

**Design of an Energy Efficient and Economical Actuator for
Automobile Windows**

by
Keith V Durand

Submitted to the Department of Mechanical Engineering
in partial fulfillment of the requirements for the degree of

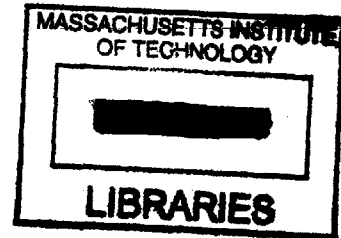
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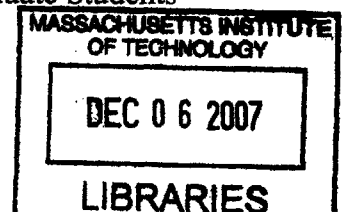
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Abstract

This thesis describes the design and analysis of an efficient, yet low cost, drum driven window actuation system for an automotive power window. The design uses a novel approach that involves using cables to both actuate and constrain the window. Test units were constructed and cycled well beyond the typical life expectancy of a window actuator, demonstrating that this technology is robust enough to be considered as a replacement for current mechanisms. Variations on the design that may produce either better longevity or higher efficiency are also presented, along with the tradeoffs that must be considered.

Thesis Supervisor: Alexander H Slocum
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Chapter 1

Introduction

The drive to increase fuel efficiency has led automotive designers to seek lighter and stronger materials and more efficient components. Engine blocks are commonly aluminum, body stampings are increasingly thinner (with more complex stiffeners), and high strength steels are commonplace. Even the wiring harness has been subject to scrutiny. 42 volt systems may become the new standard¹, allowing the use of thinner wire and smaller components. With metals prices at all time highs, reducing the amount of copper in a car is also an economic advantage.

This research was initiated due to a major automaker's desire to eliminate much of the typical wiring harness. A further goal was eliminating wired connections between the doors and the rest of the car. Aparna Jonnalagadda's SM thesis² details the design and analysis of an energy harvesting mechanism for use in an automobile door. The hope is that vibrations induced from driving may be enough to power the accessories contained in a typical automotive door. Energy harvesting would have the added benefit of reducing vibration inside the passenger compartment, a common goal of automakers.

The challenge of a wireless door is powering the various actuators with only a small amount of energy. These actuators typically include power windows, power door locks, and power mirrors. Current power window designs consume far more energy per cycle than any other actuator in the door. To make a wireless door feasible, the energy consumption of the power window must be greatly reduced. While efficiency can

¹ <http://lees-web.mit.edu/public/In_the_News/A%20New%2042-Volt%20Standard.htm> (2007)

² Aparna Jonnalagadda, MIT SM thesis, January 2007

effectively be “bought,” a cost-sensitive application such as an automotive door requires the final solution to be low in cost and easy to produce.

Energy harvesting is risky. There simply may not be enough energy in vibrations to power current window actuators with any reasonable duty cycle; a backup plan of using an inductive coupling to link the door to the body was conceived. No matter the power source, the goal of this research is clear—a window actuator that uses energy as efficiently as possible, while maintaining reasonable cost goals, must be designed.

Chapter 2

Prior Art

The power window actuator has changed little in the last decade. This could be interpreted as the actuator reaching a level of refinement not worthy of further attention, or it could mean that the power window is ripe for a clean-sheet design. In fact, current actuator designs are highly refined. Cost and durability are adequate—but they are very inefficient. This inefficiency is largely intentional; it provides for a locking action that prevents the window from being forced open. For the purposes of this project, it is important to identify where and why these inefficiencies exist and what can be done to improve efficiency without sacrificing functionality.

2.1. Efficiency and Backdriveability

Efficient mechanisms tend to be backdriveable. This is undesirable for an automotive window, as this means that the window could be forced open, or the window could potentially open due to vibrations from the road. This leads to a requirement that any replacement window mechanism must not be backdriveable. Low efficiencies also imply that a mechanism will not be backdriveable, though non-backdriveability can be accomplished in ways other than inefficiency; this will be discussed later.

