



Office of the NASA Chief Engineer

NASA TECHNICAL STANDARD

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Superseding NASA-STD-5017A w/Change 1

METRIC/SI (ENGLISH)

DESIGN AND DEVELOPMENT REQUIREMENTS FOR MECHANISMS

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DOCUMENT HISTORY LOG

Status	Document Revision	Change Number	Approval Date	Description
Baseline			2006-06-13	Initial Release
Revision	A		2015-07-31	General revision. Language for nearly every requirement was altered for clarity. Some requirements were deleted or combined with other requirements, and new requirements have been added based on lessons learned with the document and engineering state of the art. Rationale was added for all requirements. Appendix A, Best Practices for Mechanisms, was added for guidance.
	A	1	2016-05-31	Revalidated w/Administrative/Editorial Changes—This NASA Technical Standard was reviewed and no technical changes resulted. Administrative changes to number requirements, add a Requirements Compliance Matrix as an appendix, and conform to the current template were made, along with editorial corrections.
Revision	B		2022-12-06	Significant changes were made to this NASA Technical Standard. It is recommended that it be reviewed in its entirety before implementation. Key changes: • The torque and force margin section was revised to more closely align with AIAA S-114A-2020. Some terminology was also altered to facilitate technical understanding and communication of torque and force margin. • The indication of status requirement was reinstated, in a slightly altered form, based on user feedback. • Many requirements had wording changes or changes to the rationale or guidance to improve clarity or address interpretation issues based on user

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				feedback. Some requirements on clearances were split into multiple “shall” statements to clarify intent. • A requirement on lubrication process was deleted because it is redundant to requirements in NASA-STD-6016. • Requirements to treat quick release pins, threaded interfaces actuated in service, and mechanisms that release degrees of freedom as mechanisms were deleted in favor of explicitly identifying such devices as mechanisms in the document scope. • Pressurized Electroslag Remelting (PESR) was added as an acceptable steelmaking process for bearings used in high precision, low torque ripple, or long- life applications. • New content on force/torque margin calculation, minimizing stress relaxation in springs, design best practices for dry film lubrication use, and redundant rotating surfaces was added to the Best Practices in Appendix A, which was otherwise extensively edited for clarity and readability.
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FOREWORD

This NASA Technical Standard is published by the National Aeronautics and Space Administration (NASA) to provide uniform engineering and technical requirements for processes, procedures, practices, and methods endorsed as standard for NASA programs and projects, including requirements for selection, application, and design criteria of an item.

This Standard establishes uniform design, development, and verification requirements for mechanisms and mechanism components whose correct operation is required for safety or mission success.

Requests for information may be submitted via “Email Feedback” at <https://standards.nasa.gov/>. Requests for changes to this Standard should be submitted via MSFC Form 4657, Change Request for a NASA Engineering Standard.

Original Signed by Adam West for

December 6, 2022

Ralph R. Roe, Jr.
NASA Chief Engineer

Approval Date

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DESIGN AND DEVELOPMENT REQUIREMENTS FOR MECHANISMS

1. SCOPE

1.1 Purpose

This Standard establishes common NASA design, development, and test requirements for mechanisms whose operation is required for safety or mission success.

1.2 Applicability

1.2.1 This Standard is applicable to space flight mechanisms that are designed, built, or acquired by or for NASA, though it may also serve as a useful guidance document for other systems such as ground support equipment (GSE). Valves, ordnance-actuated mechanical devices, quick-release pins (also known as push in and pull [PIP] pins), threaded interfaces designed to be actuated in service, and interfaces designed to release mechanical degrees of freedom are subject to failure modes typical of mechanisms and are therefore considered mechanisms subject to the requirements of this document. See Appendix A, section A.2.8, for more information on quick release pins. This Standard does not address human factors requirements. Adherence to this Standard does not in and of itself exempt a mechanism from any fault tolerance or hazard control requirements. The requirements and best practices in this Standard may serve as a useful basis for evaluating rationale for variances to fault tolerance requirements that may be proposed for mechanisms.

1.2.2 This Standard is approved for use by NASA Headquarters and NASA Centers and Facilities, and applicable technical requirements may be cited in contract, program, and other Agency documents. This language applies to the Jet Propulsion Laboratory (a Federally Funded Research and Development Center), other contractors, recipients of grants, cooperative agreements, or other agreements only to the extent specified or referenced in the applicable contracts, grants, or agreements.

1.2.3 References to “this Standard” refer to NASA-STD-5017B; references to other documents state the specific document information.

1.2.4 Verifiable requirement statements are designated by the acronym “DDMR” (Design and Development of Mechanisms Requirement), numbered, and indicated by the word “shall.” This Standard contains 90 requirements. To facilitate requirements selection by NASA programs and projects, a Requirements Identification Matrix is provided in Appendix C.

1.2.5 Explanatory or guidance text is indicated in italics beginning in section 4. The terms “may” or “can” denote discretionary privilege or permission, “should” denotes a good practice and is recommended but not required, “will” denotes expected outcome, and “is/are” denotes descriptive material or a statement of fact.

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1.3 Tailoring

Tailoring of the requirements in this Standard for application to a specific program or project is acceptable when formally approved by the delegated NASA Technical Authority in accordance with NPR 7120.5, NASA Space Flight Program and Project Management Requirements, and documented in program or project requirements.

2. APPLICABLE DOCUMENTS

2.1 General

2.1.1 Documents listed in this section contain provisions constituting requirements of this Standard as cited in the text. Latest issuances of cited documents apply unless specific versions are designated. Obtain approval from the delegated NASA Technical Authority to use a version other than as designated.

2.1.2 Access applicable documents at <https://standards.nasa.gov> or obtain documents directly from the Standards Developing Body or other document distributors.

Note: References are provided in Appendix B.

2.2 Government Documents

NASA

NPR 7120.5, NASA Space Flight Program and Project Management Requirements

2.3 Non-Government Documents

American Bearing Manufacturing Association (ABMA)

ANSI/ABMA/ISO 3290-1, Rolling Bearings - Balls - Part 1: Steel balls

ASTM International

ASTM F2094/F2094M, Standard Specification for Silicon Nitride Bearing Balls

ASTM F2215, Standard Specification for Balls, Bearings, Ferrous and Nonferrous for Use in Bearings, Valves, and Bearing Applications

2.4 Order of Precedence

2.4.1 The requirements and standard practices established in this Standard do not supersede or waive existing requirements and standard practices found in other Agency documentation.

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2.4.2 Conflicts between this Standard and other requirements documents will be resolved by the delegated NASA Technical Authority.

3. ACRONYMS, ABBREVIATIONS, AND DEFINITIONS

3.1 Acronyms and Abbreviations

°C	degrees centigrade
°F	degrees Fahrenheit
AA	arithmetic average
ABEC	Annular Bearing Engineering Committee
ABMA	American Bearing Manufacturing Association
AGMA	American Gear Manufacturers Association
ANSI	American National Standards Institute
ASME	American Society of Mechanical Engineers
ASTM	ASTM International (Organization formerly known as American Society for Testing and Materials)
BLDC	brushless direct current
CEVM	consumable electrode vacuum melted
cm	centimeter
CMG	control moment gyroscope
CR	controlled radius
CVD	chemical vapor deposited
DC	direct current
DDMR	Design and Development for Mechanisms Requirement
DFL	dry film lubricant
EC	electronically commutated
EHD	elastohydrodynamic
EMC	electromagnetic compatibility
EMF	electromotive force
EMI	electromagnetic interference
EVA	extravehicular activity
FS	factor of safety
GN2	gaseous nitrogen
GSE	ground support equipment
HDBK	handbook
HRC	Rockwell C hardness
HST	Hubble Space Telescope
Hz	Hertz
ISO	International Organization for Standardization
ISS	International Space Station
JPL	Jet Propulsion Laboratory
ksi	thousand pounds per square inch
MAC	multiply alkylated cyclopentane

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MIL	military
mm	millimeter
MPa	megapascals
MSFC	Marshall Space Flight Center
NASA	National Aeronautics and Space Administration
NLGI	National Lubricating Grease Institute
Pa	pascal
PAO	polyalphaolefin
PESR	pressurized electrosag remelting
PFPE	perfluoropolyalkylether
PIP	push in and pull
PRC	process
PRF	performance
PTFE	polytetrafluoroethylene
PVD	physical vapor deposited
R _a	average roughness
R _{sk}	roughness profile skewness
RP	Reference Publication
RQMT	Requirement
SI	Système Internationale, or metric system of measurement
SP	Special Publication
STD	Standard
VAR	vacuum arc remelted
VIM	vacuum induction melted

3.2 Definitions

3.2.1 Definitions of Variables

σ	standard deviation
τ	motor torque
f_{error}	frequency of strain wave gear drive output error
FS_a	safety factor applied to the torque or force required to achieve a specified acceleration of the driven component
FS_f	safety factor applied to the “fixed” resisting torques that are not strongly influenced by environmental conditions and cycles
FS_v	safety factor applied to the “variable” resisting torques whose values may change with environmental conditions and cycles
I	current drawn by a motor
K_{crit}	life test criticality factor
K_{cycle}	the factor applied to number of cycles within a defined cycle range when calculating mechanism life test cycles
K_f	factor applied to each individual fixed resistive torque in a torque margin calculation
K_{lub}	lubrication factor

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K_m	motor constant
K_t	torque constant
K_v	factor applied to each individual variable resistive torque in a torque margin calculation
K_v	factor applied to each individual variable resistive torque in a torque margin calculation
N	harmonic of interest
N_{cycle}	the number of cycles within a defined cycle range when calculating mechanism life test cycles
P	resistive power loss
R_t	resistance across motor terminals
T_a	the torque required to achieve a specified acceleration of the driven component
T_{avail}	the minimum available torque generated by the driving or holding component
T_{delta}	magnitude of the difference between total resisting torque measured during an assembly-level torque margin test and the sum of the resisting torques for all contributors measured during unit-level torque margin tests, $T_{res\,assy} - T_{res\,unit}$
T_f	individual fixed resistive torques that are not strongly influenced by environmental conditions and cycles
$T_{res\,assy}$	total resisting torque measured during an assembly-level torque margin test
$T_{res\,unit}$	sum of the resisting torques for all contributors measured during unit-level torque margin tests
T_v	individual resistive torques that may vary over environmental conditions and cycles

3.2.2 Definitions of Terms

Bearing Preload: The equal and opposite axial load on each of two bearings or bearing sets mounted on a common axis.

Cogging Torque: See “Detent Torque.”

Cold Welding: A phenomenon in which similar adjacent metal surfaces molecularly bond to one another given sufficient cleanliness, time, and contact pressure.

Contact Ellipse: The area of contact between the ball and raceway that occurs as a result of elastic deformation of both parts under load.

Coulomb Friction Torque: The parasitic torque in a bearing due only to the sliding friction generated by the relative motion of the moving parts in a bearing. *Note: Generally, this torque is measured at a low enough speed that viscous drag on the bearing from the liquid lubricant is negligible.*

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Deployable: A component that is moved from a stowed position on the spacecraft to an extended position while remaining connected to the spacecraft.

Design Factor of Safety: A multiplying factor to be applied to limit loads or stresses for the purposes of analytical assessment.

Detent Torque: The amount of magnetic torque that a motor produces to resist motion when it is not energized. Also known as “cogging torque.”

Dimensional Analysis: A systematic calculation addressing how the dimensions or clearances of a part or assembly vary due to the accumulated effects of specified dimensions, manufacturing tolerances, environmental exposure, and applied loads.

Dynamic Clearance: The minimum distance between two entities when the entities are in motion and subjected to service environments.

Dynamic Torque (or Force) Margin: A measure of the excess torque (or force) available to provide a required minimum acceleration to a body.

Functional Test: A test performed to assess the operability of the item under test, but typically not assessing compliance with all specified requirements and often performed at the conclusion of one environmental test to screen for damage before proceeding with the next environmental test. See Performance Test.

Hard Preload: A bearing preload approach in which two bearings or bearing sets are axially constrained in opposition to each other by a precision mounting structure (shaft and housing) whose axial stiffness is greater than that of the bearings, so as to elastically deflect the rolling elements into the raceways by a small, controlled amount and provide the equal and opposite axial load on each of the two bearings or bearing sets. Often called “Solid Preload.” *Note: Implementation is characterized by custom-ground bearing ring faces being clamped against each other, sometimes with precision shim washers or stiff precision spacers in between.*

Holding Torque (or Force) Margin: A measure of the excess torque (or force) available to maintain position in the presence of disturbances.

Kickoff Spring: A spring intended to overcome forces present during initial separation of a contacting interface.

L0.05 Life: The life at which 0.05 percent of the bearings in an application can be expected to have failed due to rolling contact fatigue or, alternatively, the life at which 99.95 percent of the bearings will still be operating. *Note: The L0.05 life of the bearing is theoretical and may not represent actual service life of the bearing.*

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Lubricant: A material with low shear resistance that reduces friction and wear. *Note: Lubricants can include gases, reaction films, liquids, and solids.*

Mechanical Stop: A feature intended to prevent a mechanism component from extending beyond a prescribed travel limit by physically impeding motion of the component, also known as a hard stop.

Mechanism: An assembly in which one mechanical part moves relative to another mechanical part.

Microstepping: A method of achieving smoother motion, smaller step angles, or more precise positioning of a stepper motor by using a controller to rotate the stator magnetic field through an arbitrary stepping angle that is less than the cardinal step size.

Motor Constant: A figure of merit used to evaluate a motor's ability to transform electrical power to mechanical power and compare the relative efficiencies and output power capabilities of different motors, defined as

$$K_m = \frac{\tau}{\sqrt{P}} \quad \text{or} \quad K_m = \frac{K_t}{\sqrt{R_t}}$$

here K_m is the motor constant, τ is the motor torque, P is the resistive power loss, K_t is the torque constant, and R_t is the resistance across the motor terminals.

Performance Test: A test conducted to demonstrate operation to all specification requirements.

Protoflight Testing: A strategy where no dedicated qualification test article exists and verification testing is performed on flight hardware, typically exposing flight hardware to environments above design environments but below qualification environments or any other limit detrimental to the test unit.

Pull-in Torque: The maximum constant torque for a given speed, inertial load, and controller under which a stepper motor will accelerate from rest to operating speed, stop, or reverse direction in synchronism with input pulses (i.e., without loss of steps). *Note: Pull-in torque is determined using 100 percent pulse duty cycle unless defined otherwise.*

Pull-in Torque Margin: A measure of the excess torque (or force) available to accelerate a stepper motor from rest to operating speed, stop a stepper motor, or reverse stepper motor direction in synchronism with input pulses (i.e., without loss of steps) for a given speed, inertial load, and controller.

Pull-out Torque: The torque at which a stepper motor begins to lose synchronization as its torque load is increased while operating at its desired speed. *Note: Typically, a curve plotting torque versus step rate (or rotor speed) is produced. This curve represents the maximum torque that the stepper motor can supply to a load at any given speed. Any torque or speed required that*

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exceeds this curve will cause the motor to lose synchronization. Pull-out torque is affected by drive voltage and phase switching techniques.

Quick-Release Pin: A pin with a fast-acting retention and release mechanism built into the pin. *Note: Also known as a PIP (push in and pull) pin. Quick release pins come in a variety of forms but generally utilize a spring-loaded central shaft to actuate one or more retention balls that are retained in the housing via swages.*

R_a, Surface Roughness: The arithmetic average of the absolute value of the departure of the filtered roughness profile measured from the mean line. *Note: R_a values are normally specified in micrometers or microinches.*

R_{sk}, Roughness Profile Skewness: A measure of the asymmetry of the surface profile about the mean. A negative skewness indicates that a greater percentage of the profile is above the mean line and a positive value indicates that a greater percentage is below the mean line.

Separation Nut: A segmented nut in which the segments are held together for retention of a bolt and then allowed to release through a mechanical action that is triggered on command. The release may be triggered via pyrotechnics or, when low source shock is necessary, non-explosive means such as shape memory alloys.

Service Life: The period beginning with manufacture of a component and ending with completion of its specified use. All significant loading cycles or events during testing, transportation, storage, launch preparation, lift-off, ascent, on-orbit operations, entry, descent, landing, recovery, refurbishment, reacceptance, and reuse are considered part of service life.

Servomechanism: An automatic device that uses error-sensing negative feedback to correct the performance of a mechanism.

Soft Preload: A bearing preload approach using deflected spring or diaphragm elements whose axial stiffness is less than that of the bearings to provide the equal and opposite axial load on each of two bearings or bearing sets. Often called “Spring Preload.” *Note: Implementation is characterized by one bearing ring or set of rings being free to translate axially in a housing or on a shaft under the action of the spring or diaphragm elements.*

Solid Preload: See “Hard Preload.”

Spring Preload: See “Soft Preload.”

Static Clearance: The minimum distance between two entities when the entities are at rest.

Static Torque (or Force) Margin: A measure of the excess torque (or force) available to overcome resistance to motion.

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Step Integrity: The ability of a stepper motor to achieve the number of steps commanded without loss or gain of steps.

Step Stability Analysis: An analytical method of quantifying stepper motor system performance by evaluating dynamic response to step commands.

Torque Constant: The ratio of the motor torque to the current drawn by the motor, defined as

$$K_t = \frac{\tau}{I}$$

where K_t is the torque constant, τ is the motor torque, and I is the current drawn by the motor.

Torque (or Force) Margin: A measure of the excess torque (or force) of a mechanism available to impart, maintain, or prevent motion, or to provide required acceleration.

Torque Ripple: A periodic variation in torque as an element rotates.

Tribological Coating: A coating applied to a surface for the purpose of reducing friction or increasing wear resistance.

Truncation: The condition created when applied loads cause a portion of a bearing ball's contact ellipse to extend beyond the edge of the raceway.

Yield Design Load: The product of the design limit load and the yield factor of safety.

4. REQUIREMENTS

4.1 Dimensional Analysis

4.1.1 [DDMR 1] A dimensional analysis of all moving parts and intentional interference-fit parts **shall** be performed and documented to ensure that specified functional performance is maintained under worst-case environmental conditions and configurations.

[Rationale: Dimensional analysis is important for ensuring internal and external clearances and establishing manufacturing tolerances. Dimensional analysis should be performed as part of the design process prior to part manufacturing. Establishing the tolerances via a documented dimensional analysis helps one to understand the effects of tolerances and other factors, avoids tolerance stack-up conditions that can adversely affect hardware performance, ensures useful parts are manufactured, and allows for easy review and revision later.]

4.1.2 [DDMR 2] The dimensional analysis **shall** account for the following:

- a. Manufacturing, assembly, and alignment tolerances.
- b. Temperature- and temperature-gradient-induced deformations.
- c. Deflections due to vibration.

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- d. Deflections due to external loads.
- e. Deflections due to operational loads.
- f. The full range of adjustability of the mechanism parts.

[Rationale: Dimensional analysis is sensitive to these factors.]

Guidance: *These factors are similar to those to be considered for clearances. Additional factors may be appropriate for consideration based on individual applications. Verification that these factors have been considered is expected to include line items in the analysis documentation for each factor in the dimensional analysis, or if a particular factor does not apply in a given situation, rationale for its inapplicability.*

4.2 Clearances

4.2.1 [DDMR 3] Static and dynamic clearance requirements between mechanism components and any other structure, component, thermal covering, and field of view **shall** be established.

4.2.2 [DDMR 4] Required static and dynamic clearances between mechanism components and any other structure, component, thermal covering, and field of view **shall** be maintained throughout the service life and life test of the mechanism.

4.2.3 [DDMR 5] Internal mechanism clearance requirements **shall** be established.

4.2.4 [DDMR 6] Required internal mechanism clearances **shall** be maintained throughout the service life and life test of the mechanism.

[Rationale: Maintaining clearances within and around mechanisms is necessary both to maintain proper mechanism function and to prevent the mechanism from causing problems with other systems. Establishing clearances allows one to both design the mechanism to maintain the clearances and then verify the existence of those clearances in the design and on the flight hardware.]

4.2.5 [DDMR 7] The established clearance requirements **shall** account for the following:

- a. Manufacturing, assembly, and alignment tolerances.
- b. Temperature- and temperature-gradient-induced deformations or instabilities.
- c. Deflections due to vibration.
- d. Deflections due to external loads, including gravity effects.
- e. Deflections due to operational loads.
- f. Deflections due to changes in pressurization, including thermal blanket billowing.
- g. Motion of cable harnesses, tubing, and sensor wiring.
- h. Environments arising from transportation.
- i. The full range of adjustability of the mechanism parts.

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[Rationale: Because many of the factors affecting the overall dynamic clearance are not present when inspections are performed, these effects have to be accounted for and included in the static clearance specified on the drawings. Tolerancing, thermal expansion effects, and deflections are the most important factors to consider in establishing the clearances. Thermal blanket behavior is notorious for causing unexpected interferences with mechanisms. Motion under transportation loads is often not considered, but clearances are important in that situation, too.]

Guidance: Additional factors may be appropriate for consideration based on individual applications. Verification that these factors have been considered is expected to include line items in the analysis documentation for each factor in the clearance analysis or, if a particular factor does not apply in a given situation, provide rationale for its inapplicability.

4.2.6 [DDMR 8] Clearance measurements **shall** be performed on the highest level of assembly possible.

[Rationale: To verify the proper clearances exist, inspections of all the established clearances have to be made on the as-built hardware after installation or assembly of the components of interest.]

Guidance: The measurement of each clearance should be made when the mechanism is in the configuration that generates the worst case for that clearance. If clearances cannot be directly measured, positional measurements that allow clearance to be calculated may be substituted.

4.3 Torque and Force Margins

Guidance: Though the term “torque margin” is used in this section for brevity, all references to torque margin also apply to linear motion with “force” replacing “torque.”

Torque margin is a measure of the excess capability of a mechanism to impart, maintain, or prevent motion, or to provide required acceleration. Torque margin is intended to ensure that a mechanism retains reserve torque that can be applied in the event of an unforeseen effect that increases resistive torque or reduces motive torque within the mechanism, similar to the factor of safety used in a structural margin calculation.

Torque margin is not applicable to mechanisms that must provide a specific torque value within a narrow tolerance range to perform their basic function, such as a spring holding a relief valve closed or an ejection mechanism requiring a specific (not minimum) separation velocity.

Torque margin is defined in Equation 4-1 as follows:

$$\text{torque margin} = \frac{T_{\text{avail}}}{\sum FS_f T_f + \sum FS_v T_v + \sum FS_a T_a} - 1 \quad (\text{Equation 4-1})$$

T_{avail} is the minimum available torque generated by the driving or holding component (e.g., spring, motor, pyrotechnics, solenoids, heat-actuated devices, pneumatic or hydraulic systems, brake). The minimum available torque from an energized motor is typically understood to

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include “magnetic losses” such as hysteresis and eddy current drag torques, “mechanical losses” such as bearing, brush, and windage drag torques, and “torque ripple” caused by detent (cogging) torque, asymmetry in motor design and construction, commutation strategy, etc.

For static and dynamic torque margins, T_v terms are the individual maximum “variable” resisting torques whose values may change with environmental conditions and cycles (e.g., friction torque, viscous drag torques, wire harness torque due to flexing or set). For holding torque margins, T_v is used to denote the individual maximum disturbing torques regardless of their degree of variability.

T_f are the individual maximum “fixed” resisting torques that are not strongly influenced by environmental conditions and cycles (e.g., vehicle maneuver-induced torques, return spring torques, unbalanced pressure loads limited by relief mechanisms).

T_a is the torque required to achieve a specified acceleration of the driven component.

FS_v , FS_f and FS_a are safety factors applied to each individual torque in the denominator of Equation 4-1 prior to summation. Table 1, Minimum Safety Factors for Torque Margin Equation 4-1, specifies the minimum safety factors to be used in Equation 4-1. The safety factor applied should be higher than given in Table 1 if:

- a. The designs involve an unusually large degree of uncertainty in the characterization of resistive torques.*
- b. Torque margin testing is not performed in the required environmental conditions or is not repeatable and has high variability.*
- c. Torque margin testing is performed only at the component level.*

There are many forms of torque margin equations in use in various standards. Each can be reformulated to appear like the others; the only difference among them is in the magnitude and nature of the different safety factors applied to the terms. This form was chosen due to its relative simplicity, its ability to handle several torque margin calculations with a single equation, and its suitability for application of maturity-based safety factors.

The required reserve torque is included in the equation in the form of the safety factors, so a torque margin greater than or equal to zero indicates that requirements are met. Setting the FS_f , FS_v , and FS_a safety factors to unity represents the torque at which no reserve is available.

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Table 1—Minimum Safety Factors for Torque Margin Equation 4-1

Source of Torque Data	FS_v	FS_f	FS_a
Theory or analysis	3.00	1.50	1.25
Development test at expected environmental extremes	2.50	1.35	1.15
Qualification test			
Lot acceptance test at expected environmental extremes			
Acceptance test of flight hardware at ambient conditions	2.00	1.25	1.10
Acceptance test of flight hardware at expected environmental extremes			
Test evaluation of one-spring-out case ¹	1.00	1.00	1.00

¹ When the torque margin of spring-driven mechanisms that utilize multiple springs in parallel to provide torque is evaluated by test for cases in which one of the multiple springs fails (the “one-spring-out case”), a minimum FS_f , FS_v , and FS_a of 1.0 may be used. Prior to a spring failure, the factors in the first three rows of Table 1 apply. Note that because using multiple springs in parallel for redundancy is distinct from designing a system to tolerate a failure of some portion of a single spring, the torque margin for a failure-tolerant single spring must be evaluated with the same factor after failure as prior to failure. The one-spring-out factors do not apply to any case other than the use of redundant springs in parallel.

Guidance: The “Theory or analysis” safety factors listed in Table 1 are not intended to be used, to borrow the terminology of NASA-STD-5001, *Structural Design and Test Factors of Safety for Spaceflight Hardware*, as “design factors” in a “no-test option.” Such an approach is prohibited by the requirement to verify torque margins by test in section 4.3.c. The “Theory or analysis” safety factors are intended to compensate for higher uncertainty in initial sizing and calculations until torque values can be obtained by test. This approach allows the required torque to be reduced as confidence in the system characteristics grows through testing. The values FS_v , for FS_f , and FS_a are chosen such that higher factors are applied to torque sources with higher variability.

When assessing the torque margin of failure tolerant mechanisms which utilize multiple springs in parallel to provide torque, it is important to recognize that to show a positive torque margin after a spring failure, the torque margin without a spring failure can be excessive. For those cases, reduced conservatism in the safety factors is appropriate. Redundancy will often be achieved with elements such as redundant motor windings or redundant actuators. These elements do not normally work together at full power to provide torque or force. These types of implementations still require using the FS_f , FS_v , and FS_a values from the first three rows of Table 1. Note that if the failed actuator produces a resistive torque after failure, this torque must be included in the torque margin calculation for the failure case.

Note that because the use of multiple springs in parallel for redundancy is distinct from designing a system to tolerate a failure of some portion of a single spring (e.g., a broken coil of a guided helical compression spring designed and restrained per the guidance in Appendix A, section A.2.5, one leaf in a multi-leaf spring) and still maintain functionality with reduced performance. The mass penalty in this scenario is not as great as adding an entirely redundant

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spring, so the one-spring-out safety factors are not appropriate in such a case. See Appendix A, section A.2.5 for a further discussion on spring redundancy.

4.3.1 General Torque Margin Requirements

4.3.1.1 [DDMR 9] All torque margin requirements **shall** be applied under worst-case conditions throughout the mechanism's life, including throughout life testing.

Guidance: Worst-case conditions are defined as those that result in a combination of minimum driving torque and maximum resisting torque over the range of qualification environmental limits (e.g., temperature, pressure, acceleration, vibration, radiation, humidity), operating limits (e.g., supply voltage, motor/controller parameters), and design parameters (e.g., material properties, manufacturing tolerances, spring force variability).

Only those worst-case combinations that can occur simultaneously need to be considered. For example, if the minimum driving torque occurs at hot conditions and the maximum resisting torque occurs at cold conditions, then the two should not be combined to calculate a torque margin. As another example, if the minimum driving torque and maximum resisting torque do not occur at the same point in a mechanism's range of motion, the two should not be combined to calculate a torque margin.

The following are examples of conditions that should be accounted for when calculating torque margin:

- Material property variations
- Dimensional tolerances
- Environmental conditions
- Frictional effects
- Changes in static and kinetic friction due to storage time or exposure to vacuum
- Alignment effects
- Latching forces
- Return springs
- Wire harness loads (flex forces and long-term set)
- Damper drag
- Thermally induced distortions
- Load-induced distortions
- Variations in lubricity
- Fluid pressure on the elastomers in viscous dampers
- Supply voltage, voltage drops, current limit
- Motor controller parameters (gains, sampling rate, bandwidth)
- Acceleration due to vehicle motion or maneuvers that can retard motion
- Loading due to vibroacoustic environment
- Effects of exposure to propellant or propellant vapor, directly or via permeation
- Magnetic saturation (see Appendix A, section A.2.1.1)
- Torque required when swapping between redundant electronics

Whether or not to include inertial loads as resistive torques in a torque margin calculation depends on the type of torque margin being calculated and the duration of the actuation event compared to the duration of the inertial load. Inertial loads arise when inertias are accelerated. Quasi-static accelerations such as acceleration due to launch vehicle thrust have durations long enough to warrant inclusion in all torque margin calculations. However, transient accelerations such as accelerations due to vibration or shock create may generate a small enough response in some mechanisms that they may be neglected in some torque margin calculations.

