A STRUCTURED METHOD FOR THE CLASSIFICATION AND SELECTION OF ACTUATORS FOR SPACE DEPLOYMENT MECHANISMS

J.F. Cuttino†, D.D. Newman‡, J.K. Gershenson§, and D.E. Schinstock*

† Metrology and Precision Engineering Laboratory, Department of Mechanical Engineering, The University of Alabama, Tuscaloosa, AL
‡ Sverdrup Technology, Inc., Huntsville, AL
§ Life-cycle Engineering Laboratory, Department of Mechanical and Aerospace Engineering, Utah State University, Logan, UT
* Department of Mechanical Engineering, The University of Tulsa, Tulsa, OK

Abstract

This paper presents a method for quickly comparing vastly different actuators based on specified performance criteria for a given application. The research grew out of the need for a technique to select actuators for space deployable mechanisms, where stringent size constraints, expensive weight considerations, and a wide selection of potential actuator types made it very difficult to compare one actuator to another without extensive analysis of each system.

The method expedites the initial design selection by developing a set of expressions for various actuators that relate performance criteria such as weight and volume to the work required for the specific task. These work-based expressions, which are derived from manufacturers’ data and theoretical calculations, provide a common ground for comparison and provide insight for the optimization of kinematic configurations. The process provides a quick, quantitative methodology for identifying the strongest candidate for a given kinematic configuration prior to conducting exhaustive analyses to the systems. A benchmark kinematic system typical of a space antenna deployment system is used to illustrate the generalized method.

Keywords: actuator, design, selection, optimization

Introduction

The selection of actuators for a given application can be a complex issue. Often, selections are made based purely on a designer’s familiarity with particular systems. Many industries lack the resources or the time to conduct a detailed analysis of every possible actuator alternative, and even when the resources are available, determining a structured method for comparing two unrelated actuator types is very difficult. Therefore, actuators are often chosen for convenience while less familiar or new alternatives are overlooked. In light of this, the goal of this research was to develop a quantitative, preliminary selection method to analyze potential actuator alternatives. The generalized, structured method is shown in Figure 1.

The actuator selection method is a step by step method that begins by performing a kinematic analysis of the mechanism. The output of the kinematic analysis is the load and displacement histories of the mechanism. Next, a set of primary requirements is used with the histories and information on the mechanism to perform a primary requirements analysis on potential actuator types. The primary requirements are those which must be met and cannot be traded off. Each actuator type is checked for adherence to the primary requirements resulting in a set of candidate actuator types. The previously determined load and displacement histories, combined with generalized information about how specific actuator types work to meet loads and displacements, are also used to conduct the load matching. Load matching is the process of matching general actuator load characteristics to the load...
profile required by the application. This step identifies a preliminary actuator specification such as peak torque or load.

Data obtained from various actuator manufacturers are combined with the governing physical relationships of an actuator to form work-based relationships for secondary requirements. Secondary requirements are those, such as weight and volume in the case of space deployment mechanisms, which must be traded off to select the best overall actuator for the design. The result is a set of relationships that measure the performance indicators of each of the secondary requirements as a function of work. These work-based relationships are combined with the previously calculated actuator work capacities to measure the performance of each actuator relative to the secondary requirements. This process of secondary requirements analysis uses weighted evaluations to calculate the utility of each actuator type. The result is a small set of top-performing actuator types with their key parameters determined.

The small set of remaining actuator type candidates is finally analyzed against a set of tertiary requirements. The tertiary requirements analysis uses more qualitative requirements such as supplier relationship, track record in application, and design effort to select the best actuator type for the application.

This research provides enormous gains by providing an efficient, quantitative method for identifying the most appropriate actuators quickly. Rather than specifically sizing each potential actuator and then determining its applicability through costly iteration and analysis, this method relies on the creation of curves to provide information on actuator volume and weight as a function of the work history required of the actuator. These curves provide actuator applicability information to the designer without the time consuming process of specifically sizing a particular actuator. Secondary

Figure 1: Actuator selection method.
requirements analysis, through work-based relationships, provides a structured, quantitative method for selecting the strongest actuator alternatives.

To illustrate the application of the structured actuator design method, an example was used based on a common configuration used in space structures for deploying antennae. The system is shown schematically in Figure 2.

![Deployable Space Antenna System](image)

**Figure 2: Example deployment mechanism and its critical components.**

The deployment system is made up of eight rib pivot stations, two of which are highlighted in Figure 2. The eight pivot stations are positioned around the circular rib at equal intervals. The driving mechanism consists of a single ball screw actuator that transmits force through the push rod, providing a rotational moment about the rib pivot station. The pivot station is driven from a stowed angle of -90° to a deployed angle of 10 degrees past the horizontal.

A driving design requirement for the deployment system is the ability to overcome a 3000 in-lbs resistive torque at each individual pivot station throughout the deployment. This exceptionally high resistive torque is specified to overcome friction or snags in the mesh antenna encountered during deployment. The high resistive torque also provides ample driving force to overcome gravitational and inertial effects during ground testing. Additionally, the actuator is required to be contained in a cylindrical envelope with a diameter of 10 inches about the axis of the ball screw.

