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DESIGN AND DEVELOPMENT REQUIREMENTS FOR MECHANISMS

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FOREWORD

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

This Standard is approved for use by NASA Headquarters and NASA Centers, including Component Facilities and Technical and Service Support Centers.

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, corrections, or additions to this Specification should be submitted via "Feedback "in the NASA Technical Standards System at https://standards.nasa.gov/.

Original Signed By: ***2015-07-31***

Ralph R. Roe, Jr. Approval Date
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1. SCOPE

1.1 Purpose

The purpose of this Standard is to establish common National Aeronautics and Space Administration (NASA) design, development, and test requirements for mechanisms whose operation is required for safety or mission success.

1.2 Applicability

This Standard is applicable to space flight mechanisms, including valves and ordnance-operated mechanical devices 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). 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.

This Standard is approved for use by NASA Headquarters and NASA Centers, including Component Facilities and Technical and Service Support Centers, and may be cited in contract, program, and other Agency documents as a technical requirement. This Standard may also apply to the Jet Propulsion Laboratory or to other contractors, grant recipients, or parties to agreements only to the extent specified or referenced in their contracts, grants, or agreements.

Requirements are numbered and indicated by the word "shall." Explanatory or guidance text is indicated in italics beginning in section 4.

1.3 Tailoring

Tailoring of this Standard for application to a specific program or project shall be formally documented as part of program or project requirements and approved by the Technical Authority.

2. APPLICABLE DOCUMENTS

2.1 General

2.2 Government Documents

None.

2.3 Non-Government Documents

None.

2.4 Order of Precedence

This Standard establishes requirements for mechanisms whose operation is required for safety or mission success but does not supersede nor waive established Agency requirements found in other documentation.

2.4.1 Conflicts between this Standard and other requirements documents shall be resolved by the responsible Technical Authority.

3. ACRONYMS AND DEFINITIONS

3.1 Acronyms and Abbreviations

°C	degrees centigrade
AA	arithmetic average
ABEC	Annular Bearing Engineering Council
ABMA	American Bearing Manufacturing Association
AGMA	American Gear Manufacturers Association
BLDC	brushless direct current
CEVM	consumable electrode vacuum melted
CMG	control moment gyroscope
DC	direct current
DFL	dry film lubricant
EC	electronically commutated
EHD	elastohydrodynamic
EMC	electromagnetic compatibility
EMI	electromagnetic interference
EVA	extravehicular activity
FS	factor of safety
GSE	ground support equipment
HRC	Rockwell C hardness
HST	Hubble Space Telescope
Hz	Hertz
ISS	International Space Station
ksi	thousand pounds per square inch
MAC	multiply alkylated cyclopentane
MPa	megapascals
NASA	National Aeronautics and Space Administration
NLGI	National Lubricating Grease Institute
PCVD	physical chemical vapor deposited
PIP	push in and pull
PFPE	perfluoropolyalkylether

PTFE	polytetrafluoroethylene
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 harmonic drive output gear error
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
K_m	motor constant
K_t	torque constant
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_{avail}	the minimum available torque or force generated by a mechanism at worst case environmental conditions at any time in its life
T_f	individual fixed resistive torques that are well-known and not strongly influenced by friction, temperature, life, or other highly variable phenomena.
T_v	individual resistive torques that may vary over environmental conditions and life

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.

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 between the balls and the raceways. *Note: Generally, this torque has to be assessed at low speeds because the effect of drag on the bearing from the liquid lubricant starts to affect the torque at higher speeds.*

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.

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

Dynamic Torque: The torque necessary to achieve a required acceleration of a mechanism.

Hard Preload: A bearing preload approach in which the bearings are clamped together without spring or diaphragm elements.

Holding Torque: The torque necessary to prevent motion of a mechanism under external load.

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.*

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

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.*

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

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

where 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.

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

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.

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

Spring Preload: See "Soft Preload."

Starting Torque: The torque necessary to initiate motion in a mechanism.

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

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

Structural Fastener: A fastener that is used for structural purposes only, is installed on the ground only, and whose configuration is not altered during flight.

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 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.

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

4. REQUIREMENTS

4.1 Tolerancing

a. Dimensional tolerances on all moving parts and intentional interference-fit parts shall be established and documented via a dimensional analysis to ensure that proper functional performance is maintained under all natural and induced environmental conditions and configurations.

Tolerancing and dimensional analysis is important not only for ensuring external clearances, but also for ensuring proper mechanism function in the first place. Tolerancing is too often considered as an afterthought of the design during the drawing creation phase and established without a thorough understanding of the tolerance drivers. Establishing the tolerances via a documented dimensional analysis helps drive the understanding of the effects of tolerances and other factors, and allows for easy review and revision later.

b. The dimensional analysis shall account for the following:

(1) Manufacturing, assembly, and alignment tolerances.
(2) Temperature.
(3) Temperature gradients.
(4) Vibration.
(5) Deflections due to external loads.
(6) Deflections due to operational loads.
(7) Adjustability and rigging of the mechanism parts.

The factors to be considered in the dimensional analysis are similar to those 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

a. Static and dynamic clearance requirements between mechanism components and any other structure, component, thermal covering, and field of view shall be established and maintained.

b. Internal mechanism clearance requirements shall be established and maintained.

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. The necessary clearances required have to be established to enable design and verification, and the design has to maintain those clearances.

c. The established clearance requirements shall account for the following:

(1) Manufacturing, assembly, and alignment tolerances.
(2) Temperature.
(3) Temperature gradients.
(4) Vibration.
(5) Deflections due to external loads, including gravity effects.
(6) Deflections due to operational loads.
(7) Deflections due to pressurization or depressurization effects, including thermal blanket billowing.
(8) Motion of cable harnesses, tubing, and sensor wiring.
(9) Environments arising from transportation.
(10) Adjustability and rigging of the mechanism parts.

Because many of the factors affecting the overall (dynamic) clearance are not present when inspections are performed, these effects have to be accounted 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. 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, rationale for its inapplicability . Motion under transportation loads is often not considered but clearances are important in that situation too.

d. Clearance measurements shall be performed on the highest level of assembly possible.

In order to verify that 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. 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

Torque margin is defined as follows:

$$\boldsymbol{torque\ margin = \frac{T_{avail}}{\sum K_f T_f + \sum K_v T_v} - 1} \tag{4-1}$$

T_{avail} is the minimum available torque generated by the mechanism at worst case environmental conditions at any time in its life.

T_f are the individual fixed resistive torques that are well-known and not strongly influenced by friction, temperature, life, or other highly variable phenomena.

T_v are the individual resistive torques that may vary over environmental conditions and life.

K_f and K_v are factors applied to each individual resistive torque prior to summation per table 1, Minimum Torque/Force/Energy Margin Factors.

Table 1 – Minimum Torque/Force Margin Factors

Origin of Factor	K_f	K_v
Value Obtained via Theory or Analysis	1.5	3.0
Value Obtained via Test of Flight-Like Hardware	1.25	2.0
Value for One-Spring-Out Case[1]	1.0	1.0
[1]Spring-driven mechanisms that utilize multiple springs nominally working together to provide torque may utilize a minimum K_f and K_v of 1.0 for the cases in which one of those springs fails. Prior to failure, the nominal (non-failure) factors still apply.		

For linear devices, "Force" replaces "Torque" in the above equation and descriptions T_{avail} represents torques from actuators such as motors, springs, pyrotechnics, solenoids, heat actuated devices, and other devices. Examples of fixed torques, represented by T_f, include accelerated inertias, motor detent torques, and unbalanced pressure loads limited by relief mechanisms; all other resistive torques tend to be variable enough that a higher factor is more appropriate and thus fall under T_v. Examples include static or dynamic friction, alignment effects, wire harness torques, damper drag, and variations in lubricant effectiveness, including degradation or depletion of lubricant over life.

This single equation can be used to calculate holding torque margin, starting torque margin, dynamic torque margin, and pull-in torque margin (for stepper motors).

For holding torque margin, T_{avail} is the actuator torque, while T_f and T_v are the torques that tend to disturb the mechanism.

For starting torque margin, T_{avail} is the actuator torque, while T_f and T_v are the resistive torques.
For dynamic torque margin, T_f is the torque required to accelerate an inertia by a given amount, and T_v are the resistive torques.

For pull-in torque margin, T_{avail} is the pull-in torque at a given drive rate, T_f is the maximum detent torque, and T_v is the total friction torque seen by the motor.

a. All calculated force and torque margins shall account for worst-case credible combinations of factors at end of life.

Because spacecraft mechanisms are exposed to many factors in combination that can deplete margin, calculation of force and torque margins also have to account for these factors in combination. It is recommended that the following considerations be included in margin calculations as they reflect phenomena that are frequently found to cause problems in margin calculation:

- *Environmental conditions.*
- *Frictional effects.*
- *Possible changes in static and dynamic friction due to storage time.*
- *Alignment effects.*

- *Wire harness loads.*
- *Damper drag.*
- *Thermally induced distortions.*
- *Load-induced distortions.*
- *Variations in lubricity.*
- *Fluid pressure on the elastomers in viscous dampers.*
- *Supply voltage, motor, and controller parameters.*
- *Acceleration due to vehicle motion or maneuvers that can retard motion.*
- *Loading due to vibroacoustic environment.*

b. The starting torque or force margin shall be greater than zero at all points of travel.

c. Dynamic torque or force margin shall be greater than zero at all points of travel.

d. Holding torque or force margin shall be greater than zero at all points of travel.

A positive torque or force margin ensures that the mechanism retains reserve torque or force that can be applied in the event of an unforeseen effect that robs motive force from the mechanism.

e. If motors are used in the system, T_{avail} shall be measured at multiple points over the range of motion with the minimum supplied motor voltage and at the output of the prime mover, not including gear heads or gear trains affixed to the motor or within the mechanism.