For example, the static torque margin calculation for an actuator whose inertia is large may be able to neglect the inertial loads due to high-frequency vibration exposure if the vibratory forces cannot accelerate the inertia enough to generate significant resistive loads. However, exposure to the same high-frequency vibration may be important in the holding torque calculation of a valve because, even if transient, vibratory forces can induce small motions that lead to undesirable effects such as valve leakage or valve seat damage.

4.3.1.2 [DDMR 10] Mechanisms that use torque multipliers or reducers (e.g., gearboxes, strain wave gear drives, lever arms) **shall** meet torque margin requirements at both the input and output of multiplying or reducing devices.

[Rationale: Torque margins must be calculated at both positions because torque multipliers and reducers are not 100 percent efficient, and because some resistive torques can have a more detrimental effect on torque margin when considered prior to multiplication. Basing torque margin on the overall output can give a false impression of the true torque margin.]

4.3.1.3 [DDMR 11] All torque margins **shall** be verified during an acceptance test at the highest possible level of assembly.

[Rationale: Torque margins are intended to ensure that the mechanism retains reserve torque that can be applied in the event of unforeseen circumstances. Therefore, as with any other capability of the mechanism, the minimum torque margin is verified prior to placement into service.]

Guidance: *Often, it is not possible to test the torque margin directly in which case the individual torque terms in Equation 4-1 are to be individually acceptance tested and the torque margin calculated using these tested values. Attempts to calculate torque margins at higher levels of assembly (subsystem or vehicle) can be more complex because of the potential inability to discriminate between different torque sources in the denominator of Equation 4-1. See Appendix A, section A.2.1.2, for a methodology to calculate torque margin at higher levels of assembly.*

4.3.2 Static Torque Margin

[DDMR 12] Static torque margin per Equation 4-1 and Table 1 **shall** be greater than zero within the mechanism's full range of motion.

Guidance: Static torque margin is a measure of the excess torque available to overcome resistance to motion. T_{avail} is the minimum available torque generated by the driving component. Static friction and static resisting torque values are used for T_v and T_f in this calculation. T_a and FS_a are not used in the static torque margin calculation.

4.3.3 Dynamic Torque Margin

[DDMR 13] Dynamic torque margin per Equation 4-1 and Table 1 **shall** be greater than zero within the mechanism's full range of motion.

Guidance: Dynamic torque margin is a measure of the excess torque available to provide a required minimum acceleration to a body. Such a requirement can take many forms such as a specified acceleration, a specified minimum time, or a specified minimum velocity. This requirement is only applicable to mechanisms that are required to provide a minimum acceleration to a body.

T_{avail} is the minimum available torque generated by the driving component. Friction and resisting torque values appropriate for the specified motion profile are used for T_v and T_f in this calculation, and the torque required to achieve the minimum acceleration is used for T_a . Values of mass and moment of inertia used in dynamic torque margin calculations may need to include allowances for system growth over the duration of a program.

4.3.4 Holding Torque Margin

[DDMR 14] Holding torque margin per Equation 4-1 and Table 1 **shall** be greater than zero at the specified positions within the mechanism's full range of motion.

Guidance: Holding torque margin is a measure of the excess torque available to maintain position in the presence of disturbances. This requirement is applicable to mechanisms that must maintain position in the presence of external disturbances. T_{avail} is the minimum powered or unpowered holding torque intentionally provided to hold position from sources such as brakes, springs, motors, pneumatic or hydraulic systems, or detents. Incidental, unreliable, and/or uncharacterized contributors to holding torque (e.g., joint friction, harness bending, blanket rubbing) should be excluded from T_{avail} . The maximum disturbing torque values are used for T_v in this calculation. T_v is used in the holding torque margin calculation for two reasons. First, though the disturbing torques may not be highly variable in nature, the resistive torque of the T_{avail} term can contain a high degree of variability. Second, using a mechanism to hold position is inherently more variable than using a structure; therefore, a higher factor of safety than would be applied to a structure is appropriate. T_f and T_a are not used in the holding torque margin calculation.

4.3.5 Stepper Motor Margin

4.3.5.1 [DDMR 15] Stepper motor torque margin **shall** be greater than zero within the mechanism's full range of motion, using the appropriate allowable margin calculation method per Table 2, Stepper Motor Margin Calculation Methods.

Table 2—Stepper Motor Margin Calculation Methods

Case	Allowable Margin Calculation Methods
When all of the following conditions are met: - the resisting torque due to friction is much greater than the torque due to the driven inertia - the natural frequency of the mounting base and driven inertia are greater than the stepping frequency - the driven inertia is not driven into a hard stop or spring stop - the stepper motor winding current flows for the full period of the step - a pull-in torque test is performed in the operating environments - The stepping frequency is not under closed-loop control	- Pull-in torque margin analysis per Equation 4-1 and Table 1 - Torque margin analysis per Equation 4-1 and Table 1 with T_{avail} derived from a step stability analysis
All other cases	Torque margin analysis per Equation 4-1 and Table 1 with T_{avail} derived from a step stability analysis

[Rationale: The dynamic response to a commanded step can create step instability. The bulleted cases in Table 2 are intended to bound the circumstances in which stable stepping can be expected. When those circumstances are not in place, a step stability analysis is needed to guarantee stable stepping.]

Guidance: Pull-in torque margin is a measure of the excess torque available to accelerate a stepper motor from rest to operating speed, stop a stepper motor, or reverse stepper motor direction in synchronism with input pulses (i.e., without loss of steps) for a given speed, inertial load, and controller. For pull-in torque margin, the pull-in torque for a given speed is used for T_{avail} , and the total resisting torque seen by the stepper motor is used for T_v . For torque margin using a step stability analysis, the minimum torque produced under stable stepping is used for T_{avail} and the total resisting torque seen by the stepper motor is used for T_v . T_f and T_a are not used when calculating stepper motor torque margin.

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A further discussion of stepper motor performance and stability analysis can be found in Appendix A, section A.2.4.2.

4.3.5.2 [DDMR 16] When stepper motor detent torque is used to maintain rotor position in the presence of vibratory disturbances, detent stiffness and motor damping **shall** be considered when determining the holding force margin.

[Rationale: The spring-damper nature of the detent torque requires special consideration when used in a holding torque application.]

4.3.6 Servomechanism Margins

[DDMR 17] For servomechanism applications, control system performance margins **shall** be calculated in addition to meeting the applicable static torque margin, dynamic torque margin, holding torque margin, and stroke margin requirements.

[Rationale: Servomechanism performance requirements may require motor performance that far exceeds that required by the torque margin equation 4-1 alone. Though applicable margins should still be calculated and documented, servomechanisms need a control system performance analysis, e.g., phase and gain margin analysis, to fully assess performance and margin. The specific control system margins for a given application are outside of the scope of this document and therefore cannot be specified. It is important to successful servomechanism functionality that they be calculated and evaluated by control system experts.]

4.4 Stroke Margin

Stroke margin is defined in Equation 4-2 as follows:

$$\text{stroke margin} = \frac{\text{output stroke of the actuator}}{\text{stroke required to achieve desired function}} - 1 \quad (\text{Equation 4-2})$$

4.4.1 [DDMR 18] Stroke margin requirements **shall** be established for all mechanisms.

[Rationale: Stroke margin helps guarantee enough travel exists in a mechanism to accomplish its function in the presence of uncertainty. Different margins are appropriate for different mechanisms, preventing the specification of a single number for margin; but in any case, the appropriate margin for the application should be assessed and a requirement for it established early enough that it may be evaluated by all stakeholders.]

Guidance: *In applications where stroke of the driven member is physically limited by a mechanical stop, this requirement still applies to the actuator, i.e., the actuator itself is to have stroke remaining when the mechanical stop is reached by the driven member. Continuously rotating mechanisms, e.g., solar array drives with slip rings, are exempt from this requirement.*

4.4.2 [DDMR 19] All stroke margins **shall** account for worst-case credible combinations of the following:

- a. Environmentally induced distortions.
- b. Misalignments.
- c. Dimensional tolerances.
- d. Load-induced distortions

*[**Rationale:** Like force margin, stroke margin ensures that adequate travel is available to account for unforeseen effects, which can increase the stroke required. A variety of factors, including those specified in the requirement, can affect the dimensions of the assembly and stroke needed. Therefore, these various factors are important considerations when determining the necessary travel for linear actuators, which include pin-pullers.]*

Guidance: Worst-case conditions are defined as those that result in the minimum stroke over the range of qualification environmental limits (e.g., temperature, pressure, acceleration, vibration, radiation, humidity), and design parameters (e.g., material properties, manufacturing tolerances).

In applications where stroke of the driven member is physically limited by a mechanical stop, this requirement still applies to the actuator, i.e., the actuator itself is to have stroke remaining when the mechanical stop is reached by the driven member. Assurance that a linear actuator remains engaged prior to actuation is covered by holding force margin (see section 4.3.6), tolerancing (see section 4.1), and adherence to structural requirements.

4.5 Electrical Bonding and Grounding

4.5.1 [DDMR 20] Bearings **shall** not be used to carry electrical current.

4.5.2 [DDMR 21] Gears **shall** not be used to carry electrical current.

*[**Rationale:** These requirements are meant to preclude bearings and gears from being used as part of an intentional electrical distribution circuit or to carry other currents such as ground return currents, lightning currents, or plasma-induced surface currents. These components are not designed to carry electrical current, and their geometry and lubrication make it difficult for them to do so. Other current paths should be provided.*

Guidance: Electrical currents produced by unintentional charging of bearing-supported hardware should be considered when evaluating this requirement. One should not assume that bearings and gears are insulative or conductive.

4.5.3 [DDMR 22] Mechanisms **shall** include electrical bonding and ground paths between moving and stationary parts sufficient to meet electromagnetic environmental effects requirements.

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[Rationale: It can sometimes be difficult to achieve an adequate bond or ground path when the interfaces are in motion, especially when bearings and gears are involved. Care should be taken to ensure that the bonding and grounding scheme is able to perform as intended.]

Guidance: Verification of this requirement is expected to consist of an analysis or test that demonstrates that an adequate bond or ground path exists to meet electromagnetic environmental effects requirements.

4.6 Lubrication

Guidance: Mechanism lubrication is considered to be a process subject to the requirements of NASA-STD-6016. Mechanism performance can be sensitive to the lubrication process parameters such as the lubricant selected, its quantity, location, application method, etc. The wrong lubrication process parameters can negatively affect mechanism performance by increasing drag and heating while decreasing torque margin and efficiency. Additionally, too much lubricant can create contamination problems while too little lubricant can reduce mechanism life.

4.6.1 [DDMR 23] All surfaces in contact for which friction under relative motion negatively affects performance of the mechanism **shall** be lubricated.

[Rationale: Lubrication is one of the most important factors in successful mechanism design and operation. All contacting surfaces that are expected to move with respect to one another should be lubricated in some way, regardless of material choices, load, or life requirements.]

Guidance: Use of dissimilar metallic materials for the contacting surfaces, though strongly encouraged, is not an equivalent to or substitute for lubrication and does not meet the intent of this requirement. However, special cases can arise in which the use of lubricant may be more detrimental than beneficial in which case this requirement may be a good candidate for tailoring. If dissimilar metallic materials are used, consideration should be given to the potential for galvanic corrosion.

Refer to Appendix A, section A.2.2, for a discussion of lubricant selection and factors that should be considered. If a wet-lubricated mechanism contains interfaces in which friction is desirable, such as brakes, clutches, friction drives, friction joints, etc., barrier films should be used to keep wet lubricants from migrating to those interfaces. See Appendix A, section A.2.2.1, for more information on barrier films.

4.6.2 [DDMR 24] The selection of lubricants for mechanisms **shall** include the following considerations:

- a. The effect of the service environments (ground storage conditions, pressure, temperature, radiation, dust, humidity, salt spray, etc.) on lubricants.
- b. Creep properties of wet lubricants.

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- c. Viscosity index (variation of viscosity with temperature) of wet lubricants.
- d. Elastohydrodynamic (EHD) film thickness if operating in the EHD lubrication regime.
- e. Protection against galling and friction welding in the boundary lubrication regime.
- f. Outgassing and breakdown products from wet lubricants that could contaminate sensitive surfaces.
- g. Polymerization of wet lubricants, particularly due to high contact pressures.
- h. Lubricant purity and filtration level.
- i. Lubricant depletion for wet lubricants. (This is typically handled through a “lubrication loss analysis” which considers evaporation, creep/migration, consumption, and fling.)
- j. Lubricant wear-out for dry lubricants.
- k. Generation and management of dry lubricant wear debris.
- l. Compatibility of each lubricant with other lubricants and other materials (substrates, platings, coatings, and preservative or shipping oils).
- m. Exposure to propellant or propellant vapor.
- n. Interaction with pyrotechnic materials and their reaction products (blowby).
- o. Temperature limits of the lubricants.
- p. Corrosion protection of the mechanism.
- q. Contact stress.
- r. Coefficient of friction needed.
- s. The effect of wear debris on wet lubricant properties.

[Rationale: There are numerous factors that have to be considered to make a proper choice of lubricant.]

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Guidance: *Not all of these considerations will apply to every case. Verification that these factors have been considered is expected to include a line item discussing each factor or the rationale for inapplicability of each factor in the compliance assessment for this Standard.*

4.6.3 [DDMR 25] An evaporative loss analysis **shall** be performed to show that 90 percent of the initial liquid lubricant quantity remains at end of life, not including lubricant degradation.

[Rationale: *Because life testing evaluates cycle life and not calendar life, evaporative effects on lubricant availability typically cannot be evaluated with a life test alone. Equations for mass loss due to evaporation are based on mass loss of lubricant in vacuum tests and analytical idealizations of labyrinth geometry and as such may underpredict mass loss for complex labyrinth designs.]*

4.7 Indication of Status

4.7.1 [DDMR 26] A direct indication of the critical states of each mechanism **shall** be provided.

[Rationale: *Knowledge of a mechanism's critical state, i.e., any state necessary for adequate operational use or hazard control is important for mission success. Direct measurement ensures that reliable information is provided.]*

Guidance: *The necessary state information will vary depending on the mechanism, the concept of operations, and any hazards posed by failure to achieve or maintain the intended state. Examples of state indications include the mechanism's condition, the mechanism's position, indications of reaching predetermined configurations, current draw, electrical continuity, etc.*

Direct indication, as opposed to indirect indication, is achieved when the mechanism function of interest is itself indicated unambiguously rather than some other related state. For example, measuring the position of a component in a latch drivetrain gives an indirect indication of the latched state that could be erroneous if the drivetrain has structurally failed or has enough backlash or flexibility to allow the shaft to turn while the latch itself remained disengaged. Measuring the position of the latch pawl provides a direct indication of a latched state. Evaluation of the proposed means of state indication is recommended to determine whether a proposed state indication is direct or indirect.

Positioning mechanisms should always provide absolute positioning capability, either in the form of absolute position indication or the ability to command the device to a specific known position. This allows for periodic calibration or recovery in the event of an anomaly that introduces a discrepancy between the commanded and true positions.

4.8 Structural Requirements

4.8.1 [DDMR 27] Mechanisms classified as failure tolerant **shall** meet all structural requirements using full design factors of safety after failure of the mechanism to operate.

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[Rationale: To be considered failure-tolerant, a mechanism has to meet all performance requirements after any failures commensurate with its required level of failure tolerance. Structural requirements or the redistribution of loads caused by the failure are sometimes overlooked or reduced design factors of safety are sometimes erroneously applied. While it is often permissible to use a reduced statistical bound on the loads after a failure (e.g., using 2σ loads instead of 3σ loads), the design factor of safety should not be reduced.]

4.8.2 [DDMR 28] Engineering analyses **shall** account for the structural mounting boundary conditions, including:

- a. Stiffness.
- b. Mounting alignment tolerances.
- c. Temperature-induced distortions.
- d. Load-induced distortions.
- e. Interface friction.

[Rationale: Often, the structural analysis of a mechanism does not consider interface properties or assumes a rigid interface. Ignoring these properties can lead to failure in service when the interfaces create different environments than were considered in the analysis.]

Guidance: Verification of the accounting for these items is expected to include a discussion of each of these items and how they were modeled, or why they were not modeled, in the structural analysis documentation. Stiffness is an important factor in mechanism performance and can also be a significant contributor to offloading mechanism functional loads. A detailed analysis of stiffness and related loads through all ranges of motion will help identify problems early in the development cycle.

4.8.3 [DDMR 29] The mechanism **shall** remain functional after exposure to stall conditions at any point in its travel.

[Rationale: Mechanisms frequently stall torque or force, intentionally or unintentionally, either in testing or in flight. Designing the mechanism with enough structural and thermal capability to withstand stall ensures that the mechanism is undamaged by this exposure and allows steps to be taken to recover mechanism function.]

Guidance: Where stall duration is important, it is recommended that actuation force/torque stall conditions be applied for at least one minute when assessing compliance. It is recommended that stall conditions be applied for a duration of two minutes during qualification testing and one minute during acceptance testing under worst-case environmental conditions.

The program-specified factors of safety apply to this stall condition; i.e., the factors of safety applied for structural analysis are not to be reduced for this situation. Mechanism function is defined by the mechanism's specification. The stall condition load case should include the worst-case force that could be applied in that condition, which could include transient loading. In

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cases in which no recovery would be possible after a mechanism stall regardless of mechanism functionality, this ability may no longer be useful and may be a candidate for tailoring.

4.8.4 [DDMR 30] Non-jamming mechanical stops **shall** be incorporated into all mechanisms where exceeding required range of motion will result in detrimental effects to the mechanism or larger system.

[Rationale: Stops ensure that mechanisms do not travel farther than intended and cause problems or end up in an unrecoverable state. Soft stops such as software logic, open-loop control, and limit switches can be unreliable, so mechanical stops are important for maximum reliability.]

Guidance: Every case must be assessed for hazards, but examples of mechanisms for which over-travel commonly does not result in detrimental effects include filter wheels and gimbals with slip rings.

4.8.5 [DDMR 31] Mechanism components **shall** maintain a positive margin of safety with the full design factors of safety applied when subjected to worst-case transient loads from mechanical stop impact.

[Rationale: The impact against the mechanical stop can create elevated loads on other parts of the mechanism in addition to the stops themselves, and these loads have to be accounted for in the structural analysis. The contact of mechanical stops is often rapid enough that static analysis approaches can be unconservative and dynamic analysis will be necessary.]

Guidance: A bounding worst-case load would include impact at maximum speed combined with stall torque, including the effects of controller faults.

4.8.6 [DDMR 32] If manipulator systems, payload operations, extravehicular or intravehicular activities, or other situations presenting a risk of inadvertent contact are present, exposed mechanism components, protective shrouds and covers, and mounting structure **shall** be designed to accommodate inadvertent impact loads from these sources.

[Rationale: Designing for this possibility will ensure adequate margins against deformation that could cause a binding or jamming condition or inadvertent operation of the mechanism.]

Guidance: These cases are considered design loads, and therefore analysis of these cases will use full factors of safety. The particular load to be accommodated will be determined by the individual program.

4.9 Bearings

4.9.1 [DDMR 33] Ball bearings used in high precision or low torque ripple applications **shall** utilize raceways that meet Annular Bearing Engineering Committee (ABEC) 7, 7P, or 7T tolerances (or better) in accordance with American Bearing Manufacturing Association (ABMA)

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standards (documents depend on application and are accessible at <https://www.americanbearings.org/>).

4.9.2 [DDMR 34] Nonstandard ball bearings or thin section ball bearings where ABMA tolerances do not apply that are used in high precision or low torque ripple applications **shall** have the manufacturer's precision level most nearly equivalent to ABEC 7.

4.9.3 [DDMR 35] Ball bearings used in high precision or low torque ripple applications **shall** utilize steel balls of ANSI/ABMA/ISO 3290-1, Rolling Bearings - Balls - Part 1: Steel balls, grades 3 through 10, inclusive, or ASTM F2215, Standard Specification for Balls, Bearings, Ferrous and Nonferrous for Use in Bearings, Valves, and Bearing Applications, grades 3 through 10, inclusive, or ceramic balls of ASTM F2094/F2094M, Standard Specification for Silicon Nitride Bearing Balls, grades 3C through 10C, inclusive.

4.9.4 [DDMR 36] Ball bearings used in high precision or low torque ripple applications **shall** utilize a raceway surface finish of 2.0 microinches arithmetic average (AA) or better.

[Rationale: Bearing tolerances and raceway surface finish can have a strong impact on the performance of bearings where precision or low torque ripple is needed. ABEC 7 tolerances on the raceways have been shown to be adequate for these applications in most situations. However, one has to be careful to also specify the ball grade and raceway surface finish, which are not covered by the ABEC rating. Balls of poor grade can negate the benefits of tight bearing tolerances.]

Guidance: While not required, it should be noted that smoother raceway finishes can help a bearing transition to the EHD regime at lower speeds and can also help to extend life. Note that the above conditions are necessary to achieve high precision, low torque ripple, and/or long-life applications but may not be sufficient.

4.9.5 [DDMR 37] Ball bearings used in high precision, low torque ripple or long-life applications **shall** utilize material that has been refined using one of the following processing sequences:

- a. Air melting followed by Vacuum Arc Remelting (VAR, also known as Consumable Electrode Vacuum Melting or CEVM).
- b. Vacuum Induction Melting (VIM) followed by VAR.
- c. VIM followed by Pressurized Electroslag Remelting (PESR).

[Rationale: Vacuum melting and remelting or pressurized electroslag remelting improves the cleanliness of bearing steels by minimizing the size and number of non-metallic inclusions. When located at the contact surfaces, non-metallic inclusions create pits which degrade surface finish and increase torque ripple. When located at or below the contact surfaces, non-metallic

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inclusions are stress raisers from which fatigue cracks and spalls originate, shortening fatigue life.]

Guidance: Note that the above conditions are necessary to achieve high precision, low torque ripple, and/or long-life applications but may not be sufficient.

4.9.6 [DDMR 38] The mean Hertzian contact stress on the most highly loaded element in a rolling element bearing **shall** remain less than or equal to the appropriate values in Table 3, Allowable Contact Stress for Bearing Materials Under Non-Operational Yield Design Loads, when subjected to the non-operational yield design load.

4.9.7 [DDMR 39] For materials other than those listed in Table 3, an allowable contact stress **shall** be determined.

Table 3—Allowable Contact Stress for Bearing Materials Under Non-Operational Yield Design Loads

Bearing Material	Hardness Range	Mean Hertzian Contact Stress	
		Quiet Running	Non-Quiet Running
440C Steel	58-62 HRC	2310 MPa (335 ksi)	2760 MPa (400 ksi)
52100 Steel	60-63 HRC	2480 MPa (360 ksi)	2960 MPa (430 ksi)
M50 Steel	62-64 HRC	2480 MPa (360 ksi)	2960 MPa (430 ksi)
M62 Steel	66-69 HRC	3790 MPa (550 ksi)	4070 MPa (590 ksi)

NOTE: For hybrid bearings using silicon nitride rolling elements with steel rings, the allowable contact stress will be that of the steel used.

[Rationale: Stressing the bearing raceway material beyond its elastic limit creates a permanent indentation known as a brinell, which can be detected on the torque trace of the bearing and is unsuitable for quiet running applications. The values in the table for 440C and 52100 in high precision, low torque ripple applications were established by experiment and correspond to a brinell depth of 0.00003 times the ball diameter for 440C and 0.00005 times the ball diameter for 52100 (Leveille & Murphy, 1973). The values for M62 were established experimentally using ball-on-plate indentation tests with a brinell depth threshold of 0.00005 times the ball diameter based on the previous work on 440C and 52100 (Park, et al., 1998). Values for M50 are conservative limits based on hardness comparison with 52100. It is recommended that the above references be consulted to understand the configurations and bearing hardnesses tested before utilizing these limits.]

The values in the table for non-quiet running applications are based on the load that produces brinell depths of 0.0001 times the ball diameter, as found in ABMA standards and elsewhere. Experience shows that this brinell depth can be tolerated in most bearing applications without affecting fatigue life.]

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Guidance: *It should be emphasized that the allowables in Table 3 are for the loading of non-rotating bearings only, e.g., random vibration loads during launch, or loads on a bearing intended to release a degree of freedom. The allowables in Table 3 may not be appropriate for operating stresses (stresses experienced during intentional rotation of the bearing that would be suitable for inclusion in a life calculation) which will generally be lower and depend on a number of factors. See Appendix A, section A.2.3.1.2, for guidelines.*

The yield design load is specified in this requirement to allow the use of a yield factor of safety to account for uncertainty in determining the loading conditions on bearings. Because the result of exceeding these allowables is a self-limiting indentation leading to an increase in torque ripple and/or reduction in bearing operating life, a factor of safety such as what is typically applied to a structural design is not necessarily warranted. Each bearing application should be examined to determine the appropriate yield factor of safety given the nature of the loading conditions and the consequences of the resulting brinells.

4.9.8 [DDMR 40] Bearing fatigue life calculations **shall** be based on the L0.05 life when subjected to maximum time-varying loads consistent with the conditions under which the L0.05 life was determined.

[Rationale: Use of the L0.05 life has been shown through testing to effectively eliminate fatigue life failures during life testing and service for space applications.]

4.9.9 [DDMR 41] The upper and lower extremes of the ball bearing contact ellipses **shall** be contained by the raceways.

[Rationale: Understanding the contact geometry and stress in a bearing is important for obtaining required bearing performance and life. Truncation of the contact ellipse can cause significant increases in stress that are not accounted for in traditional bearing analyses and can have correspondingly large impacts on load-carrying capacity and life.]

Guidance: *See Appendix A, section A.2.3.1.2, for more on contact ellipse truncation.*

4.9.10 [DDMR 42] All ball bearings **shall** be preloaded with the following exceptions:

- a. Four-point (gothic arch) bearings.
- b. Deep groove ball bearings for which it can be shown that the absence of preload on the deep groove ball bearing is not detrimental to the performance of the mechanism.
- c. Wave generator bearings in strain wave gearing.

[Rationale: Preloading a ball bearing eliminates free play, reduces runout of the rotating member, increases axial and radial stiffness, prevents fretting damage, eliminates impact

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loading when exposed to a vibration environment, increases the load sharing among balls, and prevents skidding of balls.]

Guidance: *It is recognized that there are situations in which preload is not advisable, and there are many bearing preload strategies and considerations when analyzing the effects of preload levels and unloading of balls. See Appendix A, section A.2.3.1.3, for guidance.*

4.9.11 [DDMR 43] If axial sliding of a bearing ring is required to maintain preload and/or prevent axial overload, the sliding interface **shall** be designed to operate as a linear sliding bearing over the range of qualification environmental limits, including lubrication to reduce wear and sliding friction forces to acceptable levels throughout the service life.

[Rationale: *To maintain bearing preload and/or prevent axial overload, some bearing mounting schemes depend on one bearing ring or set of rings being free to translate axially in a housing or on a shaft. When bearing mounts that allow bearing rings to slide axially do not follow standard design practices for linear sliding bearings, vibration testing often damages the sliding interfaces and reduces their performance.]*

Guidance: *Mounting the bearing ring(s) on or between compliant diaphragms or structures is a preferred method of allowing axial translation because it does not involve loaded interfaces sliding over each other. Standard design practices for linear sliding bearings include selecting dimensions, tolerances, and clearances to prevent tilting and jamming, and selecting bearing stress level, materials, finishes, and lubricants to prevent galling and seizing. Finally, in bearing mounting schemes that depend on one bearing ring of a non-separable angular contact bearing being free to translate axially in a housing or on a shaft, it is important to include snubbers that prevent reverse thrust loading the bearing.*

4.9.12 [DDMR 44] Bearing preload **shall** be measured once all the assembly steps that establish or affect bearing preload have been completed.

[Rationale: *Measurement of preload needs to be verified after operations affecting bearing preload have been completed to ensure that the assembly operations did not adversely affect the preload and that the bearings function as desired in the fully assembled configuration.]*

Guidance: *It is highly recommended that the preload be measured at intermediate levels of assembly as well. Direct measurement is preferred; but if bearing preload cannot be directly measured, indirect methods of assessing preload, such as measurement of stiffness or Coulomb friction torque, may be used.*

4.9.13 [DDMR 45] Mechanisms utilizing guides or linear bearings to constrain a carriage to move in its length direction **shall** use a length-to-width ratio of 2:1 or greater, unless it can be shown by analysis that a length-to-width ratio of less than 2:1 will not cause the mechanism to bind or undergo stick-slip motion taking into account the following:

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- a. Possible friction coefficients.
- b. Contact forces.
- c. Actuating forces.
- d. Dynamically induced forces.
- e. Misalignments.
- f. Eccentric loading.