**Kinematic Analysis**

The purpose of conducting the kinematic analysis is to obtain information about the load and displacement histories required of the actuator. Those histories will be used to identify potential actuator choices. For example, a ball screw is a positive displacement device capable of overcoming loads that remain constant throughout the required travel. However, a passive actuator such as a spring will have more driving force at the beginning of travel than at the end, and it would therefore be
more appropriate for a kinematic design that gains mechanical advantage and reduces required forces during travel. This type of analysis will be used to “load match” the actuators to the kinematic design.

Load histories for kinematic designs can be analyzed using several tools including packages such as Working Model®, ADAMS Mechanisms, and some finite element analysis packages. The resulting load history analyzed in this study was determined using Working Model® software and is shown in Figure 3. This curve represents the maximum load requirement of a single pivot station for the actuator. The input force for the benchmark system reaches a maximum of approximately 700 lbs at most of its deployment range.

![Figure 3: Required actuator force vs. pivot station rotation for the example deployment mechanism.](attachment:figure3.png)

**Actuator Type Selection Based on Primary Requirements Analysis**

The first step in paring down the nearly infinite number of potential actuators is to eliminate those actuator types that do not have the capability to meet the primary requirements of the situation. These primary requirements, or "musts," include travel distance, force capabilities, and, in this case, physical size and deployability in space. This step makes the following comparisons more manageable and requires no analysis. For the space antennae deployment system example, the paring left six actuator types for analysis and comparison in the following steps. These six were standard and multistage pneumatic cylinders, high output paraffin actuators, linear and spiral springs, and permanent magnet brush motors.

**Work-based Relationships for Secondary Requirements**

Secondary requirements are those that can be analyzed by trade-off. They therefore fall below primary requirements in terms of necessity because we are looking to maximize the product in terms of a combination of all of these parameters collectively. In addition, they are not "yes or no" parameters, the candidate actuators possess varying, measurable levels of each of these parameters. Secondary requirements for space deployment mechanisms include those of weight and volume, both measurable quantities. While other requirements may be added, our list was capped by the customer in this development exercise. For each actuator type, there are infinite combinations of these parameters. This method, therefore, uses work-based relationships to quickly develop comparable measurements of secondary requirements for each actuator type. The concept behind work-based relationships is to reduce each requirement's performance variable to a function of work. The required
work capacity for each actuator type will later be determined by load matching general actuator characteristics to the required load profiles for the particular application.

Actuators such as those considered in this paper fall into one of two categories, force (or torque) actuators such as electric motors, or displacement actuators such as piezoelectric actuators. When designing a limited displacement system, it is convenient to consider the work requirements of the system. It therefore makes sense to match the work capacity of the actuators with the work requirements of the system. To overcome the fact that force actuators have an “infinite” work capacity, only the work required of a certain actuator for a specified range of motion determined by the application is considered. For example, in the case of a rotary electric motor used to generate a linear displacement, the specifics of the transmission are included to calculate the required work capacity over the range of deployment. This work capacity is then used with the empirical charts presented in the paper to deduce the required volume and weight relationships.

The following sections present the relationships that have been developed relating volume and weight to the required work capacity for the actuators being considered. The relationships have been generated using theoretical expressions for simple actuators such as springs and manufacturers’ data for complex actuators such as motors and pneumatic cylinders.

Pneumatic Cylinders: Pneumatic actuators such as the one shown in Figure 4 are commonly used in industry and space applications. Typical pneumatic cylinders have a travel distance of up to 12 inches and bore sizes of up to 8 inches in diameter. At these dimensions, the actuators are capable of work capacities up to approximately 90,000 in-lbs with volumes ranging from 10 in$^3$ to 1400 in$^3$.

An interesting alternative is offered by FABCO-AIR and is shown in Figure 5. This design utilizes multiple stages in order to reduce the required cylinder diameter and therefore shows strong potential for space applications with specific size constraints. The cylinder diameter is kept to a minimum by placing multiple piston heads in a cascaded arrangement. Air flows from each stage via an orifice to the adjacent stage. The increase in force resulting from this arrangement at a constant pressure is proportional to the number of stages used. The force output and volume vs. piston area remain the same, so that the performance characteristics are very similar to the standard configuration shown in Figure 4.

![Figure 4: Pneumatic cylinder (FABCO-AIR, 1998).](image)

![Figure 5: Three stage multistage pneumatic cylinder (FABCO-AIR, 1998).](image)

Work output for pneumatic cylinders is proportional to the actuator volume as shown in Equation (1). In this equation, $W_p$ is the work being done by the cylinder, $P$ is the cylinder pressure and $V$ is the ideal volume of the cylinder.

$$W_p = PV$$  \hspace{1cm} (1)
Figure 6 shows the relationship between work and volume for both standard and multistage cylinders at 150 psi. The data used to generate the chart is from two independent manufacturers, Bimba and Fabco-Air, and is assumed to be representative of most pneumatic cylinders. The variation that is present is due to the differences between the manufacturer tolerances and general cylinder design.