The torque margins must be calculated at the motor output because the resistive torques present in the gear heads can drive the minimum margin to be at the motor output rather than at the gear head output. Basing torque margin on the gear head output can give a false impression of the true torque margin.

f. Stepper motor stability margin from a step stability analysis shall be greater than zero.

A further discussion of stepper motor performance and stability analysis can be found in Appendix A. When stepper motor detent torque is used to maintain the position of the motor, the holding torque margin may be calculated via equation 4-1.

g. When stepper motor detent torque is used to maintain the position of a motor in the presence of vibratory disturbances, detent stiffness and motor damping shall be considered when determining the holding force margin.

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

h. All torque and force margins shall be verified during an acceptance test at the highest possible level of assembly.

Torque and force margins are intended to ensure that the mechanism retains reserve torque or force that can be applied in the event of an unforeseen effect that reduces motive force from the mechanism. Therefore, as with any other capability of the mechanism, the minimum torque or force margin must be verified as intact prior to placement into service.

In practice, it is often difficult to test-verify the margin directly because of the difficulty of ensuring that worst-case parameters are all in effect or the inability to measure certain values separate from others. This often drives some portion of the verification to depend on calculation. In these cases, the margin should be calculated by using the worst stack-ups of tested factors and adjusting for factors not present in the test. The result has to be greater than zero.

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

The required conservatism is included in the equation in the form of the factors, so a margin of zero indicates that requirements are met. A positive margin indicates that torque, force, or energy above that which is required for conservatism exists. In order to evaluate margin over the minimum torque, force, or energy required to operate in with no conservatism included, the equation is calculated with all K_f *and* K_v *values set to unity.*

The theoretical/analytical factors listed in table 1, Minimum Torque/Force/Energy Margin Factors, are not intended to be used as "untested factors," i.e., factors to be used when attempting to meet torque margin requirements by analysis only, without any testing to verify margin. Such an approach is prohibited by the requirement to test-verify margins in section 4.3.h. These factors are intended as higher uncertainty factors to be used in sizing and calculations until test-obtained values can be obtained for the torque values to which the factors are applied. This allows the required torque to be reduced as confidence in the system characteristics grows through testing.

When assessing failure tolerant cases, it is important to recognize that mechanisms that utilize multiple springs nominally working together to provide force or torque would end up delivering a much greater margin under normal operating conditions in order to show a positive margin after a failure. This margin can be excessive. For those cases, reduced conservatism in the factors is appropriate. The "one-spring-out" case may describe a missing spring or a broken coil as appropriate for the application. Redundancy will often be achieved with elements such as redundant motor windings, velocity-summed motor arrangements, or torque-summed motors that run at half current nominally. These elements do not work together at full power to provide force or torque. These types of implementations still require the nominal (non-failure case) K_f *and* K_v *values from the table. Note that if the failed actuator produces a resistive torque after failure, this torque has to be included in the margin calculation for the failure case.*

4.3.1 Servomechanism Margins

For servomechanism applications, performance margins shall be documented.

Servomechanisms may require motor performance that far exceeds that required by this torque margin equation in order to meet performance requirements, so the typical margin requirement alone is not sufficient. Though other applicable margins such as starting force margins should still be calculated and documented, servomechanisms will need a control system performance analysis, e.g., phase and gain margin analysis, to fully assess performance and margin.

4.4 Stroke Margin

Stroke margin is defined as follows:

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

a. Stroke margin shall be documented for all linear mechanisms.

Stroke margin helps to guarantee enough travel exists in a mechanism to accomplish its function in the presence of uncertainty. A ten percent stroke margin is frequently employed.

b. All stroke margins shall account for worst-case credible combinations of the following:

(1) Environmental conditions.
(2) Thermally induced distortions.
(3) Load-induced distortions.
(4) Mounting alignments.
(5) Tolerances.

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. 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 (4.3.d), tolerancing (4.1), and adherence to structural requirements.

4.5 Electrical Bonding and Grounding

a. Bearings shall not be used to carry electrical current.
b. Gears shall not be used to carry electrical current.

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 makes it difficult for them to do so. Other current paths should be provided. Electrical currents produced by unintentional charging of bearing-supported hardware should be considered when evaluating this requirement.

c. Mechanisms shall include electrical bonding and ground paths between moving and stationary parts sufficient to meet electromagnetic environmental effects requirements.

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, since they are prohibited from carrying current. Care should be taken to ensure that the bonding and grounding scheme is able to perform as intended. 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 without using prohibited current paths.

4.6 Lubrication

a. All surfaces in contact that affect the performance of the mechanism while in relative motion shall be lubricated.

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. Use of dissimilar metallic materials for the wear surfaces, though strongly encouraged, is not an equivalent to or substitute for lubrication and does not meet the intent of this requirement. Refer to Appendix A, for a discussion of lubricant selection and factors that should be considered.

b. The selection of lubricants for mechanisms shall include the following considerations:

(1) Lubricant property changes in storage or in a space environment.

(2) Creep properties of wet lubricants.

(3) Viscosity versus temperature properties of wet lubricants.

(4) Elastohydrodynamic (EHD) film thickness if operating in the EHD lubrication regime.

(5) Outgassing or potential breakdown products from wet lubricants that could cause contamination, such as on optical or thermal control surfaces.

(6) Possibility of polymerization of wet lubricants, particularly due to high contact pressures or contaminants.

(7) Required purity of the lubricant.

(8) Lubricant depletion (lubrication loss analysis) for wet lubricants or lubricant wear-out for dry lubricants.

(9) Dry lubricant debris generation.

(10) Compatibility of the lubricant with other materials, particularly other lubricants if used, during ground testing as well as in service.

(11) Operating temperature limits of the mechanism and the lubricant.

(12) Corrosion protection of the mechanism.

(13) Protection against galling and friction welding of the mechanism.

(14) Contact stress.

(15) Run-in requirements, such as rate of speed, load, and time duration.

(16) Coefficient of friction of the tribological system.

(17) The effect of other environments on the tribological system, such as humidity and salt spray.

There are numerous factors that have to be considered in order to make a proper choice of lubricant. Not all of these considerations 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.

c. An evaporative loss analysis shall be performed to show that ninety percent of the initial lubricant quantity remains at end of life, not including lubricant degradation.

Because life testing evaluates cycle life and not calendar life, evaporative effects on lubricant availability typically cannot be evaluated with a life test alone.

d. The lubricant application process for each application shall be specified in the engineering documentation.

The quantity of lubricant used and method of application can be almost as important as the presence of lubricant in the first place. Too much can impede mechanism performance or create contamination problems, and too little can result in reduced life or inadequate performance.

4.7 Structural Requirements

a. Mechanisms classified as failure tolerant shall meet all structural requirements after failure of the mechanism to operate using full design factors of safety.

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.

b. Engineering analyses shall account for the structural mounting boundary conditions, including:

(1) Stiffness.
(2) Mounting alignment tolerances.
(3) Temperature-induced distortions.
(4) Load-induced distortions.
(5) Interface friction.

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. 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.

c. Mechanism components shall maintain positive margins of safety under actuation force/torque stall conditions.

Many things can cause a mechanism to reach its stall torque or force. Designing the mechanism with enough strength to withstand stall ensures that the mechanism is undamaged by this situation and allows steps to be taken to recover the mechanism functionality. The usual factors of safety apply; i.e., the factors of safety are not to be reduced for this situation.

d. 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.

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.

Examples of a mechanism for which over-travel is impossible include filter wheels and gimbals with slip rings.

e. Mechanism components shall maintain a positive margin of safety with the appropriate factors of safety applied when subjected to worst-case transient loads from mechanical stop impact.

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. A bounding worst-case load would include impact at maximum speed combined with stall torque.

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

Designing for this possibility will ensure adequate margins against deformation that could cause a binding or jamming condition or inadvertent operation of the mechanism. Analysis of these cases should use full factors of safety. The particular load to be accommodated will be determined by the individual program.

4.8 Bearings

a. Ball bearings used in high precision or low torque ripple applications shall utilize raceways that meet Annular Bearing Engineering Council (ABEC) 7, 7P, or 7T tolerances (or better) in accordance with Anti-Friction Bearing Manufacturing Association (ABMA) standards.

b. 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.

c. Ball bearings used in high precision or low torque ripple applications shall utilize balls of ABMA grade 10 or better.

d. 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.

e. Ball bearings used in low torque ripple or long life applications shall utilize material that has been consumable electrode vacuum melted (CEVM), vacuum induction melted (VIM), and/or vacuum arc remelted (VAR).

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. Vacuum melting improves the cleanliness of bearing steels by eliminating 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 inclusions are stress raisers from which fatigue cracks and spalls originate, shortening fatigue life. While not required, it should be noted that smoother raceway finishes can help a bearing transition to the EHD regime at lower speeds and thus can also help to extend life. It should be stressed that the above conditions are necessary to achieve high precision, low torque ripple, and/or long life applications but may not be sufficient.*

f. 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 2, Allowable Contact Stress for Bearing Materials Under Non-Operational Limit Loads, when subjected to the non-operational limit load.

g. For materials other than those listed in table 2, an allowable contact stress shall be determined.

Table 2—Allowable Contact Stress for Bearing Materials Under Non-Operational Limit Loads

Bearing Material, Typical Hardness Range	**Mean Hertzian Contact Stress—High Precision, Low Torque Ripple Applications**	**Mean Hertzian Contact Stress—Other Applications**
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)
VIM CRU20 Steel, 66 HRC minimum	3790 MPa (550 ksi)	4070 MPa (590 ksi)

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

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 VIM CRU20 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.

It should be emphasized that these allowables are for non-operational loadings only, e.g., random vibration loads during launch, and are NOT appropriate allowables for operating stress levels, which will generally be much lower and depend on a number of factors. See Appendix A, for guidelines.

A yield factor of safety is used to account for uncertainty in determining the loading conditions on bearings.
If the design limit loads are derived from mass acceleration curves, it is acceptable to analyze the bearing using only the static loads derived from those curves.

h. 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.