In a typical window actuator, the worm gear reduction provides a mechanical element that cannot be backdriven. Earlier examples used a sliding link that would become nearly vertical when the window was closed, preventing a downward force from driving the mechanism. Such linkages can offer both efficiency and a changing mechanical advantage that locks the window in the raised position. However, automakers have strayed from this design, likely in an effort to reduce weight and cost.

2.2. Inefficiencies in Current Mechanisms

Figure 2-1 highlights the main sources of power loss in a typical window mechanism. The worm gear reduction is quite lossy, with an estimated efficiency of 40%. Furthermore, the cable slides over plastic blocks at both ends of the rail, along the entire length of the rail, and on the cable sheath.

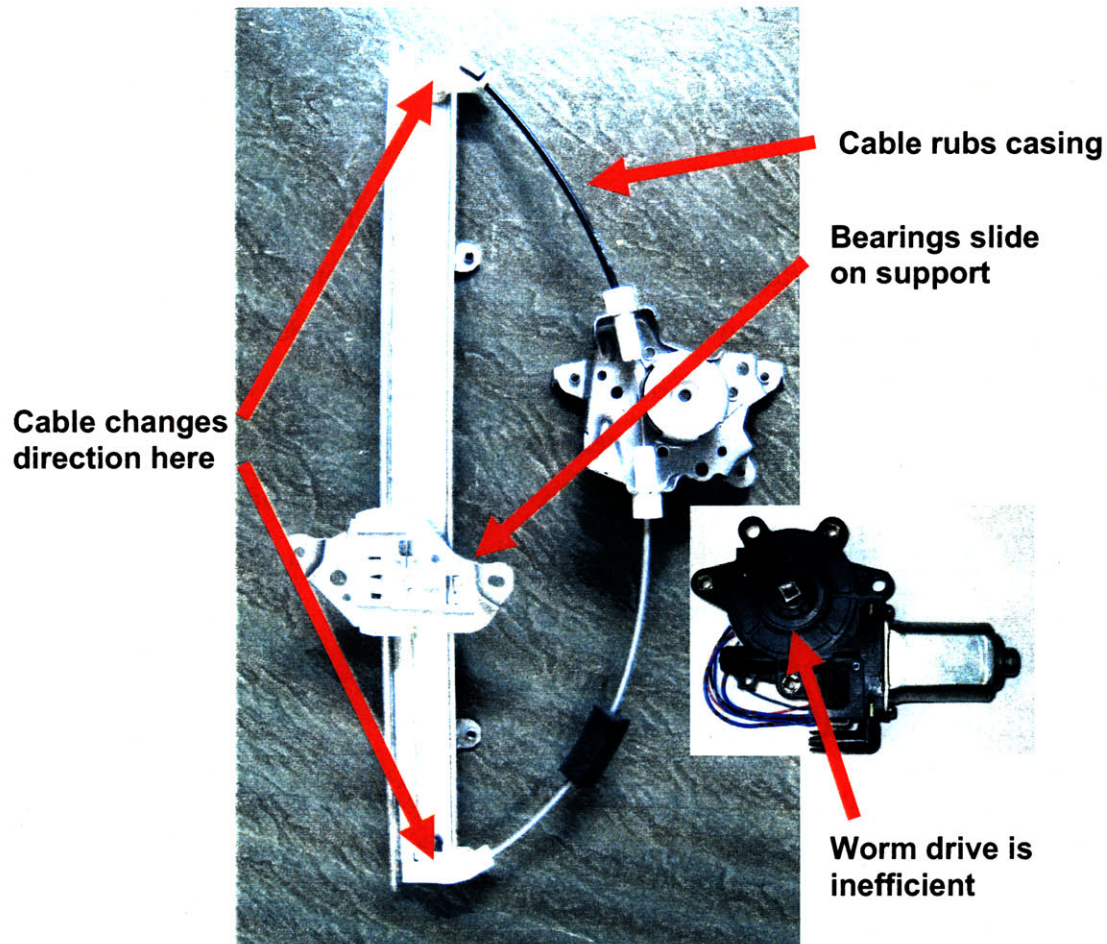


Figure 2-1: Power sinks in typical window mechanisms

The “capstan effect³” easily explains why changing a cable’s direction of travel over a plastic slider or with a cable sheath is inefficient. With a cable sliding over a cylinder, the tension on the output cable is calculated as

$$T_2 = T_1 \cdot e^{-\mu\theta} \quad (2-1)$$

³ Slocum, Alexander H. *Fundamentals of Design*. 2000.
<http://pergatory.mit.edu/2.007/lectures/final/Topic_05_Power_Transmission_Components.pdf> (2007)

T_I is the tension on the input cable, μ is the coefficient of friction of the cable on the capstan, and θ is the wrap angle. For the upper half of the mechanism, a μ of 0.1 (lubricated steel on plastic) and a wrap angle of π are reasonable estimates.