Figure 6 shows a linear relationship as predicted by Equation (1). An $R^2$ value of 0.9855 shows good correlation of the curve fit and the data, and it indicates that the volume to work relationship for both types of cylinders are essentially the same. However, according to Equation (1), the slope in Figure 6 should be $1/P$, or 0.0067 in$^3$/lb. The actual slope is 0.015 in$^3$/lb, roughly twice that of the ideal. This value represents the additional volume of the actuator hardware necessary to constrain the pressure and convert it into useful work. Note that the volume includes only the cylinder without supporting hardware, since supply tanks and compressors can take on an infinite number of variations.

Figure 7 contains a plot of the actuator weight vs. the work capacity for the standard and multistage pneumatic cylinders. As with volume, there is an apparent increase in weight with an increase in the work capacity of the cylinder. Curves were fit to derive the empirical expressions for volume and weight as a function of required work. These expressions are given by the following equations.

$$V = 0.016 in^3/in-lb \times W_p + 70 in^3$$

$$W_t = 0.033 lb/(in-lb)^{0.63} W_p^{0.63}$$

**Compression Springs:** Spring driven actuators have the advantage of being time tested and well established as a sound means of providing input. They have been used extensively for applications ranging from simple latching mechanisms and hatch doors to the sole means of deploying satellite antennas as was done with the Engineering Test Satellite VI (Meguro, 1996). The satellite consisted of two main reflectors 2.5 m and 3.5 m in diameter that were both deployed using torsional spring actuators.

The volume and weight relationships to work for a compression spring are easily derived. The shell volume, $V_{shell}$, can be expressed as shown in Equation (4). Here, the product $Nd$ represents an approximation of the solid height of the spring at full compression, where $N$ is the number of coils and $d$ is the wire diameter. $D$ is the mean diameter of the spring.

$$V_{shell} = \frac{\pi}{4} D^2 N d$$

Equations (5) and (6) represent the work capacity, $W_{CS}$, at full deflection, and the number of coils, $N$, respectively. Here, $G$ is the modulus of rigidity for the material being used, $S_T$ is the torsional...
stress in the wire (determined using Figure 8), and \( \delta \) is the deflection of the spring. \( F_{\text{max}} \) is the load at full deflection, and \( D \) is the mean diameter of the spring.

\[
W_{CS} = \frac{1}{2} F_{\text{max}} \delta
\]  
\( (5) \)

\[
N = \frac{Gd\delta}{\pi S_T D^2}
\]  
\( (6) \)

The recommended wire diameter, \( d \), is given by Equation (7).

\[
d = \sqrt[3]{\frac{2.55 F_{\text{max}} D}{S_T}}
\]  
\( (7) \)

Using Equations (5) through (7), the shell volume in Equation (4) can be expressed in terms of the work capacity of the compression spring as seen in Equation (8).

\[
V_{\text{shell}} = \frac{G}{2S_T F_{\text{max}}} \left( \frac{2.55 F_{\text{max}} D}{S_T} \right)^2 W_{CS}
\]  
\( (8) \)

**Figure 8:** Recommended design stresses, \( S_T \), Oil-Tempered Steel Wire, ASTM A 229 (MB Grade), Compression and Extension Springs (Carlson, 1978, p. 146, Fig. 84).

The weight is given by calculating the volume of the material times the density of the material. This expression is found to be

\[
W_t = \frac{\rho \pi Gd}{S_T DF_{\text{max}}} \left( \frac{2.55 F_{\text{max}} D}{S_T} \right)^2 W_{CS}
\]  
\( (9) \)

In this equation, \( \rho \) is the density of the material and \( d \) is the diameter of the wire.

**High Output Paraffin (HOP) Thermal Actuators:** The High Output Paraffin actuator, (HOP), was developed by Maus Technologies, now Starsys Research, in 1985 as an alternative to conventional actuators used in space. A cutaway of a typical HOP actuator is shown in Figure 9 (Starsys, 1998). These actuators convert heat to mechanical work through the expansion of the paraffin contained in the housing. The paraffin is heated to its actuation temperature through the use of internal resistance heating elements, resulting in a 15% expansion of the paraffin. The hydrostatic pressure is transmitted through the rubber boot, extending the actuator rod (Tibbits, 1988). Typical HOP actuators are capable of producing forces between 50 lbs and 1000 lbs and strokes of...
approximately 4 inches. According to the manufacturer, however, the process is scalable, so longer strokes may become available.

Figure 9: High Output Paraffin Actuator (Starsys Research).

Figure 10 and Figure 11 show the relationships between the work capacity of a typical HOP actuator and the actuator volume and weight, respectively. The curves shown here were extrapolated from data obtained from a study performed by Starsys Research (Tibbits, 1988), assuming ideal scalability. Correspondence with a Starsys representative revealed that the process is scalable, but sealing problems may arise as the stroke length increases.

Figure 10 indicates that the actuator volume is proportional to the required work capacity. The linear curve fit reveals a slope of 0.0214 in³/lb with an $R^2$ value of 0.9999. The following expression can therefore be deduced to yield an approximation of the volume as a function of required work capacity, $W_{HOP}$,

$$V = 0.021 \text{ in}^3/\text{in-lb} \times W_{HOP}$$

Similarly, Figure 11 indicates that the weight of the structure can be estimated as

$$W_t = 0.0023 \text{ lb/in-lb} \times W_{HOP}$$

with an $R^2$ value of 0.9955.