Experience indicates that the L0.05 life is sufficient to avoid fatigue life problems in life testing and service for space applications

i. The upper and lower extremes of the ball bearing contact ellipses shall be contained by the raceways.

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. See Appendix A for more on contact ellipse truncation.

j. All ball bearings shall be preloaded with the following exceptions:

(1) Four-point (gothic arch) bearings.

(2) 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.

Bearing preload eliminates free play, reduces runout of the rotating member, increases axial and radial stiffness, prevents fretting damage, reduces impact loading during vibration, increases the load sharing among rotating elements, and prevents ball skidding. 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 for guidance.

k. If axial sliding of a bearing ring is required to maintain preload, sliding shall be facilitated by methods such as a tribological coating or a lubricated sleeve.

Some preload schemes depend on a spring and one bearing ring that is free to slide axially. In such a case, sliding at this interface is required to achieve the desired preload and has to be guaranteed by design.

l. Bearing preload shall be measured once all the assembly steps that establish or affect bearing preload have been completed.

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

m. Mechanisms utilizing guides or linear bearings 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:

(1) Possible friction coefficients.
(2) Contact forces.
(3) Actuating forces.
(4) Dynamically induced forces.
(5) Misalignments.
(6) Eccentric loading.

For mechanisms utilizing a set of linear bearings or guides, the length is defined as the distance between guide points or bearing centers along the axis of motion and the width is the lateral spacing between the guides or centerlines of the bearings, 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 linear bearing shaft or guide rail, the length is the effective length of the linear bearing or guide and the width is the moment arm defined by 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 linear bearing because shaft deflections over the length of the bearing can also cause binding problems.

Mechanisms guided by linear devices are sensitive to the geometry of the supports. 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 undergo stick-slip motion. Ratios less than 1:1 are *suitable only for high-precision, low friction applications such as linear bearing systems. The effective bearing length is often not the overall length of the bearing, but the length of the portion of the bearing carrying the load, often called a "load zone." Few manufacturers publish information about their bearings' load zones, so it is often necessary to contact the company or make a conservative estimate of the load zone of a specific bearing. The same uncertainty factors used in the force margin analysis (e.g., on friction) should be applied to any bearing ratio analysis performed. Further information on bearing ratios can be found in "Demystifying the 2:1 Ratio and the Stick-Slip Phenomenon: A Technical Whitepaper Explaining the Theory Behind the Binding Ratio and How It Relates to Stick-Slip" (Schroeder, 2010).*

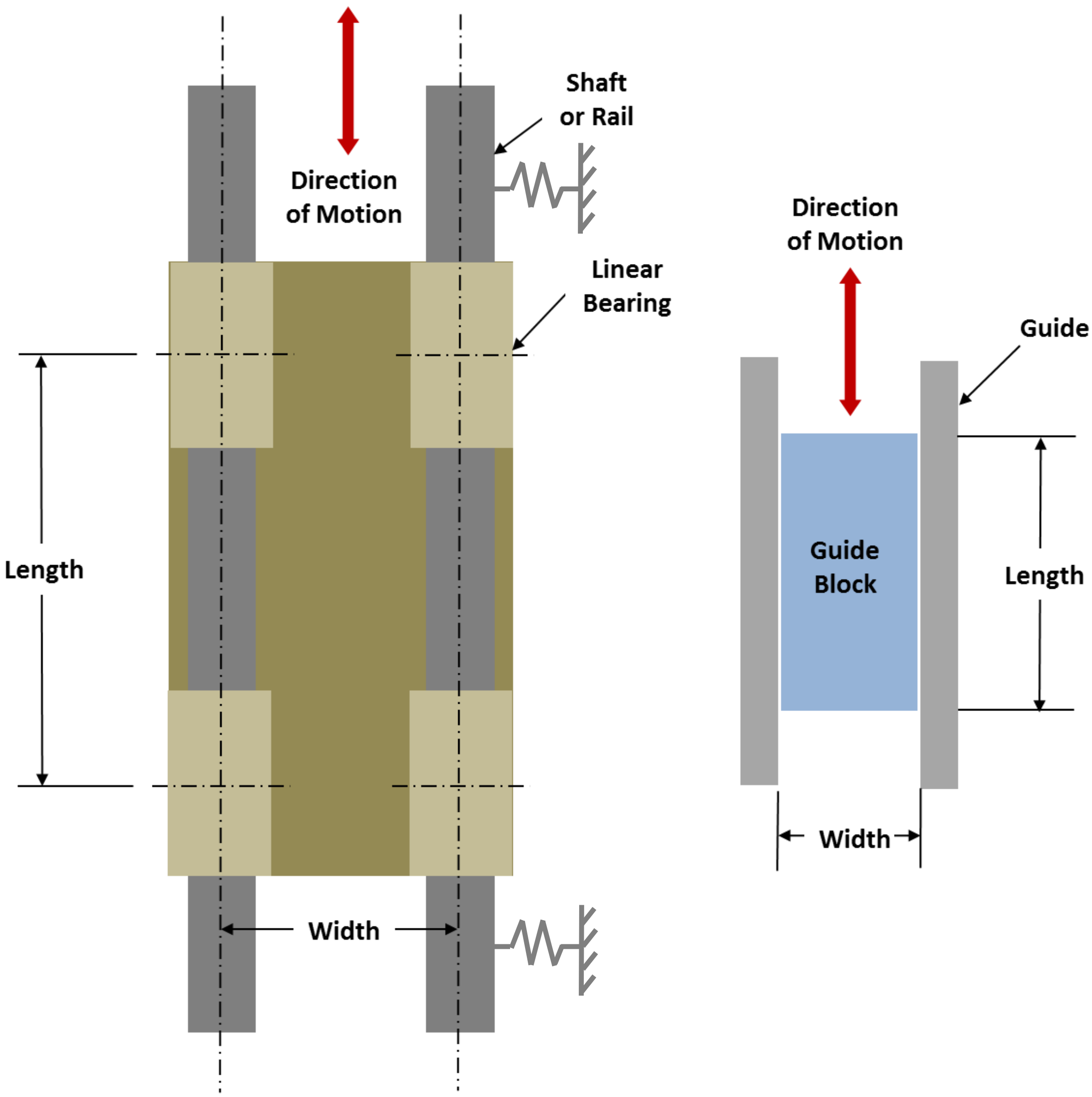

Figure 1—Illustration of Length and Width for Various Examples of Mechanisms Using a Set of Guides

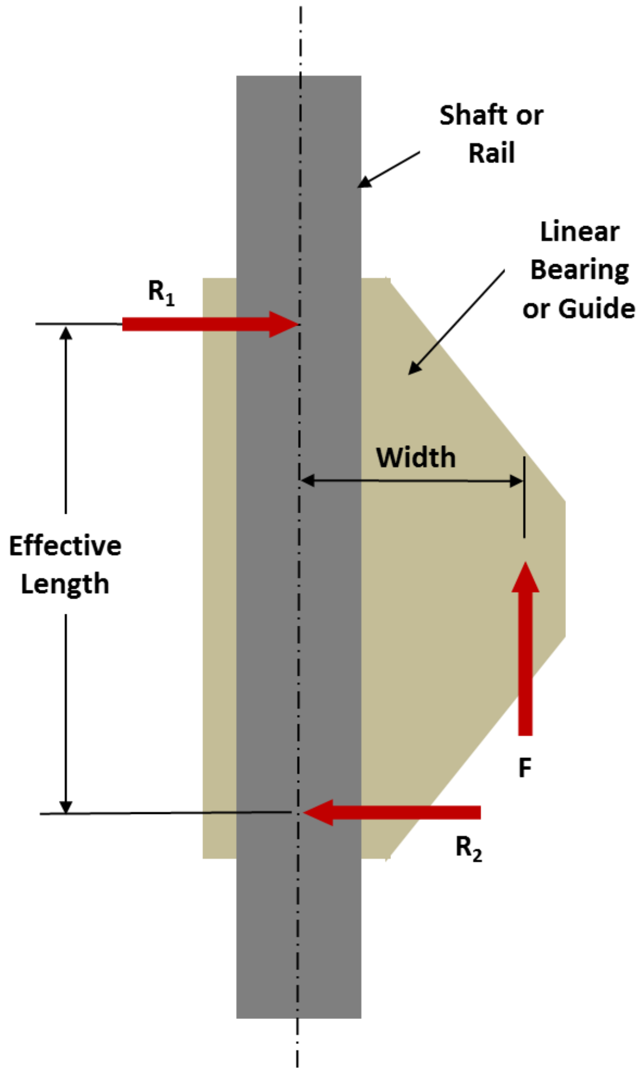

Figure 2—Illustration of Moment Arm, Effective Length, and Width for a Single Linear Bearing Application

4.9 Motors

4.9.1 Electronically Commutated Brushless Motors

a. Each electronically commutated (EC) brushless motor shall have the following characteristics measured (or calculated from measured values):

(1) Torque constant (K_t).
(2) Motor constant (K_m).
(3) Torque versus speed curve using flight-representative drive electronics.
(4) Phase resistances.
(5) Phase inductances.
(6) End-to-end commutated torque performance.

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.

b. The minimum measured torque output from each EC brushless motor shall be:

(1) Greater than 80 percent of peak output torque.
(2) Verified using flight-representative drive electronics.

The output torque parameters are characterized with a torque profile test. 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. See Appendix A for guidance on torque profile testing techniques.

4.9.2 Stepper Motors

a. Each stepper motor shall have the following performance characteristics measured:

(1) Powered breakaway torque.
(2) Unpowered (detent) torque versus angle for a full rotation.
(3) Pull-in torque with representative inertia, friction loads, and step rates.
(4) Pull-out torque with representative inertia, friction loads, and step rates.
(5) Step accuracy.
(6) Detent to powered torque null alignment.
(7) Phase resistances.
(8) Phase inductances.

b. Each stepper motor shall have the rotor polar inertia calculated.

c. Stepper motor testing shall either:

(1) Use drive electronics with a pulse duration, peak voltage, and drive pulse shape that is identical to those of the flight drive electronics, or

(2) Verify that the performance of the stepper motor is not affected by the differences in drive pulse between test and flight drive electronics.