*[**Rationale:** The performance of mechanisms utilizing guides or linear bearings is geometry-sensitive. Length-to-width ratios greater than 2:1 rarely have problems unless friction coefficients are abnormally high. Ratios less than 2:1 can be used successfully but require careful analysis and characterization of parameters to ensure the system does not bind or become susceptible to stick-slip motion. Ratios less than 1:1 are suitable only for high-precision, low-friction applications such as recirculating-ball linear bearing systems.]*

***Guidance:** For mechanisms utilizing a set of guides or linear bearings, the effective length is defined as the distance between guide points or linear bearing centers along the direction of motion, and the width is the distance between the rails or centerlines of the bearings perpendicular to the direction of motion, as depicted in Figure 1, Illustration of Length and Width for Various Examples of Mechanisms Using a Set of Guides. For mechanisms utilizing a single rail, the effective length is the effective length of the linear bearing or guide and the width is the distance from the load application point to the centerline of the rail as depicted in Figure 2, Illustration of Moment Arm, Effective Length, and Width for a Single Linear Bearing Application.*

A length-to-width ratio greater than 2:1 requires no analysis. However, care should be taken when using large length-to-width ratios within a mechanism because rail deflections over the length of the linear bearing can also cause binding problems.

The effective bearing length of an individual linear bearing is often not the overall length of the linear bearing, but the length of the portion of the linear bearing carrying the load, often called a “load zone.” Few manufacturers publish information about their linear bearings’ load zones, so it is often necessary to contact the manufacturer or make a conservative estimate of the load zone dimensions. The same uncertainty factors used in the force margin analysis (e.g., on friction) should be applied to any analysis performed to examine susceptibility to stick-slip motion. Further information on length-to-width ratios can be found in Schroeder (2010).

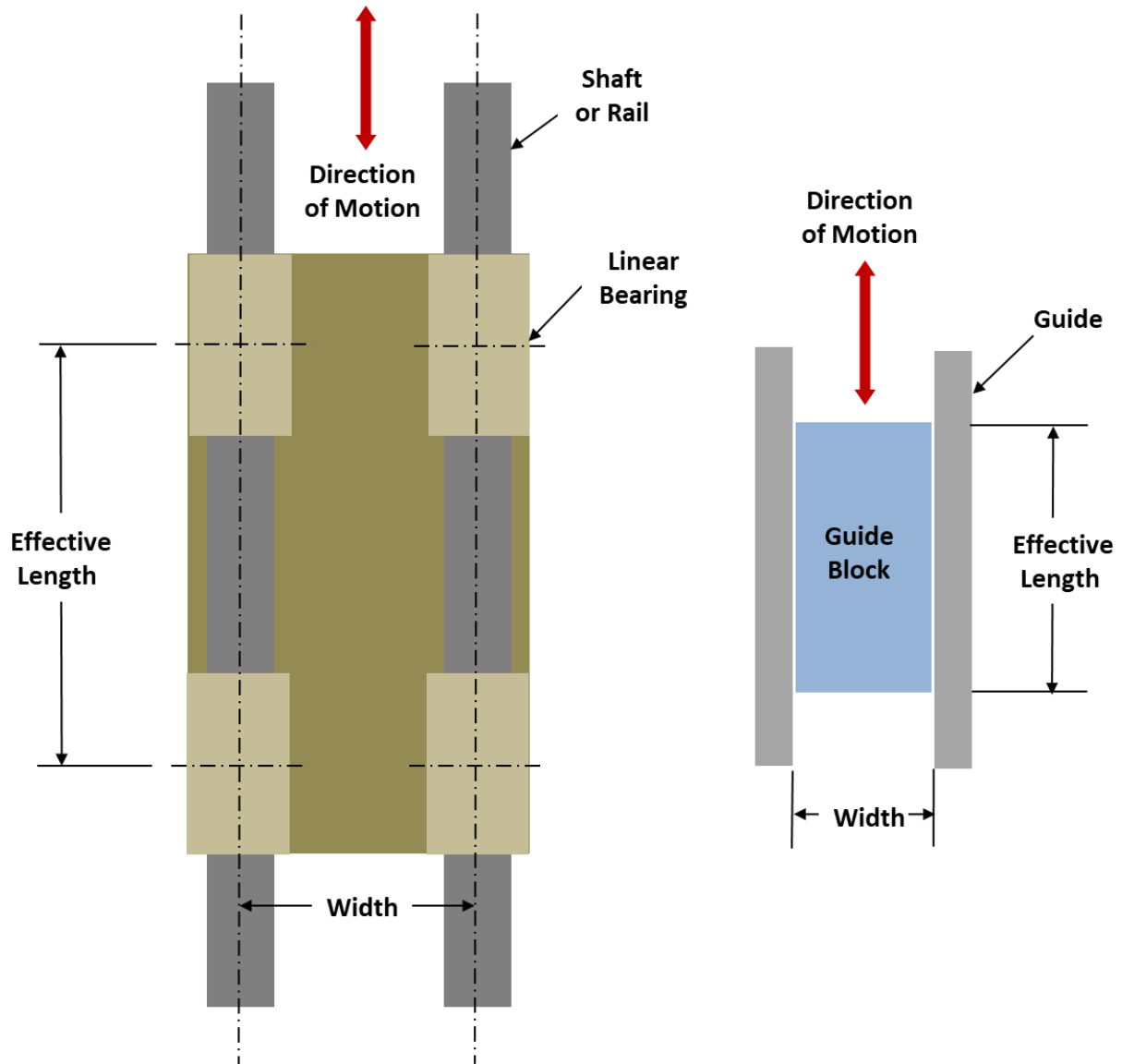


Figure 1—Illustration of Length and Width for Various Examples of Mechanisms Using a Set of Guides

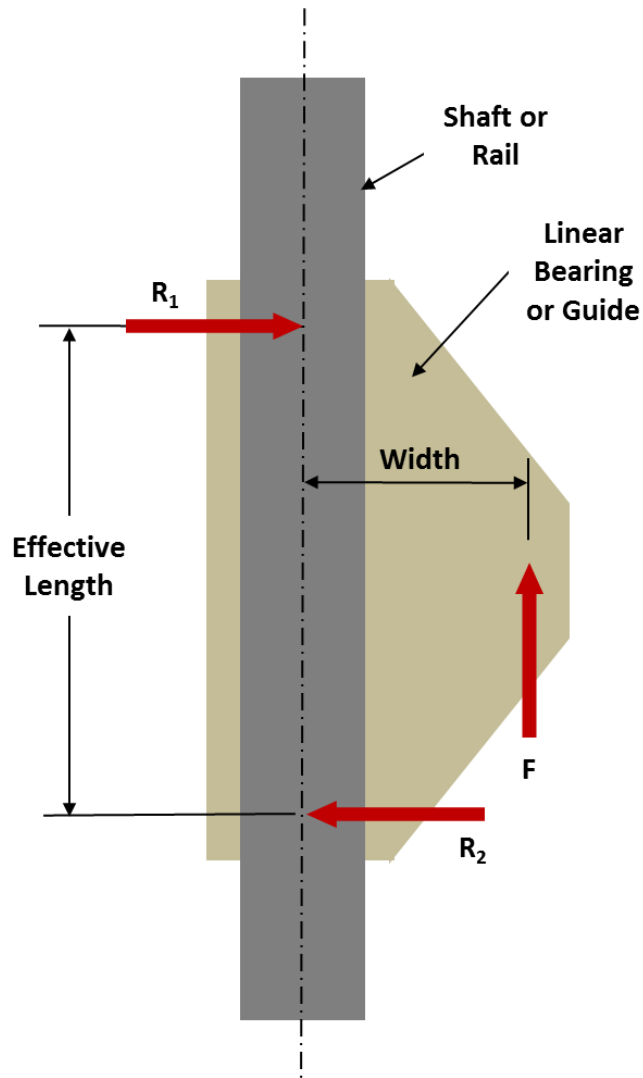


Figure 2—Illustration of Moment Arm, Effective Length, and Width for a Single Linear Bearing Application

4.10 Motors

4.10.1 Electronically Commutated Brushless Motors

4.10.1.1 [DDMR 46] Each electronically commutated (EC) brushless motor **shall** have the following characteristics measured (or calculated from measured values):

- Torque constant (K_t).
- Motor constant (K_m).
- Torque versus speed curve using flight-representative drive electronics.
- Phase resistances.

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- e. Phase inductances.
- f. End-to-end commutated torque performance.

[Rationale: These motor characteristic measurements are critical to understanding how the motor will perform in a mechanical system. Torque versus speed testing verifies the combined performance of the motor, feedback, and controller. This testing should use a flight-like driver and commutation position sensor so that it is known as early as possible how the motor and motor driver perform as a system. EC brushless motor performance can be heavily influenced by the motor driver used to power the motor and by alignment of the commutation position sensor to the motor.]

4.10.1.2 [DDMR 47] The minimum measured torque output from each EC brushless motor **shall** be verified using flight-representative drive electronics.

[Rationale: This testing is necessary because a typical dynamometer test run will not reveal “torque holes” where the commutation electronics switch motor current from winding to winding. A motor whose initial position lies in one of these holes may not start under a load.]

Guidance: The output torque parameters of an EC brushless motor are characterized with a torque profile test. See Appendix A, section A.2.4.3, for guidance on torque profile testing techniques. Flight-representative drive electronics are electronics that produce the flight drive waveform when connected to the motor. These electronics typically share the same schematic design as the flight electronics but may use off-the-shelf components. However, caution is needed to ensure that off-the-shelf electronic components perform the same as the flight electronics components. For example, the gate capacitance of a radiation-hardened field effect transistor can be up to an order of magnitude different from an off-the-shelf field effect transistor, which would change the switching time and the response of the mechanism. Care also should be taken to consider the configuration and lengths of harnessing as they may affect performance.

4.10.2 Stepper Motors

4.10.2.1 [DDMR 48] Each stepper motor **shall** have the following performance characteristics measured:

- a. Powered breakaway torque.
- b. Unpowered (detent) torque versus angle for a full rotation.
- c. Pull-in torque with representative inertia, friction loads, and step rates.
- d. Pull-out torque with representative inertia, friction loads, and step rates.
- e. Step accuracy.
- f. Detent to powered torque null alignment.
- g. Phase resistances.
- h. Phase inductances.

[Rationale: Stepper motors operate differently from electronically commutated brush motors so they need to have different characteristics measured. Measuring these motor characteristics for stepper motors allows one to understand how the motor will perform in a mechanical system.]

Guidance: All operational tests for stepper motors are sensitive to the inertia and stiffness of loads coupled to the motor shaft, including measurement sensor load parameters. Reaction torque sensors are often employed to minimize frictional and inertial shaft loading from the sensor; however, the compliance of the load sensor can also impact test results. Detent to powered torque null alignment is the difference between the angular position the rotor takes with respect to the stator when unpowered and when powered to the matching cardinal step.

4.10.2.2 [DDMR 49] Each stepper motor **shall** have the rotor polar inertia calculated.

4.10.2.3 [DDMR 50] Stepper motor testing **shall** utilize flight or flight-representative drive electronics.

[Rationale: Both pull-in and pull-out torque tests require a motor driver to perform. For similar reasons as the brushless motor, it is best to use a motor driver that is as flight-like as possible for these tests, produce the flight drive waveform when connected to the motor (pulse duration, peak voltage, and drive pulse shape) and have similar characteristics for shunting diodes, output impedance, voltage drop, and current limiting.]

Guidance: Flight-representative drive electronics are electronics that produce the flight drive waveform when connected to the motor. These electronics typically share the same schematic design as the flight electronics but may use off-the-shelf components. However, caution is needed to ensure that off-the-shelf electronic components perform the same as the flight electronics components. For example, the gate capacitance of a radiation-hardened field effect transistor can be up to an order of magnitude different from an off-the-shelf field effect transistor, which would change the switching time and the response of the mechanism. Care also should be taken to consider the configuration and lengths of harnessing as they may affect performance. Any differences in the drive electronics between the test configuration and flight configuration should be documented in the test report.

4.10.3 Brush Motors

4.10.3.1 [DDMR 51] The maximum allowable temperature limits of the motor windings and other materials in the assembly **shall** be established.

4.10.3.2 [DDMR 52] Brush motor temperature limits **shall** not be exceeded for the worst operational cases in the worst-case qualification environments.

[Rationale: Brush motors are easier to overheat than other types of motors, and the temperature difference between the windings and case, where temperature can be measured, is quite large.]

Guidance: *Temperature margin to the motor limits should be incorporated to encompass uncertainty in the test and/or analyses. Locked-rotor stall should be considered in the determination of maximum temperature rise in the motor. The testing and analysis should consider the margined torque output of the motor.*

4.11 Springs

4.11.1 [DDMR 53] Springs **shall** be failure tolerant unless spring failure can be shown to be non-credible.

[Rationale: *Springs are a common mechanism component as well as a common source of problems. Spring redundancy can greatly improve mechanism reliability.]*

Guidance: *There are two ways to achieve redundancy in a spring: (1) a second spring can be used, and (2) use of a spring that retains functionality after one coil or element of the spring (e.g., a single conical spring in a stack) is fractured or otherwise compromised. Note that this last option generally requires use of a compression spring and that in the case of coil springs, the wire diameter and coil pitch have to be such that the two spring halves cannot thread into each other after a fracture.*

Determining that a spring failure is not credible requires demonstrating that adequate life and stress margins exist on the part. This could be accomplished with a combination of stress analysis, fatigue analysis, fracture control methods, and testing. However, given the size of many springs used in mechanisms, fracture approaches are often not feasible.

More information on spring use and design is available in Appendix A, section A.2.5.

4.12 Gears

4.12.1 [DDMR 54] Gear trains **shall** demonstrate that they meet performance requirements, accounting for the worst-case combinations of the following conditions, throughout the service life of the mechanism:

- a. Tooth pitting, brinelling, bending fatigue, and bending strength under nominal and peak operating loads.
- b. Impact tooth loads from maximum combined axial, radial, and moment loads sustained during the full life cycle of the mechanism.
- c. Backlash.
- d. Effects of temperature and temperature gradients on quality of lubrication and gear contact pattern.
- e. Effects of tooth geometry.

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- f. Undercutting and tooth profile modifications.
- g. Gear mounting, misalignment, and face load distribution.
- h. Variation in operating center distance.

[Rationale: These parameters, which are often overlooked in design and analysis, can influence gear train strength and wear.]

Guidance: Verification that these factors have been accounted for is expected to include line items for each factor in the gear train stress and wear analysis. If a particular factor does not apply in a given situation, rationale for this inapplicability is to be provided in the analysis documentation.

4.13 Dampers

4.13.1 [DDMR 55] Viscous dampers, including damper fluids, **shall** have a cleanliness requirement established.

[Rationale: The cleanliness of damper components is essential due to the small clearances within a damper. The cleanliness of damper fluid can have a great impact on fluid performance and life, and frequently this aspect of procurement is ignored.]

Guidance: Verification of this requirement is expected to include a documented plan for achieving the required cleanliness levels that are specified. It should be stressed that cleanliness is also essential for the assembly area and tooling used.

4.13.2 [DDMR 56] Viscous dampers **shall** be evacuated to a pressure of 0.13 Pa (1×10^{-3} Torr) or less before filling with fluid.

[Rationale: Air entrapment in dampers can negatively affect performance of the dampers.]

4.13.3 [DDMR 57] All viscous dampers exposed to vacuum in service **shall** have their deadband measured in vacuum.

[Rationale: Measuring the deadband in a vacuum confirms that the damper has not been underfilled and does not contain a non-degassed damper fluid.]

4.14 Separable Interfaces

4.14.1 [DDMR 58] Separation systems utilizing separation nuts or frangible nuts **shall** retract the bolt beyond the separation plane without reliance upon preload or gravity.

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[Rationale: The conversion of preload strain energy to kinetic energy of the bolt is highly dependent on configuration and environments and is rarely reliable enough to depend upon for bolt extraction beyond the separation plane. Similarly, because of unknowns in overall acceleration levels and the effects of dispersions in orientation, the effectiveness of gravitational acceleration is also not dependable. Dedicated bolt retractors or other actuation sources are necessary to ensure retraction.]

4.14.2 [DDMR 59] All interfaces in deployment and jettison mechanisms designed to separate in service **shall** use kickoff springs to ensure first motion.

[Rationale: Joints designed to separate often experience a wide range of phenomena that can create a tendency to keep the joint together, such as lubricant degradation, adhesion, undesirable motion of thermal blankets, and other retarding effects. As a result, the initial separation of the joint can be completely inhibited, force margins may be negated, or the available energy may be consumed. Adding kickoff springs dedicated to overcoming these effects ensures that the joint separates and enough force or energy remains in the separation to meet requirements.]

4.15 Pulleys

4.15.1 [DDMR 60] All pulleys **shall** use pulley guards that extend to the tangency points of the cable.

[Rationale: Pulley guards enclose some portion of a cable on the portion of a pulley in contact with the cable. This prevents the cable from moving out of the plane of the pulley under a disturbance. Making sure that the guard extends all the way to the tangency points of the cable (the points at which the cable leaves the surface of the pulley, see Figure 3, Pulley Guard Illustration) derives the maximum benefit from the presence of the pulley guard.]

Guidance: Be aware that some pulley systems can have changing tangent points and reversing motion. A reverse motion at a different tangent location without a pulley guard may allow the cable to unseat from the pulley groove.

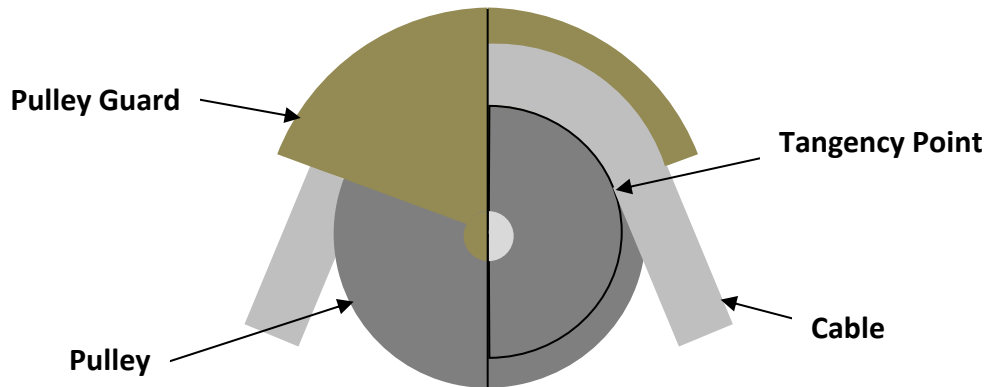


Figure 3—Pulley Guard Illustration

4.16 Switches

4.16.1 [DDMR 61] Switch mounting, orientation, and actuation **shall** be such that the switch cannot physically impede mechanism travel.

[Rationale: It is possible to arrange microswitches or their actuation devices such that they can unintentionally interfere with mechanism motion. An improper set-up or adjustment of a switch toward the end of the switch's allowable range can also impede mechanism motion. The arrangement of the switch has to be examined over the switch's entire possible range of adjustment (not just the procedural limits specified) to ensure that this does not happen.]

4.16.2 [DDMR 62] The worst-case maximum travel of switch actuating mechanisms **shall** not damage the switch.

[Rationale: Microswitches typically have very small ranges of motion and can be damaged when exercised beyond their small limits. To avoid such damage, the switch should be actuated in such a way that these limits are not exceeded.]

Guidance: Actuating switches in a way that prevents damage to the switch can be accomplished in a variety of ways, such as cam profiles or spring-loaded levers that release from the microswitch when depressed.

4.17 Fasteners

4.17.1 [DDMR 63] All retaining rings used **shall** be multiple-turn spiral-wound retaining rings.

[Rationale: Retaining rings that are not multiple-turn spiral-wound rings, especially smaller ones, can be easily opened beyond their elastic limit when installed and can subsequently become dislodged from their groove. Spiral retaining rings, due to their 360-degree design, are much more robust to failures of this nature.]

Guidance: *Improper installation techniques can still yield spiral rings, so care is still necessary with their installation.*

4.17.2 [DDMR 64] Set screws **shall** not be used to transmit torque between a shaft and a component mounted on the shaft.

[Rationale: *Set screws have to be highly preloaded and dig into a shaft to work in this way, creating surface imperfections and generating stresses that are difficult to quantify and often cannot be tolerated in low-margin aerospace applications. In addition, repeated loading invariably leads to a loosening of the fit as material around the set screws is deformed, regardless of whether the set screw backs out or retains its original position.]*

4.18 Heritage Mechanisms

4.18.1 [DDMR 65] The design of previously qualified mechanisms or mechanism components **shall** undergo a qualification program for use in a new application unless all performance requirements and environments of the new application are enveloped by the performance requirements and environments of the previous application.

[Rationale: *The use of part designs or entire mechanism designs that have been qualified for another purpose, commonly referred to as “heritage hardware,” is often encouraged due to perceived savings in cost and schedule. Great care must be exercised in utilizing such hardware or the anticipated savings can quickly disappear. A common mistake is selecting a design that was qualified for performance requirements or environments that do not envelope all of the new performance requirements and environments and then failing to re-qualify the design for the new application. Many lessons learned involving the use of heritage mechanisms are documented in the NASA Lessons Learned database and the proceedings of Aerospace Mechanisms Symposium and European Space Mechanisms and Tribology Symposium.]*

Guidance: *In practice, it is difficult to find designs that can be truly said to be previously qualified for a new application because of the wide variety of unique environments and missions encountered by spacecraft. Even if a subset of environments and performance requirements are enveloped, the effect of the unenveloped environments and performance requirements on the performance of the mechanism would be missed if they were not included in the qualification program.*

The extent of the qualification program for the new application is left for the responsible engineering organization to determine. Depending on the application, an appropriate requalification could range from a delta qualification to a full requalification.

4.19 Mechanism Testing

Guidance: *The test campaign requirements for a mechanism are typically outlined in a dedicated test specification such as SMC-S-016, Test Requirements for Launch, Upper-Stage, and Space Vehicles. Detailed mechanism testing activities within the overall test campaign*

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should follow Section 8 of AIAA S-114A-2020, *Moving Mechanical Assemblies for Space and Launch Vehicles*. The requirements below reflect certain aspects of testing that are important to capture for mechanisms.

Ordnance-actuated mechanical devices are also considered to be "pyrotechnic devices" and are therefore subject to the qualification requirements in applicable pyrotechnic standards or requirements documents.

4.19.1 Performance Testing

4.19.1.1 [DDMR 66] All mechanism functions **shall** be exercised during performance testing.

[Rationale: Sometimes there is a desire to test only critical functions or nominal operations, but all functions of the mechanism have to be tested to verify performance of the system, including back-up or redundant provisions. This applies regardless of whether the performance testing is conducted during qualification, life, or acceptance testing.]

Guidance: Performance testing includes the measuring of torque and force margins.

4.19.2 Qualification Testing

4.19.2.1 [DDMR 67] Each mechanism design **shall** be subjected to environmental qualification testing that exposes the mechanism to all environments, including fluid media, that it will experience in service.

***Rationale:** Qualification testing ensures that the design is sufficient to meet the requirements of the mechanism, with margin, in the operating environments it is expected to experience. The types of environmental qualification testing required for mechanisms are typically established by the program. A recommended set of tests is included in Appendix A, section A.2.11. The program also typically specifies required margin, test order, tolerances, and general methods to be used. Note that the intent of the qualification program is to accumulate all environmental test exposures on a common qualification unit (versus spreading the environments across multiple qualification units) to ensure that effects of cumulative exposure are accounted for. Behavior under environmental conditions can be very hard to predict, and omission of qualification testing can significantly increase the risk of in-service failure.*

4.19.2.2 [DDMR 68] All service configurations of the mechanism **shall** be subjected to environmental qualification testing in the mechanism's appropriate operating or non-operating state.

[Rationale: The hardware configuration (e.g., launch, on-orbit, landing) can affect many aspects of the system such as load path, stiffness, clearances, slop in joints, and radiative view factor. It is important that the qualification testing be performed on the mechanism while the mechanism is in the configuration in which it will experience the environments and perform its functions;

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otherwise, differences such as those stated may cause the unit to behave differently in test than it would in service, leading to false confidence in the hardware or unrealistic test failures.]

4.19.2.3 [DDMR 69] Mechanism qualification testing **shall** be conducted with mounting interface boundary conditions that replicate the flight boundary conditions, including the following:

- a. Stiffness.
- b. Mounting alignment and tolerances.
- c. Thermal distortions.
- d. Distortions induced by installation or externally applied loads.

*[**Rationale:** Testing with improper representation of boundary conditions can give misleading results due to factors such as additional loads, different frequency response, and extra joint slop.]*

***Guidance:** Appropriate boundary conditions can sometimes be difficult to implement in test, and compromises often have to be made between interface fidelity and the level of assembly needed to achieve good fidelity. Verification that these influences have been accounted for is expected to include explicit description of how these boundary conditions are addressed in the test plan, or if the condition does not apply in a given situation, the rationale for this inapplicability.*

4.19.2.4 [DDMR 70] Qualification units **shall** utilize flight-representative drive electronics.

*[**Rationale:** Using different drive electronics than will be used in flight can generate non-flight-like behavior, which may give false confidence in the performance of the mechanism. For this reason, the qualification test units, including the life test unit, has to utilize flight-representative drive electronics that produce the flight drive waveform when connected to the motor.]*

***Guidance:** Flight-representative drive electronics are electronics that produce the flight drive waveform when connected to the actuator. These electronics typically share the same schematic design as the flight electronics but may use off-the-shelf components. However, caution is needed to ensure that off-the-shelf electronic components perform the same as the flight electronics components. For example, the gate capacitance of a radiation-hardened field effect transistor can be up to an order of magnitude different from an off-the-shelf field effect transistor, which would change the switching time and the response of the mechanism. Care also should be taken to consider the configuration and lengths of harnessing and the software used as they may affect performance. Any differences in the drive electronics between the test configuration and flight configuration should be documented in the test report.*

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4.19.2.5 [DDMR 71] Inspections **shall** be conducted both at the start of qualification testing and at the conclusion of qualification testing.

4.19.2.6 [DDMR 72] Performance tests **shall** be conducted both at the start of qualification testing and at the conclusion of qualification testing, prior to the performance of testing with known destructive potential (e.g., a static test to ultimate loads).

[Rationale: Pre-test performance tests and inspections establish baseline condition and performance. Post-qualification performance tests and inspections allow the residual effect of the environmental test to be determined when compared to the pre-test baseline.]

Guidance: *At a minimum, one post-test performance test and inspection should be performed at the end of the suite of qualification tests, but additional tests or inspections inserted into the test flow, for example between random vibration testing and thermal-vacuum testing, or even between axes of random vibration tests, allow for easier identification of a problematic environment when anomalies occur. Depending on the nature of the mechanism and the failure modes of interest, disassembly of the mechanism may be required for effective post-qualification inspection.*

Note that it is not necessary to verify function after exposure to tests of a potentially destructive nature such as a static load test to ultimate load. A functional test would be expected after a test to the yield load, but generally an ultimate load test would only need to show that rupture or unstable failure did not occur.

Performance testing of an ordnance-actuated mechanical device via initiation of its energetic charge may be destructive. In that case, the performance test should only be conducted at the conclusion of a unit's qualification sequence and any applied pyrotechnic standards should be consulted for applicable pyrotechnic testing requirements.

4.19.2.7 [DDMR 73] Pass-fail criteria and rationale for those criteria **shall** be established for all qualification tests prior to the start of qualification testing.

[Rationale: Pass-fail criteria have to be established prior to testing to avoid rationalization of anomalous behavior detected during testing. This also helps streamline the decision making and documentation process when anomalies are encountered during the test.]

4.19.3 Life Testing

Guidance: *Life testing may not be meaningful for ordnance-actuated mechanical devices or other mechanisms that are consumed or must be refurbished after every actuation. Such devices*

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typically must use lot testing approaches. Pyrotechnic devices are subject to the testing requirements levied in applicable pyrotechnic standards or requirements documents.

4.19.3.1 [DDMR 74] Life testing **shall** be performed on all mechanism functions to verify that all service life requirements have been met.

[Rationale: Sometimes there is a desire to test only certain functions or nominal operations, but all functions of the mechanism have to be life tested to verify life of the system, including back-up or redundant provisions. Typical life concerns include fatigue limits, deterioration of lubrication, excessive wear, and deterioration during extended quiescent periods. See Appendix A, section A.2.11.1, for more guidance on life testing.]

4.19.3.2 [DDMR 75] Mechanisms whose failure could result in a loss of human life **shall** be life tested to a number of cycles determined by multiplying the total of all operational cycles and ground cycles (including test cycles, installation cycles, and maintenance cycles) by a minimum life test factor of 4.0.

4.19.3.3 [DDMR 76] All other mechanisms **shall** be life tested to a number of cycles determined by multiplying the total of all operational cycles and ground cycles (including test cycles, installation cycles, and maintenance cycles) by a minimum life test factor of 2.0.

[Rationale: It is difficult to assess mechanism reliability with a single life test. The uncertainty provided by unit-to-unit scatter, which remains unquantified with a single test, is mitigated by use of a life test factor. Higher factors create more confidence that subsequent builds will possess the required service life. A factor of 2.0 is consistent with the factor used on past successful robotic space flight missions, and a factor of 4.0 is consistent with previous human spaceflight implementations typically requiring higher reliability.]