Figure 10: HOP actuator: Volume vs. work capacity ($W_{HOP}$).

Figure 11: HOP actuator: Weight vs. work capacity ($W_{HOP}$).

**Spiral Springs:** Other possible actuation alternatives include the use of a spiral spring or a spiral spring and lead screw assembly. This type of configuration may prove feasible due to the high torque and large displacement capabilities of the spiral spring.

A typical spiral spring is shown in Figure 12 along with a cross section of the spring material showing the material thickness, $t$, and width, $b$. Devices of this type normally consist of a flat strip of metal coiled inside of a circular case. One end of the metal strip is connected to an arbor located in the center of the case while the other end is clamped or securely fastened.

Equations were developed to relate the spiral spring’s volume and weight to its work capacity at full deflection, $W_{SS}$. These equations are limited to springs that contain two or more active coils and that maintain clearance between the coils when fully wound. In addition, the outer end must be such that it is securely fastened to restrain any rotation.
Spiral springs exhibit a nonlinear torque throughout deployment as shown in Figure 13, where the torque is expressed in terms of percent of the maximum torque, $T_{\text{max}}$, and the angular deflection is in terms of percent of the maximum allowable angular deflection, $\theta_{\text{max}}$. An estimation of the spiral spring work capacity, $W_{SS}$, is easily obtained using Equation (12). This relationship was developed by integrating the area under the curve in Figure 13.

$$W_{SS} = 0.738T_{\text{max}}\theta_{\text{max}}$$

Equation (13) gives the inner diameter, $ID$, of the case enclosing the spring with respect to the material thickness, $t$, active length, $L$, and the arbor diameter, $D_{\text{arbor}}$. As a rule of thumb, the arbor diameter should be 12 to 25 times the thickness of the spring material. For the purposes of this study, the arbor diameter was taken to be a conservative value of 20 times the thickness.

$$ID = \sqrt{2.55tL + (D_{\text{arbor}})^2}$$

Equation (14) gives the bending stress, $S_b$, for the spiral spring as related to the torque, $T_{\text{max}}$, material width, $b$, and material thickness, $t$. The number of revolutions, $\theta_{\text{max}}$, can be expressed using Equation (15), where $E$ is the modulus of elasticity of the spring material.

$$S_b = \frac{6T_{\text{max}}}{bt^2}$$

$$\theta_{\text{max}} = \frac{S_bL}{\pi Et}$$

Solving for the active length, $L$, and using the relationships $V_{\text{shell}} = \pi(ID)^2b/4$ and $D_{\text{arbor}} = 20t$ results in Equation (16), which expresses the shell volume of the spring mechanism with respect to maximum number of spring revolutions, maximum torque, and the geometric and material properties of the material.

$$V_{\text{shell}} = \frac{2.55\pi^2Et^4b^2}{24T_{\text{max}}\theta_{\text{max}}} + 200\pi bt^2$$
The next step in this development is to relate the volume of the spring to its work capacity. This is accomplished using the relationship for the material thickness, $t$, seen in Equation (17). Substitution of Equations (12) and (17) into Equation (16) results in Equation (18).

$$t = \sqrt{\frac{6T_{\text{max}}}{S_p b}}$$  \hspace{1cm} (17)

$$V_{\text{shell}} = \frac{3.83\pi^2 E}{0.738S_p^2} W_{SS} + 600\pi S_b T_{\text{max}}$$ \hspace{1cm} (18)

Equation (19) relates the volume of the spring to the work capacity of the spiral spring, $W_{SS}$. Here, $E$ is modulus of elasticity and $S_p$, the maximum bending stress, is given by Equation (14).

$$V_{\text{shell}} = \frac{5.19\pi^2 E}{S_p^2} W_{SS} + 600\pi S_b T_{\text{max}}$$ \hspace{1cm} (19)

A thickness can be chosen that will remain within the allowable stresses using Figure 14 based on a preferred material thickness, $b$, and a known torque at maximum deflection, $T_{\text{max}}$. It should be noted that this figure is correct only for SAE 1095 steel. The set of curves in Figure 14 was developed based on empirical data for a spring width of $b=1$ inch (Carlson, 1978, p.226). The curves adjacent to the $b=1$ inch curve were plotted based on the principle that an increase in torque is proportional to the increase in $b$. In other words, an increase from $b=1$ inch to $b=2$ inches will double the torque for a given thickness.

The weight for the spiral spring, $W_t$, can be expressed similarly using Equation (20). This equation represents the cross section of the spring material multiplied by the active length, $L$. Multiplication by the material density, $\rho$, yields the weight of the spring.

$$W_t = bL\rho$$ \hspace{1cm} (20)

**Figure 14: Spiral spring: Thickness vs. Torque (SAE 1095 polished tempered and blued).**

Again using Equation (12), the weight of the spiral spring as a function of work capacity is given by Equation (21). Here, $W_{SS}$ is the work capacity, $E$ the modulus, $\rho$ the density, and $S_p$ the maximum bending stress given by Equation (14).

$$W_t = \frac{6\pi E\rho}{(0.738)S_p^2} W_{SS} = \frac{25.54 E\rho}{S_p^2} W_{SS}$$ \hspace{1cm} (21)

The equations developed above apply to rotary motion. However, several applications dictate the use of such actuators in conjunction with a transmission device such as a lead or ball screw to
convert the angular motion into linear motion. This can be done easily with knowledge about the transmission.