Stepper motors operate differently from electronically commutated brush motors so they need to have different characteristics measured. Measuring these motor characteristics for stepper motors allow one to understand how the motor will perform in a mechanical system. 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 in order to minimize frictional and inertial shaft loading from the sensor; however, the compliance of the load sensor can also impact test results.*

Both pull-in and pull-out torque tests require a motor driver to perform. For similar reasons as the brushless motor, it would be best to use a motor driver that is as flight-like as possible for these tests. It is essential that the driver have similar characteristics for shunting diodes, output impedance, voltage drop, current limiting and/or limited pulse width (if employed).

4.9.3 Brush Motors

a. The maximum allowable temperature limits of the motor windings and other materials in the assembly shall be established.

b. Brush motor temperature limits shall not be exceeded for the worst operational cases in the worst-case environments.

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. 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.

4.10 Springs

Springs shall be failure tolerant unless spring failure can be shown to be non-credible.

Springs are a common mechanism component as well as a common source of problems. Spring redundancy can greatly improve mechanism reliability. There are two ways to achieve redundancy in a spring, as follows: (1) a second spring can be used, (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 can 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 and other steps have to be taken to demonstrate reliability.
More information on spring use and design is available in Appendix A.

4.11 Gears

Gear trains shall have analysis demonstrating positive margins of safety for strength and wear, accounting for the following conditions:

a. Tooth pitting, brinelling, and bending stresses 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.

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

4.12 Dampers

a. Viscous dampers, including damper fluids, shall have a cleanliness requirement established.

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. 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.

b. All viscous dampers shall be filled while under vacuum to preclude entrapment of air.

Air entrapment in dampers can affect performance of the dampers.

c. All viscous dampers exposed to vacuum in service shall have their deadband measured in vacuum.

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

4.13 Separable Interfaces

a. Separation systems utilizing separation nuts or frangible nuts shall extract the bolt without reliance upon preload or gravity.

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. 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 extraction.

b. All interfaces in deployment and jettison mechanisms designed to separate in service shall use kickoff springs to ensure first motion unless first motion.

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 this effect ensures that the joint will separate and enough force or energy remains in the separation to meet requirements.

4.14 Pulleys

All pulleys shall use pulley guards that extend to the tangency points of the cable.

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.

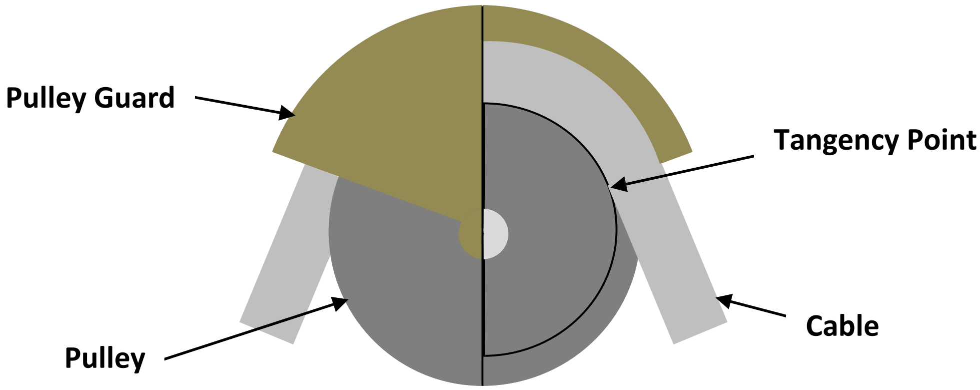

Figure 3—Pulley Guard Illustration

4.15 Switches

a. Switch mounting, orientation, and actuation shall be such that the switch cannot physically impede mechanism travel.

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.

b. The worst-case maximum travel of switch actuating mechanisms shall not damage the switch.

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. This can be accomplished in a variety of ways, such as cam profiles or spring-loaded levers that release from the microswitch when depressed.

4.16 Fasteners

a. All retaining rings used shall be multiple-turn spiral-wound retaining rings.

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. Improper installation techniques can still yield spiral rings however, so care is still necessary.

b. Set screws shall not be used to transmit torque between a shaft and a component mounted on the shaft.

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.

4.17 Quick-Release Pins

Quick-release pins, sometimes referred to as "pip-pins," shall be considered individual mechanisms and are subject to the requirements established in this document.

Quick release pins were originally designed for ground applications but because of their ease of use have seen use in space flight applications, particularly in human-rated spacecraft. The pins themselves are small mechanisms and subject to common mechanical and structural failure modes. Most off-the-shelf pins are not designed to withstand space flight environments and past use has resulted in a wide array of problems with virtually every component of the pins. Quick release pins can be used successfully if designed expressly for this purpose and subjected to the same rigor as any other mechanism though they are never recommended for zero-fault-tolerant applications. See Appendix A for more information.

4.18 Released Degrees of Freedom

Interfaces designed to release mechanical degrees of freedom shall be considered mechanisms subject to the requirements established in this document.

Features whose sole purpose is to release degrees of freedom (e.g., spherical bearings in struts) are actually mechanisms and subject to failure modes typical of mechanisms, despite their structural appearance. These interfaces are often overlooked when mechanism requirements are applied and verified.

4.19 Threaded Interfaces

Threaded interfaces designed to be actuated in service shall be considered mechanisms subject to the requirements established in this document.

Actuated threaded interfaces (e.g., crew-actuated bolts assembling space station components) are actually mechanisms and subject to failure modes typical of mechanisms, despite their structural appearance. These interfaces are often overlooked when mechanism requirements are applied and verified. Past oversights of this kind have led to high-profile failures in service.

4.20 Heritage Mechanisms

The design of previously qualified mechanisms or mechanism components shall undergo a qualification program for use in a new application unless all environments of the new application are enveloped by the environments of the previous application.

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 has to be exercised in utilizing such hardware or the anticipated savings can quickly disappear. One of the most common mistakes is selecting a design that was qualified for an environment that is not enveloped by all of the new environments but failing to re-qualify the design for the new environment. In practice, it is difficult to find designs that can truly be said to be previously qualified for a new application because of the wide variety of unique environments and missions encountered by spacecraft. Even if certain environments are enveloped, the effect of the environment on the performance of the mechanism would be missed if it 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.

4.21 Performance Testing

All mechanism functions shall be exercised during performance testing.

Sometimes there is a desire to test only critical functions or nominal operations, but all functions of the mechanism have to be tested in order 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.

4.22 Qualification Testing

a. Each mechanism design shall be subjected to environmental qualification testing that exposes the mechanism to all environments that it will experience in service.
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. The program also typically specifies required margin, test order, tolerances, and general methods to be used. Behavior under environmental conditions can be very hard to predict and omission of qualification testing can significantly increase the risk of in-service failure.

b. All service configurations of the mechanism shall be subjected to environmental qualification testing in the mechanism's appropriate operating or non-operating state.

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, 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.

c. Mechanism qualification testing shall be conducted with mounting interface boundary conditions that replicate the flight boundary conditions, including the following:

(1) Stiffness.
(2) Mounting alignment and tolerances.
(3) Thermal distortions.
(4) Load-induced distortions.

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. 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.

d. Qualification units shall utilize flight-life electronics.

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 drive electronics with the same characteristics as the flight unit. Care also should be taken to consider the configuration and lengths of harnessing and the software used as they may affect performance.

e. Inspections shall be conducted both at the start of qualification testing and at the conclusion of qualification testing.

f. Performance tests shall be conducted both at the start of qualification testing and at the conclusion of qualification testing.

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. 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.

g. Pass-fail criteria and rationale for those criteria shall be established for all qualification tests prior to the start of qualification testing.

h. Inspection procedures for all qualification tests shall be established prior to the start of qualification testing.

Pass-fail criteria and inspection procedures 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.

4.22.1 Design Life Testing

a. Design life testing shall be performed on all mechanism functions to verify that all design life requirements have been met.

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

b. Mechanisms whose failure could result in a loss of human life shall be life tested to a number of cycles no less than four times the total of all operational cycles plus the total of all ground cycles (including test cycles, installation cycles, and maintenance cycles).

c. All other mechanisms shall be life tested to a number of cycles no less than two times the total of all operational cycles plus the total of all ground cycles (including test cycles, installation cycles, and maintenance cycles).

The life test factor is 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 in order to meet the life test requirement?

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.

d. Design life verification testing shall include a number of cycles at the expected operating environmental extremes, loads, and speeds that is representative of the number of cycles at those conditions expected in the service life of the mechanism.

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. Appendix A contains recommendations on how to allocate cycles of the life test to different environments.

e. The design life test shall include functioning bearings under the maximum predicted operational contact stress for the number of cycles predicted in flight multiplied by the appropriate life test factor.

Bearing life is highly dependent on contact stress, thus it is necessary to ensure that flight-like contact stresses are present in the life test to avoid misleading test results.

f. Design life verification testing shall include testing of mechanical stops by intentionally running the mechanism into the stops during each test cycle.

Mechanical stops perform a function of the mechanism and have to be life tested. Often mechanical stops are not contacted under nominal testing conditions and so their ability to withstand loading over the life of the mechanism may not be evaluated in ground testing. However, in service, conditions may be such that the stops are contacted consistently regardless of whether or not the mechanism reached its mechanical stops during ground testing. For this reason, the mechanical stops have to be intentionally tested during a life test. In some cases where 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 stops during the design life test of the assembly. In these situations, life testing of the stops may be conducted separately at a subassembly level.

g. The life test shall be performed on the environmental qualification unit.

Performing the life test on the environmental qualification unit helps ensure that environmental effects on the mechanism life are accounted for.

h. A performance test shall be conducted during both the first and last cycles of the design life test.