Guidance: *The minimum life test factors specified above are intended to address mechanical life and wear concerns, not vibration and fatigue life concerns, though in certain cases the mechanical motion and life are driven by vibration and the cycle count and magnitudes have to be derived from the vibration environment. It is highly recommended that spare cycles be added to the total cycles required to allow troubleshooting or execution of extra cycles without exceeding the mechanism's certified life.*

An example calculation for determining the total number of cycles required in the life test follows:

A filter wheel mechanism with a mechanical detent used on an orbiting observatory is required to perform 1500 cycles over its life. It undergoes run-in testing, one cycle of initial performance testing, one cycle of performance testing after the random vibration test, one cycle of performance testing at each hot and cold extreme of a thermal-vacuum test comprised of four hot-to-cold excursions, and one cycle of performance testing prior to flight. The design lead wishes to incorporate ten contingency cycles as margin for unexpected actuations. How many cycles have to be applied to the life test unit to meet the life test requirement?

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Run-in of the example mechanism requires 5 percent of the operational cycles, in this case 75 cycles. The rest of the tests add another 11 cycles. There are no other cycles required for installation, maintenance, or any other activity, so the total number of cycles expected on the mechanism is 1586 and 10 cycles of margin brings the final total to 1596 cycles. The failure of the filter wheel is not safety-critical, so the total number of cycles is multiplied by 2. The resulting number of required cycles in the life cycle test is therefore 3192 cycles.

4.19.3.4 [DDMR 77] Life testing **shall** include a number of cycles at the expected operating environmental extremes, loads, ranges of motion, and speeds that is representative of the number of cycles at those conditions expected in the service life of the mechanism.

[Rationale: Temperature and vacuum conditions can both have significant effects on component life due to effects on lubrication, friction, and material properties. Not properly including these environments in the life tests can lead to test results that are not indicative of how the hardware will perform in service. In addition, the extent of motion must be accurately represented. For example, small or dithering motions can be more severe because they can wipe away liquid lubricant and create debris dams at the ends of the range of motion that prevent the flow of oil back into the contact zone. A life test that exercised only the full range of motion for such parts could give a false impression of life. Such mistakes have resulted in mission failures.]

Guidance: Appendix A, section A.2.11.1, contains recommendations on how to allocate cycles of the life test to different environments.

4.19.3.5 [DDMR 78] The life test **shall** include the operation of bearings under the maximum predicted operational contact stress for the number of cycles predicted to experience maximum operational contact stress in flight multiplied by the life test factor specified by section 4.19.3.2 or section 4.19.3.3 of this Standard.

[Rationale: Bearing life and lubricant life are highly dependent on contact stress; it is necessary to ensure that flight-like contact stresses are present in the life test to avoid misleading test results.]

4.19.3.6 [DDMR 79] Life testing **shall** include intentionally running the mechanism into the mechanical stops during each test cycle.

[Rationale: Mechanical stops perform a function of the mechanism and must be life tested. Often mechanical stops are not contacted under nominal testing conditions, so the ability of the mechanical stops to function over the life of the mechanism may not be evaluated in ground testing. Mechanical stop contact loads may be a driving load case not only for the mechanical stop but for other parts of the mechanism. In service, conditions may be such that the mechanical stops are contacted consistently regardless of whether or not the mechanism reached its mechanical stops during ground testing. For this reason, the mechanical stops need to be intentionally tested during a life test.]

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Guidance: Mechanical stops are often called "hard stops" but may be made compliant using components such as springs or preloaded spring plungers in order to limit the magnitude of contact loads.

When stops are not normally contacted due to the use of software stops or other means, it may be impractical to run the mechanism into the mechanical stops during an assembly-level life test. In these situations, life testing of the mechanical stops may be conducted separately at a subassembly level in place of assembly-level testing. When subassembly-level life tests are used it is recommended that the assembly-level life test run the mechanism into the mechanical stops for at least the first cycle before continuing the test using the nominal range of motion. This serves as a preconditioning step to help avoid false life test results should a mechanical stop contact occur.

4.19.3.7 [DDMR 80] The life test **shall** be performed on a unit that has been exposed to all environments and levels expected to be encountered by the flight unit, including those encountered during testing.

[Rationale: Performing the life test on a unit that has been exposed to environments helps ensure that environmental effects on the mechanism life are accounted for.]

Guidance: For programs with a qualification unit, it is often most convenient to utilize this qualification unit for life testing. For programs utilizing a protoflight strategy, a qualification unit is not available and a dedicated life test unit is needed. Whereas qualification units are exposed to qualification levels prior to running the life test, a dedicated life test unit need not use qualification levels for the environmental exposure. The dedicated life test unit exposure levels should, however, envelope the ground test and operational environments seen by the protoflight unit, including ground test environments.

4.19.3.8 [DDMR 81] A performance test **shall** be conducted during both the first and last cycles of the life test.

[Rationale: Beginning-of-life performance tests establish baseline condition and performance. End-of-life performance tests allow the residual effect of the life test to be determined when compared to the beginning-of-life baseline.]

4.19.3.9 [DDMR 82] Pass-fail criteria for life tests and performance tests **shall** be established prior to the start of life testing.

[Rationale: Pass-fail criteria have to be established prior to testing to avoid rationalization of anomalous behavior detected during testing. This also helps streamline the decision making and documentation process when anomalies are encountered during the test.]

4.19.4 Acceptance Testing

Guidance: *The requirements in this section are intended to be applied to flight mechanisms. The best practice is to also perform acceptance testing on qualification units before beginning the qualification test program.*

Acceptance testing may not be meaningful for ordnance-actuated mechanical devices or other mechanisms that are consumed or must be refurbished after every actuation. Such devices typically must use lot testing approaches. Pyrotechnic devices are subject to the testing requirements levied in applicable pyrotechnic standards or requirements documents.

4.19.4.1 [DDMR 83] Each mechanism **shall** be subjected to environmental acceptance testing that exposes the mechanism to all environments that it will experience in service.

[Rationale: *Acceptance testing ensures that the individual hardware build is sound and capable of meeting its performance requirements in the operating environment it is expected to experience.]*

Guidance: *The types of environmental acceptance testing required for mechanisms are typically established by the program. A recommended set of tests is included in Appendix A, section A.1.12. The program also typically specifies required margin, test order, tolerances, and general methods to be used. Omission of acceptance testing can significantly increase the risk of in-service failure.*

4.19.4.2 [DDMR 84] All service configurations of the mechanism **shall** be subjected to environmental acceptance testing in the mechanism's appropriate operating or non-operating state.

[Rationale: *The hardware configuration can affect load path, stiffness, clearances, slop in joints, radiative view factor, and many other aspects of the system. It is important that the acceptance testing be performed on the mechanism while the mechanism is in the configuration in which it will experience the environments and perform its functions; otherwise, differences such as those stated may cause the unit to behave differently in test than it would in service, leading to false confidence in the hardware or unrealistic test failures.]*

4.19.4.3 [DDMR 85] A run-in test **shall** be performed on each mechanism prior to undergoing any other acceptance testing.

[Rationale: *Run-in testing (also known as wear-in testing) serves two purposes: (1) it acts as a screen to detect material and workmanship defects that manifest themselves early in the mechanism's life ("infant mortality"), and (2) it allows the mechanism to work through initial transient behavior and reach steady-state performance. This allows for a better understanding of mechanism performance and easier observation of trends and detection of anomalies.]*

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Guidance: *If performance metrics do not appear to be leveling off, trends are not as expected, or performance levels are not as anticipated, this may be an indication of failure. Sufficient data must be obtained during the run-in test to allow determination of these important performance characteristics and their trends. Without sufficient data, for example, high enough sample frequency, actual trends may be masked. In addition, measurements of important parameters should be made as directly as possible to help avoid ambiguous results.*

4.19.4.4 [DDMR 86] The run-in test **shall** be conducted for at least 15 cycles or 5 percent of the total expected service life, whichever is greater.

[Rationale: *These run-in durations have been shown through experience to be adequate for achieving the workmanship screening goals of run-in testing in most cases; however, it may not be sufficient to achieve smooth, consistent performance in all situations.]*

Guidance: *If performance metrics do not appear to be leveling off or friction levels have not been reduced to a desired level at the conclusion of the run-in test, continued run-in may be warranted. For this reason, among others, it is recommended to include margin in the number of run-in cycles used in calculating the service life for the mechanism. The definition of what constitutes a cycle is application-dependent and has to be determined on a case-by-case basis.*

4.19.4.5 [DDMR 87] The run-in test conditions **shall** be representative of the operational loads, speed, and environment.

[Rationale: *Running-in using conditions other than those expected in service can produce data during the test that is not indicative of true performance or can even damage hardware. For example, running-in at a different speed than anticipated in service can change the lubrication regime in a wet-lubricated bearing, and running-in a molybdenum disulfide dry film lubricant at ambient pressure can generate excessive wear of the lubricant and inaccurate friction values.]*

Guidance: *If a variety of environments are expected in service, a split similar to that recommended for life testing in Appendix A, section A.2.11.1, should be used. This requirement is not intended to require application of all environments simultaneously.*

4.19.4.6 [DDMR 88] Inspection and performance tests **shall** be conducted after run-in testing prior to further acceptance testing, and at the conclusion of acceptance testing.

[Rationale: *Pre-test performance tests and inspections establish baseline condition and performance. Post-test performance tests and inspections allow the residual effect of the environmental test to be determined when compared to the pre-test baseline.]*

Guidance: *Unlike inspections after qualification testing, the post-acceptance inspections have to be made without disassembly or other breaking of configuration. At a minimum, one performance test and inspection should be performed at the conclusion of the acceptance test program; but additional tests or inspections inserted into the test flow, for example between random vibration testing and thermal-vacuum testing, or even between axes of random vibration*

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test, allow for easier identification of a problematic environment when anomalies occur. Be sure to account for these performance tests when calculating the required service life of the mechanism.

4.19.4.7 [DDMR 89] Pass-fail criteria for all acceptance tests and the rationale for those criteria **shall** be established prior to the start of acceptance testing.

[Rationale: Establishing pass-fail criteria prior to testing avoids normalization of anomalous behavior detected during testing. This also helps identify what successful performance means and helps streamline the decision making and documentation process when anomalies are encountered during the test.]

4.20 Mechanism Installation

4.20.1 [DDMR 90] Mechanisms **shall** either be designed to preclude installation in an incorrect orientation or be clearly labeled in a manner that indicates proper installation orientation.

[Rationale: There have been incidents in the past wherein mechanisms whose correct installation orientation is not overtly obvious have been installed improperly because the interface allowed this to happen.]

Guidance: To prevent improper installation, features such as keys or asymmetric mounting patterns should be utilized whenever possible. If such provisions are not possible or practical, the hardware has to be at least labeled in such a way that makes improper orientation obvious. Depiction of the correct orientation on the drawing, drawing notes, or other procedural controls that are not present on the hardware are not as effective and are not considered sufficient to meet this requirement.

APPENDIX A: BEST PRACTICES FOR MECHANISMS

A.1 PURPOSE

This Appendix presents best practices for the design, development, and testing of aerospace mechanisms that past experience has shown to create the maximum potential for successful mechanism operation in service, pitfalls to recognize and avoid, and background information to enhance understanding of the requirements in section 4 of this Standard. This Appendix contains no requirements that must be followed to adhere to this Standard.

A.2 BEST PRACTICES

A.2.1 Torque and Force Margin

In this section, references to “torque margin” also apply to linear motion, with “force” replacing “torque.”

A.2.1.1 Effect of Magnetic Saturation on Torque Margin

When determining torque or force margins on electromagnetic actuators such as motors or solenoids, one must consider magnetic saturation and current limiting effects to obtain an accurate determination of torque margin in cases where torque or force measurement at the worst-case operating conditions cannot be made. A common mistake is to assume that the relationship between output torque and voltage and current is linear throughout the range of operation. Such an assumption does not account for the nonlinear shapes of the magnetization curves of the motor materials, or for motor driver design details that limit current in nonlinear ways. Figure 4, Illustration of the Effects of Magnetic Saturation and Motor Drivers on Actuator Output, illustrates how the assumption of linear behavior can lead to incorrect output calculations.

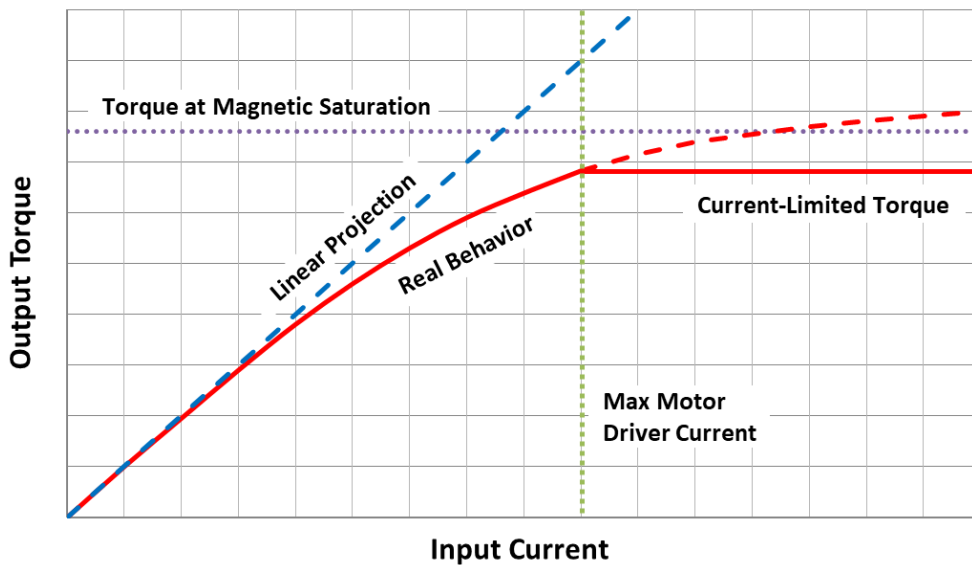


Figure 4—Illustration of the Effects of Magnetic Saturation and Motor Drivers on Actuator Output

A.2.1.2 Torque Margin Calculation at Higher Levels of Assembly

When attempts are made to calculate torque margins at higher levels of assembly (subsystem or vehicle), often only a single value of the total resisting torque can be measured with the individual resisting contributors no longer identifiable. In this case, the approach described in section 4.1 of this Standard have to be adjusted.

The following steps should be used for calculating torque margin at higher levels of assembly:

- (1) At the higher level of assembly, measure the total resisting torque, $T_{res\ assy}$.
- (2) If unit-level test data are available, sum the resisting torques for all contributors measured during the unit-level tests per Equation A-1. Apply no torque margin safety factors.

$$T_{res\ unit} = \sum T_f + \sum T_v \quad (\text{Equation A-1})$$

- (3) Find the magnitude of the difference between the measured assembly resisting torque (Step 1) and the sum of the unit level resisting torques (Step 2).

$$T_{delta} = T_{res\ assy} - T_{res\ unit} \quad (\text{Equation A-2})$$

- (a) If the difference is negative (i.e., the total resisting torque measured at the higher level of assembly has been reduced from that measured during the unit-level tests), no further calculation is required since the torque margin is inherently greater than calculated from the unit-level test data. Therefore, the torque margin that was

calculated based on unit level test data can be used as the torque margin at this higher level of assembly.

- (b) If the difference is positive, (i.e., the total resisting torque measured at the higher level of assembly has increased from that measured during the unit-level tests), add an “additional assembly resistance” term $FS_v T_{\Delta}$ to the unit level $\Sigma FS \cdot T$ terms of Equation 4-1, as shown in Equation A-3. The torque margin safety factor has to be selected from the FS_v column because the source of T_{Δ} is not known and therefore has to be considered variable.

$$\text{torque margin} = \frac{T_{\text{avail}}}{\Sigma FS_f T_f + \Sigma FS_v T_v + \Sigma FS_a T_a + \Sigma FS_v T_{\Delta}} - 1 \quad (\text{Equation A-3})$$

- (4) If no unit-level test data exists, replace the entire denominator in Equation 4-1 with an “overall assembly resistance” term $FS_v T_{\text{res assy}}$. The default torque margin safety factor FS_v is selected from the FS_v column because the source of $T_{\text{res assy}}$ is assumed to be a combination of fixed and variable terms. However, if the sources of the assembly resistances are understood and not variable, then the lower FS_f safety factor can be used instead of FS_v (see Table 1). The higher level of assembly torque margin is then calculated as shown in Equation A-4:

$$\text{torque margin} = \frac{T_{\text{avail}}}{\Sigma FS_v T_{\text{res assy}}} - 1 \quad (\text{Equation A-4})$$

A.2.1.3 Energy Margin

Energy margin is an additional value that is sometimes calculated for mechanisms. If torque/force margin requirements are followed through the entire range of motion of a mechanism, it is unnecessary to calculate energy margin. All the energy in the world will not move a mechanism if insufficient force is available. However, an application for energy margin occasionally arises such as when evaluating pyrotechnic device performance or in special cases where sufficient torques or forces cannot be applied over the entire range of motion of the mechanism and the mechanism energy needs to be assessed while coasting through this unpowered phase. Energy margin is not permitted to be used as an alternate approach to satisfy force and torque margin requirements in section 4. However, energy margin may be a useful tool for evaluating risk of a non-compliance with force and torque margin requirements.

Energy margin may be calculated using Equation 4-1. In these situations, “Energy” replaces “Torque” in the equation and descriptions. E_{avail} represents the total stored energy in the system that is available to do work, E_f is typically the energy required to accelerate a mass at a given rate or to a given velocity, and E_v is typically energy dissipated by nonconservative forces in the system. Energy margin should, of course, be greater than zero; but given the risky nature of relying upon energy margin, a higher value is often warranted to ensure that the mechanism retains reserve energy that can be applied in the event of an unforeseen effect that dissipates energy from the mechanism.

A.2.2 Lubrication

Lubricant selection for space mechanisms is an important consideration and often dictates the performance and lifetime of a given application. When hardware is available, the best and most direct method for determining lubricant life is to perform life tests in a relevant flight-like environment. At the design stage, stress cycle analysis is often employed to quickly estimate lubricant life. For example, bearing life predicted by stress cycle analysis of the bearing lubricant is more useful than fatigue life predictions of the bearing raceways since with current technology it is lubricant life, and not fatigue life, that dictates bearing performance and lifetime. The NASA Engineering and Safety Center Mechanical Systems Technical Discipline Team maintains a “Lubricated Mechanism Bearing Life Test Database” at <https://nen.nasa.gov/web/ms/lifetest-db/> that aids with the determination of an allowable value for stress cycles, sometimes referred to as the “cumulative degradation factor” (CDF). Data on CDF and stress cycle allowables for different lubricants can also be found in NASA/CR—2005-213424, Lubrication for Space Applications, and in papers by Mobley, et al. (2018) and Suffern and Parker (2018).

While many lubricants are available commercially, few have been qualified for the vacuum and wide temperature extremes of space, which preclude most lubricants from consideration. There are two basic types of lubricants to consider: wet lubricants (i.e., oils and greases) and solid, or dry, lubricants. In general, wet lubricants are preferred for most space applications provided that the thermal environments of the application fall within the performance specification of the wet lubricant. Solid lubrication is often selected when wet lubricants are unable to meet the thermal conditions specified for a given application (e.g., cryogenic environments). Care should be taken to specify the proper amount of wet lubricant to be applied. While the effects of too little lubrication may be obvious, excessive lubrication can also cause problems such as reduced clearances or high forces or torques due to viscous effects. Regardless of the lubricant chosen, metal-to-metal tribological contacts should be composed of dissimilar materials whenever the application allows.

A.2.2.1 Wet Lubrication

When considering wet lubrication, the selection of grease versus oil is important. Grease consists of a base oil that may contain additives and a thickening agent. Oil tends to creep and migrate over time, and so grease is often preferred because of its ability to remain at a given location and supply oil gradually over time. Grease consistency over the operating temperature range is critical in determining how the lubricant will perform in a given application. A typical consistency for commercially available grease, and the one used almost exclusively in space applications, is designated as National Lubricating Grease Institute (NLGI) grade 2; however, consistencies ranging from grade 000 which is the consistency of cooking oil, up to grade 6 which is the consistency of cheddar cheese, exist. To illustrate the importance of selecting a proper grease consistency, consider a grease-lubricated high-speed ball bearing. A softer grease consistency will “slump” or flow behind each ball as it races around the bearing raceway, causing the following ball to perform work on the grease to push it out of its track. This increases a bearing’s drag torque and temperature rise. If, on the other hand, the grease is stiffer and forms a channel as the balls race around the bearing raceway, the balls will not encounter large

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quantities of grease in the track. This scenario reduces a bearing's drag torque and temperature rise. It is not surprising that high-speed bearings typically prefer channeling greases to their non-channeling counterparts. Gears can be problematic applications because gear tooth meshing tends to push and wipe lubricant away from where it is needed. Greases for gears need to have consistency to both adhere to the gear teeth and to reflow back into the contact zone and reestablish lubrication. For this reason, it is often desirable to use softer greases such as NLGI grade 1. The takeaway regarding consistency is that it matters, as it is the consistency that maintains and distributes the supply of oil, and that every application may be suited to different consistency. Mechanism performance and lifetime can be maximized by properly addressing grease consistency effects via a dedicated test program.

Any lubricant selected for a given application has the potential to cause harm as a contaminant. All wet lubricants have the potential to evaporate and then condense on cold optics or contamination-sensitive components such as exposed electrical contacts. Consultation with a contamination control engineer is strongly encouraged when assessing wet lubricant volatility. Analysis and testing may be necessary to reduce risks associated with volatile contamination. Per NASA-STD-6016, Standard Materials and Processes Requirements for Spacecraft, NASA uses the testing method of ASTM E595, Standard Test Method for Total Mass Loss and Collected Volatile Condensable Materials from Outgassing in a Vacuum Environment, to evaluate the volatility of materials, including wet lubricants intended for space applications. Care should also be taken to ensure that the vent paths from volumes containing liquid lubricants do not impinge on critical surfaces.

Contamination arising from oil creep should also be considered. An oil will wet (spread on) a solid whose surface energy is higher than the oil's surface tension. An oil will not wet a solid contaminated (intentionally or unintentionally) by films having a lower surface energy than the oil. "Barrier films" are very low surface tension solids such as perfluoro polymethacrylates dispersed or dissolved in a carrier solvent. When barrier films are applied to an area of a solid's surface and the carrier solvent evaporates, that area is rendered unwettable since the oil's surface tension (17 to 25 dyne/cm for perfluoropolyalkylether (PFPE) oils at 20 °C and 32 dyne/cm for Pennzane® multiply alkylated cyclopentane (MAC) oils at 20 °C) is higher than the surface energy of the barrier film (~10 dyne/cm). Where grease and oils are employed for lubrication, a common best practice to reduce risks associated with creep is to apply barrier films at appropriate locations to both retain oil in the desired areas and prevent oil from creeping onto and contaminating sensitive areas like optics. Barrier films are ~1/8 micron thick, invisible, delicate, and easily disrupted. To protect them from incidental contact and handling damage, a best practice is to apply them in machined grooves.

An additional consideration is contamination of the lubricant itself. During the manufacture and handling of wet lubricants, it becomes contaminated with metallic debris and other particulate matter. All wet lubricants intended for spaceflight should be filtered to remove contaminants to avoid introducing them into the mechanism during application of the lubricant. Oils should be filtered through 0.45-micron filters, and grease should meet the "maximum particulate contamination per cubic centimeter of grease" requirement of MIL-G-81937, Grease, Instrument, Ultra-Clean, Metric, "Ultra-clean" grease: < 1000 particles 10 micrometers or larger, and zero

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particles 35 micrometers or larger. Once filtered, wet lubricants should be handled and introduced into the application under controlled conditions (e.g., clean room, class 100 flow bench).

A.2.2.1.1 Wet Lubrication Selection

The two most common wet lubricant chemistries in space applications are hydrocarbons and PFPEs, although other chemistries such as esters have been flown. Most hydrocarbon oils in modern space applications are typically of the low vapor pressure synthetic variety, such as the Pennzane[®] MAC oils and polyalphaolefin (PAO) oils, although applications persist for oils like the higher vapor pressure non-synthetic super-refined mineral oils like KG-80 or low-creep greases like Apiezon[®]. The International Space Station (ISS) control moment gyroscopes (CMGs) employ oil reservoirs to ensure a long-term supply of KG-80 oil, despite its higher vapor pressure.

Lubricant consumption is normal and occurs in all tribological contacts. In chemical terms, hydrocarbons are consumed in the tribocontact at a slower rate than their PFPE counterparts, leading to the general result that under equivalent conditions, the lifetime of hydrocarbons in terms of stress cycles, exceeds the lifetime of PFPEs. PFPEs should be avoided when lubricating aluminum or titanium surfaces, especially those operating in the boundary lubrication regime where sliding or fretting is anticipated. These metals are notorious for their rapid degradation of the PFPE backbone via Lewis acid catalyzed degradation. Silicone-based lubricants are not used in modern space applications as they quickly degrade into an abrasive, polymerized product, resulting in relatively short lifetimes.

Boundary lubricated counterfaces with intimate asperity contact, as found in slow speed gears, bearings, sliding contacts, and rollers, typically are best served by employing synthetic hydrocarbons formulated with boundary additives, often referred to as anti-wear additives, such as low vapor pressure phosphate esters. PFPEs typically suffer from rapid degradation in boundary conditions when oil-soluble anti-wear additives are absent. Molybdenum disulfide is an insoluble boundary additive frequently used in PFPE grease formulations. PTFE present as an insoluble thickener in PFPE grease formulations also affords some boundary protection, but often fails to prevent the PFPE oil from degrading under harsh boundary conditions.

The higher molecular-free volume of PFPEs compared to hydrocarbons renders the PFPE more susceptible to permeation by water and oxygen and hence corrosion of metallic surfaces. Neat hydrocarbon oils provide better corrosion resistance than neat PFPE oils, and oils with corrosion inhibitors added offer still better corrosion resistance than their neat counterparts. Alloys commonly used in space mechanisms such as 52100 steel and hardened tool steels such as M50 and M62 are susceptible to corrosion and require the use of corrosion-resistant wet lubricants.

PFPEs are typically selected when service environments fall below -20 °C (-4 °F). In these situations, hydrocarbons cannot be employed unless heaters are present. One drawback to the use of PFPEs is the tendency of PFPE oils to creep faster than hydrocarbon oils. The flow activation energy for a PFPE oil is approximately half that of a hydrocarbon oil, resulting in about twice the tendency to flow. One advantage to PFPEs is their low vapor pressure, which results in

decreased volatility. As stated earlier, PFPEs are typically less stable in the tribological contact for a given number of stress cycles. PFPEs have a much higher viscosity index and are stable at temperatures approaching 200 °C (392 °F) whereas their hydrocarbon counterparts are only thermally stable up to about 120 °C (248 °F). On the cold side, PFPEs have been known to perform down to the -60 °C (-76 °F) to -70 °C (-94 °F) range depending upon the application. The superior vapor pressure and thermal performance of PFPE are largely responsible for their wide use in space applications, despite the drawbacks cited earlier. Since PFPEs are known for their chemical compatibility with various elastomers and polymers, they are often employed on gaskets and O-rings to enhance sealing and prevent binding. MIL-PRF-27617G, Grease, Aircraft and Instrument, Fuel and Oxidizer Resistant, specifies performance requirements for PFPEs.

High speed applications operating in the EHD regime have benefited greatly from the use of synthetic hydrocarbons versus their non-synthetic counterparts due in part to their lower vapor pressure, but synthetic hydrocarbons have lower pressure-viscosity coefficients resulting in thinner EHD films when compared to their non-synthetic counterparts, e.g., Pennzane[®] versus Coray 100[®]. Pennzane[®]-based lubrication (oil or grease) is now commonly used in virtually all high-speed bearing applications, including reaction wheels and control moment gyroscopes operating in the EHD regime. One notable exception cited earlier is ISS CMGs, which are lubricated with recirculating KG-80 oil.

A.2.2.2 Solid Lubrication

Solid lubricants include self-lubricating composites, dry-film lubricants (DFL), and low shear strength metal films such as gold, silver, and lead. When using dry-film lubricant such as molybdenum disulfide, proper attention to surface preparation and cleanliness should be given. Considerations for polymer-bound DFLs are noted in MIL-PRF-46010H, Lubricant, Solid Film, Heat Cured, Corrosion Inhibiting. This specification describes surface preparation, application of the lubricant, and associated performance requirements for properly applied and adherent films. When using sputtered, physical vapor deposited (PVD), or chemical vapor deposited (CVD) forms of DFL, the substrates are typically pre-sputtered in vacuum with argon ions to ensure adventitious carbon and metal oxides are removed prior to application of the lubricant. Failure to properly prepare surfaces prior to application of any form of DFL will result in poor adhesion. The same is also true for deposition of metallic films.

