22 Spacecraft Subsystems V—Structures and Thermal

22.1 Spacecraft Structures and Mechanisms

Motors

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Motors are used in the majority of cases where repetitive, high-cycle, or constant motion over long duration is desired. They are also frequently used in single-cycle applications where high reliability and high torque margins are essential. With the proper reduction and feedback, motors are capable of achieving tremendous positioning accuracy and repeatability (arcminute accuracy ranges are now common). In comparison to other prime movers, motors offer:

- High torque output-to-size ratio
- Excellent mechanical reliability
- Long life (but bearing life is an issue)
- Ability to achieve high accuracies and repeatability of position
- High accelerations and high operating speeds
- Continuous duty cycles
- Operation in two directions
- Continuous stroke—no need to reset

Motors do require control electronics to operate. There will always be some development to match the application to its environment and operational scenario. This is one area to invest the time and energy in managing the development and verification.

In general, Brushless DC and Stepping motors are used for the majority of spacecraft motor applications. (See Table 22web-1.)

Table 22web-1. Brushless DC Motor Comparison to Stepper Motor.

Brushless Motor	Stepper Motor
Good for Power Limited Applications	Small angle steps recommended
Most Efficient	Harmonic drives typical
Feedback—Closed Loop	100–200 gear reduction typical
Smoother Operation (minimizes jitter)	Feedback—encoder and count steps
	Cardinal and/or micro-stepping
	Simpler control electronics—easier to develop

Motor Design Considerations

In selecting a motor for space flight applications, the following characteristics should be considered at the preliminary design level:

Specification: Also known as the Source Control Document, this is a collection of performance requirements, size and interface information, quality assurance requirements and expected deliverable items that will be submitted to the motor supplier as part of the purchase order. Development of the motor specification occurs after the type of motor has been selected.

Sizing: "Sizing" a motor refers not only to enveloping its physical volume, but to specifying the parameters which will provide the appropriate torque and speed characteristics. Generally, motor manufacturers publish

Table 22web-2. Commonly Used Motor Performance Parameters.

Description	Symbol	English Units	SI Units
Nominal Operating Voltage	V_{nom}	V	V
Motor Constant	K_m	$\frac{in-oz}{\sqrt{Watt}}$	$\frac{N-m}{\sqrt{Watt}}$
Torque Constant	K_T	$\frac{in-oz}{amp}$	$\frac{N-m}{amp}$ or $\frac{mN-m}{amp}$
Speed Constant	K_N	$\frac{RPM}{V}$	$\frac{rad}{V-s}$
Resistance, Each Phase	R	W	W
Rotor Inertia	J_M	in-oz-s^2	kg-m^2 or kg-mm^2
No-load Speed	n_0	RPM	RPM
No-load Current	I_0	Amps	Amps
Gearhead Efficiency	h	Unitless	Unitless

common motor performance parameters to permit comparison of different motors as shown in Table 22web-2. These parameters usually provide enough information for preliminary motor performance calculations. Generally, at the outset of the motor selection process, a rough idea of the available space is known, so that a size limit can be established. Also, the motion profile is typically known, or at least inferred, so preliminary starting, stopping, and running torques can be estimated. Finally, the available power is usually known or estimated. The mechanism designer's task is to select a motor within the size envelope, and not exceeding the available power, whose performance characteristics bracket the motion profile with adequate torque margin.

Conversely, if a custom motor is to be specified, the mechanism designer uses the motion profile and the available power (torque, speed, voltage, and current) to establish a range of acceptable motor constants and torque constants, which are then provided to the motor supplier; the supplier submits one or more proposed motor designs that fall within the design range and are within the limits of the supplier's ability to actually manufacture.

Case size rule of thumb: a standard high-performance BLDC motor can be expected to produce roughly 0.1 in-lbf of stall torque, at the armature (prior to the gearhead), per amp, for every inch diameter of case size (0.1 in-lbf/amp-inch). Thus, a 1 inch diameter motor with 10 amps applied to the windings can be expected to produce roughly 1 in-lbf torque. This figure is related to case length as well. A pancake motor will produce less armature torque for the same outer diameter, as it does not possess as much coil length as a "standard" form factor.

Comparison of Performance Characteristics

Assuming that the nominal supply voltage is known, the motor and torque constants, as well as the maximum continuous torque and the maximum intermittent torque can be compared to catalog values to narrow the field of acceptable motors. If catalog values are not sufficient, the mechanism designer can supply target motor constant, torque constant, resistance, loaded speed, torque-at-speed, stall torque and nominal voltage values to the vendor, who will iterate to see if a non-catalog option can be manufactured. All else being even, it is desirable to maximize the motor constant and torque constant—a high motor and torque constant indicate that the motor produces torque more efficiently, i.e., with less current and therefore fewer thermal waste. To further refine the selection, there are some other useful comparisons that can be made.

For a candidate motor/reduction combination, the speed at a known load can be calculated from

$$n_{LOAD} = (1/R_G)(K_N/K_T)(R_{MOTOR})[T_{LOAD}/(\eta_{GEARTRAIN}R_G)] \qquad (22web-1)$$

In the case of FireSat II, $T_{LOAD} = T_{MAX}$. The loaded speed value is compared to the required operating speed, in this case, n_{ARRAY}, and should be at least the desired value. The value $(K_N/K_T)(R_{MOTOR})$ is equal to the slope of the speed-torque curve for the motor, and is referred to as the speed/torque gradient—this is often published as an explicit catalog value for candidate motors.

Based on knowledge of the required loaded speed, a theoretical no-load speed for a motor can be calculated from

$$n_{0,\mathrm{theor}} = n_{LOAD}R_G + (K_N/K_T)(R_{MOTOR})[T_{LOAD}/(\eta_{GEARTRAIN}R_G)] \qquad (22web-2)$$

Where for the case of the FireSat II SADA, $n_{LOAD} = n_{ARRAY}$ in RPM, $K_T = K_{T\_min}$ in N-m/Amp, $K_N = 1/K_{T\_min}$ in RPM/V, and $T_{LOAD} = T_{MAX}$ in N-m. Resistance values can be estimated—typical winding resistances range from a few tenths of an Ohm to several Ohms. The theoretical no-load speed can be compared to catalog values for candidate motors. Candidates should have a no-load speed higher than $n_{0,theor}$.