Consider the apparatus shown in Figure 2-2. The tension in the output cable is the weight of the block, W . Any tension in the second cable does work on the load as it rises; multiplying the weight by the displacement yields the work done. Dividing the work output by the work input eliminates the cable tensions, resulting in

$$\eta = e^{-\mu\theta} \quad (2-2)$$

For the above values of μ and θ , this yields an efficiency of 73%. Anywhere the cable changes direction as it rubs against the rail, a cable housing, or a plastic slider is subject to the same high losses.

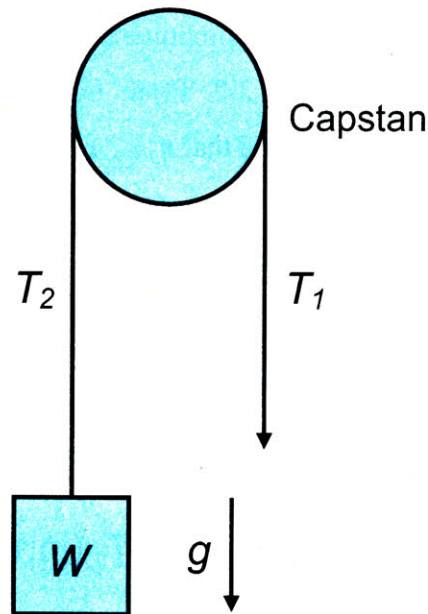


Figure 2-2: A simple model of the cable sliders

To make matters worse, the above model too simplistic. The cable in a typical actuator is preloaded to reduce excess play. This preload causes an increase in T_2 , and consequently in T_I . Now consider an apparatus as shown in Figure 2-3. Suppose the window components weigh 30 N (W) and the cable preload is 50 N (T_p). This means that the tension in the output cable of the capstan is 80 N (weight + preload), and the required

force on the input cable to raise the load is now about 110 N. Subtracting the preload gives a required effort of about 60 N to be exerted on the input cable.

Efficiency, defined as work done raising the window components against gravity divided by the work input, is now 50%, much worse than the previously calculated 73%— and that only included the capstan at the top. Add in another capstan and environmental effects that increase friction as the car gets older, and it becomes clear that routing the cable over plastic blocks and through sheathes to change its direction is responsible for significant losses in a typical power window actuator. Efficiency with preload effects added can be calculated as

$$\eta_{preloaded} = \frac{W}{(W + T_p) / \eta - T_p} \quad (2-3)$$

where W is the weight of the moving components and T_p is the preload tension in the cable. η is calculated from Equation 2-2. From Equation 2-3, it should be clear that preload always decreases efficiency, and that $\eta_{preloaded} = \eta$ if the preload tension is zero.