The use of transition metal dichalcogenides such as molybdenum disulfide require special consideration of humidity effects. In general, the polymer-bound form of molybdenum disulfide is the most stable and least sensitive to negative performance effects attributed to exposure to humid environments. Polymer-bound forms of DFL are less crystalline and have limited reactive edge sites available for reaction with water. Other polymer-bound formulations not conforming to MIL-PRF-46010H exist, including those polymer-bound forms that are prepared by electrophoretic methods. Polymer-bound forms of DFL typically result in thicknesses around 12 microns (0.0005 inches) maximum. Burnished (mechanically applied) films (up to 0.4 microns thick) and physical and chemical vapor deposited forms of molybdenum disulfide (~ 1 micron thick) are thinner films and are highly crystalline forms possessing reactive edges sensitive to humidity exposure. Pure sputtered molybdenum disulfide coatings (~ 1 micron thick) can also

exhibit sensitivity, but these have been replaced in most applications with co-sputtered nanocomposite molybdenum disulfide coatings containing nickel or antimony oxide that are considerably more robust with respect to humidity.

When molybdenum disulfide is exposed to humidified air, there is a conversion at the surface to molybdenum trioxide, a higher friction material. This brittle oxide shows a higher coefficient of friction and wear rate until it is worn away, resulting in a net loss of original molybdenum disulfide film. Therefore, operation in humid environments is not recommended and should be minimized if such loss is a concern. Exposure to humidified environments without operation results in conversion to the oxide and a higher coefficient of friction that will not recover until the oxide is worn away. For this reason, molybdenum disulfide films should be stored in anhydrous environments whenever possible, including the use of dry inert gas purging (e.g., dry nitrogen). Slip ring brushes based on molybdenum disulfide are especially vulnerable because oxidation products create electrical noise. It is, therefore, recommended that slip ring assemblies be designed with a built-in nitrogen purge port to allow internal purging. In addition, inorganic binders (phosphates and silicates) are sometimes used with DFLs. Because these binders themselves are hygroscopic, temperature and humidity should be controlled to ensure that the DFL does not soften or produce atmospheric reaction products (e.g., carbonates) that can impede performance.

Polymer-bound forms of molybdenum disulfide are well known to generate debris in service. For this reason, burnishing or run-in is normally performed in which the DFL coating is subjected to some predetermined number of cycles to wear the DFL smooth and orient the lamellar planes of the DFL. Copious DFL wear debris is produced during the burnishing process. Once a steady state is achieved and debris generation is minimized, it is desirable to clean the application to remove the debris generated. In all cases where polymer-bound DFL is employed, debris generation should be considered when assessing the life and performance of the application. The potential for negative effects attributed to the DFL debris such as contamination of optics or other debris-sensitive components, including electronics should be considered. Sputtered, PVD, and CVD coatings tend to generate a lesser quantity of debris (though the particle count may be higher).

A.2.2.2.1 Best Practices for Successful Dry Film Lubricated Mechanisms

To minimize friction and meet life requirements, space mechanisms frequently rely on thin layers or “films” of solid lubricants applied to substrates. Mechanism designers unfamiliar with the idiosyncrasies of such DFLs often experience DFL failures at short lives even where the nominal applied stress on the DFL is less than the DFL’s allowable stress. “DFL failure” results in unlubricated contact. If only one-half of a tribological pair is DFL-coated, DFL failure is defined as exposed substrate. If both halves of a tribological pair are DFL coated, DFL failure is defined as exposed substrates on both halves in corresponding contact locations. Such DFL failures drive home the point that dry film lubricated mechanisms must be designed and manufactured specifically with the unique characteristics of DFLs in mind: one cannot simply cross off “grease” on an assembly drawing, redline in “dry lube”, and expect success. Experience has shown that the root causes of DFL failures at short lives are less often related to the

particular DFL chosen, and more often related to inappropriate or unspecified (left to chance) design details, inappropriate or unspecified manufacturing processes, tolerances and geometric controls, and inappropriate surface preparation resulting in poor adhesion. This section summarizes the design and fabrication lessons learned from DFL failures and the best practices employed in DFL successes. For guidance in selecting DFLs, see Lince, 2020.

A.2.2.2.1.1 Design Best Practices

Control the size and geometry of surfaces and features that are dry film lubricated or come in contact with DFLs before applying DFLs. Specifically:

- Planes must be flat to a fraction of the DFL thickness.
- Cylinders must be cylindrical to a fraction of the DFL thickness.
- Cones must be conical to a fraction of the DFL thickness.
- Spheres must be spherical to a fraction of the DFL thickness.
- Surfaces that are misaligned, or can misalign under load, should be crowned where appropriate to avoid edge loading that over-stresses or rapidly wears out the DFL.

Part edges that either contact DFLs or are DFL-coated themselves, must not be sharp or chamfered, as such edges act like paint scrapers that remove the DFL from counterfaces. All such edges must be radiused and blended to produce a smooth curve without steps or reversals. All such edges should be individually dimensioned on the engineering drawing; generic notes to “BREAK EDGES” do not provide the level of control required for DFL applications (and are frequently ignored by both machinists and inspectors who do not appreciate how sensitive DFLs are to edge geometry). One way to ensure the desired result is to dimension edge radii with ASME Y14.5’s, Dimensions and Tolerancing “controlled radius symbol” CR instead of the conventional radius symbol, R. Notes like “NO STEPS” and “NO SHARP EDGES” are an alternative to CR. Notes like “BLEND” are acceptable as long as the extent of the blended region is dimensioned and toleranced appropriately.

Control both the surface roughness and the lay (the predominant direction of the surface pattern) of DFL-coated parts before applying DFLs; see ASME Y14.36, Surface Texture Symbols, for how to specify this on engineering drawings. A rule of thumb for long life is that the DFL thickness should be ten times the R_a surface roughness value so the asperity peaks do not prematurely poke through the DFL. Since bonded dry lubes are typically 200 to 500 microinch thick and sputtered dry lubes are typically 40 microinch thick, R_a values of ~35 microinch and ~4 microinch, respectively, are appropriate. A non-directional lay is ideal. In general, visible feed marks are cause for rejection but may be acceptable if the R_a requirement is met when measured perpendicular to the dominant lay direction and the dominant lay direction is parallel to the sliding direction. Ensure that processes used to reduce surface roughness do not degrade part geometry.

The surface roughness and the lay (the predominant direction of the surface pattern) of non-coated mating parts and counterfaces should also be controlled to ensure these non-coated surfaces pick up a DFL transfer film and do not rapidly sand away or file through the delicate

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DFL; see ASME Y14.36 for how to specify this on engineering drawings. In general, ground or EDM'ed surfaces (which act like sandpaper or teeth on a rasp) and surfaces with visible feed marks (which act like teeth on a file) are cause for rejection. An ideal surface has a non-directional lay with a texture characterized by plateaus (instead of sharp peaks) and valleys, like the surface produced by tumbling, fine honing, lapping, isotropic superfinishing, or electropolishing. An ideal profile trace will have a negative skewness parameter, R_{sk} ; see ASME B46.1 for more detail. Again, ensure that processes used to reduce surface roughness do not degrade part geometry.

Consider the effect of DFL thickness on clearances between parts. Though an individual DFL film is thin, diameters and dimensions of parts coated all over will change by twice the film thickness. This effect adds up to reduce clearances in assemblies comprised of multiple parts coated all over.

Consider the effect of temperature gradients on clearances between dry film lubricated parts. Though DFLs lack the heat-producing viscous drag of wet lubricants, DFLs also lack (a) the area-increasing menisci that wet lubricants form around contact areas, (b) the enhanced thermal coupling that occurs between wet-lubricated parts, and (c) the heat and mass transfer that occurs when part motion transports lubricant with a heat capacity back and forth between warm and cool regions. As a result, low-mass DFL-coated parts tend to run hotter than their higher-mass mating parts. For the same reasons, the thermal conductivity across dry film lubricated assemblies, like ball bearing assemblies, is lower than across wet lubricated assemblies.

Design assemblies to maintain area contact between dry film lubricated parts under load and to prevent edge contact which over-stresses or rapidly wears out the DFL. For example, one can spherically mount plain bearings such that they are free to self-align to shafts. Alternatively, one can use "elastic matching" concepts to allow bearings and shafts to deflect together and maintain area contact under load (see Blodgett, 1963, section 3.5, and Neale, 1995, page A13).

Choose dissimilar substrates with low mutual solid solubility (e.g., a steel shaft in a copper-beryllium bushing) and high galling threshold stress levels (e.g., Nitronic 60) to resist galling if the DFLs wear out. Substrate hardness is also important; the harder a substrate is, the more effectively the dry lube will perform. When not using heat-cured resin-bonded DFLs which protect a substrate from corrosion in service, choose substrates to avoid surface corrosion that can result in loss of DFL adhesion. Ensure that the substrates can withstand the environments (chemicals, vacuum, elevated temperatures, etc.) experienced during cleaning, coating, and curing without changing dimensions or material condition (temper, over-aging, hardness, embrittlement, etc.).

Manage the wear debris generated in service, especially when using bonded DFLs (and sacrificial self-lubricating composite materials). All DFLs wear in service; yet, at the end of their service lives, DFL-coated parts should still exhibit full-coverage, albeit thinner, solid lubricant films. Keep in mind that the volume of the DFL wear debris will be greater than the initial DFL volume. One must choose whether to recirculate DFL wear debris (which can extend life in some applications) or eject it from the dry film lubricated interface. If the latter, (a) incorporate design

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features that provide DFL wear debris with a safe place to accumulate (for example, reliefs, undercuts, dedicated debris-catching grooves, and by increasing tip to root clearance in gears since wear debris tends to pack into tooth roots and cause tip-to-root interference), and (b) prevent DFL wear debris from migrating (for example, with shields or labyrinths) to places that can contaminate optics or cause mechanisms to bind and jam. Service lives are often extended until space assets cease functioning. A robust design for managing wear debris becomes especially important as the DFL wears out and the volume of wear debris greatly increases.

Design the mechanism to accommodate disassembly, blowing out with pressurized gaseous nitrogen (GN₂), brushing, and/or wiping to allow DFL wear debris to be removed after burnishing or run-in.

Choose which part or parts will be DFL-coated. Practically speaking, it is easier to coat an outer diameter than an inner diameter. When using a hard/soft material combination in the tribological pair, the lowest coefficient of friction can be obtained by putting a smooth finish on the harder part and DFL-coating only the softer part. Under some sliding conditions, applying DFL to both parts may result in up to ~30 percent more life than when DFL is applied to only one part.

Dedicate a sheet of the engineering drawing, for each DFL-coated part, solely to DFL details. This sheet should specify the functional surfaces on which the DFL thickness matters, the non-functional surfaces on which overspray of any thickness is acceptable, and the surfaces which must remain free of DFL. The need for time-consuming application and removal of masks should be eliminated or at least minimized. Where masking is required, work closely with the DFL vendor to simplify part processing; specify mask dimensions with appropriately large mask tolerances and provide full-scale mask templates and/or reusable shields.

Select gear ratios, diametral pitches, pressure angles, profile shift coefficients, etc., to minimize the specific sliding ratio (the ratio of gear tooth sliding velocity to rolling velocity) when designing dry film lubricated gears.

Incorporate "load balancing" concepts in dry film lubricated epicyclic gearboxes so all planets carry equal torques and all planet bearings carry equal loads. It is also important to select appropriate design strategies for the different stages of a multi-stage planetary gearbox. Given that input stage gears see many more cycles than output stage gears in reduction gearboxes, and that lower contact stress gives longer DFL life, an input stage gear needs a lower contact stress than an output stage gear to have the same life. The longest life design strategy would be one where the DFL on an output stage gear fails first (since there is enough torque on the output stage to keep it turning as it fails). Another design strategy is to design the dry lube on all stages to fail simultaneously.

A.2.2.2.1.2 Fabrication Best Practices

The manufacture of dry film lubricated parts must be performed with the same level of attention to detail as their design. DFL materials are unique among platings and coatings; they are delicate and have uncommon application processes which require specialized training to ensure

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successful outcomes. Prepare a detailed process specification for the manufacturing and fabricating process that completely describes the steps, the precautions, and the controls for the manufacturing variables. Specify all requirements and their individual inspection methods. List clear pass and fail criteria for each inspection. To avoid process escapes, conclude each failure criterion with an instruction on what to do with parts that fail that particular inspection.

Many academic discussions of surface lay assume the microscopic surface asperities are symmetric with axes oriented normal to a surface, but in fact manufacturing processes can leave asperities shaped and oriented like sharkskin or saw teeth. In applications where (a) the non-coated mating part is produced by grinding and polishing *without* an intermediate process that produces a non-directional lay with negative roughness profile skewness (R_{sk}), and (b) the non-coated mating part rotates in only one direction in service, the longest DFL life will result when the final finishing operation leaves the asperity peaks “bent backward” or pointing opposite the direction the non-coated part rotates in service. A DFL-coated part rubbing a non-coated mating part in service should be analogous to a hand petting a dog or snake from head to tail, not tail to head. Each successive finishing operation of the non-coated surface should be carefully planned to reverse the asperity direction left by the previous operation (reverse bending the asperity peaks helps remove them). Practically speaking, a ground and polished non-coated mating part should be ground with the grinding wheel rotating opposite the direction of part rotation in service (creating asperities that point in the direction of part rotation), and then polished with either (a) the polishing wheel rotating in the same direction the part rotates in service, or (b) a polishing belt or pad translating opposite the direction the part surface translates in service, creating asperities that point opposite the direction of part rotation. For an expanded treatment of this recommendation, see Mahle, 2016, pages 21 to 23.

Prior to either applying DFLs or mating with a DFL-coated part, the parts and counterfaces should be visually inspected under magnification; all deviations from the above design practices should be corrected. Be on the lookout for dents, nicks, and scratches, as they are typically surrounded by raised edges that exceed the DFL thickness and will wreak havoc on the counterface. These raised edges are best removed with precision-ground flatstones. See section B.2.2 in this Standard for a list of useful precision-ground flatstone web references.

Store parts in ways that prevent them from bumping into each other or into hard objects.

Parts should then be fit-checked and, if necessary, reworked before applying the DFL. When the manufacturing methods used cannot produce the required size, geometry, fits, and/or finish, serialize the mating parts and then hand-fit them together (for example, by honing, lapping, bluing and scraping, running-in with kerosene as a lubricant) prior to applying the DFL.

Prepare part surfaces to maximize DFL adhesion, and then immediately apply the DFL. All part surfaces should start out visibly clean, with no foreign matter, grease, oil, tarnish, scale, corrosion or other contaminants. For sputtered DFLs, the part surfaces are typically precision cleaned with an argon ion plasma in the sputtering chamber immediately before applying the DFL. For bonded DFLs, promote adhesion by using techniques similar to those used to prepare parts for adhesively bonded structural joints. In general, these techniques include:

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- Scrupulous cleaning to remove all contaminants before blasting so as not to risk embedding them in the surface.
- Dry bead blasting low hardness metals (Cu alloys, Al alloys, Mg alloys, etc.) and dry grit blasting higher hardness metals (Ti alloys, steels, and stainless steels, etc.) to produce a uniform matte surface with a surface roughness between 20 and 35 microinch Ra. #10 (100-170 mesh or 120-150 grit) glass beads, and 120 to 400 grit aluminum oxide abrasive grit have proven effective in producing the desired surface roughness on different materials. Reuse of beads and abrasive grit causes them to break down and become contaminated, so always blast with new, dry beads and abrasive grit in a clean gun and clean blasting cabinet. Beware that bead and grit blasting thin parts may cause them to warp like Almen strips, so consider alternative approaches like wet sanding to roughen their surfaces. Take care to blast away only as much material as is necessary to obtain a matte finish; seconds of overzealous blasting can remove excessive amounts of material and change part geometries that took hours to carefully craft. Carefully set and control blasting process parameters (air pressure, gun type, nozzle type and size, blast media type and size, gun-to-work distance, speed of traverse, number of passes, etc.) to values appropriate to producing the desired surface roughness range on the particular substrate.
- Scrupulous cleaning to remove all grit and soils.
- Passivation or anodization or phosphate, oxide, chromate, or dichromate coating the substrates as applicable. Note that sealing Type II or Type III sulfuric acid anodized aluminum produces a surface film on the anodize that interferes with adhesion and that must be removed from the anodize by gentle glass bead blasting before applying DFLs. A better alternative to a MIL-PRF-8625F, Anodic Coatings for Aluminum and Aluminum Alloy, Type II Class 1 anodize is a phosphoric acid anodize process per the guidelines of SAE ARP1524A, Surface Preparation and Priming of Aluminum Alloy Parts for High Durability Structural Adhesive Bonding, or ASTM D3933, Standard Guide for Preparation of Aluminum Surfaces for Structural Adhesive Bonding (Phosphoric Acid Anodizing), substituting a bonded DFL for the primer mentioned.
- Verification of cleanliness by ASTM F22, Standard Test Method for Hydrophobic Surface Films by the Water-Break Test, water break testing before applying the DFL.

Specific surface preparation requirements for a range of substrates can be found in the following documents:

- JSC PRC-8001, Process Specification for Dry-Film Lubricant Application.
- MSFC-RQMT-1282, Requirements for Surface Preparation and Application of Dry-Film Lubricants.
- JPL D-51842, Solid Film Lubricants, Application of.
- Everlube Products Process Bulletin 3000-A, Solid Film Lubricants & Engineered Coatings, Recommended Application Procedure for Solid Film Lubricants & Coatings (2017).
- MIL-HDBK-83377, Adhesive Bonding (Structural) for Aerospace and Other Systems, Requirements for.

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- SAE AS1701, Adhesive Bonding (Structural) for Aerospace and Other Systems, Requirements for, Table 2.

Inspect bonded DFLs before curing. To minimize risk to the part, reprocess any obviously defective bonded films before curing while they are still relatively easy to strip off.

Ensure that the time-temperature sequence when sputtering DFLs and when curing bonded DFLs does not negatively alter the substrate's material properties, hardness, temper condition, etc. Tailor the time-temperature sequence as necessary (for example, lower temperatures for longer times) to both protect the substrate and fully cure bonded DFLs.

Minimize cleaning of DFL-coated parts. Significant effort has been expended to scrupulously clean the parts and then apply the DFL by a clean method, so do not carelessly allow them to become contaminated. Work closely with the DFL vendor to keep these clean DFL-coated parts clean from the moment the DFL is applied and/or cured (treat DFL-coated parts as if they were optics).

Minimize the handling of DFL-coated parts. When handling is necessary, wear clean room gloves and do not grip DFL-coated parts with metal tweezers (again, treat DFL-coated parts as if they were optics).

Burnish the DFL before placing the parts in service. Burnishing can be thought of as a run-in/wear-in/break-in process that removes excess DFL, resulting in a smooth film with aligned crystallite layers that will produce less DFL wear debris in service. It is normal for DFL to drastically change appearance (color, luster, reflectivity, etc.) when burnished, so beware of interpreting this change as a DFL failure. Whenever possible, burnish the DFL by running it against the mating part. When the mating part is not coated with DFL, such burnishing will create the desired dry lubricant transfer film on the uncoated part. If the uncoated mating part cannot be used to burnish the DFL-coated part, the desired dry lubricant transfer film can be artificially produced on the mating part at the beginning of life by charging its uncoated surface with the appropriate solid lubricant; see page 40 of NASA SP-5059, Solid Lubricants: A Survey, and page 22 of NASA SP-8063, Lubrication, Friction, and Wear. Following these practices at the component level makes performance of the assembled mechanism stabilize more quickly. Alternative methods of burnishing DFL-coated surfaces include:

- a. Rubbing with a conforming tool having a surface roughness R_a of 8 micro-inch max,
- b. Rubbing planar surfaces on a sheet coated with fine abrasives such as lapping film and rubbing contoured surfaces with a polishing cloth or abrasive polymer pad, and
- c. Brushing with a rotary nylon bristle brush mounted in a motorized tool.

Ensure that burnishing processes used do not degrade the geometry of the coated part.

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Ideally, burnish the DFL by running-in the DFL-coated parts in the expected operating environment. When burnishing, avoid extended running in air, as DFL wear rates in air are higher than in vacuum. Remove DFL wear debris after burnishing or running in, especially when using bonded DFLs. As noted in the design practices, the mechanism design must accommodate the complete disassembly, blowing out with pressurized nitrogen, brushing, and/or wiping that this step often requires.

Store DFL-coated parts and assemblies in low humidity environments to maximize DFL life. Options include desiccating jars, continuous dry nitrogen purge, or in dry-nitrogen-purged and heat-sealed bags that are then placed inside metal-foil moisture-barrier bags containing desiccant pouches and humidity indicating cards, which are then dry-nitrogen-purged and heat-sealed such that the heat seal width is at least as wide as the width of the moisture barrier bag's side heat seals.

A.2.3 Bearings

Bearings are common and frequently critical components in mechanical systems. As simple as they may seem, bearings can be a challenge to properly design and implement in a mechanical system. The body of knowledge on bearings is vast and continually increasing, and while a thorough treatment of bearings would require many volumes of text, this section presents some of the more important best practices for the selection, sizing, preloading, analysis, lubrication, and implementation of bearings within a space mechanism.

A.2.3.1 Ball Bearings

A.2.3.1.1 Ball Bearing Selection

The selection of a ball bearing can depend on many elements. The most common considerations are load capacity, stiffness, drag torque characteristics, life, and sometimes material compatibility and thermal characteristics, for example oxygen compatibility or the need for a low-temperature stabilized material in cryogenic applications).

Many factors influence the drag torque characteristics of a ball bearing, including applied load, preload, fits, ball complement, component precision, temperature, cage material and style, and lubricant. Bearing size also has a direct effect; all other influences being equal, a larger diameter bearing will have a higher drag torque.

The two most commonly encountered types of ball bearings are deep groove ball bearings and angular contact bearings. Deep groove ball bearings (also called radial bearings or Conrad bearings) are intended for applications where radial load capacity is the primary concern; however, they can accommodate limited axial loads as well. Deep groove ball bearings are designed with a radial clearance. For a given conformity, a larger radial clearance will result in a higher axial stiffness and correspondingly lower radial stiffness because of the larger effective contact angle that is created when loaded.

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Angular contact ball bearings support both a radial and axial load. The angle between the vector through the points of contact between ball and inner raceway and ball and outer raceway, relative to the plane of rotation, defines the contact angle. The higher the contact angle, the higher the axial stiffness and axial load capacity. The bearing rings in angular contact bearings are asymmetric and can only support the applied axial load in one direction. This means they can only be used in preloaded pairs. Angular contact bearings are recommended when tight position control or high stiffness is required.

Other types of bearings may be worth examining in certain situations. Super-duplex bearings are like a pair of angular contact bearings with either the inner or outer set of rings in a single piece. Because of the reduced tolerance stack-up from the single piece ring, they can provide higher precision than an equivalent pair of angular contact bearings. Thrust bearings are essentially deep groove ball bearings with a 90-degree contact angle. As such, the rings become upper and lower rings as opposed to inner and outer rings. These bearings are intended to support primarily axial loads and have to be used in conjunction with another type of bearing to adequately react radial and moment loads.

A.2.3.1.2 Ball Bearing Analysis

With current bearing material and lubricant technology, bearing life in space applications is nearly always governed by lubricant failure rather than metal fatigue. Though the fatigue life of a bearing always has to be shown to meet the minimum requirements given in section 4 of this Standard, the size of a bearing will typically be determined by the static load capacity.

Bearings should have analysis demonstrating acceptable material, mounting, preload, performance, and contact stresses. The analysis should account for maximum combined axial, radial, and moment loads sustained during ground handling, launch, on-orbit, entry, descent, landing, or other operational mode. System stiffness requirements also have to be addressed.

Rolling element bearings should exhibit a minimum hardness of Rockwell C58. However, it should be noted that the maximum mean Hertzian contact stress allowables noted in Table 3 were developed using bearings with more typical hardness values ($>$ Rockwell C60), and thus the as-produced bearing hardness should be considered when approaching these contact stress allowable values.

It is recommended that the maximum mean Hertzian contact stress in a rolling element bearing in high precision, low torque ripple, or long-life applications be limited to 120 ksi (830 Mpa) when subjected to the worst-case combined operational loads, including worst-case qualification environments. A large body of data exists to indicate that operating below this stress level will help avoid wet lubricant failure for long-life applications in the boundary lubrication regime.

Per the requirements in section 4 of this Standard, a bearing should never be designed such that the ball contact ellipse is truncated. However, late load changes or other circumstances can sometimes result in a truncated contact ellipse while in a non-operating condition well past the design phase. For certain applications such as deployment mechanisms where few cycles are

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needed, quiet running is not important, and plenty of torque is available, non-operating truncation may not have an overall detrimental effect. In this situation, a variance should address the redistribution of load and the stress concentration at the edge of the raceway along with testing that demonstrates adequate life and performance considering the truncation. Leveille and Frantz (2002) may provide a useful first step in such an assessment. Operating a bearing in a truncated condition is never recommended.

When analyzing a bearing, ball and ring tolerances can cause some of the balls to carry more load than others, resulting in an effective complement of balls that is lower than the actual complement. When the ball complement is small (on the order of 8-10 balls) or the preload is high, the effect is also small and can be neglected. However, bearings with larger ball complements or bearings that have very low load on the balls can exhibit torque characteristics more representative of bearings with smaller complements. It may be advisable to model torques as if only a portion of the complement is effective. The appropriate reduction will vary with the circumstances. This effect may also manifest itself under thermal conditions that relieve preload.

Bearings in which motion is limited to small oscillations have to be analyzed with special techniques, especially when the angle of oscillation is small enough that no ball overlap occurs in the raceways. In such cases, wet lubricants are pushed out of the loaded region during small oscillatory movements and prevented from flowing back into the contact region by debris dams. Without large rotational movement of the bearing surfaces in the raceway, there is no method to reintroduce the lubricant to the contact region. In addition, a phenomenon known as “blocking” in which a bearing experiences progressive increases in torque over the cycle life can manifest itself in bearings that undergo only oscillatory motions of larger magnitude (e.g., +/- 90 degrees). The phenomenon is not well-understood but appears to involve either ball speed variation effects and/or transverse ball creep in bearings with tight raceway conformity. For more information, see Loewenthal (1988).

Care has to be taken when analyzing bearings that use thin section rings since most bearing analysis codes assume that the bearing rings are rigid. Thin section rings have a low enough stiffness that the deflection of the rings under preload can affect the bearing fit, change the torque characteristics, change the contact stress, and reduce stiffness. Analysis of thin section rings generally requires the use of special techniques or tools.

A.2.3.1.3 Ball Bearing Preload

Preloading a ball bearing eliminates free play, reduces runout of the rotating member, increases axial and radial stiffness, prevents fretting damage, eliminates impact loading when exposed to a vibration environment, increases the load sharing among balls, and prevents skidding of balls. Ball skidding typically occurs when there is insufficient traction to keep the balls rolling in cases of high angular acceleration, and is detrimental because it creates frictional heat that can decrease lubricant life and damage the bearing raceway and ball. Preload also increases Coulomb friction torque and increases difficulty in controlling the preload tolerance. As preload increases, lubricant life decreases, wear increases, and the sensitivity of preload to temperature increases. All of these considerations have to be balanced when choosing a preload.

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There are unusual applications in which a preload is not advisable or is simply unnecessary. Examples include the following:

- A non-preloaded radial contact bearing at the opposite end of a shaft from a preloaded angular contact pair.
- Radial contact bearings on a shaft where both bearings are always loaded, such as by belt tension or a gear reaction torque.
- Bearings with low-life requirements that do not merit the complexity and expense of preloading.
- Bearing applications where the bearing's axial play cannot be allowed to shrink as it accommodates thermal deflections.

In such cases, one should expect to provide a rationale that describes why a preload adds no value or is detrimental when preparing a variance against this requirement.

Bearings can be preloaded via two methods. The first method uses a structure that is much stiffer than the bearings, usually called "hard preloading." Hard preloading is achieved by precision grinding the bearing rings or spacers between them and then clamping them together. The second approach uses a spring or other structure that is much less stiff than the bearing in conjunction with a bearing ring that slides, or other structure that is much less stiff than the bearing (with no sliding). This is usually called "soft preloading" or "spring preloading." This allows the magnitude of the preload to be controlled or adjusted. However, axial loading sufficient to overcome the preload in the direction of the low-stiffness element will be reacted only by the low-stiffness element, which can allow large displacements unless controlled in some way such as with a snubber.

For any given preload, the stiffness of the assembly will be higher with a hard preload than with a soft preload. Regardless of preload method, be sure to consider the clamping forces used on the inner and outer rings and the clamped stiffness that will result. Excessive clamping force can cause unwanted deformation of the rings and negatively affect the performance of the bearing. Insufficient clamping force can be overcome by vibratory loads and cause detrimental performance changes as well.

Bearings should be preloaded with a load calculated to withstand the operational environments with no unloaded balls, known as "gapping." Gapping under operational conditions is undesirable but may be tolerable in certain cases. However, increased component testing that verifies performance in this condition becomes necessary because it is difficult to predict the effects of gapping analytically. Testing should demonstrate lubricant lifetime, bearing component lifetime, specified functional performance, and shaft stiffness. Under non-operational environments, it may be permissible to have some balls unloaded.

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A.2.3.1.4 Ball Bearing Lubrication

Given the typical bearing materials that are currently employed, the life of a bearing in a space application is usually limited by the lubrication. Key factors in establishing the life of a lubricant include contact stress, number of stress cycles, and temperature. Other important factors affecting the performance of the lubrication system include the amount of lubricant, retainer design, and for wet-lubricated systems reservoir design, and proximity of the reservoir to the areas requiring lubrication. These factors should be examined with development testing whenever possible.