Finally, a theoretical motor speed constant, K_N in RPM/V, can be calculated from

$$K_{N,theor} = n_{0,theor}/V_{nom} \qquad (22web-3)$$

and used to compare to catalog values of K_N or to refine the calculation for $n_{0,theor}$. Candidate motors should have a speed constant close to or greater than $K_{N,theor}$.

Designing a custom reduction drive, and custom placement of the feedback device, can permit tremendous flexibility in component placement and lead to a truly mass and space optimized design, but the program must possess the financial and schedule resources to perform such a task. This is a system-level trade. Planetary and spur reduction drives are designs that can be tackled within the constraints of some programs; custom harmonic drives are not recommended; they are, with almost no exception, procured from a vendor specializing in such drives.

Motors and Torque Margin

The motor and torque constants are based on marginless calculations for required torque. Therefore, in order to establish adequate torque margin, look for motors having

- K_M higher than K_{M_min}
- K_T higher than K_{T_min}
- n_{LOAD} higher than n_{ARRAY}
- Stall torque higher than the required torque $T_{armature\_max}$ by a significant factor of safety (1.5 to 3, depending on the program)
- Load-speed ratings (here, given by $T_{armature\_max}$ and $n_{ARRAY}R_g$) that fall within the 100% duty cycle rating for the motor, for those periods when the array will be driven continuously.
- Inertia of the payload reflected back through the geartrain to the armature is not more than a few times greater than the rotor inertia—this, along with required array slewing speed, assists in selection of the appropriate gear ratio for the SADA.

Initial selection of the reduction ratio is roughly based on comparing the candidate motor's no-load speed to the desired maximum array slewing speed, n_{ARRAY}. This establishes a ballpark for the ratio. Candidate motors that meet all other requirements will generally have no-load speeds that are similar, making the ROM estimate simple. However, there is a trade to be weighed during selection of the reduction ratio. A high gear ratio increases torque margin and makes external torque disturbances more transparent to the motor. The motor operates at a higher speed—closer to its no-load speed, so the required torque is less and therefore power and temperature are reduced. However, bearing life is also reduced. Multiple stages in high-ratio drives add volume and mass to the system. A single stage, low-ratio reduction drive is more mass-efficient, but requires the motor to work harder to perform the task—consuming more power and requiring more dissipation of heat. It goes without saying that the mechanism designer should seek a highly efficient motor with a large stall torque/case diameter ratio, coupled with a highly efficient gearhead.

In the end, the designer must ensure that adequate torque margin exists to drive the mechanism. Torque margin helps account for variations in torque due to speed and temperature, as well as losses that are very difficult to model mathematically. Also, adequate torque margin ensures that the mechanism has, in some cases, sufficient torque authority to drive through jams or other torque anomalies.

Torque margin, T_{margin}, should be greater than zero, and is calculated from

$$T_{margin} = T_{available}/(T_{known}F.S._{known} + T_{variable}F.S._{variable}) - 1 \qquad \text{(22web-4)}$$

Where $T_{available}$ is the available output torque at the prime mover; stall torque is often used, though a more conservative value for the SADA would be available torque at the known speed n_{array} (which, reflected to the armature would be equal to $n_{armature} = n_{array}R_g$.

T_{known} is (are) the maximum required torque(s) that are "known," such as inertial torques.

FS_{known} is the factor of safety applied to known torques, often ranging from 1.5 to 2.

$T_{variable}$ are the worst case estimates of torques that are difficult to characterize, such as those due to friction, cogging, ripple, or thermally varying torques. In the case of the SADA, the slip ring running torque would fall into this category.

$FS_{variable}$ is the factor of safety applied to "unknown" torques, and is consequently given a higher value, often 2 to 4.

Single Axis Solar Array Drive Assembly (SADA) Sizing Example

In Sec. 14.7.2, one of the options we discussed for the FireSat II solar array configuration was a single-axis gimbal (see Table 14-28). In this example, we will size the mechanism for that option. The one axis of rotation, naturally, must be around the pitch axis. For the FireSat II operational orbit of 700 km, we can determine the average pitch rate using Eq. (9-17):

$$n = \sqrt{\frac{\mu}{a^2}} = 0.00106\,\frac{\text{rad}}{\text{s}} = 0.061\,\frac{\text{deg}}{\text{s}} \qquad \text{(22web-5)}$$

We will assume a constant rate for the circular orbit.

For a more detailed iteration, a spacecraft attitude profile should be provided to the mechanism engineer by the orbit dynamicist or GNC team. The point is that the job of the SADA mechanism engineer really begins when the spacecraft body pitch position profile is established.

As an example, we assume a single motor, since this is a low-cost system. Life, stiffness, and slewing speed will factor into whether a harmonic drive or gear reduction is used. Generally, the simplest solution is to procure a turnkey SADA that includes integrated reduction, feedback and slip rings.

Assume for FireSat II that each SADA (two total) will be an integrated unit containing a single motor, reducer, feedback device and slip ring. Assume for the first sizing iteration a stepper motor with an integrated optical encoder; a brushless DC motor would be reserved for the case that no stepper can be found that satisfies the pointing, torque and speed requirements. Generally, the calculations apply to both steppers and BLDC motors, so other options such as case size, power consumption and mass/area of motor drive electronics PWA's can be compared.

The most simplistic approach to sizing the motor is to examine the torque and speed requirements and select a motor/gearhead that can supply the required parameters at the available voltage provided by the spacecraft. For first-order estimation, assume a 100% duty cycle for the maximum torque that the motor will have to generate; this will provide an enveloping motor performance specification. Also, for first-order estimation, assume a truly linear speed-torque curve for the motor. This is actually a conservative assumption; under a PWM input, for a constant voltage, the speed-torque curve is more rectangular in shape—relatively flat speed available out to some threshold torque, after which the loaded speed drops off quickly to the stall torque.