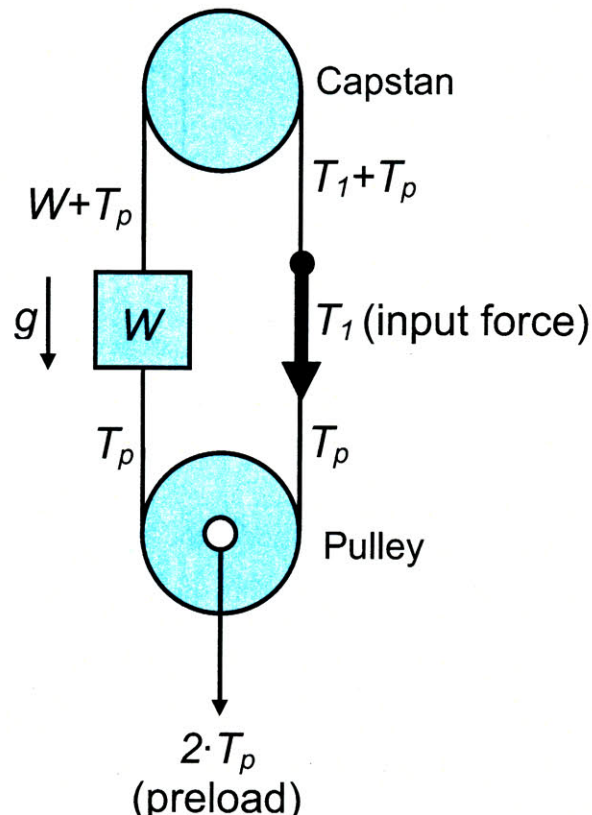


Figure 2-3: A more sophisticated capstan with preload added

The method outlined above can be extended one step further to include the effects of a capstan on the lower half of the actuator. Figure 2-4 is modified from Figure 2-3 to replace the lower pulley with another capstan.

Starting at the cable with preload tension T_p , following the cable around the lower capstan counterclockwise (against cable travel) results in a reduction in tension to $T_p \cdot \eta$. Starting at the cable with tension $W + T_p$, following the cable clockwise (with cable travel) around the capstan results in an increase in tension to $(W + T_p) / \eta$. The difference between these two calculated tensions is the required input force T_I . As before, dividing W by T_I results in a mechanism efficiency

$$\eta_{\text{minimum}} = \frac{W}{(W + T_p) / \eta - T_p \cdot \eta} \quad (2-4)$$

Using the above values of η , W , and T_p results in an efficiency of 41%. Because of the way the preload spring are arranged in the mechanism, the preload tension will decrease under load. Therefore, it can be expected that the actual efficiency will be somewhat higher than the value found with Equation 2-4.

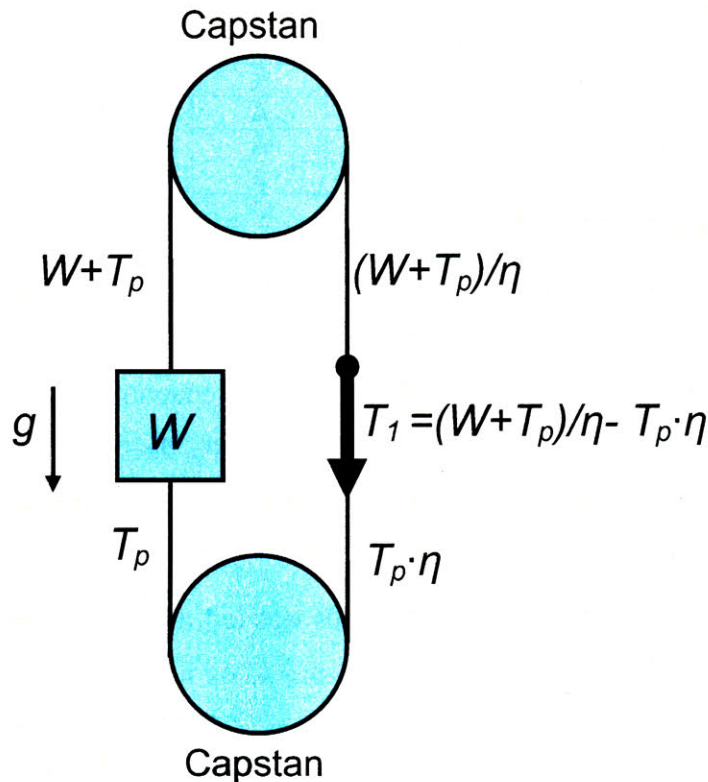


Figure 2-4: Mechanism model with two capstans

An energy budget for the existing design was created and is shown in

Table 2-1. Efficiencies were estimated for each component and drag force was measured with a spring scale. Efficiencies of each component were not measured directly, but the aggregate efficiency was validated with electrical and mechanical measurements of the actuator supplied by the sponsor. The counterbalance force, another unknown, was set to correlate the theoretical results with the different values of motor current obtained when raising and lowering the glass. Overall, the estimated power consumption correlates well with the measured power consumption of the window actuator supplied by the sponsor.