A.2.3.1.4.1 Oil or Grease Lubrication

Bearings used in vacuum applications that are lubricated with oil or grease will typically need to utilize labyrinth seals, anti-creep barriers, or both to prevent loss of base oil. Anti-creep barriers are usually applied to the surface of all components in contact with the bearing. Bearings utilizing oil or grease lubrication are typically used with a phenolic laminate cage that can act as a reservoir. However, the phenolic cage has to be properly impregnated with the oil (or base oil of the grease) used in the bearing to be able to be used in this way. If the impregnation is insufficient, they can absorb the oil (or base oil from the grease), resulting in lubricant depletion. Impregnation (or cleaning) can take some time to give the substances enough time to fully enter (or exit) the phenolic. On occasion bearing assemblies will be procured with phenolic laminate retainers that are lubricated with an oil unsuited to the application. The best course to follow in this situation is to remove the original retainers, clean the bearing components, reassemble with new retainers, then impregnate the bearing assembly with the new oil. If this is not possible, sufficient cleaning of the bearing assemblies, typically using a Soxhlet extractor, must be performed to ensure that all traces of the original oil are removed. Without proper cleaning, the original oil or even traces of the cleaning solvent can bleed out of the retainer and react with the new lubricant, causing failure. To remove residual cleaning solvent, the cleaning process should be followed by a vacuum-bake treatment.

When oil or grease is used on a bearing in a habitable volume, lubricant creep or evaporation remains a concern, not only to control lubricant depletion but also to avoid contaminating the habitable volume with the lubricant. The risk of contamination of the bearing is higher as well. As a result, labyrinth seals with anti-creep barriers or dynamic seals are usually employed on the bearings, often in conjunction with static seals in the mechanism's static interfaces. The use of dynamic seals will have an impact on the performance of the bearings and has to be considered in the bearing analysis.

The quantity of oil or grease used generally involves striking a balance between having sufficient lubricant available to meet the life performance requirements, but not so much that containment becomes difficult or high parasitic torques are encountered. For applications that require a very long life, lubricant reservoirs and replenishment may be necessary.

In bearings where obtaining adequate life depends on operation in the EHD lubrication regime, it is important to establish the presence of an EHD film via analysis or testing.

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Bearings whose motion is normally limited to small oscillations should, if possible, be subjected to periodic larger motion sweeps to re-wet the contact surfaces.

For guidelines on the selection of bearing lubricants, see section A.2.2 in this Standard.

A.2.3.1.4.2 Solid Lubrication

Solid lubrication encompasses dry film lubrication and sacrificial cage materials. Optimum performance is often achieved with a combination of the two, self-lubricating cages used with DFL-coated raceways (coating the balls/rollers is usually not desirable). The DFL helps with overall performance and endurance and is especially important during the early portion of the service life when sufficient transfer of lubricant from the cage to the raceways has not yet occurred. Though lubricant creep is not a concern for bearings utilizing solid lubrication, labyrinth seals or shielded bearings are often employed to keep any solid debris generated by the bearing contained within the bearing and to keep any external contamination from entering the bearing. The volume of debris generated over the life of the bearing can be considerable and affect bearing performance when the required life is long, so this may need to be accommodated in the design.

Solid lubricants bound in sacrificial cages are released and transferred to the raceways via the balls during operation of the bearing. Run-in is essential to the proper performance of solid-lubricated bearings.

A.2.3.1.5 Ball Bearing Implementation

Forces applied to the bearings during installation should never be transferred through the bearing balls.

Balls can exhibit many different types of flaws. Balls used in critical ball bearing applications should be subjected to 100 percent inspection.

Because solid lubricant performance depends strongly on the operating environment, it is strongly recommended that solid-lubricated bearings in critical applications be run in the expected operating environment of the bearing.

A.2.3.2 Roller and Needle Bearings

Roller bearings use a cylindrical roller instead of a ball. Roller bearings with a high length-to-diameter ratio are generally termed “needle bearings” and tend to be the most common type of roller bearing used in aerospace applications because of their compact size.

Because of the cylindrical rolling elements, roller bearings can only take load in axes perpendicular to the roller axis; however, the roller orientation and geometry can be arranged to allow loads to be taken in a variety of bearing axes. Radial roller bearings take radial loads,

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thrust roller bearings take thrust loads, and tapered roller bearings allow loads to be reacted in both axial and radial directions. The rollers in the bearing can be straight-sided or crowned. Crowning the rollers reduces the edge contact stresses.

Radial roller bearings can be provided with or without an inner ring. A compact type of needle bearing without an inner ring is known as a “drawn cup” bearing and relies on an appropriately hardened shaft as the inner raceway with rollers that are captive within the outer ring. Roller bearings can also be supplied in the form of only the cage and roller assembly using the shaft and housing as the raceways. This gives the smallest possible envelope for the bearing but generally requires special materials and treatment of the bearing surfaces.

In general, with the exception of preload practices, the ball bearing best practices apply to roller bearings. In roller bearings, preload can be achieved by utilizing a pair of thrust roller bearings but such a system will have very low radial stiffness. Preloading of a radial thrust bearing can only be accomplished through control of the degree of interference with the mounting shaft or housing. As a result, roller bearings are often implemented with no preload.

A.2.3.3 Spherical Plain Bearings

Spherical plain bearings (also called monoballs) consist of a ball in an outer ring that allows rotational motion in all three axes, though the motion in two axes is limited by the geometry of the ball’s through hole and the outer ring. These bearings are used in low-speed applications in which there is a tilt with respect to the primary axis of rotation or applications that require released rotational degrees of freedom.

Rod-end bearings are a type of spherical bearing with a typically low-precision spherical bearing, limited race width, and an integral threaded interface. Rod-end bearings are not intended to react axial loads.

Spherical plain bearing applications are usually constrained by the energy dissipated at the sliding interface between the ball and the raceway, typically quantified with a “PV” factor that is the product of the mean projected contact pressure and the sliding velocity. Preload can only be applied via an oversized ball or by swaging or staking the outer ring. The swaging/staking method typically results in high variability as the process is difficult to control precisely. As a result, spherical bearings are often employed with no preload in applications where low precision is acceptable.

A.2.3.4 Plain Bearings

Plain bearings, sometimes called bushings, are typically metallic or polymeric sleeves or washers that facilitate sliding motion. Plain bearings are often used in applications where obtaining the lowest friction is not important. Depending on the material chosen, they can be self-lubricating or can be used with greases. If grease is used, appropriate anti-migration features should be employed. Smooth finishes are desirable for the sliding surfaces in plain bearing applications; average roughnesses in the range of 0.1 to 0.3 μm (4 to 12 μin) are recommended. Plain bearings should be interference fitted into or fastened onto their housings to avoid motion between the

bearing and the housing. Attempting to use a loose fit between plain bearing outside diameter and housing bore to provide a measure of redundancy is not recommended. For shaft diameters over approximately 6 mm, diametral clearances between plain bearing bore and shaft outside diameter of approximately 0.0010 to 0.0015 millimeter per millimeter of shaft diameter are typically employed.

A.2.3.5 Linear Bearings

Linear bearings are bearings designed to provide free motion in one direction. Linear bearing systems often consist of multiple linear bearings supporting a carriage that moves on one or more rails that constrain the motion to a single linear axis. Linear bearings can take many forms and may utilize plain bearings, rollers, stationary balls, or recirculating balls to facilitate motion. Where rolling elements are used, the normal Hertzian contact stress requirements apply. Linear bearings that incorporate rolling elements should be preloaded like other rolling element bearings; and depending on the type of linear bearing chosen, a choice between hard and soft preloads may be available.

Linear bearings are susceptible to binding or stick-slip motion when subjected to moments or when poor geometric characteristics exist, even if well lubricated. See requirement 4.9.13 in this Standard. Pairs of linear bearings can also be susceptible to problems if the parallelism of the rails is not tightly controlled. As an alternative, consider statically determinate arrangements in which one rail is allowed to float.

A.2.4 Motors

Motors frequently present challenges for mechanism engineers. Much of this section was adapted from Sevilla (2010).

A.2.4.1 DC Motor Types and Selection

Electric motors come in both rotary and linear form. Rotary motors used in space actuators are typically one of the following types:

- Brush direct current (DC) motors.
- Electronically commutated brushless DC motors, usually referred to simply as “brushless DC (BLDC) motors.
- Stepper motors.

Motor applications are defined by the performance requirements on the motor and can be roughly divided into three categories:

- Deployment Applications.
- Incremental Positioning Applications.
- Servo Applications.

Selection of the appropriate motor type requires matching the characteristics of the motor to the application for which the motor is intended.

A.2.4.1.1 Motor Characteristics

A.2.4.1.1.1 Brush DC Motor Characteristics

On the positive side, brushed motors can produce high output torques, have low cabling mass, and are self-commutating, demanding only a simple electrical interface that usually only requires a DC voltage to operate. However, they have several drawbacks.

Brushed motors are limited-life items and can be considered consumables. They exhibit wear and performance changes over their operating life, which can be sensitive to small changes in the operating parameters or environment. As a result, operating cycles have to be monitored during ground testing, and motor run-in has to be carefully controlled to avoid extreme variations in operating life. Highly varying or intermittent load requirements can result in brush and commutator degradation and reduced life of the motor. Proper material selection is critical; brushes designed for use in terrestrial environments are unsuitable for space applications because in a moisture-free environment typical terrestrial brush materials become abrasive and quickly wear away the brush or the commutator. Even very low levels of moisture in the environment can improve these wear characteristics, so misleading ground test results are a danger. Metallic brushes are less sensitive to moisture but are far more sensitive to cold temperatures because of the necessity for wet lubricants to extend life of the brushes. Success has been achieved in vacuum applications using brushes comprised of a combination of copper or silver, molybdenum disulfide, and graphite in appropriate amounts; the molybdenum disulfide reduces wear during operation in vacuum while the graphite provides wear reduction during atmospheric testing.

Brushed motor rotors are poor at dissipating heat so their temperature can exceed the allowable temperatures of the polymers and solders used, particularly in vacuum applications, and often need overheating protection algorithms that result in additional cost and development risk. “Simple” brushed motor implementations may get complicated as a result. In addition, brushed motors have a history of poor performance in cold operating conditions. This stems from the complicated and poorly understood temperature-dependent interactions between brushes and commutators that lead to good “commutator film” formation. Similarly, brushed motors are more susceptible to damage from thermal cycling than brushless motors because brushed motors contain more potting and soldered or welded terminations that, due to differences in coefficient of thermal expansion, can become highly stressed by temperature changes.

Brushed motors require a reversal of polarity to reverse motor direction which may necessitate a circuit design that provides protection against accidental polarity reversal. Input current on brushed motors varies with the rotor position and the number of commutator bars in contact with the brushes. Brushed motors are also sources of significant electromagnetic emissions and can cause electromagnetic interference (EMI)/electromagnetic compatibility (EMC) problems.

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Though they have significant drawbacks, when the load requirements are well-defined and consistent and a comprehensive qualification program has been implemented, brushed motors have provided reliable service in space flight operations. When assessing heritage of an existing brushed motor, it should be recognized that nearly any change from the successful heritage performance requirements can represent a significant development risk for the new application. Brush wear can vary with the speed, load, thermal environment, pressure, and atmospheric chemistry.

See NASA Preferred Reliability Practice No. PD-ED-1229, Selection of Electric Motors for Aerospace Applications, for more information on the use, characteristics, idiosyncracies, and limitations of brushed motors.

A.2.4.1.1.2 Electronically Commutated Brushless DC Motor Characteristics

Electronically commutated BLDC motors can provide the same torque capability as brush motors without the drawbacks of brush motors. BLDC motors will exhibit unpowered holding torque unless designed to produce zero cogging torque. They are relatively easy to size and test for torque margin and exhibit consistent changes in performance over variations in temperature or voltage. They also exhibit very long life, limited only by the life of the bearings or bearing lubricants.

The main drawback to BLDC motors is that they require a feedback encoder and expensive drive electronics. New electronic designs carry development and cost risks and require torque-versus-current relationships to be established across the operating temperature range to characterize the commutation and verify adequate motor torque output over the entire rotary range. The torque performance of a BLDC motor is also sensitive to the switching performance of the commutation. Systems of BLDC motors can require a large amount of cabling if a centralized set of drive electronics is used (four to five wires for the motor and five to seven wires for the encoder) and introduce the possibility of EMI through the cables. Distributed drive electronics present a simple uncoupled interface to the flight system and eliminate “surprises” when transitioning from component to system testing but can have packaging challenges. Once a BLDC motor and its drive electronics have been fully developed, they provide robust, consistent performance through their mission life.

Qualification testing of BLDC motor designs has to include the “torque profile” characterization test to identify “torque holes” where commutation electronics switch motor current from winding to winding, and to verify proper end-to-end design of windings, magnetics, the position encoder, and the commutation switching logic. In such a test, motor output torque is measured quasi-statically by restraining the output shaft and rotating the output over 360 degrees in both directions. The torque at the minimum torque positions in a well-designed motor should not have a magnitude less than 80 percent of the torque at the maximum torque positions.

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A.2.4.1.1.3 Stepper Motor Characteristics

Stepper motors provide precise incremental motion, unpowered holding torque, and good heat dissipation. Compared to BLDC motors, stepper motors require fewer cables (three to five wires, more if an encoder is used) and, like BLDC motors, they have a long life that is limited by the life of the bearings and the bearing lubricants.

On the negative side, stepper motors can be difficult to size properly. Their performance is highly dependent on the drive electronics and, unlike brushed and BLDC motors, the torsional stiffness of the drivetrain and the moment of inertia being driven. Stepper motors do not have a single torque-speed curve but instead have pull-in and pull-out torque-step rate curves; and they do not exhibit linear input current-to-torque curves or linear speed-to-torque curves. Performance also depends highly on the shape and timing of the electrical drive pulse. The output motion of a stepper motor is not necessarily synchronized to the input pulse and requires additional encoder feedback to ensure step integrity (the ability to achieve the number of steps commanded without loss or gain of steps). Because a stepper motor's torque margin has to be calculated differently than other systems and because this calculation can be more difficult to perform, stepper motors are often incorrectly sized for the application.

Stepper motors always operate at maximum input power, regardless of resistive torque, which can make anomaly investigations difficult. Flight-representative drive electronics are crucial for accurate results during stepper motor qualification testing, creating a significant problem if they are not available in time to support the testing. This is particularly true if microstepping is implemented. Since stepper motors are most commonly driven open-loop, unanticipated motor performance due to motor controller issues is the most common cause of ground test and flight anomalies. Stepper motor applications may require a basis for initializing prior to counting steps, which is usually implemented by use of a reference signal from an encoder, resolver, or other "home position" sensor.

Stepper motors are used extensively in spacecraft and instrument systems and provide reliable performance when sized properly and verified by a comprehensive test program. The design of stepper motor applications needs to consider the inherent damping provided by the winding configuration. Two-phase or four-phase wound stepper motors behave similarly to very lightly damped spring-mass systems. The three-phase wye winding configuration provides increased damping due to back-electromotive force (EMF) phasing producing a dissipative current in two of the three windings. In addition, this effect can be utilized to assist with unpowered damping torque if the drive electronics can short the winding leads together.

When assessing heritage stepper motors for a new application, be on the lookout for changes in the ratio of torque that accelerates a moment of inertia to torque that overcomes friction and changes in the design of the drive electronics (e.g., a different stepper motor controller from that used on the heritage motor).

See the 37th Aerospace Mechanisms Symposium paper “Establishing Adequate Performance Margin for Space Flight Stepper Motor Mechanisms” by David Marks for more information on the use, characteristics, idiosyncracies, and limitations of stepper motors.

A.2.4.1.2 Motor Application Classes

A.2.4.1.2.1 Deployment Applications

In deployment applications, or any application in which a motor must move a mass from one point to another and perhaps back again, precise control of speed is usually not necessary; and speed is usually free to vary with torque, time, and/or distance. The motion is constrained through the use of switches, control electronics, and mechanical hardstops. Common implementations are deployments, restraint and release devices, and instrument covers. This application is typically the least demanding with respect to flight system cabling and control.

While a brushed motor can usually meet the performance requirements for the deployment application class, BLDC motors can do this job as well and are commonly implemented because of their long life and consistent performance, particularly when drive electronics are present for other BLDC motors already.

Stepper motors are sometimes used in deployment application classes when stepper drive electronics already exist within the flight system. However, sizing stepper motors for such an application class can be difficult; and their performance is highly dependent on the electrical drive pulse and the driven load. Stepper motors in this application class should be able to supply output torque above what is necessary to accelerate the driven inertia (and be tested to verify that they can). Use of flight stepper motor drive electronics in component testing is critically important to prevent anomalies during flight system integration testing.

A.2.4.1.2.2 Incremental Positioning Applications

Incremental positioning applications involve positioning the motor output shaft at discrete positions that are not at the bounds of the range of travel. One example of this application category is an instrument scan mirror that requires dwell time at numerous positional locations. Not only is precise rotational positioning required, but also controlled angular velocities and accelerations are typically necessary as well. Unpowered holding torque is typically required at each positional state.

Incremental positioning applications are most often filled by the lowest overall cost solution: stepper motors. This is because stepper motors naturally produce incremental rotary motion, can be driven open-loop which eliminates the need for a position encoder, and utilize stepper motor controllers which are simpler than servo controllers.

A.2.4.1.2.3 Servo Applications

Servo applications are applications that require precise output control that can only be obtained with a closed-loop electrical control system. Such outputs may include target positioning, acceleration/deceleration rates, load compensation, odometry, output torque, input current control, input current limiting, fault detection, and fault response. This class of application always requires positional feedback, which may or may not be accomplished with the device used for motor commutation. Examples of servo applications include actuation of a robotic appendage and high-precision pointing devices. Servo applications require the most development effort and the most flight system resources.

Specialty motors are sometimes necessary for servo applications, such as motors with ironless rotors for when exceptional acceleration is required. BLDC motors are typically chosen for servomechanism applications. Brush motors can also be employed if conditions allow, but development and mission risk will be higher.

When closed loop velocity control is required, stepper motors can use pulse train control to achieve the desired velocity as long as it is within the pull-in torque range. BLDC motors can use either a tachometer to directly produce a velocity feedback signal, or a position encoder and timer to derive a velocity feedback signal.

A.2.4.1.3 Motor Selection Guidelines

Motor selection for an application is typically based on a trade study involving a survey of similar heritage applications. Consideration should be given to the full system development costs and risks in the trade, rather than just the procurement costs. The following evaluations are recommended when selecting a motor for use on a flight program:

- a. Identify the motor application class. Though iteration may be required as part of the system engineering process to trade the capabilities and costs of different motor application classes when more than one approach can be taken; once that process has been completed, it is straightforward to identify the appropriate motor application class from section A.2.4.1.2.

Stepper motors are not recommended for use in applications requiring minimal torque ripple. “Zero-cogging” motor designs and “cogging compensation” commutation strategies can be used in a BLDC motor to minimize torque ripple.

Increasing care is warranted as motor output torques approach the inch-ounce range and below. As output torques get into this regime, they become subject to significant torque margin erosion from effects that do not scale with a reduction of motor size, such as particulate contamination from foreign objects and debris, and increased friction or viscosity at lower operating temperatures.

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b. Examine the environments and conditions under which the motor will operate. The motor operating environments and conditions include pressure conditions, atmospheric chemistry, thermal conditions, the motor torque profile, and the life required.

Brushed motors have been used successfully in a number of spacecraft flight systems but have difficulty functioning consistently and reliably in a vacuum; therefore, the brushed motor option should be discarded if a long operating life in vacuum is required. Proper simulation of the mission environment is essential for the brush-to-commutator sliding interface, as a falsely successful life test can be obtained with extremely low levels of moisture in the test environment. When used in Mars environmental conditions, brush motors have demonstrated excellent but variable wear life. Brushed motors also require accurate replication of expected torque cycles since brush wear (and potential damage to the commutator) is highly dependent on the magnitude and duration of the electrical input as well as the operating environment.

c. Evaluate development costs and risks for each motor type. Developmental cost (and cost risk) depends on the motor type selected and may be substantial when considering the potential failure modes and characterization testing necessary for each type of motor. A commercial motor design will have a relatively low procurement cost. Costs will increase when modifying a commercial design and certifying it for flight, and the more changes, the higher the costs. Custom designs cost still more. And, of course, costs can skyrocket if the program suffers development, qualification, or acceptance test failures. This is one of the major problems with using brush motors, since brush motors are generally designed for terrestrial applications that are very different than space applications. The development risks of brushed motor applications should not be underestimated. The most likely risks to be realized when implementing brushed motors are unexplained excessive brush wear and inadequate torque margin as a result of rotor overheating.

Procurement costs for a commercial brushed motor can be quite low; flight system costs are higher for brushless motors because of the need for drive electronics and encoders. In servo applications, drive electronics and motor feedback devices are required regardless of motor type; so the motor type selected for the application may be dominated by the other costs of developing a servo system.

BLDC motors require a rotor position feedback encoder regardless of application. However, non-servo applications generally do not require as precise (or complex) an encoder as most servo applications do, potentially reducing cost.

d. Evaluate mission risk for each motor type. It is necessary to evaluate mission risk in addition to development risk because some types of performance anomalies will not be detected during flight system integration and only discovered after launch. Most of the knowledge about the performance of a motor is acquired at the motor level (e.g., in dynamometer testing) and at the mechanism or instrument level testing. Often, little knowledge about the motor is acquired during flight system tests other than electrical compatibility and simple confirmation of function.

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In general, mission risk when implementing brushed motors should be considered moderate to high due to the inherent variability in brushed motor life, the difficulty in accurately characterizing and predicting in-flight performance, and the extreme sensitivity in brushed motor life to the presence of moisture during testing.

Stepper motors cannot provide in-flight monitoring of motor health by themselves. The lack of direct in-flight knowledge of output motion of a stepper motor is a risk. This problem (for any motor type) can be mitigated in a servo application when a motor encoder is implemented or when secondary telemetry exists. All BLDC and stepper motors have to be qualified using flight-representative drive electronics. The unavailability of flight electronics at the time of motor qualification or acceptance testing has repeatedly caused in-flight performance problems. Implementation of BLDC motors can ensure robust performance and can provide unambiguous in-flight health monitoring. When each BLDC motor is controlled by individually mounted drive electronics, nearly all sources of mission risk involving flight system compatibility are eliminated because the motor and electronics configuration remains unchanged throughout motor qualification, mechanism testing, instrument testing, and flight.

Additional information may be found in Preferred Reliability Practice PD-ED-1229, Selection of Electric Motors for Aerospace Applications (https://extapps.ksc.nasa.gov/Reliability/Documents/Preferred_Practices/1229.pdf).

A.2.4.2 Stepper Motor Performance Analysis

A stepper motor rotor behaves like a rotary magnetic spring-mass-damper system subject to forced vibration from the electrical drive pulses. Its stability is influenced by a wide variety of motor, driver and load parameters, and can be evaluated with a Monte Carlo simulation of the combinations of parameter values or with a worst-on-worst deterministic analysis. To envelope all conditions which affect output torque and synchronicity, the following parameters and their ranges should be considered in the analysis:

- Motor inductance and its variations due to manufacturing tolerances.
- Motor resistance and its variations due to manufacturing tolerances and thermal conditions.
- Input voltage and current characteristics (including variations in pulse shape, timing, and operational states such as tracking and slewing).
- Step angle size and its step-to-step variations.
- Unpowered detent torque and its step-to-step variations due to manufacturing tolerances.
- Voltage constant and its variations due to manufacturing tolerances and thermal conditions.

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- Powered holding torque and its variations due to manufacturing tolerances, thermal conditions, and current tolerances.
- Rotor damping and its variations due to manufacturing tolerances, thermal conditions, and changes over life.
- Viscous damping of motor bearings, lubricant, and magnetic structure, and their variations due to manufacturing tolerances, thermal conditions, and aging.
- Variations in rotor friction (from motor bearings) due to manufacturing tolerances, thermal conditions, and changes over life.
- Rotor inertia and its variations due to manufacturing tolerances.
- Motor rotor-to-gear train deadband and its variations due to manufacturing tolerances and thermal conditions.
- Geartrain inertia, stiffness, and deadband, and their variations due to manufacturing tolerances.
- Geartrain friction and damping, and their variations due to thermal conditions and aging.
- Geartrain torque and its variations due to transmission error or strain wave gear drive two-cycle torque variation.
- Load inertia and its variations due to manufacturing tolerances.
- Modal properties of the driven inertia.
- Stop stiffness and its variations due to manufacturing tolerances.
- Modal properties of the base, including base motion disturbances.
- Test conditions, test equipment effects, and “one g” effects.

Where the holding capability of a stepper motor due to its inherent magnetic detent is used to hold a load in position, the holding torque margin should be evaluated using the unpowered detent torque at each discrete rotor position for both the clockwise and counterclockwise direction. If the holding capability of the unpowered stepper motor is marginal, a trickle current may be applied to the appropriate motor windings to increase the holding torque. The trickle current should be disconnected when the motor is energized to reduce its effect on running torque.

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A.2.4.3 Torque Profile Test for Electronically Commutated Motors and Drive Electronics

In a torque profile test, the electronically commutated motor without any gearheads or gearboxes attached has its stall torque and detent torque mapped as a function of rotor position around a full 360 degrees of rotation. The stall torque is quasi-statically measured while the rotor is slowly rotated (less than one revolution per minute) in the commanded direction with nominal operating voltage applied to the motor drive electronics. This test is performed in both commanded directions of rotation to demonstrate that there are no anomalous torque conditions such as excessively low torque valleys or non-symmetrical torque variations. In the event that heating of the stator poses a significant performance variation, stopping the test to allow cooling is acceptable. The test is performed with a constant input current, operating the motor at a torque level substantially larger than the motor detent torque.

A.2.5 Springs

Springs are a common mechanism component and come in a wide variety of forms. Examples include helical extension and compression springs, torsion springs, Belleville washers, disc springs, diaphragm springs, wave washers, wave springs, leaf springs, urethane springs, gas springs, constant-force springs, and garter springs. Their simplicity can give a false impression of reliability. To efficiently store energy, spring materials must have high strengths. Consequently, spring materials tend to have low fracture toughness, making it crucial to adhere to best practices for spring design such as those listed in the Engineering Guide to Spring Design available from <https://www.asbg.com/engineering-guide-to-spring-design.aspx> and the appendix of the now-cancelled MIL-STD-29A, Spring, Mechanical, Drawing Requirements for. Note that ASME Y14.13M should NOT be considered a replacement for MIL-STD-29A as a source of spring design information because the scope in ASME Y14.13M is limited to information necessary for the creation of spring drawings and does not contain design information.

Compression springs are generally preferred over tension and torsion springs because they will usually retain some measure of performance after fracturing while the others often cannot. However, to depend on this type of failure tolerance, it has to be ensured that broken halves of coil springs cannot thread into one another after breaking. This requires the wire diameter of the spring to be larger than the spacing between the coils, or the ends of the spring to be fixed to prevent rotation, as well as lateral restraint of the spring (like with a concentric rod within the spring or a concentric tube around the spring, which also prevents buckling under nominal operation and environments). Helical compression springs wound from wire should have closed and ground end coils for interfaces whenever the spring's wire diameter, spring index, and spring rate allow. The interfaces and restraint of leaf springs should be designed to avoid stress concentrations, for example, by rounding sharp corners or keeping mounting holes away from highly stressed areas.

Extension springs can often be replaced with compression spring alternatives; in commercial practice, this takes the form of what are termed "Drawbar Springs." If extension springs must be

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used, it is recommended they be specified with close-wound coned ends and swivel hoops. This arrangement increases the fatigue life compared to an extension spring with hook or loop ends.

Springs should be designed or selected using the factors of safety (FS) shown in Table 4, Factors of Safety for Springs, at the maximum operating stress.

Table 4—Factors of Safety for Springs

Spring Application	Yield FS	Ultimate FS
Safety-critical springs	1.65	2.0
Mission-critical springs	1.5	1.65

If torsion springs are used, they should be under load in the direction of winding (i.e., the load should tend to further wind up the coil) at all times. Temperature affects the stiffness of springs and should always be considered in their design or selection.

Springs often see a large number of small cycles, either through their function in the mechanism or due to environmental cycling such as temperature cycles or vibration. These cycles can accumulate and lead to unexpected fatigue failure, so all springs should be analyzed or better yet tested against fatigue failure.

A.2.5.1 Avoiding Stress Relaxation in Springs

Stress relaxation is defined as the loss of load or available deflection that occurs when a spring is held in a preloaded state and/or cycled. Spring applications sensitive to stress relaxation must employ springs engineered to minimize or eliminate this effect.

The primary factors influencing the degree of stress relaxation experienced by a spring are the spring temperature, the spring material, the residual stress state of the spring in its free state, and the magnitude of the stress in the deflected state.