Load Torque Estimation

Maximum load torque is the sum of the torque due to maximum acceleration of the solar array and frictional losses in the geartrain and slip rings, which will be assumed to be constant for first-order estimation. In reality, geartrain frictional torque will vary significantly with temperature if it is wet-lubricated with grease, as the viscosity increase with colder temperatures will cause parasitic torque to increase. The designer must ensure that an appropriate grease is chosen that is capable of operating in the expected temperature range. See Sec. 22.6.1.4 for details on lubrication.

$$T_{max} = I_{yy\_array}\alpha_{SADA\_MAX} + T_{SLIP\_RING} + T_{FRICTION} \qquad \text{(22web-6)}$$

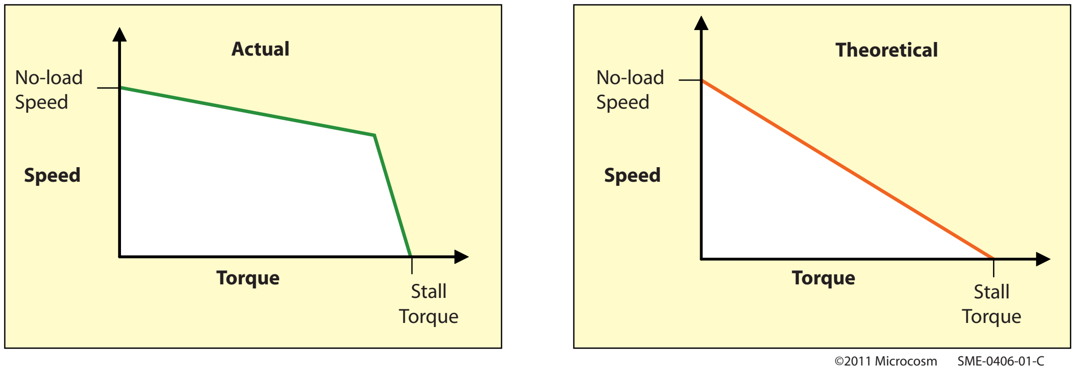

Fig. 22web-1. Torque-Speed Curve for a BLDC Motor.

This is the required torque at the output of the SADA—that is, at the output shaft of the reduction drive, not at the armature of the motor. That torque is reflected back to the armature of the motor through the ratio of the reduction drive. $T_{FRICTION}$ covers miscellaneous small losses, such as running torque in the output bearings of the SADA and shaft seal drag torque, if seals are used in the assembly.

To calculate inertia of each array, assume the single-axis design for FireSat II with 1.4 m^2 total array area. If half of the area is used by each array, and 24 inch wide by 0.5 inch thick panels are used, then the mass moment of inertia about the solar array drive axis is

$$I_{yy\_array} = \left(\frac{m_{array}}{12}\right)\left(w_{array}{}^2 + t_{array}{}^2\right) \quad \text{(22web-7)}$$

Using 2.8 kg/m^2 to estimate panel mass, the mass of the arrays—panels only—is 3.92 kg. Double the mass to account for hinges, launch lock seats, fasteners, and other mounting structure, then halve the mass to estimate the per-array mass—3.92 kg per array. Using these values gives I_{yy_array} = 0.121 kg-m^2.

For a well-designed precision mechanism, frictional torque will be low, on the order of 10^{-2} N-m; estimate the slip ring running torque to be a maximum 0.113 N-m (1.0 in-lbf) which can be acquired from vendor discussions or catalogs. Then:

$$T_{max} = (0.121 \text{ kg m}^2)(1.66 \times 10^{-6} \text{ rad/s}^2) + 0.113 \text{ N-m} = 0.113 \text{ N-m} \quad \text{(22web-8)}$$

Essentially, the torque required to accelerate and decelerate the array is negligible compared to the frictional torque in the slip ring. Since frictional losses are always present, the duty cycle for this torque load will be 100%.

$$T_{armature\_max} = I_{armature}\alpha_{armature} + T_{max}/\eta_{GEARTRAIN} R_g + T_{RIPPLE} + T_{COGGING} + T_{BEARINGS} + T_{DETENT} + T_{ENCODER} \cdots \quad \text{(22web-9)}$$

The inertial torque due to acceleration of the armature itself is usually very small and for first-order estimation can be ignored. Assume cogging, ripple, bearing, hysteresis and detent torques are also small. The main torques, then, that the armature needs to overcome are:

1. The inertial acceleration of the solar array and drag on the slip ring reflected back through the geartrain, and
2. The running torque of the geartrain itself.

If the geartrain losses are known, then this calculation is easier—typically, however, running torque in a geartrain will be highly dependent on the ratio and construction of the drive, and it requires actual measurement from the back of the motor armature. Geartrain torque losses can be estimated by choosing (conservatively) an efficiency of the geartrain, $\eta_{GEARTRAIN}$, and the gear ratio R_g. The motor, then, roughly, needs to at least be able to generate, at its armature,

$$T_{armature\_max} = T_{max}/\eta_{GEARTRAIN} R_g \quad \text{(22web-10)}$$

at the supplied input voltage. Gearhead efficiency will decrease with decreasing temperature and speed. Worst-case assumptions for efficiency should be used, or the running friction of each stage should be calculated for a more accurate value.

Assume for the FireSat II SADA, a gearhead efficiency of 65% with a ratio of 300:1. The efficiency can be gathered from a vendor, but 65% is nominal estimate. The gear ratio is really determined by the step size required by the system. A nominal motor might be a 3 deg step or angle. The 300:1 ratio will drive the step size down to 0.01 deg. Ratios of 200:1 or 300:1 are common. These gear reducers are usually provided by the motor manufacturer and are paired with the motor systems. Getting a generic mass estimate is difficult. Typically, one will specify the performance characteristics and then seek out vendors for solutions. The armature torque that the motor will need to maintain at a 100% duty cycle becomes:

$$\begin{aligned} T_{armature\_max} &= T_{max}/\eta_{GEARTRAIN} R_g \\ &= (0.113 \text{ N-m})/(0.65 \times 300) \\ &= 5.8 \text{ x } 10^{-4} \text{ N-m} \end{aligned} \quad \text{(22web-11)}$$

For this example, this value is compared to the maximum continuous-duty torque rating for the motor, as it represents the torque to overcome during continuous slewing (recall that start and stop accelerations were negligible compared to drag torque). For high-torque applications where start and stop inertial accelerations are much higher, the maximum torque would be compared to the stall torque of the motor rather than the continuous-duty torque. An even more accurate torque comparison is performed by devising the entire cyclic torque profile, calculating an RMS torque value, and comparing the RMS torque to the maximum continuous-duty torque rating for the motor.