Energy to raise and lower was calculated from the running current and based on the window raising or lowering in 4 seconds each way.

Table 2-1: Power budget for current technology

Weight of moving components	27	N
Travel	0.5	M
Drag force	70	N
Counterbalance	15	N
Travel time (each way)	4.0	sec
Motor efficiency	68	%
Gear efficiency (worm)	45	%
Mechanism efficiency	45	%
Current (raise) (at 13.6 VDC)	5.5	A
Current (lower) (at 13.6 VDC)	3.9	A
Energy to raise	300	J
Energy to lower	210	J
Total energy (per cycle)	510	J

A second power budget, shown in Table 2-2, was created with higher target mechanical efficiencies of 80%, easily achievable with reduction of sliding contact forces and velocities. The motor efficiency was estimated at 68% due to a desire to use low cost motors. Even the highest quality DC motors will have efficiencies of around 80%— and cost many times more.

Drag force on the window from the seals was left the same, as reducing this force is outside the scope of this work. However, it should be noted that in this model energy consumption is linear with drag force. Large gains could be made from improving the seals and window run. One possibility would be the use of inflatable seals that deflate when the glass moves.

In the second energy budget, a counterbalance force exceeding the weight of the glass was proposed. This was done to ensure that the window would be easier to raise than lower in the event of a battery or capacitor running low. If energy storage capacity is not a concern, then there would be no reason to use such a large counterbalance. No matter the counterbalance force, the total energy consumption per cycle is the same; the counterbalance only affects the energy split between raising and lowering.

Table 2-2: Projected power budget for rolling bearing and actuator

Weight of moving components	32	N
Travel	0.5	m
Drag force	70	N
Counterbalance	50	N
Travel time	4.0	sec
Motor efficiency	68	%
Gear efficiency (planetary)	80	%
Mechanism efficiency	80	%
Current (raise) (at 13.6 VDC)	1.1	A
Current (lower) (at 13.6 VDC)	1.9	A
Energy to raise	60	J
Energy to lower	100	J
Total Energy (per cycle)	160	J

2.3. Proper constraint of the window

Every rigid body must be constrained with six constraints. An analysis of current window mechanisms should yield this result. Having more constraints often results in large forces and large frictional losses. For this discussion, SAE convention will be used for the coordinates— x points in the direction of car travel, y points to the right of the driver, and z points down into the pavement. Roll is defined to be about the x axis, pitch is about the y axis, and yaw is about the z axis.

The travel of the glass is largely in the z direction; other motions will be ignored. The constraint on this coordinate is provided by the actuator. Motion in the x direction is prevented by sliding bearings in the carriage that connects the glass to the rail. Motion in pitch is prevented in a similar manner.

The window run acts to prevent movement in the y and yaw motions. The run also acts as a stop in the x direction. Adequate clearance must be designed into the window run such that the glass is not overconstrained in this direction. Doors are not made

accurately enough to allow both a window run and some kind of central bearing and actuator to compete for constraints.

Just as certain motions are permitted by the window run, some motions of the carriage must be left unconstrained. The bearings allow for some movement in the y direction and in yaw.

In summary, the role of any window mechanism is to constrain the window in the plane of the glass. Any other motions should not be hindered by the mechanism. The directions that the mechanism should be soft or stiff in will be important to later sections. Figure 2-5 shows the directions that the window mechanism should be stiff in.

Note: Any non-planar motion should be unhindered (“soft”)