The efficiency of a lead or ball screw is calculated using Equation (22), where \( l \) is the lead of the screw, \( F \) is the load being driven by the screw assembly, and \( T \) is the torque applied to the screw. Efficiencies for ball screws are defined as a ratio of the output linear work, \( W_{\text{lin}} \), to the input rotary work, \( W_{\text{rot}} \), and generally fall in the range of 0.9 – 0.95.

\[
\eta = \frac{W_{\text{lin}}}{W_{\text{rot}}} = \frac{Fl}{2\pi T}
\]  

(22)

\( W_{\text{lin}} \) represents the work required of the actuating system by the application of interest. Substituting \( W_{\text{lin}} \) for \( W_{\text{ss}} \) in Equations (19) and (21), expressions for volume and weight for spiral springs combined with a lead or ball screw are given by

\[
V_{\text{shell}} = \frac{5.19\pi^2E}{\eta S_b^2} W_{\text{lin}} + 600\pi \frac{T_{\text{max}}}{S_b}
\]  

(23)

\[
W_{\text{t}} = \frac{25.54E\rho}{\eta S_b^2} W_{\text{lin}}
\]  

(24)

Note that in this case, the weight of the transmission is neglected due to the wide variety of designs available. The engineer must take this into account when designing the actuating system.

**Permanent Magnet Brush Motors:** Permanent magnet (PM) motors are used extensively and have proven to be very reliable in space applications ranging from cooling fans and latch mechanisms to solar array deployment actuators and antenna pointing mechanisms. Brush motors are generally used for short duration applications where the brush life is not a limiting factor.

Figure 15 and Figure 16 contain data obtained from Kollmorgen (1998), a popular manufacturer of DC motors, that relate volume and weight to the continuous torque, \( T_{\text{cont}} \), for typical brush motors. The torque and angular velocity data used to produce these charts were taken from motor/amplifier combinations suggested by Kollmorgen. In Figure 15, the curve fit shows a slope of 6.5 in^2/lb, with an \( R^2 \) value of 0.96. The volume for the brush motor alone can therefore be estimated as

\[
V = 6.5 \text{ in}^2/\text{lb} \times T_{\text{cont}}
\]

(25)

where \( V \) is given in \( \text{in}^3 \) and \( T_{\text{cont}} \) in \( \text{in-lbs} \). Figure 16 shows a similar relationship for motor weight as a function of continuous torque. Here, the curve fit yields the expression

\[
W_{\text{t}} = 0.65 \text{ in}^{-1} \times T_{\text{cont}}
\]

(26)

Assuming the torque to be constant, the rotary work provided by the actuator over a specified angular displacement is given by \( W_{\text{rot}} = T_{\text{cont}} \times \theta_{\text{max}} \) where \( \theta_{\text{max}} \) is the rotation of the actuator. Substituting into Equations (25) and (26) the volume and weight of the motor are expressed in terms of the rotary work, \( W_{\text{rot}} \), and \( \theta_{\text{max}} \) as follows.
Substituting Equation (22) to account for a linear transmission, Equations (27) and (28) can be converted for linear applications as given by Equations (29) and (30). These equations apply for the application of ball screws, lead screws, and other transmissions used to convert the rotary work into linear work.

\[
V = 6.5 \text{in}^3/\text{in} - \text{lb} \frac{W_{\text{rot}}}{\theta_{\text{max}}} \\
W_t = 0.65 \text{lb/in} - \text{lb} \frac{W_{\text{rot}}}{\theta_{\text{max}}} 
\]

Summary of Actuators: Table 1 summarizes the expressions relating volume and weight to the required work capacity of each actuator. For each actuator, the relationship between volume and work capacity, weight and work capacity, and the nature of the derivation is given.

**Load Matching**

Once kinematic alternatives and potential actuators have been selected, there is enough information to define the physical constraints imposed by the system, loading requirements of the kinematic alternatives, and work capacities of the potential actuators that will meet the requirements of the kinematic designs. These steps constitute the process of “load matching” actuators to a particular application. By examining the load requirements of the kinematic system, characteristics of actuators that meet the requirements can quickly be determined and compared.

To match the capabilities of various actuator types to the work requirements of the application, two definitions of work are discussed. The term actuator work capacity refers to the maximum amount of work that the actuator is capable of producing over a given displacement, regardless of the input required by the application. In contrast, the required work capacity for a selected actuator is the input required by the application. The required work capacity is equal to the load history determined by the kinematic analysis times any factor of safety that the engineer wishes to apply.
### Table 1: Expressions relating volume and weight to required work capacity.