Power Estimation

The maximum power required is the work done in rotating the solar array, plus the I^2R losses in the windings of the motor, where I_{MOTOR_MAX} is the current required to produce $T_{armature\_max}$ and R_{motor} is the winding resistance:

$$P_{MAX\_SADA} = T_{max}\omega_{array} + I_{MOTOR\_MAX}{}^2R_{motor} \tag{22web-12}$$

The motor torque constant, K_T, is equal to the torque produced per amp at the armature,

$$K_T = T_{armature}/I_{motor} \tag{22web-13}$$

which is expressed in N-m/Amp. Note that torque constant is equal to the back EMF constant K_B and is the inverse of the speed constant K_N (expressed in rad/s-V).

The actual maximum motor current drawn can be calculated from a known torque constant and the known maximum armature torque:

$$I_{MOTOR\_MAX} = T_{armature\_max}/K_T \tag{22web-14}$$

Substituting into the power equation,

$$P_{MAX\_SADA} = T_{max}\omega_{array} + [T_{max}/(\eta_{GEARTRAIN}R_gK_T)]^2(R_{motor}) \tag{22web-15}$$

Knowing T_{max} and ω_{array}, which is set to the orbit angular rate, catalog values for K_T, R_{motor}, $\eta_{GEARTRAIN}$ and R_g can be used to calculate the power consumed—which can be compared to the available power for the SADA subsystem. These estimates are preliminary only—they do not take into account thermal effects on the motor resistance, for example.

Torque Margin

This being a preliminary sizing exercise, details such as backlash and step accuracy of the motor are not considered here. It is assumed that the solar arrays are forgiving, i.e. their pointing requirements will fall well within the default capability of any well-engineered stepper motor (or brushless DC motor) drive system.

Estimating SADA Mass

Typically, given the target motor performance characteristics, the vendor will provide the engineer with a first-order mass estimate based on similar drives that they have manufactured. Generic mass estimates are difficult for motor systems. Masses are very dependent upon the vendor and their suite of hardware. A quick request to industry can produce rough order of magnitude mass estimate.

22.6.1.2 Stored Energy Devices

A stored energy device is any mechanical or structural component that can change its shape, store that change as strain energy, and then convert that energy to work at some later time, returning its shape toward the zero state.

Springs & Flexures

Undoubtedly, the most common stored energy devices are springs, and to a lesser extent, flexures, which are essentially leaf springs with customized cross-sections. All springs and flexures operate on the principle that a beam in bending or torsion absorbs strain energy when *elastically* deformed, and can release that energy when allowed to return to its free state. Springs and flexures are so commonly used, in part, because their performance is for all intents and purposes linear, with a high degree of repeatability and predictability. Note however, that for such repeatability and predictability—and to maintain a linear force-displacement curve, the spring or flexure must be operated in the material's elastic range. For additional detail, three recommended resources for spring design are Wahl [1963]; MIL-STD-29, Spring Manufacturer's Institute [1986].

Elastically Coilable Booms

Elastically Coilable Booms are lattice trusses which are a stored energy devices commonly used in antenna and instrument deployment. The boom is normally procured as an assembly with opposing interfaces for mounting the deployed item and mounting to the spacecraft. Coilable booms consist of three or more composite longerons which can be coiled down onto one another, wrapping the entire assembly into a tightly stacked helical package. When the stowage restraint is released, the longerons uncoil into to their free (un-bent) state, uncoiling and extending away from the spacecraft. The truss is fully deployed from the base outwards as the boom deploys, providing stiffness and control throughout deployment. Deployment rate is typically controlled by paying out a lanyard attached to the tip plate off a reel at the base, although a number of options for deployment management are available if retraction is necessary or if the tip is not tolerated to rotate during deployment. The truss material is typically fiberglass composite, although graphite may be employed in lightweight or thermal-stability-critical applications.

Elastically Coilable Booms stow very compactly (typically 1–2% of their deployed length), are very thermally stable, deploy repeatably and precisely, are easily ground-testable over unlimited deployment cycles, readily accommodate substantial cable harnesses along their longerons. Some considerations in application can include the characteristic end of deployment accelerations imparted to the tip, and the above-mentioned tip rotation during deployment, although these aspects can be mitigated by additional mechanism.

Tape Hinges

Tape hinges are commonly used in solar array and antenna deployment. They are single-cycle, self-rigidizing devices. Think of a retractable tape measure—bend the tape in half, and it snaps back into position and becomes rigid. A tape hinge operates in exactly the same manner: a strip of metal (or composite) with curved cross-section is bent along its length (the stowed configuration); when a launch restraint is released, and often with the help of a kick-off spring, the hinge tries to straighten itself. Once it is straight, the cross-section at the bend, which was straight through the bend radius, resumes its natural curved shape. Once snapped into its natural (cupped) shape, the tape possesses the strength and stiffness afforded by the cross-sectional moment of inertia, allowing it to resist on-orbit deployed loads. Tape hinges may be augmented with secondary latching features to increase this resistance to re-folding, although such complications tend to defeat the elegant simplicity of the tape hinge. Such high-strength hinge applications may be better served by a standard lug/clevis hinge configuration.

22.6.1.3 Gearing/Power Transmission

The category of power transmission includes all of those components which connect the prime mover to the driven item. Much of the work in mechanism design centers around coupling the driver to the driven item, and, not surprisingly, many mechanism failures occur in this area. Power transmission is most often where good and poor mechanism designs are distinguished, as the better designs almost unilaterally result from the simplest connection between the prime mover and the driven item.