Stress relaxation is an exponential function of both temperature and stress. Stress relaxation increases with increasing spring temperature. To minimize the effects of stress relaxation, the spring temperature should be kept as low as possible. However, stress relaxation is not solely a high-temperature phenomenon; stress relaxation can occur even at cryogenic temperatures if the stress is high enough. Stress relaxation increases with increasing spring stress; to avoid stress relaxation, the working stress in the spring should be reduced as much as possible. A maximum stress of 50 percent of the spring material's elastic limit, not yield strength, is recommended.

Spring materials show different susceptibilities to stress relaxation. The two most commonly employed materials for springs in applications sensitive to stress relaxation are 17-7PH and Elgiloy®, with 17-7PH being least susceptible. In high-temperature applications, the coarser grain size of the material, the less susceptible it is to stress relaxation.

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Residual stresses resulting from spring manufacture are detrimental to relaxation resistance. Shot peening, a process that intentionally imparts residual stress and is commonly used to increase fatigue resistance of springs, should not be performed on springs in applications sensitive to stress relaxation. After spring manufacture, stress relief at the highest practical temperature is recommended. "Presetting," in which springs are intentionally yielded by compressing to their solid height, and "heat setting," in which springs are simultaneously exposed to heat and stress, are beneficial. The effect of these processes is to reduce the magnitude of a residual stress or establish a different state of residual stress that increases resistance to stress relaxation.

A.2.6 Gears

All gears should comply with the standards of the American Gear Manufacturers Association (AGMA).

Where pitchline velocities are low and cycles are high, hunting tooth gear ratios should be used to distribute wear. Complete hunting occurs when every tooth on a gear mates with every tooth on the other gear. This occurs when the numbers of teeth have no common factors higher than one. Partial hunting occurs when any tooth on one gear mates with N teeth on the other gear, where N is the number of teeth divided by the highest common factor. The gear tooth numbers should be selected, within the limits of the gear ratio requirement, to maximize the number of revolutions before the same two teeth mesh.

Select gear ratios, diametral pitches, pressure angles, profile shift coefficients, etc., to optimize performance and to avoid undercuts and narrow top lands on gear teeth. Maximum resistance to gear wear and pitting is obtained by profile shifting to balance and minimize the specific sliding ratio (the ratio of gear tooth sliding velocity to rolling velocity) over a tooth mesh cycle. This strategy will also increase gear train efficiency and lubricant life, especially dry film lubricant life. Where pitchline velocities are high, maximum resistance to scuffing is obtained by profile shifting to minimize the flash temperature peaks along the line of action. This strategy will also increase lubricant effectiveness and life. Maximum bending resistance is obtained by profile shifting until the ratio of the bending strength geometry factors equals the ratio of the allowable bending stresses. This strategy will increase gear tooth bending fatigue life. For gear sets with pinions having at least 25 teeth, the profile shifts for balanced specific sliding, flash temperature and bending fatigue life are all nearly the same.

Spur gear contact ratios should be greater than 1.4 for power transmission gearing. Whenever possible, straddle mount gears between support bearings. Avoid mounting gears on cantilevered shafts whose deflection results in a non-uniform load distribution across the face of the teeth. Gear tooth contact patterns should be checked upon initial assembly to establish that the pattern is well-centered over the tooth flank and that edge loading is not present.

Aluminum gears are not recommended except in light-duty, limited-life applications where tooth wear and their thermal expansion can be accommodated, and where compatibility with the selected lubricant can be established. Aluminum gears may be hard anodized to improve wear and galling resistance, provided the contact stresses will not cause the hard anodize to fail.

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Precision gear sets, such as those used in fine-pointing mechanisms, should use anti-backlash gearing. For critical applications, AGMA quality level 12 or better should be considered. Where gears are required to be matched sets, the gears should be identified and marked as such.

A.2.6.1 Strain Wave Gear Drives

Strain wave gear drives should never be assembled or operated in a dedoidal condition where the flexspline is not concentrically engaged with the circular spline. Strain wave gear drives are known to have transmissibility errors which create a cyclic output disturbance torque whose frequency is a multiple of their input shaft speeds, even when driven at perfectly constant input shaft speeds. This disturbance torque can drive a torsional resonance, especially when the input is provided by a stepper motor. Stepper motor-driven mechanisms containing strain wave gear drives should avoid stepping at rates that will couple with the load resonances. For strain wave gear drives driven directly by stepper motors, the disturbance torque frequency at the output is defined as

$$f_{\text{disturbance torque}} = \frac{\text{Step Rate} \cdot \text{Step Angle} \cdot N}{360} \text{ Hz} \quad (\text{Equation A-1})$$

where $f_{\text{disturbance torque}}$ is the frequency of the strain wave gear drive output disturbance torque, *Step Rate* is the step rate in electrical pulses per second, *Step Angle* is the angle a stepper motor rotor rotates through during a full step (in degrees), and *N* is the harmonic of interest, most usually 2, 4, or 9. Since the second harmonic is the greatest disturbance, many times it is sufficient to analyze for $N = 2$ only.

A.2.7 Fastening

Fastening is an important aspect of virtually any mechanism that is assembled. Poor fastening practices can have an adverse impact on mechanism performance or prevent operation outright. Fastening requirements and practices are out of the scope of this document but NASA-RP-1228, Fastener Design Manual, and NASA-STD-5020, Requirements for Threaded Fastening Systems in Spaceflight Hardware, are excellent resources. All threaded fasteners in space mechanisms should adhere to the requirements and practices in NASA-STD-5020.

A.2.8 Locking Quick Release Pins

A.2.8.1 Locking Quick Release Pin History

Quick release pins come in locking and non-locking styles. Non-locking quick release pins, also called detent pins, are not discussed here. Locking quick release pins, also known as PIP pins, have built-in fast-acting retention and release mechanisms. They come in a variety of forms: single-acting (push to release) or double acting (push or pull to release) with a wide range of handle designs and other features to choose from; but generally they all utilize a spring-loaded plunger and one or more retaining balls, which are captured in clearance holes (called sockets) by swaging or staking the shank material around the hole. In the default plunger position, the

plunger prevents the balls from translating radially inward when external loads are applied to the retaining balls. In the pushed/pulled plunger positions, the plunger allows the retaining balls to translate radially inward when external loads are applied to the retaining ball. When translating from the default position to the pushed/pulled position, the plunger acts like a cam and forces its follower—the retaining balls—to translate radially outward but does not preload the retaining balls against the swages/stakes. Figure 5, Diagram of a Typical Double-Acting Locking Quick Release Pin, depicts a cross-section of a representative double-acting locking quick release pin that illustrates the principle.

Locking quick release pins were originally designed for use in non-critical, remove-before-flight ground applications on aircraft. Their speed and convenience have led to their adoption in spaceflight applications involving crew interfaces. Unfortunately, these applications are far beyond the original design applications and have resulted in a history of failure.

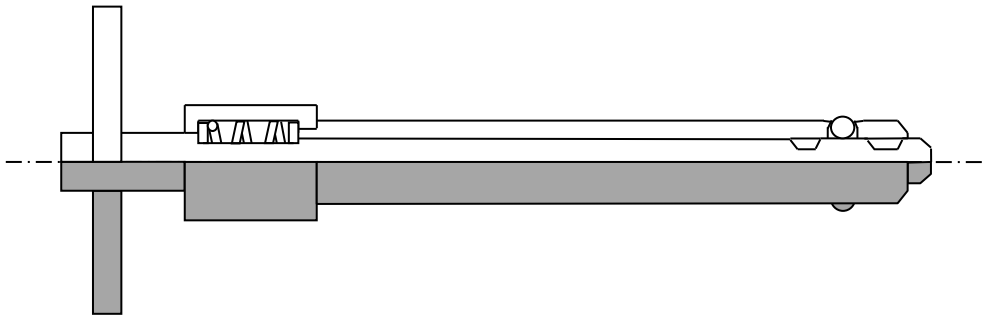


Figure 5— Diagram of a Typical Double-Acting Locking Quick Release Pin

A.2.8.2 Locking Quick Release Pin Failures in Spaceflight

Though several documented inadvertent releases of locking quick release pins were noted, no serious documented failures occurred in the space program until 1990. In that year, NASA began environmental testing of the extravehicular activity (EVA) Development Flight Experiments payload, which contained locking quick release pins. During vibration testing, several retaining balls vibrated out of their sockets; during cold temperature vacuum testing, the wet lubricant in the pins froze and seized the plungers.

After these failures, several design changes were proposed for locking quick release pins to make them more reliable for space use:

- Use of four retaining balls instead of two to improve retention in the event of a single ball release.
- Double action.
- PTFE-coated tethers.

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- Welded handles and tether rings.
- Addition of dry film lubricants.
- Addition of a hitch pin clip.

More detail is provided in Skyles (1994).

The addition of hitch pin clips, which are self-retained shear pins placed through cross holes in both the plunger and tubular shank of the locking quick release pin to prevent inadvertent actuation of the pin and act as a secondary means of pin retention in the event of ball loss, would turn out to be controversial. Hitch pin clips proved to be difficult to install with gloved hands and presented snag hazards during EVAs. Several instances of inadvertently pulled hitch pin clips were encountered on Hubble Space Telescope (HST) servicing missions 3A and 3B.

In 1994, these and other problems prompted the Space Shuttle Safety Review Panel to establish a policy in which locking quick release pins had to be treated as mechanisms in their own right, requiring the same engineering rigor and review practices as other mechanisms. Continuing problems led to the policy being revised in 2000 to prohibit locking quick release pins from being used in zero-fault-tolerant applications.

In 2001, a set of pins that had incorporated these improvements exhibited some other problems during preparation for HST servicing mission 3B. Pins with a two-piece welded spindle/button construction were found to be prone to fracture. Such pins that were used only in contingency scenarios were launched with hitch pin clips; others with planned uses were flown with a system of hook-and-loop flags to secure the pins in the event of the loss of the retaining balls.

Vibration testing of the same pins uncovered a defect that caused the retaining balls to stick in the protruded position. The ball staking and shank boring processes were inconsistent, resulting in the protrusion of the balls to vary widely. In addition, the retention balls had a rough surface finish. The combination allowed side loads to jam the balls into the stakes under vibration loads. A redesign being infeasible at that point, an acceptance test procedure was developed to screen for this failure mode in an attempt to reduce risk. Nine of the forty-eight pins used in the mission, which came from multiple vendors, failed the test and five more barely passed.

Afterward, a new vendor was contracted to build forty-six custom pins for the project. A series of meetings and site visits by HST and NASA personnel ensured that the new hardware was built and tested to the highest possible standards. Design enhancements included a single-piece spindle/button design, an H-1025 heat treat of the 15-5 PH steel used in the pins for greater ductility, molybdenum disulfide lubrication, a drive-out feature, specially designed EVA-friendly handles, and hitch pin clips. The manufacturer also performed additional verification, including extra dimensional inspections and staking strength tests on 100 percent of the pins. All subsequent qualification and acceptance testing (thermal, random vibration, and “stick-ball” tests) was successful.

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In 2004, a double-acting locking quick release pin on the ISS mobile transporter rail experienced a ductile fracture prior to flight due to an overload from an inadvertent impact. The investigation revealed a deficient design of the spring housing that allowed tolerance stack-ups between an external groove and an internal thread, combined with poor process control, to produce an unacceptably thin wall. The spring housing and other parts of the pin were redesigned by the prime contractor and the pin manufacturer to address the deficiencies, and random vibration testing was added to the part specification, creating a new part number approved for use in certain space applications. One hundred and twenty-four discrepant pins were replaced on various parts of the ISS.

A.2.8.3 Current Locking Quick Release Pin Best Practices

Properly designed and constructed pins can be used successfully in non-critical shear applications. Locking quick release pins are essentially shear pins and should not be used in critical applications where the retaining balls would have to react significant tensile loads along the axis of the pin.

Locking quick release pins are not recommended for applications that control hazards. Even with the most up-to-date pin designs, three failure modes remain and will need to be addressed when considering their use, especially in safety-critical applications:

- a. Loss of retaining balls.
- b. Failure of the shank.
- c. Failure of the spring housing.

Due to the arrangement of the balls and the subsequent need to use swaging or staking around them, loss of retaining balls is nearly impossible to eliminate with design and manufacturing. The usual control is to provide a back-up pin retention feature such as a hitch pin clip or some other means of ensuring that the pin will not fall out under environmental conditions if the balls disappear. Failures of the shank and spring housing can be eliminated if the locking quick release pin vendors adhere to best practices in design and manufacturing. A continuing risk is that locking quick release pin manufacturers can change their internal designs at any time without notification, so the detailed design of the mechanism and the change history should be investigated thoroughly and frozen by contract prior to the use of any pin. Lastly, locking quick release pin structural margins must be assessed in each application; due to the internal features of the locking quick-release pin, dependable allowables are not always available, are dependent upon application, and will not cover situations such as inadvertent contact loads.

A.2.9 Redundant Rotating Surfaces

Providing redundant rotating surfaces can be an efficient way to add fault tolerance to cylindrical, revolute, and spherical joints. Redundant rotating surfaces exist when free motion remains after a seizure between one pair of surfaces that rotate with respect to one another. Common-cause failures such as axial clamping of the joint members should be considered when determining whether redundancy truly exists.

It is not recommended to add redundant rotating surfaces to joints utilizing rolling element bearings because such bearings are usually designed to be securely seated in a housing or on a shaft and adding radial clearance to the joint can decrease the reliability of the rolling element bearing.

It is generally better to improve the reliability of a rolling element bearing by increasing its size and lowering its contact stress than it is to implement series redundant rolling element bearings in joints. In comparison, series redundant rolling element bearings have reduced stiffness, less accuracy, larger mass and volume, and reduced thermal conductivity from inner to outer ring. Larger bearings will exhibit higher drag torques, but this is readily addressed by using higher torque motors. Do not be stingy with motor torque; in many instances, high torque margin can be just as effective as redundant rotating surfaces.

A.2.10 Inspection

Inspection of both constituent parts and assembled mechanisms is an important part of the fabrication and testing of flight mechanisms. Parts made of materials known to be susceptible to material defects—for example, castings, weldments, fiber composites, laminates, and honeycombs—should be carefully inspected prior to acceptance. Parts that represent single-point failures—independent elements of a system, the failure of which would result in loss of objectives, hardware, or crew—should be inspected for defects. Fracture control requirements, if applicable, may mandate inspection of parts for defects. Highly stressed parts should be carefully examined for defects.

Springs, due to their mass-produced nature, frequently contain flaws. Improper heat treating is a common cause; for this reason, springs should have their heat treat condition tested on an unstressed location whenever feasible. In cases where this cannot be done, heat treat lots may be evaluated by inspecting per a sampling plan.

Inspection of assemblies should be performed both before and after environmental, performance, and life tests, prior to installation, and to the extent practical, after installation into test hardware or the flight vehicle. For most aerospace mechanisms, virtually every aspect of the mechanism can be critical in some way; and the appropriate inspections will vary widely depending on the application. Many inspections can be mandated via a drawing requirement. Inspections for the following items are recommended:

- Cleanliness.
- Handling damage.
- Corrosion.
- Critical clearances and dimensions (especially of wiring harnesses and multilayer insulation).

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- Proper thread engagement.
- Proper fastener torque/preload (where relaxation is not a concern or is accounted for, this can be accomplished easily with verification during assembly coupled with torque striping—a technique that seems to be underemployed).
- Proper installation of thread locking adhesive, safety wire, or safety cable.
- Condition of electrical and fluid connectors.
- Wiring harness retention.
- Spring failure.
- Leakage of fluid-filled components.
- Excessive wear, clearance, backlash, or free play.
- Lubricant condition and availability.
- Evidence of unintentional contact between parts.
- Generated debris.
- Evidence of yielding or cracking.

The method of inspection will vary depending on what is being inspected, the effect of the inspection on the parts inspected, and access to the parts. Non-destructive techniques include visual inspections, visualization under magnification (optical or scanning electron microscopy), X-ray, N-ray, eddy current inspection, magnetic particle inspection, and dye penetrant inspection. Destructive techniques such as sectioning are not often used, being reserved mainly for development activities or failure investigations.

Photographs of the parts and assemblies should generally be taken any time any type of inspection is performed, even if limited to quick visual scans, to document the condition of the unit. Verification that photographs exist of each part showing all surfaces should be made prior to the installation of that part into the next level of assembly. Photographs should also be taken of the assembly from all angles necessary to indicate its condition prior to installation into test equipment and after installation into test equipment. Special attention should be paid to surfaces and components that are expected to deflect, wear, or have the possibility of moving during the test. All externally visible screws should be photographed with enough detail to be able to see a change in orientation, if not torque-striped. The same set of photographs should be taken before and after any change in configuration.

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A.2.11 Qualification Testing

Qualification testing is conducted to verify that a mechanism's design, materials, and manufacturing processes meet specification requirements. Qualification testing includes tests that verify hardware functions during and after exposure to the specified environments. The testing typically stresses the hardware beyond the design conditions to ensure that positive design margins exist. The test margins applied during qualification are typically program-specified.

All mechanisms are required to undergo qualification testing, and this should be done at the mechanism level of assembly whenever possible. At the project's discretion, however, such testing may be performed at higher levels of assembly, after any other testing that may affect mechanical operation to confirm proper performance and to ensure that no degradation has occurred during the previous tests. All functions of the mechanism have to be included in qualification testing, including redundant modes of operation.

Qualification testing should utilize hardware that is as flight-like as possible, including any drive electronics and controllers. Qualification tests should use the same test hardware that is expected to be used in acceptance testing. This includes, test fixtures, test facilities, test support equipment, and any resident firmware and software.

The qualification test should assess design requirements levied in the mechanism's specification. The specification should include all worst-case service environments and operational cycles for the mechanism's service life, which includes acceptance tests (including acceptance retests), tests at higher levels of assembly, mission operations, and ground operations. Qualification tests should be performed for each mechanism operation at nominal-, low-, and high-energy levels. To establish that functioning is proper for normal operations, the nominal test should be conducted under the most probable conditions expected during normal flight. A high-energy test and a low-energy test should also be conducted to prove positive margins of strength and function. Adverse interaction of potential extremes of parameters such as temperature, friction, spring forces, stiffness of electrical cabling or thermal insulation, and spin rate should be addressed. Parameters to be varied during the high- and low-energy tests should include all those that could substantively affect the operation of the mechanism as determined by the results of analytical predictions or development tests. Worst-case torque or force margins should be determined by simulating the lowest motive force combined with the highest resistance under the worst-case qualification environmental conditions.

The types of environmental qualification testing required for mechanisms are typically established by the program, but a recommended set of tests is given in Table 5, Recommended Environmental Qualification Test Matrix for Mechanisms. Mechanisms that include unique types of components may require other types of tests in addition to these. The recommended test sequence is listed in Table 6, Recommended Testing Sequence; however, there is no single correct order to ensure maximum effectiveness applicable to all mechanisms. The order of testing is most valid if it is in agreement with the order in which the environments will be encountered by the flight hardware during its service life.

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Table 5—Recommended Environmental Qualification Test Matrix for Mechanisms

Performance	Static Loads	Leak	Shock	Random Vibration	Acoustic Vibration	Sinusoidal Vibration	Thermal Cycle/ Thermal Vacuum	Thermal Gradient	Depressurization/ Repressurization	Climatic	Electromagnetic Compatibility	Life
R	AN	AN	AN	R	AN	AN	R	AN	AN	AN	AN	R
Legend: R - Required Test. The test indicated is to be performed as part of the qualification program. AN - Test As Needed. The indicated test is to be performed if the environment is present for the specific application and is not enveloped by other required tests.												

Table 6—Recommended Testing Sequence

Test	Acceptance	Qualification
Run-in	X	(1)
Performance	X	X
Leak	X	X
Shock	X	X
Random Vibration	X	X
Acoustic Vibration	X	X
Sinusoidal Vibration	X	X
Thermal Cycle/Thermal Vacuum	X	X
Thermal Gradient	X	X
Depressurization/Repressurization	X	X
Climatic		X
Electromagnetic Compatibility		X
Life		X
Static Loads		X

(1) Run-in testing is not listed in the qualification sequence because it is a workmanship test, and it is assumed that the qualification unit undergoes acceptance testing prior to qualification testing. However, if for some reason acceptance testing is not performed on the qualification unit, the qualification unit should still be run in prior to qualification testing

Gravity compensation via offload fixtures may be used to test deployable systems that have insufficient structural margin and/or torque margin in a one “g” field. When gravity compensation is required, the loads from uncompensated gravity effects should be less than

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10 percent of the operational loads as a guide, but more stringent constraints may be warranted. Uncompensated gravity of 0.1 g is usually achievable and acceptable for separation tests and for comparative measurements of appendage positioning if the direction is correct, i.e., the net shear and moment imposed during measurements acts in the same direction as it would in flight, thereby causing any mechanism with backlash to assume the correct extreme positions. The assisting or resisting effects of the offload fixtures should be characterized to show that the goals of the test are met (e.g., measure torque margin, demonstrate deployment, replicate worst-case loads). Offload fixtures should not introduce degree-of-freedom constraints on deployables, e.g., a vertical deflection limitation on an air-bearing table.

To reduce the risk of workmanship-related failures during the qualification test program, it is recommended that an acceptance test be performed on the qualification hardware prior to beginning the qualification test program.

A.2.12 Life Testing

Mechanisms often experience a wide range of environments in service and the temperature at which the mechanism will perform any given cycle is usually not precisely known, so adequately representing the entire range in life testing can be a challenge. A common approach is to conduct 50 percent of the cycles at nominal expected operating conditions (which are not necessarily room-temperature conditions), 25 percent at the maximum expected operating temperature, and 25 percent at the minimum expected operating temperature. If the make-up of the cycle temperatures in service is more accurately known, the split during the life test should reflect that. The pressure during operational cycles is usually well understood, so if the hardware will experience reduced atmospheric pressure or vacuum during a portion of its service life and the mechanism contains vacuum-sensitive components such as lubricants and material combinations susceptible to galling and cold welding, then a representative fraction of the total cycles should be performed in the appropriate pressure conditions. The order in which the environments are applied in the test should follow the order that the mechanism will experience the environments in service as closely as possible.

When determining the calculated number of life cycles, it is important to consider the number of cycles at component assembly, performance baseline testing, cycles expended during and after exposure to environments (e.g., thermal-vacuum and random vibration) as well as those cycles needed at higher levels of assembly to demonstrate the performance of the entire instrument or flight system (e.g., system calibrations or comprehensive performance tests), up to and including launch site operations if applicable. It is often advisable to add some margin to the calculated number of life cycles to allow for anomaly investigations or other unforeseen needs. This margin allows for the mechanism to be functioned during investigation without using up flight cycle life and exceeding the life certification. This margin is added before applying any test factors.

Mechanisms often have reliability requirements assigned to them, and sometimes there is confusion regarding whether the reliability number should be applied to the expected life before or after the life test factor is applied. Unless otherwise specified, the reliability should be applied to the expected life before the life test factor is applied.

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The life test factor utilized in this Standard is a result of the aerospace industry's common practice of building only one life test unit. If only one unit is tested, and it is tested only to the number of cycles it is expected to see, there is no information available to establish confidence that another unit will last as long. To account for unit-to-unit variation, a factor has to be added to the number of cycles in the test to demonstrate a degree of robustness in the design. Another possible approach, especially in applications where life requirements approach the state of the art, is to build multiple life test units and test them to a reduced factor. There is no standardized methodology for this approach, so the reduced factors and the number of life test units have to be assessed on a case-by-case basis to determine an acceptable plan.

It is important to accurately represent the demands on a mechanism during the life test, including driven inertias. In some cases, it may be more appropriate to substitute a dummy load for the actual driven member so long as the dummy load provides a reasonable representation of the dynamic characteristics of the actual driven hardware (such as inertia, stiffness, free play, and natural frequencies).

There is often a desire to replace integrated, mechanism-level life testing with component-level life testing for schedule and cost purposes among others. While component life can provide valuable information, it does not take the place of integrated testing. Interactions between components often create conditions that cannot be replicated or even anticipated in component-level tests.

Life tests for long-term missions often require accelerated testing. This needs to be approached carefully with knowledge of the failure modes of the hardware and potential undesirable consequences of increased speed. For example, changing the rotational speed of a bearing may change the lubrication regime in which the bearing operates. For boundary and mixed lubrication regimes, the most likely failure mechanisms will be wear and wet lubricant breakdown, not fatigue. A bearing that normally operates in a boundary lubrication or mixed lubrication regime should never be operated in the EHD lubrication regime during an accelerated test. In the EHD regime, no appreciable wear of the balls and raceways occurs so the life increases and their failure mechanism transitions to material fatigue instead. Therefore, while life test acceleration by increasing speed may be considered, other speed-limiting factors need also be considered. For example, at the speed at which EHD lubrication is attained, one has to evaluate bearing cage instability which may produce excessive wear of the cage and would in turn produce contaminants that could degrade the performance of the bearings. Additionally, thermal issues may arise related to increased power dissipation for higher speed operation, like increased bearing thermal gradients, which should be thoroughly evaluated. Even performing the test at flight speeds with the dwells removed can mask failure modes when the quiescent durations, including storage time, are not included in the life test; continuous cycling can, for example, prevent cold welding that would otherwise manifest itself during expected periods of inactivity. For all these reasons, the life test should be run as nearly as possible using the in-service speeds and duty cycles. In some cases, it may not be possible to accelerate the test at all. One should be careful not to perform a test that is unrepresentative of the design conditions.

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Certain dry-film lubricants, particularly molybdenum disulfide-based lubricants, perform better and degrade less quickly in a vacuum than they do in an atmospheric environment. For this reason, life verification tests performed in atmospheric conditions may be conservative in this respect; but by the same token, this increase in lubricity can have undesirable effects such as changes in achieved preload of EVA bolts. Conversely, the increased friction in ambient pressure can accelerate wear and generate debris. All effects of atmosphere versus vacuum on the mechanism in question need to be considered when selecting the life test environment.

A thorough post-life test inspection of the life test unit is necessary to assess the hardware for anomalous conditions or indications of failure. Such inspections may include physical dimensional inspection of components, high magnification photography, lubricant analysis, scanning electron microscope analysis, or other techniques. If it is known that disassembly of the life test unit for inspection will be undesirable, design inspection ports into the mechanism that allow inspection of lubricants, gear teeth, bearings, and wear components without having to completely disassemble the mechanism. Photographic documentation of the life test article should begin at the component inspection and acceptance level and continue through the full assembly to act as a baseline for comparison. The critical areas of parts that may be subject to fatigue should be inspected to determine whether cracks have initiated. Where lubrication is used, it may be prudent to measure lubricant loss, degradation, distribution, and condensed outgassed constituents for wet lubricants and accumulation of lubricant debris for solid lubricants.

A.2.13 Acceptance Testing

Acceptance testing is used to verify that the manufacturing and assembly process has been accomplished in an acceptable manner and that the mechanism performs within specified parameters. Acceptance testing includes functional tests and performance tests that verify hardware meets its requirements during and after exposure to the specified environments.

The types of acceptance testing required for mechanisms are typically established by the program, but a recommended set of tests is given in Table 7, Recommended Environmental Acceptance Test Matrix for Mechanisms. Run-in is not included in this list because it is treated separately from other acceptance tests in this Standard; but note that as required in section 4, the run-in test is to be performed prior to all other acceptance tests. Mechanisms that include unique types of components may require other types of tests in addition to these. The recommended test sequence is listed in Table 6; however, there is no single correct order to ensure maximum effectiveness applicable to all mechanisms. The order of testing is most valid if it is in agreement with the order in which the environments will be encountered by the flight hardware during its mission life.

All mechanisms have to undergo acceptance testing, and this should be done at the mechanism level of assembly whenever possible. At the project's discretion, however, such testing may be performed at higher levels of assembly, after any other testing that may affect mechanical operation to confirm proper performance and to ensure that no degradation has occurred during the previous tests.

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Table 7—Recommended Environmental Acceptance Test Matrix for Mechanisms

Performance	Leak	Shock	Random Vibration	Acoustic Vibration	Sinusoidal Vibration	Thermal Cycle/ Thermal Vacuum	Thermal Gradient	Depressurization/ Repressurization
R	AN	AN	R	AN	AN	R	AN	AN
Legend: R - Required Test. The test indicated is to be performed as part of the acceptance program. AN - Test As Needed. The indicated test is to be performed if the environment is present for the specific application and is not enveloped by other required tests. Note: Run-in is not included in this matrix because it is not an environmental test. Run-in is performed prior to all environmental acceptance testing.								

Regardless of the level of assembly of the acceptance testing, mechanical function tests should be performed after integration into the spacecraft to demonstrate freedom-of-motion of all appendages and other mechanical devices whose operation may be affected by the process of integrating them with the payload. The tests have to demonstrate proper release, motion, and lock-in of each device, as appropriate, to ensure that no tolerance buildup, assembly error, or other problem will prevent proper operation of the mechanism during mission life. Unless the design of the device dictates otherwise, mechanical testing may be conducted in ambient laboratory conditions. The testing should be performed at an appropriate time in the payload environmental test sequence and, if any device in the vicinity of the mechanism (or the mechanism itself) is subsequently removed from the payload, the testing has to be repeated after final reinstallation of the device.

All functions of the mechanism have to be included in acceptance testing, including redundant modes of operation, with the obvious exception of single-use hardware such as ordnance for which no refurbishment is possible. Acceptance tests must be structured to detect workmanship defects that could affect operational performance.