Pre-test performance tests and inspections establish baseline condition and performance. Post-test performance tests and inspections allow the residual effect of the design life test to be determined when compared to the pre-test baseline.

i. Inspection procedures and pass-fail criteria for design life test and performance tests shall be established prior to the start of design life testing.

Inspection procedures and 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.

4.23 Acceptance Testing

a. Each mechanism shall be subjected to environmental acceptance testing that exposes the mechanism to all environments that it will experience in service.

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. 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. 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.

b. All service configurations of the mechanism shall be subjected to environmental acceptance testing in the mechanism's appropriate operating or non-operating state.

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.

c. A run-in test shall be performed on each mechanism prior to undergoing any other acceptance testing.

Run-in testing (also known as wear-in testing) serves two purposes. First, it acts as a screen to detect material and workmanship defects that manifest themselves early in the mechanism's life. Second, 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. 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 has to 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.*

d. The run-in test shall be conducted for at least 15 cycles or 5 percent of the total expected service life, whichever is greater, unless the mechanism has demonstrated the capability to perform in a consistent and controlled manner with fewer cycles.

These run-in durations have been shown through experience to be adequate for achieving the goals of the testing. However, it may not be sufficient in all situations. 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 cycles used in calculating the design 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.

e. The run-in test conditions shall be representative of the operational loads, speed, and environment.

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 bearing and running-in a molybdenum disulfide dry film lubricant at ambient pressure can generate excessive wear of the lubricant and inaccurate friction values. If a variety of environments are expected in service, a split similar to that recommended in Appendix A should be used. This requirement is not intended to require application of all environments simultaneously.

f. Inspection and performance tests shall be conducted after run-in testing prior to further acceptance testing, and at the conclusion of acceptance testing.

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. Unlike inspections after qualification testing, the post acceptance inspections have to be made without disassembly or other breaking of configuration. At a minimum, one post-test performance test and inspection should be performed at the end of the suite of acceptance 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. Be sure to account for these performance tests when calculating the required design life of the mechanism.

g. Pass-fail criteria, the rational for those criteria and inspection procedures for all acceptance tests shall be established prior to the start of acceptance testing.

Inspection procedures and pass-fail criteria have to be established prior to testing to avoid normalization of anomalous behavior detected during testing. This also helps streamline the decision making and documentation process when anomalies are encountered during the test.

4.24 Mechanism Installation

Mechanisms shall either be designed to preclude installation in an incorrect orientation or be clearly labeled in a manner that indicates proper installation orientation.

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. To prevent this, 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. Drawing notes or other procedural controls that are not present on the hardware are not as effective and are not considered sufficient.

APPENDIX A

BEST PRACTICES FOR MECHANISMS

A.1 Purpose

The purpose of this appendix is to present best practices for the design, development, and testing of aerospace mechanisms. The contents of this appendix do not constitute requirements that have to be followed in order to adhere to this Standard, but rather practices 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.

A.2 Best Practices

A.2.1 Torque and Force Margins

When determining torque or force margins on electromagnetic actuators such as motors or solenoids, one must consider magnetic saturation effects to obtain an accurate calculation of margin in cases where measurement of output at the worst-case operating condition cannot be made. A common mistake is to assume that the relationship between output torque or force and voltage or current is linear throughout the range of operation, which does not account for the non-linear shape of the magnetization curve. Figure 4, Illustration of the Effect of Saturation on Actuator Output, illustrates how the assumption of linear behavior can lead to incorrect output calculations.

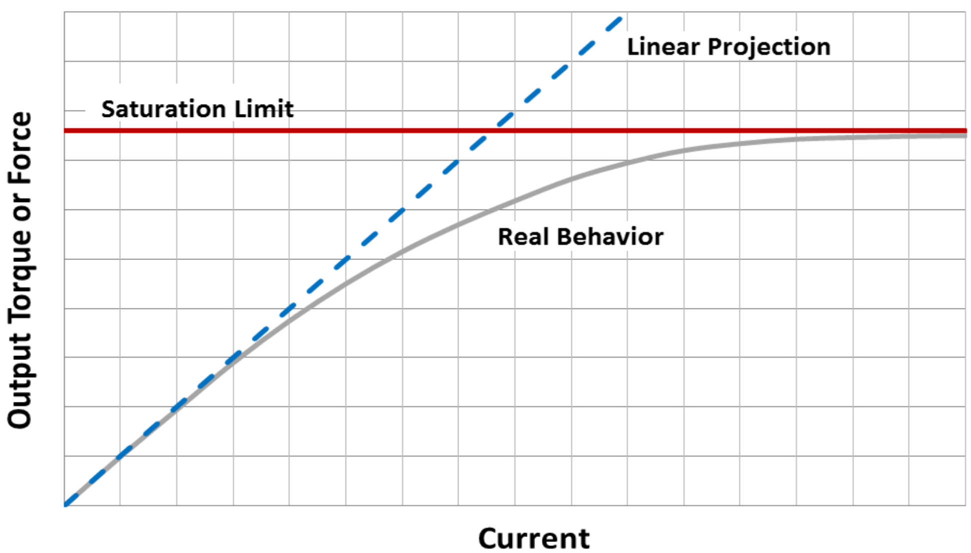

Figure 4—Illustration of the Effect of Saturation on Actuator Output

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.

If it becomes necessary to evaluate energy margin, equation 4-1 may still be used. 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 lifetime of a given application. The best and most direct method for determining lubricant life is by performing life testing in a relevant flight-like environment. Stress cycle analysis is often employed as an empirical approach to quickly assess lubricant life. Bearing life is best predicted by performing stress cycle analysis on the lubricant used in the bearing and is more useful than fatigue life predictions since with current technology it is lubricant life, and not fatigue life, that dictates bearing performance and lifetime.

While there are many lubricants available commercially, few have been qualified in space where vacuum conditions and wide temperature extremes preclude most lubricants from consideration. There are two basic types of lubricants to consider: wet lubricants (i.e., oils and greases) and solid 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 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), although many applications have employed solid lubrication over wet lubrication for various reasons. Care should be taken to specify the proper amount of lubricant to be applied. While the effects of too little lubrication may be obvious, excessive lubrication can also cause problems such as high forces or torques due to viscous effects or reduced clearances. Regardless of the lubrication chosen, metal-to-metal tribological contacts should be composed of dissimilar materials whenever the application allows.

When considering wet lubrication, the selection of grease versus oil is important. The composition of grease consists of a base oil that may contain additives and a thickening agent used to form the bulk. Grease is often preferred to oil because of its ability to maintain an oil supply at a given location versus the tendency of oil to creep and migrate. Grease consistency is affected by the quantity of oil present in a formulation and hence is critical in determining how the lubricant will perform in a given application. A typical consistency for commercially available grease is designated as National Lubricating Grease Institute (NLGI) grade 2, however other consistencies exist. One important point to note is that grease consistency can be adjusted by varying the quantity of oil present in the formulation. NLGI grades higher (stiffer) than NLGI grade 2 can be obtained by using less oil in the grease formulation, while NLGI grades lower (softer) than NLGI grade 2 can be obtained by using more oil in the formulation. To illustrate the importance of this point, consider a high-speed ball bearing application where the high-speed bearing is lubricated with grease. If the consistency of the grease is such that the grease slumps, falling behind each ball as it passes around the bearing raceway, the balls have to continuously perform work on the grease in order to push the grease out of their path. This causes a drag torque and results in elevated heating. If, on the other hand, the grease is stiffer and forms a channel as the balls pass around the bearing, the balls will not encounter large quantities of grease in the track. This scenario would produce low drag and reduced heating. It is not surprising that high-speed bearings typically prefer channeling greases to their non-channeling counterparts. Gears can be problematic applications because there is an appreciable amount of sliding that occurs as the gear teeth mesh. Greases employed on gears need to have proper consistency so that they can adhere to the gear, but can also 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, although NLGI grade 2 is still almost exclusively used in space applications. 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. Unfortunately, testing is often necessary to deduce effects attributed to consistency, but if properly addressed, application performance and lifetime can be maximized.

When grease and oils are employed for lubrication, barrier films are normally applied to adjacent surfaces to retain oil where desired, prevent oil creep, and to protect sensitive areas from contamination. Oils have very low surface tensions as compared to the surface energies of solids, and it is for this reason that oils wet (spread) on solids when applied. Wetting is prevented when contamination or boundary films of lower surface tension than the lubricating oil are present. Barrier films are very low surface tension solids such as perfluoropolymethacrylates that are dispersed or dissolved in a carrier solvent and are typically applied by brushing or dabbing surfaces and allowing the solvent to evaporate. Areas coated with barrier film are rendered unwettable since oils have higher surface tensions than the barrier film. However, barrier coatings may be less effective with perfluoropolyalkylether- (PFPE) based lubricants than with hydrocarbon-based lubricants because of the lower surface tension of PFPEs compared to hydrocarbon lubricants. Barrier films may still provide some creep protection with PFPEs and require case-by-case evaluation.

Any lubricant selected for a given application has the potential to cause harm as a contaminant. All wet lubricants have some volatility that has the potential to contaminate optics or other sensitive components. The consultation of a contamination engineer is strongly encouraged when assessing lubricant volatility. Volatile components can also form friction polymers in undesired locations such as electrical contacts and moving parts, although this is less common with the use of modern low-outgassing materials. 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 lubricants intended for space applications. Contamination arising from oil creep should also be considered. As discussed previously, a common best practice to reduce risks associated with creep is to introduce barrier films at appropriate locations to prevent oil from migrating to sensitive areas while maintaining the oil in the location desired for lubrication performance. Care should also be taken to ensure that the vent paths from volumes containing liquid lubricants do not impinge on critical surfaces.

An additional consideration is contamination of the lubricant itself. During the manufacture and handling of wet lubricants, it is not uncommon to encounter metallic debris and other soft particulate matter. All wet lubrication intended for spaceflight should be filtered to remove contaminants and avoid their introduction into the application during lubrication. Oils should be filtered through 0.10 micron filters and grease should be filtered through 0.25 micron filters. Once filtered, lubricants should be handled and introduced into the application under controlled conditions (e.g., clean room, class 100 flow bench).