One of the primary design objectives in selecting and sizing power transmission components is to make them as transparent to the mechanism as possible—that is, to minimize force, torque and frequency perturbations coming from the components themselves. A "seamless" connection between the prime mover and the driven element makes for a much easier analysis, and furthermore, a much less complicated control law and plant dynamics model. Of course, in practice, there is no such thing, but the mechanism designer strives for this objective by implementing the following derived requirements on power transmission components:

- Low mass (equals low inertia)
- High stiffness
- Positively located and aligned
- Low parasitic drag
- Minimal backlash or free play

Table 22web-3 enumerates the more common power transmission components seen in spacecraft. It is by no means exhaustive. Many mechanisms feature combinations of several different power transmission types working together. Gears are separated out as a subsection following the table.

Gears

Gears and gear drives are one of the most common ways to multiply torque (at the expense of speed), multiply speed (at the expense of torque), transform rotational to linear motion or vice-versa, and change the normal planes of rotating axes. When properly implemented, they are reliable and long-lived. They are highly tolerant of shock loads and can be made to run quietly and with almost zero backlash.

A multitude of reference material exists regarding gear design and use. An excellent collection of technical papers has come from the NASA Lewis Research Center, including *Publication 1152*, which provides a thorough introduction to gearing and an extensive bibliography pointing to other references. Also useful are publications from the American Gear Manufacturer's Association (AGMA) [2005], ANSI/AGMA [2003], *Machinery's Handbook* [2008], which conveniently contains many of the standard AGMA equations and rules-of-thumb for gear sizing and selection, and Shigley and Mitchell [1993].

22.6.1.4 Bearings, Lubrication, and Life

This section discusses the general category of components that are used to release degrees of freedom in a mechanism; i.e., components which permit the prime mover to actually impart motion to the load. Proper mechanism design requires managing and releasing DOF's in a controlled manner—making sure to permit the types of motion that are necessary, and constraining the other types in such a way so as not to impact the performance of the mechanism

For the purposes of this text, *bearing* will refer to any rolling-element bearing, and *bushing* will refer to any component which reduces friction through sliding surface contact.

Bearings

Rolling element bearings are the principal means for releasing rotational degrees of freedom in rotary systems. Their use requires dutiful handling and exacting machining tolerances, but when properly used, they can afford nearly frictionless rotation for many thousands of hours. They are almost always preferred over bushings for rotating applications, with the caveats that they occupy more space and are less forgiving of heavy shock loading than bushings. Compared to a bushing-suspended rotating assembly, rolling element bearings, and in particular ball bearings, offer tighter location control, better stiffness, less jitter, considerably less parasitic drag, longer life, and superior speed capability.

Bearings must be designed to withstand the vibration and dynamic loads expected during launch. These loads

Table 22web-3. Common Power Transmission Devices for Spacecraft.

Drivetrain Component	Description	Design Considerations
Linkages & Bellcranks	Any combination of rods or bars, usually pinned and allowed to pivot in one or more directions, that connect the prime mover to the driven element. In a bellcrank, two or more pin joints are located at varying distances from a central pivot. The prime mover is connected to one joint (via a link); the driven element to the other joint (via a link). By varying the geometry, any number of motions can be created, including force-to-torque conversion, force vector direction changes, and force or torque multiplication. They are also very useful for transmitting forces under relative motion—when the prime mover's position changes relative to the driven element	• Accurate free-body analysis is essential to characterize loading at different joints and to ensure that the force vectors don't produce a net moment counter to the desired direction of rotation • Pin joints are usually sliding-surface rotation, unless bearings are used at the joints • All degrees of freedom should be released except those along the line-of-action and those required to fasten the assembly together. Spherical bearings and rod ends are very useful for this purpose
Harmonic Drives	Sometimes considered a type of gear drive, it is technically a category unto itself. Harmonic drives consist of a wave generator, a flexspline, and a circular spline. In a reduction drive, the most common setup is to use the wave generator as the input and the flexspline as the output. The wave generator is a series of rollers or an elliptically-shaped ball bearing that rides inside of and deforms the flexspline as a rotating ellipse. The flexspline, almost universally having two less teeth than the circular spline, engages a small region of its external teeth with the internal teeth of the circular spline at the opposing ends along the semimajor axis of the ellipse. All other areas of the flexspline are disengaged from the circular spline teeth. As the wave generator rotates, the semimajor axis of the ellipse rotates, so the deformed shape of the flexspline rotates. The two opposing areas of engagement crawl around the inside of the circular spline, imparting rotational motion to the flexspline. Like a planetary geartrain, a harmonic drive can be used as a speed reducer, speed increaser, or a differential, with the input and output being any combination of circular spline, flexspline, and wave generator	• Very high accuracy and repeatability • Essentially zero backlash, because the flexspline teeth are preloaded against the circular spline teeth • Very high reduction ratios per stage. In the same size package, a single harmonic drive stage can achieve up to 50 times more reduction than a single planetary stage • Requires high mounting accuracy • High cost • Life expectancy can be significantly less than with a planetary • Not as shock-resistant as a planetary drive
Cams & Eccentrics	The term "eccentric" can be generally applied to any rotating item that transfers motion at a point (or points) located in a plane normal to the primary axis of rotation, but with the point being non-coincident with that axis. A cam is a type of eccentric. Eccentrics are commonly used to transform rotation into oscillatory or intermittent translation. They can also be used to convert uniform rotation into sinusoidal, wobbling, or intermittent rotation (geneva drives)	• Unless space permits a roller, most cams are frictional devices • High-speed or high-load situations can induce heavy misalignment loads into the shaft support bearings; on cams, high speed can cause liftoff—some type of backing spring is used in this case to maintain preload on the cam surface • Eccentric and helical slot drives may require free-body analysis at multiple locations along the displacement profile to capture the worst-case loads, since the displacement of the slot-driven component may not be linear with respect to the driving wheel
Dampers	Dampers are used to slow, restrict, or control motion that might otherwise be violent, such as in unraveling of a deployable truss or a tape-hinged solar array. By performing resistive work over their displacement stroke, they can also be used to filter vibration and jitter that might otherwise propagate to a sensitive optical instrument. In theory, they produce a resistive force (linear dampers) or torque (rotary dampers) that is proportional to the *velocity* of the driven item. Dampers may use friction elements (disc clutches), permanent magnets (hysteresis and eddy current dampers), or a viscous fluid through which a vaned shaft must rotate or a perforated disc must translate. They can be rigged to operate only during a certain phase of the mechanism travel	• In practice, damper behavior is somewhat nonlinear • Significant hysteresis can be encountered • Passive damping is preferred over active damping due to complexity and reliability issues • Magnetic dampers tend to be the most predictable and thermally forgiving in terms of accuracy, but require a larger size to exert the same damping rate compared to a viscous damper • When used as a shaft coupling, a rotary damper becomes a torque converter