<table>
<thead>
<tr>
<th>Actuator Type</th>
<th>Volume vs. Work Capacity</th>
<th>Weight vs. Work Capacity</th>
<th>Derivation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pneumatic</td>
<td>( V = 0.016 \text{in}^3/\text{in-lb} \cdot W_p + 70 \text{in}^3 )</td>
<td>( W_t = 0.033 \text{lb/(in-lb)}^{0.63} \cdot W_p^{0.63} )</td>
<td>Empirical, Mfg. Data</td>
</tr>
<tr>
<td>Compression Spring</td>
<td>( V_{shell} = \frac{G}{2S_t F_{max}} \left( \frac{2.55 F_{max} D}{S_T} \right)^{\frac{3}{2}} )</td>
<td>( W_t = \frac{\rho \pi G_d}{S_T D F_{max}} \left( \frac{2.55 F_{max} D}{S_T} \right)^{\frac{3}{2}} W_{CS} )</td>
<td>Closed Form</td>
</tr>
<tr>
<td>High Output Paraffin Actuator</td>
<td>( V = 0.021 \text{in}^3/\text{in-lb} \cdot W_{HOP} )</td>
<td>( W_t = 0.0023 \text{lb/in-lb} \cdot W_{HOP} )</td>
<td>Empirical, Mfg. Data</td>
</tr>
<tr>
<td>Spiral Spring (rotary motion)</td>
<td>( V_{shell} = \frac{5.19 \pi^2 E}{S_b^2} W_{SS} + 600 \pi \frac{T_{max}}{S_b} )</td>
<td>( W_t = \frac{25.54 E_p}{S_b^2} W_{SS} )</td>
<td>Empirical and Closed Form</td>
</tr>
<tr>
<td>Spiral Spring (linear transmission)</td>
<td>( V = \frac{5.19 \pi^2 E}{\eta S_b^2} W_{lin} + 600 \pi \frac{T_{max}}{S_b} )</td>
<td>( W_t = \frac{25.54 E_p}{\eta S_b^2} W_{lin} )</td>
<td>Empirical and Closed Form</td>
</tr>
<tr>
<td>Permanent Magnet Brush Motor (rotary motion)</td>
<td>( V = 6.5 \text{in}^3/\text{in-lb} ) ( \frac{W_{rot}}{\theta_{max}} )</td>
<td>( W_t = 0.65 \text{lb/in} ) ( \frac{W_{rot}}{\theta_{max}} )</td>
<td>Empirical, Mfg. Data</td>
</tr>
<tr>
<td>Permanent Magnet Brush Motor (linear transmission)</td>
<td>( V = 6.5 \text{in}^3/\text{in-lb} ) ( \frac{W_{lin}}{\eta \theta_{max}} )</td>
<td>( W_t = 0.65 \text{lb/in} ) ( \frac{W_{lin}}{\eta \theta_{max}} )</td>
<td>Empirical, Mfg. Data</td>
</tr>
</tbody>
</table>

As shown in Figure 3, loads seen by the actuator throughout the deployment of the mechanism can vary greatly depending on the kinematic design. The shape of the load profile helps to determine the suitability of an actuator for a given application and can be used to arrive at a quick analysis of the feasibility of different actuators.

Actuators considered in this study fall into two basic categories, “active” and “passive.” Active actuators include the pneumatic cylinder (for the purposes of this study, it is assumed that a sufficient supply of air is available to ensure constant pressure throughout the deployment), HOP actuator, and the motor and lead screw assembly. Load profiles describing the work capacities of these actuators are generally rectangular. Actuators of this type typically rely on an external source to provide the driving energy. Passive actuators such as the compression spring and the spiral spring and lead screw assembly have force profiles that vary as the actuator travels through its range of motion. These actuators typically rely on internal energy sources such as the energy stored as a result of spring deflection.

Figure 17 shows the relationships between the input required by the particular application and the actuator work capacities. Starting with the application force requirements, the three types of actuator capacity profiles are drawn such that they never fall below the force requirements of the mechanism. The minimal vertical distance from any point on the required mechanism force curve to a corresponding point on a particular actuator work capacity curve dictates the safety factor the designer wishes to incorporate. The required rectangular profile of an active actuator can be readily drawn...
knowing the maximum required force, $F_{\text{max-A}}$, needed by the mechanism and the displacement needed from the actuator, $x_{\text{max-A}}$, as seen in Figure 17. Values for the required work capacities of active systems can be found using Equation (31).

$$W_{\text{cap}} = F_{\text{max-A}} \times x_{\text{max-A}}$$  \hfill (31)

![Figure 17: Actuator force profiles for compression springs, HOP actuators, and pneumatic actuators.](image)

The work capacities for compression springs are described by a triangular force profile due to the decrease in available force as the springs extend. The slope of the line, or spring constant, can be chosen to minimize the total work required of the spring to deploy the mechanism. This can be accomplished by selecting a pivot point $(x_p, y_p)$, around which the actuator work capacity profile may rotate slightly without intersecting the required force profile. Given the coordinates $(x_p, y_p)$ of the pivot point, the intersections with the travel axis, $x_{\text{max-P}}$, and the force axis, $F_{\text{max-P}}$, that minimize the area of the triangle (required work) are given by Equation (32). Extending a line from $x_{\text{int}}$ through the pivot point, and to the resulting intersection with the force axis, $F_{\text{max}}$, results in a force profile for the compression spring that minimizes the amount of work needed to deploy the mechanism. The work capacity of the compression spring, $W_{CS}$, is represented by the area under the curve and is given by Equation (33).