The two classes of wet lubricants commonly employed for space applications are hydrocarbons and PFPEs, although other chemistries such as esters have been flown. Synthetic hydrocarbons including Pennzane®, a multiply alkylated cyclopentane (MAC) oil, and polyalphaolefins are typically encountered in modern space applications versus their non-synthetic higher vapor pressure counterparts, although applications persist in which oil reservoirs are employed to provide a long-term supply of oil, such as the International Space Station (ISS) control moment gyroscopes (CMGs), which are lubricated with KG-80 oil. In general, under equivalent conditions, hydrocarbons will outperform PFPEs with respect to operating lifetime for a given application where lifetime is quantified in terms of stress cycles. 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. 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, a well-established degradation pathway for PFPEs.

While each application should be considered independently, there are some general rules of thumb that can be employed when considering lubrication for a given application. Boundary lubrication occurs when there is intimate asperity contact between moving parts. Examples of boundary lubrication applications include gears and bearings operating at slow speeds, sliding contacts, and rollers. These applications typically are best served by employing synthetic hydrocarbons formulated with boundary additives, often referred to as anti-wear additives. Commonly employed boundary additives used in space formulations include phosphate esters of varying composition possessing low vapor pressure to prevent evaporation in a vacuum. Phosphate esters also afford some corrosion protection and are superior to PFPEs in this regard. The higher molecular free volume of PFPEs compared to hydrocarbons renders the PFPE more susceptible to permeation by water and hence corrosion of metallic surfaces. Care should be taken with respect to corrosion when using alloys such as 52100 steel and hardened tool steels such as M-50 and Crucible 20. These alloys are typically best served by coating them with hydrocarbon oils containing corrosion inhibitors.

PFPEs typically suffer from rapid degradation in boundary conditions when oil-soluble anti-wear additives are absent. Molybdenum disulfide is one boundary additive frequently used in PFPE formulations. Polytetrafluoroethylene (PTFE) present as a thickener in PFPE formulation also affords some boundary protection, but experience has proven that it often fails to prevent the PFPE oil from degrading under harsh boundary conditions. PFPEs typically see service when thermal environments to be witnessed by a given application fall below -20 °C. In these situations, hydrocarbons cannot be employed unless heaters are present rendering PFPEs as the only wet option available. One drawback to the use of PFPEs is the tendency of the oil to creep. When compared to a hydrocarbon, the flow activation energy for a PFPE is approximately one half that of a hydrocarbon resulting in about twice the tendency to flow. One advantage to PFPEs is the superior vapor pressure and decreased volatility. As stated earlier, PFPEs are typically less stable in the tribological contact for a given set of stress cycles. PFPEs have a much better viscosity index and are stable at temperatures approaching 200 °C versus their hydrocarbon counterparts that are thermally stable up to about 120 °C. On the cold side, PFPEs have been known to perform down to the -60 °C to -70 °C 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. PFPEs are also known for their compatibility with various elastomers and polymers, often employed on gaskets and o-rings to enhance sealing and prevent binding. MIL-PRF-27617, 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 improved thermal stability, i.e., 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 an active oiler using KG-80 oil.

Solid lubricants include dry-film lubricant (DFL) and other sacrificial solid materials 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-46010, Lubricant, Solid Film, Heat Cured, Corrosion Inhibiting. This specification describes surface preparation, application of lubricant, and associated performance requirements for properly applied and adherent films. When using sputtered and physical chemical vapor deposited (PCVD) forms of DFL, the substrates are typically pre-sputtered in vacuum with argon 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 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-46010 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 in). Burnished (mechanically applied) and physical chemical vapor deposited forms of molybdenum disulfide are thinner 0.25 to 0.5 micron films and are highly crystalline forms possessing reactive edges sensitive to humidity exposure. Pure sputtered molybdenum disulfide coatings 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 possesses a higher coefficient of friction and will be present 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., sodium carbonate) that can impede performance.

Polymer-bound forms of molybdenum disulfide are well known to generate debris as the DFL coating is burnished under the applied loads and cycles witnessed by the application. For this reason, pre-burnishing or run-in is normally performed where the application is subjected to some predetermined number of cycles to orient the lamellar planes of the DFL and the debris that accompanies 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 the run-in, if possible. 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 and PCVD coatings tend to generate a lesser quantity of debris (though the particle count may be higher).

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.

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.

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 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 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 thus have to be used in conjunction with another type of bearing in order 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, 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 max mean Hertzian contact stress allowables noted in table 2 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 environments. A large body of data exists to indicate that operating below this stress level will help avoid lubricant failure for long-life applications in the boundary lubrication regime.

Per the requirements in section 4, 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 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 in light of the truncation. *An Approach to Predicting the Threshold of Damage to an Angular Contact Bearing during Truncation*, (Leveille & 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, and 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, lubricant is pushed out of the loaded region during small oscillatory movements and without large rotational movement of the bearing surfaces in the raceway, there is no method for redistribution of lubricant. 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 bearing has many effects. Bearing preload eliminates free play, reduces runout of the rotating member, increases axial and radial stiffness, prevents fretting damage, reduces impact loading during vibration, increases the load sharing among rotating elements, and prevents ball skidding. Preload also increases friction torque, and increases difficulty in controlling the preload tolerance. As preload increases, lubricant life decreases, wear increases, and the sensitivity of preload to thermal effects increases. All of these considerations have to be balanced when choosing a preload.

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 don’t 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.

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, 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 in order 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 cotton cloth-reinforced phenolic 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.

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.

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.

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 design 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. However, 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.

Lubricants bound in sacrificial cages are released and transferred to the raceways via the balls during operation of the bearing. Run-in is therefore 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, 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 that facilitate sliding motion. Plain bearings are often used in applications where obtaining low 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. Very smooth finishes are desirable for the sliding surfaces in plain bearing applications. Roughnesses in the range of 0.1-0.3μm (4-8 μin) are recommended. Plain bearings are not intended to be used with loose fits and should be pressed into their housings to avoid motion between the bearing and the housing. Attempting to use a loose fit in the housing to provide a measure of redundancy is not recommended. Diametral clearances of 0.0015 mm per millimeter or shaft diameter are typically employed and as such, preloads are not typically employed in plain bearings.

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 contract 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 4.8.m. 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 determinant 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 the excellent white paper “Brush Motors and Brushless Motors - Flight System Design Considerations” (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, brush 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.

Brush 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 in order 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; motors designed for terrestrial use typically use moly/graphite or carbon/metallic compound brushes, but these are not suitable for space applications because they behave much differently in the absence of moisture. In a vacuum or extremely dry operating environment, the graphite acts as an abrasive and quickly wears away the brush or wears through the rotor contacts. Even very low levels of moisture in the environment can improve these wear characteristics, so misleading ground test results are a danger. Metallic-compound 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 provides wear reduction during operation in vacuum while the graphite provides wear reduction during atmospheric testing.

Brush motors can suffer from thermal problems, particularly in vacuum applications, and often need overheating protection algorithms that result in additional cost and development risk.

“Simple” brush motor implementations may get complicated as a result. In addition, brush motors often exhibit poor performance under cold operating conditions and are more susceptible to damage from thermal cycling than brushless motors.

Brush motors require a reversal of polarity to reverse motor direction. This can lead to increased circuit design complexity in order to provide fault tolerance against polarity reversal. It can be difficult to make an accurate determination of input current in brush motors. They are also sources of significant electromagnetic emissions and can cause electromagnetic interference (EMI)/electromagnetic compatibility (EMC) problems.

Though they have significant drawbacks, when the load requirements have been well-defined and consistent and a comprehensive qualification program has been implemented, brush motors have provided reliable service in space flight operations. When assessing heritage of an existing brush 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 of the use and limitations of brush motors.

A.2.4.1.1.2 Electronically Commutated Brushless DC Motor Characteristics

Electronically-commutated BLDC motors can provide high output torque under all conditions, are insensitive to the driven inertia, and provide unpowered holding 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, being limited only by the life of the bearings or lubrication.

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 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 will verify proper end-to-end design of windings and 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 minimum torque positions in a well-designed motor should not have a magnitude less than 20 percent of the high torque positions.

It is worth noting that a BLDC motor driven by a centralized controller may differ in performance between component and integrated system testing, and that the use of distributed drive electronics results in less potential for these types of surprises.

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 lubrication.
On the negative side, stepper motors can be difficult to size properly and their performance is highly dependent on the inertia being driven and the drive circuit. The speed and torque performance limits of stepper motors can be variable, and they do not exhibit linear input current-to-output 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. 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 can sometimes end up oversized for the application.

Stepper motors always operate at maximum input power, regardless of load. The lack of a relationship between input current and output motion can make anomaly investigations difficult. Flight-representative drive electronics are crucial for accurate results during 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 with the motor controller 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 simple sensor.

Stepper motors are used extensively throughout most 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-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, changes in the ratio of driven inertia to the friction torque load and changes in the design of the drive electronics (e.g., a different stepper motor controller from that used on the heritage motor) are the primary concerns.

A.2.4.1.2 Motor Applications

A.2.4.1.2.1 Deployment Applications

Applications categorized here as "deployment" applications encompass more than deployment mechanisms; they are in fact any application for which the motor performs a very simple task: producing a certain torque and moving a load from one point to another and perhaps back again. Precise control of speed is usually not necessary, and speed is usually allowed to vary with the load. The rotary motion is constrained externally through the use of switches, avionics control, and mechanical hardstops. Typical implementations are deployments, instrument covers, and latches. This application is typically the least demanding with respect to flight system cabling and control.

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 controlled angular velocities and accelerations are typically necessary as well. Unpowered holding torque is typically required at each positional state.

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.