Table 22web-3. Common Power Transmission Devices for Spacecraft. (Continued)

Drivetrain Component	Description	Design Considerations
Clutches & Brakes	Clutches and brakes are used to selectively couple and decouple two coaxial rotating shafts. They can be used to stop motion, start motion, or restrict motion to one direction (overrunning roller and sprag clutches). Torque limiters are in-line, rotating mechanical fuses that decouple a rotating input source (drive element) from the corresponding output (driven element) when the resistive torque load of the output reaches some pre-determined level. Conversely, they are also used to couple an input and an output source at a pre-determined torque level, ensuring that there is enough available input torque to turn the output and thereby preventing stalling of the input source. Sensing of the resistive torque may be mechanical or electrical (usually mechanical), and any number of decoupling methods may be found including friction plates, mechanical detents, spring-loaded pawls, hydraulic and pneumatic means, solenoid-engaged friction devices, passive magnetic means such as eddy current and hysteresis, active magnetic means such as magnetic particle	• Each type of clutch carries its own set of rules for use. Many clutches can exert equivalent amounts of torque in the same form factor, but environmental conditions may prevent the use of one or more types. For example, friction clutches generate contamination and require recalibration over time; mechanical detent clutches can be very accurate but introduce cyclic shock loads into the system; permanent magnet clutches can magnetize a set of ball bearings and reduce bearing life by attracting metal contamination • Electric brakes and clutches require additional control electronics to support their use • Some clutches are not automatically re-engaging, and others require a finite rotational displacement before re-engagement; the mechanism should be tolerant of such features, if applicable • Space-qualified clutches and brakes are almost exclusively custom devices, meaning long lead times and high cost of implementation
Belts	Belts transmit rotary motion from one axis to another. They normally operate in a plane, but can also shift rotary motion to orthogonal planes when bent in a half-figure-eight. Belts considered for space mechanisms are usually metal or polymeric, with integral teeth to prevent slippage	• Polymeric belts incorporate intrinsic damping capabilities and shock forgiveness due to their flexibility • Attaching a driven item to the belt itself converts rotary motion to linear motion with less mass (and correspondingly less accuracy) than a rack and pinion or leadscrew mechanism
Flexshafts	Flexshafts are, as the name implies, flexible lengths of shafting that can transmit rotation in nonlinear directions without the use of universal joints and multiple segments. Normally constructed of braided or coiled cable	• Can achieve very unique rotary power transmission geometries • No joints to wear out • Relative motion between the elements of the shaft cause friction and drag • Due to the construction, flexshafts can exhibit torque windup which introduces backlash and spurious resistance torques—for this reason, they are not preferred for precise positioning—however, when used according to the load rating, can be essentially backlash-free • Minimum allowable bend radii must be observed—and in certain instances, multi-DOF shaft couplings can achieve better bend radii while maintaining constant angular velocity
Couplings	Shaft couplings are used to compensate for misalignment between a driving and driven element. Without them, prohibitive manufacturing tolerances would have to be used in structural components to assure proper alignment of parts, and even then, the slightest deviation can induce enormous side loads in shaft bearings. Most couplings are in-line devices that accommodate small deviations in axial position, angle and parallelism between two shafts that in theory should lie along the same line. They also accommodate differential thermal expansion and contraction between what are frequently steel drivetrain components mounted in aluminum or titanium housings. Examples are Oldham, Schmidt, multi-disc, spider, jaw, helical-beam, bellows, splines, and crowned-gear types. Other couplings allow for two axes of rotation to occupy significantly different directions in a plane. These include the common spider universal joint (Canham joint), K-couplings, constant-velocity joints, and polygon shafts	• Use of a coupling will ordinarily add some amount of parasitic friction or cyclic eccentric loading to the mechanism, which must be accounted for • The way in which a coupling is mounted is often overlooked. Most industrial clamp hubs are not sufficient to transmit torque without slipping when mounted to precise, ground shaft surfaces often found in spacecraft mechanisms • The torque rating of a coupling is usually greater than the capability of its means of attachment to the shaft—analysis should consider both failure of the coupling and failure of the mount • Space-qualified shaft couplings are difficult to find off-the-shelf; therefore, most implementations are custom designs. The program must build in schedule margin for manufacture of these devices, as they are usually long-lead

are estimated working in concert with the spacecraft loads, structures, and dynamics analysts. For slow speed, low precision application, the maximum mean Hertz stress should be kept below 500 kpsi. High-speed and high precision applications should limit the maximum mean Hertz stress to 340 kpsi. Bearing preload should be sufficient to remove radial pay and assure ball/race traction under orbital operating conditions. Preload should not be sized to prevent ‘gapping’ under launch loads as this will results in accelerated lubricant degradation and reduce life in orbit.

The authoritative resource on ball bearings is Jones, [1946]. Other references are Palmgren [1959], and the work of Hertz [1887], whose equations are published in most mechanical engineering texts. Almost all bearing manufacturers now publish extensive engineering addendums to their bearing catalogs which conveniently include equations and coefficients that have been tailored to the specific manufacturer’s fabrication techniques. Though not standalone, they are important and very useful supplements to the theoretical texts.