Acceptance testing is performed on flight hardware by definition, but care should be taken to ensure that any drive electronics and controllers used to support acceptance testing are also flight units or as flight-like as possible. The acceptance testing of mechanisms that are part of deployable or movable systems should be conducted with the mechanism attached to the movable system whenever possible. In some cases, it may be more appropriate to substitute a dummy load for the driven member so long as the dummy load provides a reasonable representation of the dynamic characteristics of the actual driven hardware (such as inertia, stiffness, free play, and natural frequencies). It is also highly recommended to perform acceptance testing on qualification hardware.

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A.2.14 Protoflight Testing

Protoflight refers to a strategy where no test-dedicated qualification article exists and all production hardware is intended for flight. Typically, protoflight testing exposes all flight hardware to environments at qualification magnitudes for acceptance durations. In cases where a qualification test would generally be required without a corresponding acceptance test, the protoflight test is the same as the qualification test. However, sometimes adjustments need to be made to the test duration or levels to prevent unnecessary wear or consumption of life. Adjustment of qualification test parameters to avoid such erosion of capability have to be reviewed carefully to avoid incurring undue risk.

Because the hardware is used for flight, testing that is intended to demonstrate service life or ultimate strength capability cannot be performed on protoflight hardware. A protoflight approach therefore carries a higher technical risk than a full qualification test program. The protoflight approach should thus be used with caution and used only for low-risk applications since no design margin for fatigue, wear, or yield is demonstrated.

The risk generated by a protoflight approach should be mitigated using measures such as increased development testing (including component-level life testing), increased component-level qualification testing, and use of higher factors of safety.

APPENDIX B: REFERENCES

B.1 PURPOSE

This Appendix provides additional information on references mentioned in the text of this Standard. The latest issuances of cited documents apply unless specific versions are designated.

B.2 REFERENCES

B.2.1 Government Documents

Department of Defense

MIL-G-81937, Grease, Instrument, Ultra-Clean, Metric

MIL-HDBK-83377, Adhesive Bonding (Structural) for Aerospace and Other Systems, Requirements for

MIL-PRF-8625F, Anodic Coatings for Aluminum and Aluminum Alloy

MIL-PRF-27617G, Grease, Aircraft and Instrument, Fuel and Oxidizer Resistant, Amendment 1

MIL-PRF-46010H, Lubricant, Solid Film, Heat Cured, Corrosion Inhibiting

MIL-STD-29A (Superseded), Spring, Mechanical, Drawing Requirements for
Note that ASME Y14.13M should NOT be considered a replacement for MIL-STD-29A as a source of spring design information because the scope in ASME Y14.13M is limited to information necessary for the creation of spring drawings and does not contain design information.

SMC-S-016, Test Requirements for Launch, Upper-Stage, and Space Vehicles

NASA

JSC PRC-8001, Process Specification for Dry-Film Lubricant Application

MSFC-RQMT-1282, Requirements for Surface Preparation and Application of Dry-Film Lubricants

NASA/CR-2005-213424, Lubrication for Space Applications

NASA-RP-1228, Fastener Design Manual

NASA SP-5059, Solid Lubricants: A Survey

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NASA SP-8063, Lubrication, Friction, and Wear

NASA-STD-5001, Structural Design and Test Factors of Safety for Spaceflight Hardware

NASA-STD-5020, Requirements for Threaded Fastening Systems in Spaceflight Hardware

NASA-STD-6016, Standard Materials and Processes Requirements for Spacecraft

PD-ED-1229, Preferred Reliability Practice, Selection of Electric Motors for Aerospace Applications (https://extapps.ksc.nasa.gov/Reliability/Preferred_practices.html)

B.2.2 Non-Government Documents

American Institute of Aeronautics and Astronautics (AIAA)

AIAA S-114A-2020, Moving Mechanical Assemblies for Space and Launch Vehicles

American Society of Mechanical Engineers (ASME)

ASME B46.1, Surface Texture (Surface Roughness, Waviness, and Lay)

ASME Y13.6, Surface Texture Symbols

ASME Y14.5, Dimensioning and Tolerancing

ASME Y14.36, Surface Texture Symbols

ASTM International

ASTM D3933, Standard Guide for Preparation of Aluminum Surfaces for Structural Adhesives Bonding (Phosphoric Acid Anodizing)

ASTM E595, Standard Test Method for Total Mass Loss and Collected Volatile Condensable Materials from Outgassing in a Vacuum Environment

ASTM F22, Standard Test Method for Hydrophobic Surface Films by the Water-Break Test

Jet Propulsion Laboratory

JPL D-51842, Solid Film Lubricants, Application of

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SAE International

SAE ARP1524A, Surface Preparation and Priming of Aluminum Alloy Parts for High Durability Structural Adhesive Bonding

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APPENDIX C: REQUIREMENTS IDENTIFICATION MATRIX

C.1 PURPOSE

Due to the complexity and uniqueness of space flight, it is unlikely that all of the requirements in a standard will apply. The Requirements Identification Matrix below contains this Standard's technical authority requirements and may be used by programs and projects to indicate requirements that are applicable or not applicable. Enter "Yes" in the "Applicable" column if the requirement is applicable to the program or project or "No" if the requirement is not applicable to the program or project. The "Comments" column may be used to provide specific instructions on how to apply the requirement, specify proposed tailoring, or provide an explanation/justification when not applicable.

Table 8—Requirements Identification Matrix

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Section	Description	Requirement in this Standard	Applicable (Enter Yes or No)	Comments
4.1.1	Dimensional Analyses	[DDMR 1] A dimensional analysis of all moving parts and intentional interference-fit parts shall be performed and documented to ensure that specified functional performance is maintained under worst-case environmental conditions and configurations.		
4.1.2	Dimensional Analyses	[DDMR 2] The dimensional analysis shall account for the following: a. Manufacturing, assembly, and alignment tolerances. b. Temperature- and temperature-gradient-induced deformations. c. Deflections due to vibration. d. Deflections due to external loads. e. Deflections due to operational loads. f. The full range of adjustability of the mechanism parts.		
4.2.1	Clearances	[DDMR 3] Static and dynamic clearance requirements between mechanism components and any other structure, component, thermal covering, and field of view shall be established.		
4.2.2	Clearances	[DDMR 4] Required static and dynamic clearances between mechanism components and any other structure, component, thermal covering, and field of view shall be maintained throughout the service life and life test of the mechanism.		
4.2.3	Clearances	[DDMR 5] Internal mechanism clearance requirements shall be established.		

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Section	Description	Requirement in this Standard	Applicable (Enter Yes or No)	Comments
4.2.4	Clearances	[DDMR 6] Required internal mechanism clearances shall be maintained throughout the service life and life test of the mechanism.		
4.2.5	Clearances	[DDMR 7] The established clearance requirements shall account for the following: a. Manufacturing, assembly, and alignment tolerances. b. Temperature- and temperature-gradient-induced deformations or instabilities. c. Deflections due to vibration. d. Deflections due to external loads, including gravity effects. e. Deflections due to operational loads. f. Deflections due to changes in pressurization, including thermal blanket billowing. g. Motion of cable harnesses, tubing, and sensor wiring. h. Environments arising from transportation. i. The full range of adjustability of the mechanism parts.		
4.2.6	Clearances	[DDMR 8] Clearance measurements shall be performed on the highest level of assembly possible.		
4.3.1.1	General Torque Margin Requirements	[DDMR 9] All torque margin requirements shall be applied under worst-case conditions throughout the mechanism's life, including throughout life testing.		
4.3.1.2	General Torque Margin Requirements	[DDMR 10] Mechanisms that use torque multipliers or reducers (e.g., gearboxes, strain wave gear drives, lever arms) shall meet torque margin requirements at both the input and output of multiplying or reducing devices.		
4.3.1.3	General Torque Margin Requirements	[DDMR 11] All torque margins shall be verified during an acceptance test at the highest possible level of assembly.		
4.3.2	Static Torque Margin	[DDMR 12] Static torque margin per Equation 4-1 and Table 1 shall be greater than zero within the mechanism's full range of motion.		

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Torque margin is defined in Equation 4-1 as follows:

$$\text{torque margin} = \frac{T_{avail}}{\sum FS_f T_f + \sum FS_v T_v + \sum FS_a T_a} - 1 \quad (\text{Equation 4-1})$$

Table 1 – Minimum Safety Factors for Torque Margin Equation 4-1

Source of Torque Data	FS_v	FS_f	FS_a
Theory or analysis	3.00	1.50	1.25
Development test at expected environmental extremes	2.50	1.35	1.15
Qualification test			
Lot acceptance test at expected environmental extremes			
Acceptance test of flight hardware at ambient conditions	2.00	1.25	1.10
Acceptance test of flight hardware at expected environmental extremes			
Test evaluation of one-spring-out case ¹	1.00	1.00	1.00

¹ When the torque margin of spring-driven mechanisms that utilize multiple springs in parallel to provide torque is evaluated by test for cases in which one of the multiple springs fails (the “one-spring-out case”), a minimum FS_f , FS_v , and FS_a of 1.0 may be used. Prior to a spring failure, the factors in the first three rows of Table 1 apply. Note that because using multiple springs in parallel for redundancy is distinct from designing a system to tolerate a failure of some portion of a single spring, the torque margin for a failure-tolerant single spring must be evaluated with the same factor after failure as prior to failure. The one-spring-out factors do not apply to any case other than the use of redundant springs in parallel.

4.3.3	Dynamic Torque Margin	[DDMR 13] Dynamic torque margin per Equation 4-1 and Table 1 shall be greater than zero within the mechanism’s full range of motion.		
4.3.4	Holding Torque Margin	[DDMR 14] Holding torque margin per Equation 4-1 and Table 1 shall be greater than zero at the specified positions within the mechanism’s full range of motion.		
4.3.5.1	Stepper Motor Margin	[DDMR 15] Stepper motor torque margin shall be greater than zero within the mechanism’s full range of motion, using the appropriate allowable margin calculation method per Table 2, Stepper Motor Margin Calculation Methods.		

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Table 2 – Stepper Motor Margin Calculation Methods				
Case		Allowable Margin Calculation Methods		
When all of the following conditions are met: - the resisting torque due to friction is much greater than the torque due to the driven inertia - the natural frequency of the mounting base and driven inertia are greater than the stepping frequency - the driven inertia is not driven into a hard stop or spring stop - the stepper motor winding current flows for the full period of the step - a pull-in torque test is performed in the operating environments - The stepping frequency is not under closed-loop control		- Pull-in torque margin analysis per Equation 4-1 and Table 1 - Torque margin analysis per Equation 4-1 and Table 1 with T_{avail} derived from a step stability analysis		
All other cases		Torque margin analysis per Equation 4-1 and Table 1 with T_{avail} derived from a step stability analysis		
4.3.5.2	Stepper Motor Margin	[DDMR 16] When stepper motor detent torque is used to maintain rotor position in the presence of vibratory disturbances, detent stiffness and motor damping shall be considered when determining the holding force margin.		
4.3.6	Servomechanism Margins	[DDMR 17] For servomechanism applications, control system performance margins shall be calculated in addition to meeting the applicable static torque margin, dynamic torque margin, holding torque margin, and stroke margin requirements.		
4.4.1	Stroke Margin	[DDMR 18] Stroke margin requirements shall be established for all mechanisms.		
4.4.2	Stroke Margin	[DDMR 19] All stroke margins shall account for worst-case credible combinations of the following: - Environmentally induced distortions. - Misalignments. - Dimensional tolerances. - Load-induced distortions.		
4.5.1	Electrical Bonding and Grounding	[DDMR 20] Bearings shall not be used to carry electrical current.		

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4.5.2	Electrical Bonding and Grounding	[DDMR 21] Gears shall not be used to carry electrical current.		
4.5.3	Electrical Bonding and Grounding	[DDMR 22] Mechanisms shall include electrical bonding and ground paths between moving and stationary parts sufficient to meet electromagnetic environmental effects requirements.		
4.6.1	Lubrication	[DDMR 23] All surfaces in contact for which friction under relative motion negatively affects performance of the mechanism shall be lubricated.		
4.6.2	Lubrication	[[DDMR 24] The selection of lubricants for mechanisms shall include the following considerations: a. The effect of the service environments (ground storage conditions, pressure, temperature, radiation, dust, humidity, salt spray, etc.) on lubricants. b. Creep properties of wet lubricants. c. Viscosity index (variation of viscosity with temperature) of wet lubricants. d. Elastohydrodynamic (EHD) film thickness if operating in the EHD lubrication regime. e. Protection against galling and friction welding in the boundary lubrication regime. f. Outgassing and breakdown products from wet lubricants that could contaminate sensitive surfaces. g. Polymerization of wet lubricants, particularly due to high contact pressures. h. Lubricant purity and filtration level. i. Lubricant depletion for wet lubricants. (This is typically handled through a “lubrication loss analysis” which considers evaporation, creep/migration, consumption, and fling.) j. Lubricant wear-out for dry lubricants. k. Generation and management of dry lubricant wear debris. l. Compatibility of each lubricant with other lubricants and other materials (substrates, platings, coatings, and preservative or shipping oils). m. Exposure to propellant or propellant vapor. n. Interaction with pyrotechnic materials and their reaction products (blowby). o. Temperature limits of the lubricants. p. Corrosion protection of the mechanism. q. Contact stress. r. Coefficient of friction needed. s. The effect of wear debris on wet lubricant properties.		
4.6.3	Lubrication	[DDMR 25] An evaporative loss analysis shall be performed to show that 90 percent of the initial liquid lubricant quantity remains at end of life, not including lubricant degradation.		
4.7.1	Indication of Status	[DDMR 26] A direct indication of the critical states of each mechanism shall be provided.		

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4.8.1	Structural Requirements	[DDMR 27] Mechanisms classified as failure tolerant shall meet all structural requirements using full design factors of safety after failure of the mechanism to operate.		
4.8.2	Structural Requirements	[DDMR 28] Engineering analyses shall account for the structural mounting boundary conditions, including: a. Stiffness. b. Mounting alignment tolerances. c. Temperature-induced distortions. d. Load-induced distortions. e. Interface friction.		
4.8.3	Structural Requirements	[DDMR 29] The mechanism shall remain functional after exposure to stall conditions at any point in its travel.		
4.8.4	Structural Requirements	[DDMR 30] Non-jamming mechanical stops shall be incorporated into all mechanisms where exceeding required range of motion will result in detrimental effects to the mechanism or larger system.		
4.8.5	Structural Requirements	[DDMR 31] Mechanism components shall maintain a positive margin of safety with the full design factors of safety applied when subjected to worst-case transient loads from mechanical stop impact.		
4.8.6	Structural Requirements	[DDMR 32] If manipulator systems, payload operations, extravehicular or intravehicular activities, or other situations presenting a risk of inadvertent contact are present, exposed mechanism components, protective shrouds and covers, and mounting structure shall be designed to accommodate inadvertent impact loads from these sources.		
4.9.1	Bearings	[DDMR 33] Ball bearings used in high precision or low torque ripple applications shall utilize raceways that meet Annular Bearing Engineering Committee (ABEC) 7, 7P, or 7T tolerances (or better) in accordance with American Bearing Manufacturing Association (ABMA) standards (documents depend on application and are accessible at https://www.americanbearings.org/).		
4.9.2	Bearings	[DDMR 34] Nonstandard ball bearings or thin section ball bearings where ABMA tolerances do not apply that are used in high precision or low torque ripple applications shall have the manufacturer's precision level most nearly equivalent to ABEC 7.		
4.9.3	Bearings	[DDMR 35] Ball bearings used in high precision or low torque ripple applications shall utilize steel balls of ANSI/ABMA/ISO 3290-1, Rolling Bearings - Balls - Part 1: Steel balls, grades 3 through 10, inclusive, or ASTM F2215, Standard Specification for Balls, Bearings, Ferrous and Nonferrous for Use in Bearings, Valves, and Bearing Applications, grades 3 through 10, inclusive, or ceramic balls of ASTM F2094/F2094M, Standard Specification for Silicon Nitride Bearing Balls, grades 3C through 10C, inclusive.		
4.9.4	Bearings	[DDMR 36] Ball bearings used in high precision or low torque ripple applications shall utilize a raceway surface finish of 2.0 microinches arithmetic average (AA) or better.		
4.9.5	Bearings	[DDMR 37] Ball bearings used in high precision, low torque ripple, or long-life applications shall utilize material that has been refined using one of the following processing sequences:		

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		a. Air melting followed by Vacuum Arc Remelting (VAR, also known as Consumable Electrode Vacuum Melting or CEVM). b. Vacuum Induction Melting (VIM) followed by VAR. c. VIM followed by Pressurized Electroslag Remelting (PESR).																								
4.9.6	Bearings	[DDMR 38] The mean Hertzian contact stress on the most highly loaded element in a rolling element bearing shall remain less than or equal to the appropriate values in Table 3, Allowable Contact Stress for Bearing Materials Under Non-Operational Yield Design Loads, when subjected to the non-operational yield design load.																								
4.9.7	Bearings	[DDMR 39] For materials other than those listed in Table 3, an allowable contact stress shall be determined.																								
	Table 3—Allowable Contact Stress for Bearing Materials Under Non-Operational Yield Design Loads <table><tr><th rowspan="2">Bearing Material</th><th rowspan="2">Hardness Range</th><th colspan="2">Mean Hertzian Contact Stress</th></tr><tr><th>Quiet Running</th><th>Non-Quiet Running</th></tr><tr><td>440C Steel</td><td>58-62 HRC</td><td>2310 MPa (335 ksi)</td><td>2760 MPa (400 ksi)</td></tr><tr><td>52100 Steel</td><td>60-63 HRC</td><td>2480 MPa (360 ksi)</td><td>2960 MPa (430 ksi)</td></tr><tr><td>M50 Steel</td><td>62-64 HRC</td><td>2480 MPa (360 ksi)</td><td>2960 MPa (430 ksi)</td></tr><tr><td>M62 Steel</td><td>66-69 HRC</td><td>3790 MPa (550 ksi)</td><td>4070 MPa (590 ksi)</td></tr></table> NOTE: For hybrid bearings using silicon nitride rolling elements with steel rings, the allowable contact stress will be that of the steel used.		Bearing Material	Hardness Range	Mean Hertzian Contact Stress		Quiet Running	Non-Quiet Running	440C Steel	58-62 HRC	2310 MPa (335 ksi)	2760 MPa (400 ksi)	52100 Steel	60-63 HRC	2480 MPa (360 ksi)	2960 MPa (430 ksi)	M50 Steel	62-64 HRC	2480 MPa (360 ksi)	2960 MPa (430 ksi)	M62 Steel	66-69 HRC	3790 MPa (550 ksi)	4070 MPa (590 ksi)		
Bearing Material	Hardness Range	Mean Hertzian Contact Stress																								
		Quiet Running	Non-Quiet Running																							
440C Steel	58-62 HRC	2310 MPa (335 ksi)	2760 MPa (400 ksi)																							
52100 Steel	60-63 HRC	2480 MPa (360 ksi)	2960 MPa (430 ksi)																							
M50 Steel	62-64 HRC	2480 MPa (360 ksi)	2960 MPa (430 ksi)																							
M62 Steel	66-69 HRC	3790 MPa (550 ksi)	4070 MPa (590 ksi)																							
4.9.8	Bearings	[DDMR 40] Bearing fatigue life calculations shall be based on the L0.05 life when subjected to maximum time-varying loads consistent with the conditions under which the L0.05 life was determined.																								
4.9.9	Bearings	[DDMR 41] The upper and lower extremes of the ball bearing contact ellipses shall be contained by the raceways.																								
4.9.10	Bearings	[DDMR 42] All ball bearings shall be preloaded with the following exceptions: a. Four-point (gothic arch) bearings. b. Deep groove ball bearings for which it can be shown that the absence of preload on the deep groove ball bearing is not detrimental to the performance of the mechanism. c. Wave generator bearings in strain wave gearing.																								

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4.9.11	Bearings	[DDMR 43] If axial sliding of a bearing ring is required to maintain preload and/or prevent axial overload, the sliding interface shall be designed to operate as a linear sliding bearing over the range of qualification environmental limits, including lubrication to reduce wear and sliding friction forces to acceptable levels throughout the service life.		
4.9.12	Bearings	[DDMR 44] Bearing preload shall be measured once all the assembly steps that establish or affect bearing preload have been completed.		
4.9.13	Bearings	[DDMR 45] Mechanisms utilizing guides or linear bearings to constrain a carriage to move in its length direction shall use a length-to-width ratio of 2:1 or greater, unless it can be shown by analysis that a length-to-width ratio of less than 2:1 will not cause the mechanism to bind or undergo stick-slip motion taking into account the following: a. Possible friction coefficients. b. Contact forces. c. Actuating forces. d. Dynamically induced forces. e. Misalignments. f. Eccentric loading.		
4.10.1.1	Electronically Commutated Brushless Motors	[DDMR 46] Each electronically commutated (EC) brushless motor shall have the following characteristics measured (or calculated from measured values): a. Torque constant (K_t). b. Motor constant (K_m). c. Torque versus speed curve using flight-representative drive electronics. d. Phase resistances. e. Phase inductances. f. End-to-end commutated torque performance.		
4.10.1.2	Electronically Commutated Brushless Motors	[DDMR 47] The minimum measured torque output from each EC brushless motor shall be verified using flight-representative drive electronics.		
4.10.2.1	Stepper Motors	[DDMR 48] Each stepper motor shall have the following performance characteristics measured: a. Powered breakaway torque. b. Unpowered (detent) torque versus angle for a full rotation. c. Pull-in torque with representative inertia, friction loads, and step rates. d. Pull-out torque with representative inertia, friction loads, and step rates. e. Step accuracy. f. Detent to powered torque null alignment.		

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		g. Phase resistances. h. Phase inductances.		
4.10.2.2	Stepper Motors	[DDMR 49] Each stepper motor shall have the rotor polar inertia calculated.		
4.10.2.3	Stepper Motors	[DDMR 50] Stepper motor testing shall utilize flight or flight-representative drive electronics.		
4.10.3.1	Brush Motors	[DDMR 51] The maximum allowable temperature limits of the motor windings and other materials in the assembly shall be established.		
4.10.3.2	Brush Motors	[DDMR 52] Brush motor temperature limits shall not be exceeded for the worst operational cases in the worst-case qualification environments.		
4.11.1	Springs	[DDMR 53] Springs shall be failure tolerant unless spring failure can be shown to be non-credible.		
4.12.1	Gears	[DDMR 54] Gear trains shall demonstrate that they meet performance requirements, accounting for the worst-case combinations of the following conditions, throughout the service life of the mechanism: a. Tooth pitting, brinelling, bending fatigue, and bending strength under nominal and peak operating loads. b. Impact tooth loads from maximum combined axial, radial, and moment loads sustained during the full life cycle of the mechanism. c. Backlash. d. Effects of temperature and temperature gradients on quality of lubrication and gear contact pattern. e. Effects of tooth geometry. f. Undercutting and tooth profile modifications. g. Gear mounting, misalignment, and face load distribution. h. Variation in operating center distance.		
4.13.1	Dampers	[DDMR 55] Viscous dampers, including damper fluids, shall have a cleanliness requirement established.		
4.13.2	Dampers	[DDMR 56] Viscous dampers shall be evacuated to a pressure of 0.13 Pa (1×10^{-3} Torr) or less before filling with fluid.		
4.13.3	Dampers	[DDMR 57] All viscous dampers exposed to vacuum in service shall have their deadband measured in vacuum.		
4.14.1	Separable Interfaces	[DDMR 58] Separation systems utilizing separation nuts or frangible nuts shall retract the bolt beyond the separation plane without reliance upon preload or gravity.		
4.14.2	Separable Interfaces	[DDMR 59] All interfaces in deployment and jettison mechanisms designed to separate in service shall use kickoff springs to ensure first motion.		
4.15.1	Pulleys	[DDMR 60] All pulleys shall use pulley guards that extend to the tangency points of the cable.		
4.16.1	Switches	[DDMR 61] Switch mounting, orientation, and actuation shall be such that the switch cannot physically impede mechanism travel.		
4.16.2	Switches	[DDMR 62] The worst-case maximum travel of switch actuating mechanisms shall not damage the switch.		

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4.17.1	Fasteners	[DDMR 63] All retaining rings used shall be multiple-turn spiral-wound retaining rings.		
4.17.2	Fasteners	[DDMR 64] Set screws shall not be used to transmit torque between a shaft and a component mounted on the shaft.		
4.18.1	Heritage Mechanisms	[DDMR 65] The design of previously qualified mechanisms or mechanism components shall undergo a qualification program for use in a new application unless all performance requirements and environments of the new application are enveloped by the performance requirements and environments of the previous application.		
4.19.1.1	Performance Testing	[DDMR 66] All mechanism functions shall be exercised during performance testing.		
4.19.2.1	Qualification Testing	[DDMR 67] Each mechanism design shall be subjected to environmental qualification testing that exposes the mechanism to all environments, including fluid media, that it will experience in service.		
4.19.2.2	Qualification Testing	[DDMR 68] All service configurations of the mechanism shall be subjected to environmental qualification testing in the mechanism's appropriate operating or non-operating state.		
4.19.2.3	Qualification Testing	[DDMR 69] Mechanism qualification testing shall be conducted with mounting interface boundary conditions that replicate the flight boundary conditions, including the following: a. Stiffness. b. Mounting alignment and tolerances. c. Thermal distortions. d. Distortions induced by installation or externally applied loads.		
4.19.2.4	Qualification Testing	[DDMR 70] Qualification units shall utilize flight-representative drive electronics.		
4.19.2.5	Qualification Testing	[DDMR 71] Inspections shall be conducted both at the start of qualification testing and at the conclusion of qualification testing.		
4.19.2.6	Qualification Testing	[DDMR 72] Performance tests shall be conducted both at the start of qualification testing and at the conclusion of qualification testing, prior to the performance of testing with known destructive potential (e.g., a static test to ultimate loads).		
4.19.2.7	Qualification Testing	[DDMR 73] Pass-fail criteria and rationale for those criteria shall be established for all qualification tests prior to the start of qualification testing.		
4.19.3.1	Life Testing	[DDMR 74] Life testing shall be performed on all mechanism functions to verify that all service life requirements have been met.		
4.19.3.2	Life Testing	[DDMR 75] Mechanisms whose failure could result in a loss of human life shall be life tested to a number of cycles determined by multiplying the total of all operational cycles and ground cycles (including test cycles, installation cycles, and maintenance cycles) by a minimum life test factor of 4.0.		
4.19.3.3	Life Testing	[DDMR 76] All other mechanisms shall be life tested to a number of cycles determined by multiplying the total of all operational cycles and ground cycles (including test cycles, installation cycles, and maintenance cycles) by a minimum life test factor of 2.0.		

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4.19.3.4	Life Testing	[DDMR 77] Life testing shall include a number of cycles at the expected operating environmental extremes, loads, ranges of motion, and speeds that is representative of the number of cycles at those conditions expected in the service life of the mechanism.		
4.19.3.5	Life Testing	[DDMR 78] The life test shall include the operation of bearings under the maximum predicted operational contact stress for the number of cycles predicted to experience maximum operational contact stress in flight multiplied by the life test factor specified by section 4.19.3.2 or section 4.19.3.3 of this Standard.		
4.19.3.6	Life Testing	[DDMR 79] Life testing shall include intentionally running the mechanism into the mechanical stops during each test cycle.		
4.19.3.7	Life Testing	[DDMR 80] The life test shall be performed on a unit that has been exposed to all environments and levels expected to be encountered by the flight unit, including those encountered during testing.		
4.19.3.8	Life Testing	[DDMR 81] A performance test shall be conducted during both the first and last cycles of the life test.		
4.19.3.9	Life Testing	[DDMR 82] Pass-fail criteria for life tests and performance tests shall be established prior to the start of life testing.		
4.19.4.1	Acceptance Testing	[DDMR 83] Each mechanism shall be subjected to environmental acceptance testing that exposes the mechanism to all environments that it will experience in service.		
4.19.4.2	Acceptance Testing	[DDMR 84] All service configurations of the mechanism shall be subjected to environmental acceptance testing in the mechanism's appropriate operating or non-operating state.		
4.19.4.3	Acceptance Testing	[DDMR 85] A run-in test shall be performed on each mechanism prior to undergoing any other acceptance testing.		
4.19.4.4	Acceptance Testing	[DDMR 86] The run-in test shall be conducted for at least 15 cycles or 5 percent of the total expected service life, whichever is greater.		
4.19.4.5	Acceptance Testing	[DDMR 87] The run-in test conditions shall be representative of the operational loads, speed, and environment.		
4.19.4.6	Acceptance Testing	[DDMR 88] Inspection and performance tests shall be conducted after run-in testing prior to further acceptance testing, and at the conclusion of acceptance testing.		
4.19.4.7	Acceptance Testing	[DDMR 89] Pass-fail criteria for all acceptance tests and the rationale for those criteria shall be established prior to the start of acceptance testing.		
4.20.1	Mechanism Installation	[DDMR 90] Mechanisms shall either be designed to preclude installation in an incorrect orientation or be clearly labeled in a manner that indicates proper installation orientation.		

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