$$x_{\text{max-P}} = 2x_p, \quad F_{\text{max-P}} = 2y_p$$  \hfill (32)

$$W_{CS} = \frac{1}{2} F_{\text{max-P}} x_{\text{max-P}}$$  \hfill (33)

Finally, the work capacities of spiral springs are included by transposing the relationship shown in Figure 18 to show the nonlinear relationship depicted in Figure 17. This is discussed in detail under
the section **Spiral Springs**. Again, the only constraint is that the curve depicting the actuator work capacity not intercept the curve representing the required work capacity.

![Figure 18: Spiral spring: Thickness vs. Torque (SAE 1095 polished tempered and blued).](image)

**Selection Based on Secondary Requirements**

Once the minimal actuator work capacity curves have been established in Figure 17, values for work for each type of actuator are easily obtained by calculating the areas under each curve (and using Equation (12) in the case of spiral springs). Given these values for minimal work capacity, the respective measurements for the secondary requirements for each actuator are quickly obtained using Table 1. Actuator selection charts such as the one shown in Table 2 are then used to weigh the different requirements and to acquire quantitative values for use in the trade-off analysis.

The actuator selection chart (Table 2) and trade-off analysis are based on concept selection as used in other methods such as Pugh Concept Selection (Pugh, 1991), House of Quality (Revelle, et al., 1997), and Utility Theory (Thurston and Locasciao, 1994). The objective is to quantitatively determine the degree to which various design alternatives (actuator types in this example) satisfy the guiding design requirements. Measurements of the performance indicators for the requirements are made for each alternative. Relative scores (low, 1 - high, 10) are then given to each alternative based upon these calculations by scaling the magnitudes against the best value. For example, the relative score for volume is calculated by the equation

\[
\text{RelativeScore} = 10 \times \frac{\text{Smallest Actuator Volume}}{\text{Volume of Current Actuator}}
\]

For example, if option B has a volume of 40 in\(^3\) and option A has a volume of only 20 in\(^3\) and is the smallest actuator, option B will score a 5 and option A will score a 10. The relative scoring is used to deparameterize each performance indicator for consistency in utility comparison. In addition, a utility (low, 1 - high, 10) of each secondary requirement is given based on the relative importance of that requirement in the overall design goals. This utility parameter is established by the designer based on the stated criteria of the application or the "voice of the customer." By multiplying the relative performance measurement by the relative utility of the requirement, a weighted score (low, 1 - high, 100) is given to the performance of each actuator type in terms that benefit the overall design goals. Summing the weighted scores gives a total weighted score that quantitatively reflects the relative utility of each concept (actuator type). Given the measurements used in this selection table, the
The maximum possible score for a concept will be 10 (the maximum individual relative score) times the sum total of the individual requirement utilities.

Table 2 shows the example selection chart used for the deployable space structure actuators discussed in this paper. Only the two performance indicators, weight and volume, are included since these were the two stressed by the industrial partner in this research. Other factors, such as cost and reliability, are included later in the tertiary analysis. In the table, volume is given a utility value of 10 since the volume is determined by the cargo head of the launch vehicle and can not be changed. Weight is very important as well since it has a direct impact on the cost of launch, but the relative magnitude of the actuator weight to the weight of the satellite is very small and the constraint is not as rigid as that of volume. Weight is therefore given a utility value of 7. The maximum possible score for a concept will therefore be 170.

The table accentuates the reason brush motors are used so often in space structures. The brush motor scored 170 points in the analysis and was an order of magnitude better than any of the other actuators. This result is intuitive, however, as motors are known for high force to volume ratios. The next highest scores were given by the spiral spring at 10 (6% of the score for the brush motor) and the pneumatic actuator at 8 (5%). Since the initial impetus for conducting this research was to find possible alternatives to brush motors, the brush motor, spiral spring, and pneumatic actuators were all carried forward to the tertiary analysis.

Table 2: Actuator selection chart for secondary requirements.

<table>
<thead>
<tr>
<th>Requirement Utility =&gt;</th>
<th>Volume (in³)</th>
<th>Weight (lbs)</th>
<th>Total Score</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volume</td>
<td>Relative Score</td>
<td>Weighted Score</td>
<td>Volume</td>
</tr>
<tr>
<td>Pneumatic</td>
<td>900</td>
<td>0.3</td>
<td>3</td>
</tr>
<tr>
<td>Spiral Spring</td>
<td>360</td>
<td>0.7</td>
<td>7</td>
</tr>
<tr>
<td>Compressive Spring (single)</td>
<td>620</td>
<td>0.4</td>
<td>4</td>
</tr>
<tr>
<td>Compressive Spring (quad)</td>
<td>600</td>
<td>0.4</td>
<td>4</td>
</tr>
<tr>
<td>HOP (quad)</td>
<td>1160</td>
<td>0.2</td>
<td>2</td>
</tr>
<tr>
<td>Brush Motor</td>
<td>24</td>
<td>10.0</td>
<td>100</td>
</tr>
</tbody>
</table>