Bushings

Bushings are used to allow rotational and linear motion, sometimes simultaneously. They are easier to install and far less expensive than rolling element bearings, but they can be less precise unless preloaded. Lack of precision equates to greater forgiveness in misalignment, but they are not appropriate for precision applications.

The most common bushings are porous sintered bronze which is impregnated with some type of oil. They, as well as solid bronze alloys, work well for rotational applications but are not recommended for linear applications because the pore network tends to bite the shaft and cause stiction. Rather, polymeric bushings should be used for linear applications. Because it is soft, plastic locally yields at the edges of the bushing bore when the shaft cocks, eliminating the biting phenomenon. The best materials are Vespel, Rulon, Peek, Torlon, Delrin, and in low-load circumstances, Teflon. Teflon has limited application due to its cold flowing under relatively low stress. Plastic bushings also have the advantage of being able to be used unlubricated.

Bushings are generally pressed into their housings, though in redundant systems, dual rotating surfaces are achieved by allowing the bushing to float within its housing (often a larger bushing). Bushings are usually superior to linear ball bearings in linear applications because side loads are distributed over a much larger surface area, and softer shafts can be used.

Spherical bearings, which are highly polished balls surrounded by a strip of low-friction composite material—sometimes called monoball bearings, spherical bushings, or spherical rod ends—are very common in linkages and struts. The ball incorporates a hole for fastening a bolt or pin, and the composite strip is encased in a steel ring which mounts into a housing. They are also used in kinematic mounts to accommodate thermal effects. Spherical bearings are able to impart motion under extreme static and dynamic loads. The fact that they are widely used in primary load path structures on Space Shuttle payload bay carriers speaks to their strength and reliability.

Slots & Tracks

Slots and tracks are effective ways to direct motion and transmit loads. Where space permits, they are used with ball- or roller bearing cam followers, but dowel pins work equally well provided adequate lubrication and mechanical advantage exists. Diagonal and helical slots are useful, for example, for converting rotation of a collar into linear motion of an internal collar which is pinned through the slot. In space-constrained situations, an array of slots can be a substitute for a spline, permitting relative linear motion between two coaxial shafts or collars while simultaneously transmitting torque.

The load-transmitting capability of a slot is limited by the cam follower or pin in bearing against the wall of the slot, so they are normally suitable only for low-load situations, as it is easy to brinell the slot walls and produce erratic motion. That is not to say that they are exclusively limited to low-load applications, however, a capable roller or cam follower tracking in a hard steel slot can easily handle hundreds of pounds of force.

Flexures, Flex Pivots

Flexures and flex pivots differ from bearings and bushings as they exert a resistance force or torque that is proportional to their displacement. They are useful in kinematic thermal expansion setups where the resistance force is insignificant compared to the capability of the expanding/contracting platform to deflect them. They are also useful in accommodating assembly misalignment. Introducing a flexure or flex pivot may be as simple as necking down a shaft or notching a block to provide a little bit of adjustability.

Though the resistance load needs to be considered, one distinct advantage to the deflection of a flexure is that it is accurately predictable and essentially frictionless, meaning that it can be treated as a “known” or “fixed” load in Torque and Force Margin calculations. For more information on flexures, refer to Sec. 22.1.5.3.

Lubrication

Lubrication of spacecraft mechanism components is essential to assure smooth operation and long life. NASA Lewis Research Center has produced a wealth of technical papers on the various aspects of lubrication of spacecraft systems. An excellent introductory text is Jones and Jansen [2000], which also provides an extensive bibliography of more advanced texts on the subject.

Wet Lubrication

Wet lubrication is accomplished with oils or, more commonly, greases. A space flight-rated grease is unique in that it must operate in vacuum (possess a very low vapor pressure), operate at low temperatures (low pour point and a high viscosity index), and operate in the presence of atomic oxygen. Synthetic greases are used exclu-

Table 22web-4. Common Feedback Devices.

Device	Principle of operation	Accuracy	Comments
Limit Switches	Mechanical contact switch	Fractions of a degree	Mounting and alignment is critical to system reliability
Reed Switch / Magnet	Magnetic field actuated electrical contact reed	Degrees	Glass case can be delicate High hysteresis Can be difficult to align
Hall Effect Switch / Magnet	Semiconductor magnetic switch	Fractions of a degree	Used for motor commutation Requires conditioning electronics
Potentiometer	Variable resistor	Fractions of a degree	Simple electronics Can have noise, limited life issues
Resolver	Rotary transformer with sinusoidal outputs	1 arcsec	Need to digitize output signal
Encoder	Optical emitter / detector pairs acting through coded optical mask	Sub arcsec	Can be incremental (count from known position) or absolute
LVDT	Linear Variable Differential Transducer	0.001 inch	Linear output

sively, with Nye's Pennzane, Castrol's Braycote, and Dupont's Krytox being the most common. The multiply-alkylated cyclopentane (Pennzane) grease provides the longest life and is preferred if the minimum operating temperatures can be maintained above 0 C. In applications where lower temperature operation or extremely low outgassing is required, the perfluorinated polyether (Braycote) is appropriate.

Dry Lubrication

Dry lubrication is always in a boundary regime. However, it offers distinct advantages over wet lubrication under certain circumstances. It is a requirement for cold operating temperatures with no active thermal control—more than likely its greatest attribute. It is capable of withstanding contact stresses as high as 300,000 psi—near the desired limit for quiet-running bearings. Dry lubrication, though most often seen on sliding surfaces, is especially suited to rolling contact, as it tends to be cold worked in compression rather than sheared in translation. It is at home in vacuum, and the coefficient of friction is independent of operating speed—which makes dry-lubricated systems good candidates for accelerated life testing. Certain formulations are electrically conductive, whereas liquid lubricants are generally insulating.

Its chief drawbacks are in the generation of debris as it wears and its reduced life—an order of magnitude in CDF—when compared to wet lubrication. It can also be sensitive to atmospheric humidity, forming an oxide layer that increases its coefficient of friction until it is worn off through mechanical action.