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. Often, too little weight is given to the potential developmental difficulties that might arise with the various motor types considered, especially if successful heritage designs exist. This is often the genesis of the conflict that can arise when choosing between a "simple" brush motor versus a "complex" brushless motor. The following evaluations are recommended when selecting a motor for use on a flight program.

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

A brush motor can usually meet the performance requirements for the "deployment" class of applications and some servo class applications though they are certainly not the only choice. BLDC motors and stepper 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 motors already.

Incremental positioning applications are most often filled by stepper motors, as they often provide the lowest overall cost solution despite the need for a motor controller. This is because stepper motors produce incremental rotary motion by nature and can be driven open-loop, which eliminates the need for a motor feedback encoder, resulting in a motor controller that is simpler than a servo controller.

In servo applications, drive electronics and motor feedback devices are required. The actual motor design type selected for the application may be only a secondary cost driver since procurement costs for the motor will typically be a small part of the overall flight system development costs. 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 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 a tachometer to directly sense velocity or a position encoder to derive actual velocity for feedback control.

Stepper motors are not recommended for use in applications requiring minimal 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 debris/foreign object particulate contamination and increased friction or viscosity at lower operating temperatures.

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 load profile, and the life required.

Brush motors are difficult to design to function consistently and reliably in a vacuum and should be discarded as an option if long life vacuum operation is required, but in suitable applications they have been used successfully in a number of spacecraft flight systems. 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.

Brush motors also require accurate replication of expected load cycles since the wear of the brushes (and potential damage to the commutator) is highly dependent on the magnitude and duration of the input electrical as well as the operating environment.

c. Evaluate development costs and risks for each motor type. 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 bigger the application differences, 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 applications very different than space applications. The development risks should not be underestimated. The most likely risks to be realized when implementing brush motors are unexplained excessive brush wear and inadequate torque margin as a result of rotor overheating.

Procurement costs for a commercial brush motor can be quite low. Flight system costs are higher for brushless motors because of the need for drive electronics and encoders. BLDC motors require a rotor position feedback encoder, as do any motors used in servo systems. However, the technical requirements for commutation alone do not require as precise (or complex) an encoder as most servo applications do. Therefore, 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.

BLDC motors can provide the same torque capability as brush motors without the drawbacks of brush motors. One risk unique to BLDC motors is the potential for “torque holes” as discussed in the section on BLDC motor characteristics, which can be mitigated by performing a “torque profile” test.

The use of stepper motors in deployment class applications is sometimes adopted when stepper drive electronics already exist within the flight system. However, as stated previously, sizing stepper motors for such applications can be difficult and their performance is highly dependent on the electrical drive pulse and the driven load. Stepper motors in this application should be able to (and tested to verify that they can) supply output torque above what is necessary to accelerate the driven inertia. Use of flight electronics is critically important as surprise anomalies have occurred during flight system integration testing when component testing had been performed with test-specific electronics.

Distributed drive electronics decouple the flight system from the motor-to-controller interaction, resulting in less risk of harness EMI issues and elimination of “surprises” when transitioning from component to system testing.

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 unit level (e.g., dynamometer testing) and component level testing (e.g., instrument or mechanism). Often, little knowledge about the motor is acquired during flight system tests other than electrical compatibility and simple confirmation of function.

In general, mission risk when implementing brush motors should be considered moderate to high due to the inherent variability in brush motor life, difficulty in accurately characterizing and predicting in-flight performance, and extreme sensitivity 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 brushless motors have to be qualified using flight-representative drive electronics, and historically this has been a problem with stepper motors. The unavailability of flight electronics at the time of motor qualification or acceptance testing has repeatedly caused in-flight performance problems. Implementation of electronically commutated 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 unit qualification, component testing, system testing, and flight.

A.2.4.2 Stepper Motor Performance Analysis

The discrete nature of the motor stepping drives a stepper motor rotor to behave like a magnetic spring-damper system. In general, this drives a need for analysis of the rotor position stability in response to the step commanding. This stability is influenced by a wide variety of motor, driver and load parameters. Such an analysis can be evaluated with a Monte Carlo simulation of the combinations of parameter values or with a worst-on-worst deterministic analysis. In order to envelope all conditions which affect output torque and synchronicity, the following parameters should be considered in the analysis:

- Motor inductance variations due to manufacturing tolerances.
- Motor resistance 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-to-step variations in step angle size.
- Step-to-step variations in unpowered detent torque and detent.
- Voltage constant variations due to manufacturing tolerances and thermal conditions.
- Powered holding torque variations due to manufacturing tolerances, thermal conditions, and current tolerances.

- Rotor damping variations due to manufacturing tolerances, thermal conditions, and changes over life.

- Variations in viscous damping of motor bearings, lubricant, and magnetic structure 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 variations due to manufacturing tolerances.

- Motor rotor-to-gear train deadband variations due to manufacturing tolerances and thermal conditions.

- Variations in gear train inertia, stiffness, and deadband due to manufacturing tolerances.
- Variations in gear train friction and damping due to thermal conditions and aging.

- Gear train torque variation such as transmission error or harmonic drive two-cycle torque variation.

- Load inertia variations due to manufacturing tolerances.

- Modal properties of the load.

- Stop stiffness and variations due to manufacturing tolerances.

- Modal properties of the base including base motion disturbances.

- Test conditions, test equipment effects, and "one g" effects.

In cases where the magnetic holding capability of a stepper motor is used to hold a load in position, the margin should be evaluated using the detent torque at each discrete rotor position for both the clockwise and counterclockwise direction. If the holding capability of the magnetic detent torque of the stepper motor is marginal, trickle current may be applied to the motor winding when it is not energized. The trickle current should be disconnected when the motor is energized to reduce its effect on running torque.

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 locked-rotor torque is measured while the rotor is moving in the direction of motor phasing at less than one revolution per minute with nominal operating voltage applied to the motor drive electronics. This test is performed in both commutated 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 tension and compression springs, torsion springs, Belleville washers, 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, so it is as important to adhere to good design practice with springs as it is with any other mechanism component.

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, in order 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 springs to be fixed to prevent rotation as well as enclosure or other lateral restraint of the springs (which also prevents buckling under nominal performance conditions).

Helical compression and tension springs should be designed to develop a maximum shear stress of no more than 80 percent of the allowable shear yield strength. This reduces the possibility of a reduction of potential energy due to stress relaxation. A similar margin should be considered for other types of springs.

Springs should be designed or selected using the factors of safety (FS) shown in table 3, Factors of Safety for Springs, at the maximum operating stress.

Table 3—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

Helical compression springs should have closed and ground coils for interfaces. 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.

If torsion springs are used, they should be under load in the direction of winding (i.e. the load should tend to further close the winding) at all times.
Temperature can affect the stiffness of the spring and should always be considered in the design or selection of a spring.

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 have a comprehensive fatigue analysis or test performed on them.

A.2.6 Gears

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

Hunting tooth gear ratios should be used to distribute wear. A hunting tooth gear ratio is one in which the number of teeth on the driven and driving gears is selected so that the same two teeth do not mesh with each revolution of the larger gears. The number of teeth on each gear should be selected, within the limits of the gear ratio requirement, to maximize the number of revolutions before meshing of the same two teeth.

Undercutting of spur gears should be avoided. Spur gear designs that have greater recess action than approach action are preferred. Spur gear contact ratios should be greater than 1.4 for power transmission gearing. Cantilever gear shaft mounting should be avoided to reduce non-uniform load distribution across the face of the teeth. Gear tooth wear patterns should be checked after first assembly to establish that the pattern is well-centered over the tooth flank, and that edge loading is not present.

Aluminum gears should never be considered except in light-duty, limited-life applications where tooth wear and the coefficient of thermal expansion can be accommodated, and where compatibility with the selected lubricant can be established. An anodization process may be used to improve wear resistance for acceptable aluminum gear applications, provided that the contact stresses will not cause the coating to crack.

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 Harmonic Drives

Harmonic drives should never be subjected to a dedoidal condition. A dedoidal condition exists when the flexspline is not concentrically engaged with the circular spline. Harmonic drives are known to have harmonic frequency errors which exhibit themselves by imparting a disturbance torque when driven by perfectly smooth input shaft speeds. This disturbance torque can couple into the driven load torsional resonance, especially when the input is provided by a stepper motor. Control of a mechanism containing a harmonic drive should avoid stepping at a rate that will couple with the load resonances. The gear error frequency at the output is defined as

$$f_{error} = \frac{Step\ Rate\ \cdot Step\ Size\ \cdot N}{360}\ Hz$$

where f_{error} is the frequency of the harmonic drive output gear error, *Step Rate* is the motor step rate in pulses per second, *Step Size* is the stepper motor rotor step angle (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 fasteners in space mechanisms should adhere to the requirements and practices in NASA-STD-5020.

A.2.8 Quick Release Pins

A.2.8.1 Quick Release Pin History

Quick release pins, also known as PIP (push in and pull) pins are pins with fast-acting retention and release mechanisms built into the pin. 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 central shaft to actuate one or more retention balls, which are held in via swages or staking of the housing material around the balls. Figure 5, Diagram of a Typical Quick Release Pin, depicts a cross-section of a representative quick release pin, in this case a double-acting pin, that illustrates the principle.

