of the EDU harmonic gear assembly. In order to maintain reliability for long life applications, increasing the wave generator plug size (a service provided by the harmonic drive vendor) is an acceptable technique up to the point where the low stiffness region is eliminated. This is the maximum stiffness enhancement achievable without compromising the reliability of the gear assembly for long life. Any further increase in the wave generator plug size will increase the stiffness at the cost of reducing the life of the unit. For minimal life applications, increasing the stiffness beyond this point may still be acceptable.

Stiffness enhanced harmonic gears are very sensitive to the externally applied load and test environment. Since the failure mode is galling, the presence of any gas (nitrogen, for example) severely compromises the test results. Life capabilities from previous heritage programs had been successful and so the initial gear material for the MRO gimbal actuators was considered acceptable and robust. However, some of the heritage operation had been performed in nitrogen, instead of vacuum. When enhanced stiffness is required in a harmonic gear application and it is not being used in a preload configuration represented by Figure 16, the situation is sensitive to possible galling of the harmonic gear teeth. Performing harmonic gear component operation at loads above the planned level to increase the tooth contact stresses should be considered. This will demonstrate if internal stress margin exists in the hardware. Also note that running a harmonic gear at a high input speed to reduce the operating time is not adequate. The high-speed condition may function with no incidents while low speed operation may catastrophically quit functioning. Finally, operating a unit at nominal contact stress levels to a larger number of revolutions than planned is a necessary, but not complete, margin demonstration program. A catastrophic failure may be lurking just a few megapascals away from the nominal value.

References

Harmonic Drive, LLC web site, www.harmonic-drive.com/support/principals.htm, description of harmonic gear assembly operation

Acknowledgements

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