Measurement and Feedback

Most mechanisms use some kind of feedback device to measure position, velocity or force and provide for telemetry or control. These vary from simple switches that indicate one position to precision analog and digital devices. The accuracy of the feedback device is important since it drives the overall accuracy achievable by a closed-loop device. The feedback accuracy must be several times the desired closed-loop accuracy of the mechanism. Common feedback devices are described in Table 22web-4.

Control

Mechanisms can be either open loop or closed loop control. In an open loop system, the mechanism responds to sent commands directly and there is no feedback. An open loop stepper motor is driven by a series of pulses. It moves one step in response to each pulse. Open loop control is used for relatively simple low performance mechanism.

To achieve precise position or velocity control, complex motion profiles and/or high rates required closed loop control. A feedback device is used along with (usually) a microprocessor control system to send the signals that drive the prime mover, usually a motor. The desired position, velocity or profile is loaded into the controller. It reads the feedback signal to determine the actual state on the mechanism. It then uses the error between desired and actual to drive the mechanism to the desired state. The design of a control law can be complex. Matlab and Simulink are used to ensure that the loop is stable and to simulate and optimize performance.

Power/Signal Transmission

It is frequently necessary to pass power and signal electrical connections through a mechanism. As an example, the power and telemetry generated by a solar array must be passed to the spacecraft through the Solar Array Drive. Also, the RF communication signal from and antenna must be passed through the antenna gimbal. Common devices for electrical transfer are shown in Table 22web-5.

Occasionally mechanisms also need to pass fluid through them. Flexible stainless hoses with loop geometries are typically used for this purpose.

Launch Restraint and Release

Mechanisms and deployable systems are typically restrained for launch to survive the high vibration environment and then released on orbit. Solar array panels and deployable antennas typically have several points of restraint chosen to give acceptable stiffness. Over the years, many different technologies have been used for launch restraint and release. Some of the more relevant ones are described in Table 22web-6.

Table 22web-5. Common Electrical Transfer Devices for Spacecraft Mechanisms.

Device	Principle of Operation	Comments
Slip Rings	Electrically conductive rings with sliding brushes	Gold or silver redundant contacts Typical solar array practice
Cable Wraps	Coiled flat cable assemblies that flex as a joint is rotated	Potential cable fatigue life limits Limited angle travel Lowest noise Most reliable
Roll Rings	Uses rolling ring contacts in place of sliding brushes to transfer signals	Potential Long life Limited heritage Increased envelope and cost
Rotary Waveguides	Non-contact waveguide with one side fixed and the other rotating	RF signals

Table 22web-6. Launch Restraint and Release Systems.

Device	Principle of Operation	Comments
Pyrotechnic Bolt Cutters and Pin Pullers	An explosive material is ignited which produces a pressure pulse which is used to pull a pin or cut a restraining bolt	Explosive power gives high operational margins Also gives high shock levels which can be a problem for instruments Cannot test the individual device that is flown. Lot qualification used Safety considerations—not used as much anymore
Paraffin Actuators	When heated, paraffin phase change volume increase operates a piston	Resettable and ground testable Limited operational temperature range Must be protected from overtravel/overheat
Frangible Bolts	Shape memory actuator stretches and fractures a notched bolt	Sensitive to assembly preload Limited operational temperature range Must be protected from overtravel/overheat
Burn Wire Devices	Fusible link cut by current pulse. Allows spring to unwrap, segmented nut to expand and bolt to be released	Wide temperature range Insensitive to preload variations Usually needs to be reset by supplier
Ejector Release Actuators	A shape memory wire is shortened by heating, pulling on detent balls, releasing a threaded coupler	Resettable and ground testable Limited operational temperature range Must be protected from overtravel/overheat

Cup/Cone Pairs:

A cup/cone pair consists of, most simply, a machined conical (male) section on one component and a conical cup (female) section on another component to be restrained to the former. When the conical feature is inserted into the cup, the two components are restrained axially (along the axis of the cone) and laterally (perpendicular to the axis of the cone), and angular displacements are also constrained. The conical geometry has the added benefit of providing a lead-in feature for components that are moving together from some other position—it allows for slight misalignment and has a self-centering effect on the two components as the cone is driven into the cup.

Many variants of the cup/cone interface exist; for example, the "cone" may be a spherical ball, or axes of constraint may be separated by using a V-groove instead of a conical cup, which allows for motion along the direction of the groove. These variations are employed selectively in order to design kinematic, thermally compatible couplings between components.

These actuators typically act through cup/cone pairs to provide lateral shear support. It is recommended to use a kickoff spring located at or near to the release point to ensure positive release and to guard against any tendency for the interfaces to stick.

Mechanism Manufacture, Assembly and Test

Due to the vacuum of space and the usual proximity of mechanisms to sensitive hardware such as optics, cleaning and contamination control are key elements of the assembly process. All parts are ultrasonically cleaned in solvent with additional measures reserved for critical parts such as bearings. They are then stored and assembled under clean room conditions and handled by personnel wearing appropriate protective attire. MoS_2 dry lube coatings are particularly sensitive to moisture and the humidity of assembly areas must be controlled. Mechanisms usually include electrostatic discharge (ESD) sensitive components and must be handled using ESD protective equipment with appropriately trained personnel.

Environmental testing at acceptance levels is conducted on the mechanism at the component level. Functional and performance testing involves operating a test article through all of its primary and redundant modes. Key performance parameters, such as torque margins and alignments are recorded so that they can be trended throughout the test flow. Functional testing is typically performed at the beginning and end of the flow after each environmental exposure. Abbreviated functional testing is also performed at hot and cold plateaus during thermal vacuum testing.

Early in a program, components such as bearings, gears, and flex cables can be subjected to accelerated life tests to identify show stoppers. Later in the program, more formal life testing can be conducted on full engineering or qualification units. Formal life tests should include vacuum environment, temperature cycles and representative inertias and motion profiles. Acceleration is usually required in life tests to produce results in the shortened development cycles that are common today. Acceleration should be approached carefully to avoid creating an artificially benign test condition. For example, bearing wear rates can be higher in the lubrication regimes experienced at very low speeds. Acceleration may artificially improve lubrication.