The actuator selection chart is extremely useful in selecting the final actuator candidates. However, it is not the final process. Rather than simply taking the top performer, concept selection takes into account that the relative weighting and rating process is not perfect, nor is it the last step. Under most conditions, the output of the actuator selection chart identifies a top class of actuator types rather than a single one. The case shown in Table 2 is an exception unless one realizes that the goal is to identify the most plausible alternative to brush motors. Removing brush motors from the analysis results in the condition just noted, where pneumatic and spiral spring actuators are identified as the “top class” of actuators. These top performers are so close that picking one based upon a very small difference in weighted score is a pick among equals. If possible, most concept selection methods advocate synthesizing concepts. However, this does not apply here. What is necessary is to take the top performing candidates and apply any tertiary requirements to analyze which is the best among equals.
Selection Based on Tertiary Requirements

Tertiary requirements are those less distinct, usually qualitative concerns. Their application should be viewed as a tie breaking process among equally good alternatives. The requirements are typically issues such as supplier relations, lead time, and technological risk. These requirements are applied collectively to the remaining candidate actuator types. If any major differences are realized, the better candidate is selected for implementation. If all of the top candidates are equal, the lead concept from the secondary requirements analysis is selected for implementation.

In the antennae deployment system, the customer-supplied tertiary requirements were cost, simplicity, and reliability. The top candidates, brush motor, spiral spring, and pneumatic actuators were again analyzed in a concept selection chart, but the values were this time subjective since quantitative relationships were unknown. This chart is shown in Table 3. The results of the secondary selection chart are included as the first criteria and reflect a weighting factor of 10. Cost was a driving factor in performing this research, so it was given a relative score of 8. Simplicity of the actuator system is important since it reflects reductions in labor, maintenance, and troubleshooting. However, this is of smaller importance and was therefore given a 6. Reliability is crucial for space deployable structures since one failure to deploy can render the structure useless. It was therefore given a 9.

Based on this analysis, the brush motor still comes out as the obvious best choice from most aspects, scoring 253 points. The spiral spring, however, has significant gains in the areas of cost, simplicity, and reliability, that bring its total up to 224 points. These gains are the result of the lack of required supporting hardware (such as amplifiers and power supplies) and a significant reduction in possible failure modes. In this case, it appears that the spiral spring can serve as a viable alternative to the brush motor, since slight changes in the subjective scoring (which is very dependent on application) could easily move the spiral springs ahead of brush motors.

Table 3 : Tertiary concept selection chart.

<table>
<thead>
<tr>
<th>Requirement Utility =&gt;</th>
<th>Secondary Selection</th>
<th>Cost</th>
<th>Simplicity</th>
<th>Reliability</th>
<th>Total Score</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actuator</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pneumatic</td>
<td>Weighted Score</td>
<td>Relative Score</td>
<td>Weighted Score</td>
<td>Relative Score</td>
<td>Weighted Score</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>3</td>
<td>24</td>
<td>3</td>
<td>18</td>
</tr>
<tr>
<td>Spiral Spring</td>
<td>Weighted Score</td>
<td>Relative Score</td>
<td>Weighted Score</td>
<td>Relative Score</td>
<td>Weighted Score</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>8</td>
<td>64</td>
<td>10</td>
<td>60</td>
</tr>
<tr>
<td>Brush Motor</td>
<td>Weighted Score</td>
<td>Relative Score</td>
<td>Weighted Score</td>
<td>Relative Score</td>
<td>Weighted Score</td>
</tr>
<tr>
<td></td>
<td>170</td>
<td>1</td>
<td>8</td>
<td>5</td>
<td>30</td>
</tr>
</tbody>
</table>

The fact that the tertiary analysis made the selection between brush motors and spiral springs less distinct leads to an important conclusion. In the case presented here, the tertiary analysis had a significant effect on the final selection rather than serving as simply a “tie-breaker.” This points to the need to identify similar quantitative relationships for categories such as cost and reliability and to move those requirements to the secondary analysis. This step is beyond the scope of this paper. However, the usefulness of such an effort is clear given the results from quantifying the weight and volume data.

Conclusion

A quantitative methodology has been developed to aid in the quick selection of actuators. The method eliminates unfeasible actuator types through a primary requirements analysis based on load and displacement capabilities and other "must" requirements. The method then uses the work required by a particular application as the common denominator for comparing conflicting
requirements and a multitude of both displacement and velocity type actuators. Empirical and theoretical expressions have been developed that relate characteristics of secondary requirements such as volume and weight of commercially available actuators to the required work capacity for the application of interest. This singular component has enabled the development of this method. Using these expressions, an actuator selection chart can be developed to give immediate indications of the weighted utility of candidate actuators types. The method has been applied to a space antennae deployment mechanism, but the method is applicable to any mechanism where both force and displacement actuators are options. Extensions to this method include a set of work-based weight and volume relationships for all actuator types and work-based relationships for other secondary requirements such as cost and reliability.

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