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

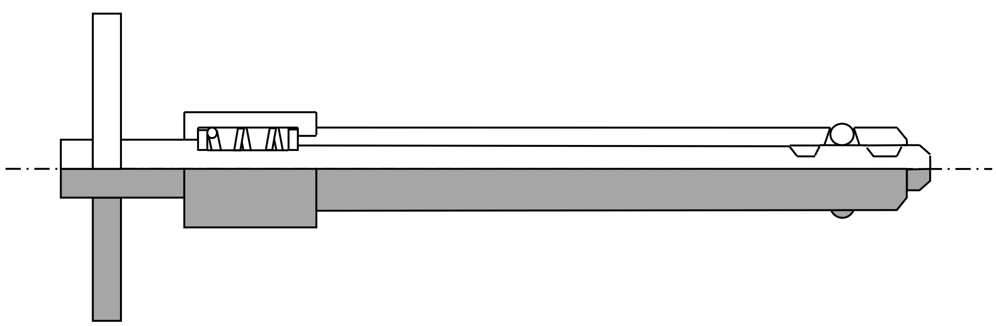

Figure 5—Diagram of a Typical Quick Release Pin

A.2.8.2 Quick Release Pin Failures in the Space Program

Though several documented inadvertent releases of 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 quick release pins. During vibration testing, several locking balls in the pins vibrated out of their sockets, and during cold temperature vacuum testing the lubricant in the pins froze and seized the pins. NASA solved these problems by using military standard pins that were quality controlled and removing all lubrication from the pins based on their single mission use and because the lubrication was mainly provided for corrosion protection in the first place.
After this failure, several design changes were proposed for quick release pins to create a more reliable pin for space use:

- Use of four retention balls instead of two to improve retention in the event of a single ball release.
- Double-action.
- PTFE-coated tethers.
- Welded handles and tether rings.
- Addition of dry film lubricants.
- Addition of hitch pin.

More detail is provided in *PIP Pin Reliability and Design* (Skyles, 1994).

The addition of hitch pins, which are self-retained shear pins that are placed through both the interior shaft and exterior housing to prevent inadvertent actuation of the pin and act as a secondary means of retention in the event of ball release, would turn out to be controversial. Hitch pins proved to be difficult to install with gloved hands and presented snag hazards during EVAs. Several instances of inadvertently pulled hitch pins 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 quick release pins had to be treated as mechanisms in their own right, requiring the same engineering rigor and review practices as other mechanisms. A continuing history of problems led to the revision of the policy that prohibited quick release pins from being used in zero-fault-tolerant applications in 2000.

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 pins; others with planned uses were flown with a system of hook-and-loop flags to secure the pins in the event of a structural failure.

Vibration testing of the same pins uncovered a defect that caused the retaining balls to stick in the extended 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 staking under vibration loads. An acceptance test procedure was developed to screen for this failure mode. 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 pins. The manufacturer also performed additional verification including extra dimensional inspections and staking tests on 100 percent of the pins. All subsequent qualification and acceptance testing (thermal, random vibration, and “stick-ball” tests) was successful.

In 2004, a double-acting pin on the ISS mobile transporter rail was found fractured prior to flight due to a ductile overload from a suspected incidental impact. The investigation revealed a deficient design of the head 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 head and other parts of the pin were redesigned by the prime contractor and the pin manufacturer to address the deficiencies and add random vibration testing 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 the ISS.

A.2.8.3 Current Quick Release Pin Best Practices

Given this history, quick release pins are not recommended for applications that control hazards, and should never be used in critical applications in which they experience axial loads. Properly designed and constructed pins can be used successfully in non-critical shear applications. Even with the most up-to-date pin designs, four failure modes have to be addressed when considering their use, as follows:

a. Loss of locking balls.
b. Failure of shank.
c. Failure of head.
d. Structural integrity under load.

Due to the arrangement of the balls and the subsequent need to use swaging or staking around them, loss of locking balls is nearly impossible to eliminate with design and manufacturing. The usual control is to provide back-up retention feature such as a hitch pin 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 head can be eliminated but depend on the pin vendors since they rely on design practices and process controls. A continuing risk is that 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 prior to the use of any pin. Lastly, quick release pin structural integrity in a particular application has to be assessed; due to the internal features of the pin, dependable allowables are not always available, are dependent upon application, and will not cover situations such as inadvertent contact loads.

A.2.9 Inspection

Inspection of both constituent parts and assembled mechanisms is an important part of the testing and preparation of flight mechanisms. Some materials used are more susceptible to material defects than others and parts made from these materials should be carefully inspected prior to acceptance. These materials include castings, fiber composites, laminates, and honeycombs. Regardless of material, any part that represents a single-point failure should be inspected for defects. This is often mandated by fracture control requirements if fracture control is utilized. Careful examination should also be made of any parts that will be highly stressed.

Springs, due to their mass-produced nature, are frequently affected by initial flaws. Improper heat treating is a common cause; for this reason, springs should have their heat treat tested on an unstressed location whenever feasible. In cases where this can't be done, heat treat evaluation via lot sampling may be useful.

Inspection of assemblies should be performed both before and after to 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 drawing requirement. Some recommended inspections are listed below.

- Cleanliness.
- Handling damage.
- Corrosion.
- Critical clearances and dimensions (especially of wire harnesses and multilayer insulation).
- 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 lock wire or safety cable.

- Condition of electrical and fluid connectors.
- Wiring harness retention.
- Spring integrity.
- Leakage of fluid-filled components.
- Excessive wear.
- Lubricant condition and availability.
- Evidence of unintentional contact between parts.
- Generated debris.

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, 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, in order 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.

A.2.10 Qualification Testing

Qualification testing is conducted to verify that 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 margins exist. The 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 in order 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 any resident firmware and software, test fixturing, test facilities, and test support equipment.

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 applicable item's service life, which includes acceptance tests (including acceptance retesting), tests at higher levels of assembly, mission operations, and ground operations. Qualification tests should be performed for each mechanical 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 analytic 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 most adverse 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 4, Recommended Environmental Qualification Test Matrix for Mechanisms. Design life testing is not included in this list because it is treated separately from other qualification tests in this document. 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 5, 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 mission life.

Table 4—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 5—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 should be provided to the extent necessary to achieve the test objectives. As a guide, the uncompensated gravity effects should be less than 10 percent of the operational loads. 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.

For testing of certain mechanical functions, however, more stringent uncompensated gravity constraints may be required.

In order 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.2.10.1 Design 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 design 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 design life test should reflect that. The pressure during operational cycles is usually much better understood, so if the hardware will experience vacuum or reduced atmospheric pressure during a portion of its design life and the mechanism contains vacuum-sensitive components, a representative fraction of the total cycles should be performed in the appropriate pressure conditions. Vacuum-sensitive components typically include lubricants and material combinations susceptible to cold welding. 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.

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.

Design 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 lubricant breakdown, not fatigue. A bearing that normally operates in a boundary lubrication or mixed lubrication regime should never be accelerated to a level where the bearing operates in the EHD lubrication regime during the test. In the EHD regime, no appreciable wear should occur and the failure mechanism should be material fatigue rather than wear. 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 retainer instability which may produce excessive wear of the retainer 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 gradients, which should be thoroughly evaluated. Even performing the test at flight speeds can mask failure modes when the quiescent durations are not included in the life test—continuous cycling can 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 on-orbit 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.

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 a vacuum on the mechanism in question need to be considered when selecting the life test environment.

As stated in the rationale for inspections of design life test hardware, a thorough inspection of the life test unit is necessary in order 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. Photographic documentation of the life test article should be made from component inspection and acceptance through full assembly to act as a baseline for comparison. The critical areas of parts that may be subject to fatigue failure should be inspected to determine whether failure has occurred. Where lubrication is used, it may be prudent to measure lubricant loss, degradation, distribution, and condensed outgassed constituents.

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.

The life test factor utilized in this document 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 this 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.

A.2.11 Acceptance Testing

Acceptance testing is used to verify that the manufacturing and assembly process has been accomplished in an acceptable manner and that the as-built unit performs within specified parameters. Acceptance testing includes integrity tests that verify hardware functions during and after exposure to the specified environments (e.g., performance tests).

The types of acceptance testing required for mechanisms are typically established by the program, but a recommended set of tests is given in table 6, 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 document but note that as required in section 4, the run-in test is to be performed prior to all other acceptance testing. 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 5; 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 in order to confirm proper performance and to ensure that no degradation has occurred during the previous tests.

Table 6—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 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.

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, in order 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 for which no refurbishment is possible, such as ordnance. Acceptance tests have to 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 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.

A.2.12 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

The purpose of this appendix is to provide additional information on references mentioned in the text of this Standard.

B.2 References

B.2.1 Government Documents

Department of Defense

Document Number	Document Title
MIL-PRF-27617G	Grease, Aircraft and Instrument, Fuel and Oxidizer Resistant, Amendment 1
MIL-PRF-46010H	Lubricant, Solid Film, Heat Cured, Corrosion Inhibiting

NASA

Document Number	Document Title
PD-ED-1229	Preferred Reliability Practice, Selection of Electric Motors for Aerospace Applications
NASA-RP-1228	Fastener Design Manual
NASA-SP-38	Advanced Bearing Technology
NASA-STD-5020	Requirements for Threaded Fastening Systems in Spaceflight Hardware
NASA-STD-6016	Standard Materials and Processes Requirements for Spacecraft
NASA/TP-1999-206988	NASA Space Mechanisms Handbook

B.2.2 Non-Government Documents

ASTM International

Document Number	Document Title
E595	Standard Test Method for Total Mass Loss and Collected Volatile Condensable Materials from Outgassing in a Vacuum Environment

Other Documents

Leveille, A.; Murphy, J. (1973). *Determination of the Influence of Static Loads on the Output Torque of Instrument Ball Bearings.* Paper presented at the International Ball Bearing Symposium. The Charles Stark Draper Laboratory, Inc.: Cambridge, MA.

Loewenthal, Stuart H. (1988). *Two Gimbal Bearing Case Studies: Some Lessons Learned.* Proceedings of the 22nd Aerospace Mechanisms Symposium. Mechanisms Education Association: Hampton, VA, 253-269.

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Schroeder, J. R. (2010). *Demystifying the 2:1 Ratio and the Stick-Slip Phenomenon: A Technical Whitepaper Explaining the Theory Behind the Binding Ratio and How It Relates to Stick-Slip.* PBC Linear, a division of Pacific Bearing Company: Rockford, IL.

Sevilla, D. (2010). *Brush Motors and Brushless Motors - Flight System Design Considerations*. White Paper, Jet Propulsion Laboratory: Pasadena, CA.

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