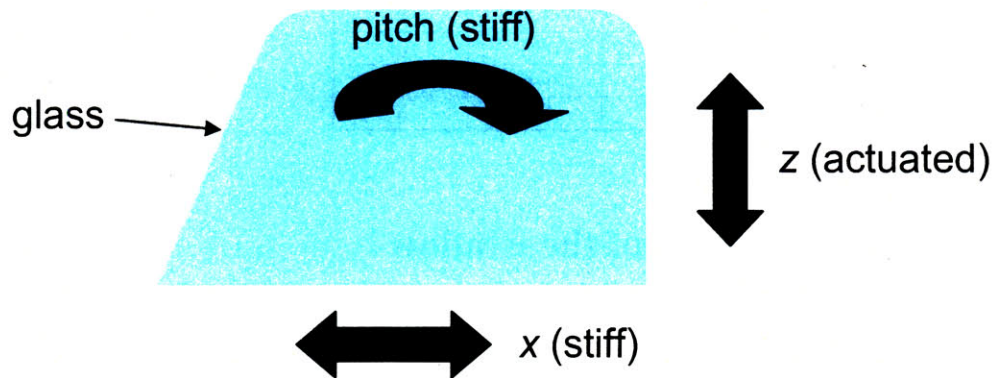


Figure 2-5: Required stiff directions of window mechanism

Chapter 3

Conceptual Designs

It was assumed that a drop-in replacement for the typical window mechanism would be desirable. Typically, a guidance rail (curved to fit the glass) would be bolted to the door, and a carriage on the rail would provide the proper constraints on motions of the glass. Likely, the window run would continue to provide constraint in y , yaw, and roll. Initially, functionality was divided between bearings and actuators. Bearings would provide for constraint in the x and pitch directions, while some kind of actuator would make the window traverse in the z direction.

3.1. Rolling Elements for Increased Efficiency

No matter whether the part in question is a bearing or actuator, efficient concepts involve replacing sliding elements with rolling elements. Figure 3-1 shows a concept for a low cost rolling element bearing. Three crowned rollers are used in a stamped steel track to support the window in the x direction and pitch. Axial float on the rollers provides compliance in the y direction and roll, while these motions are constrained by the window run.

To keep cost low, the rollers would likely be made of plastic and ride directly on steel pins attached to the carriage. Even though there is still sliding contact between the roller and the shaft, the sliding velocity is greatly reduced from a plain slider. With an OD to ID ratio of 5:1, for example, energy consumed in the bearing will be reduced by approximately 80% from that of a plastic slider.

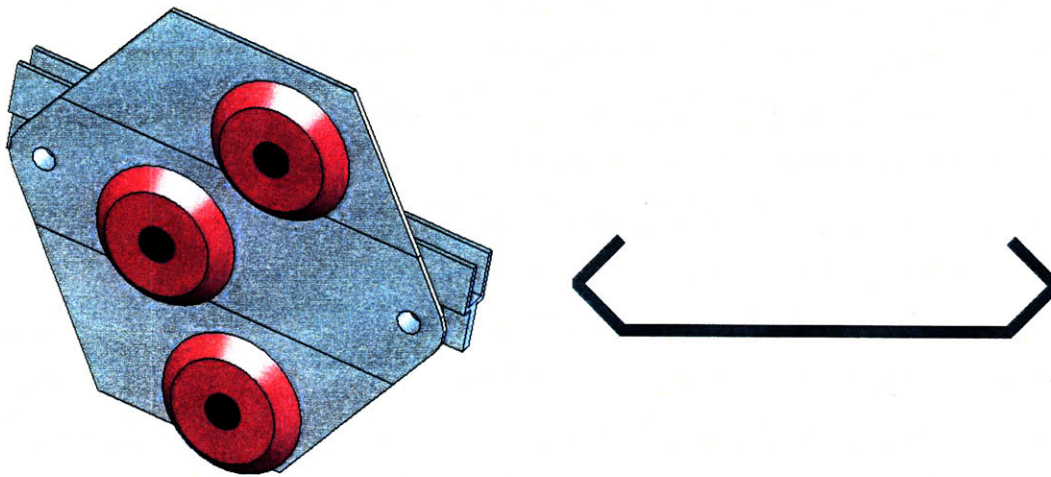


Figure 3-1: Low cost rolling element center bearing and cross section of formed rail

This bearing concept would have to be combined with a high efficiency actuator. Ballscrews and high helix leadscrews were investigated for their ability to compete with the current technology in terms of cost and reliability in harsh environments. Because the window follows a gently curved path, cables are typically used to transmit power from the motor to the glass. Any proposed actuator must be able to let the glass move in an arc. If linear actuators such as screws are used, the screw must be connected at both ends in such a way as to allow it to pivot.

If the motor is allowed to move with the glass, some new possibilities exist. First, a rack and pinion concept was explored. One possible implementation would be to form gear teeth into the guidance rail, and have the carriage (with onboard motor) climb the rail. Another alternative might be to use a friction drive— similar in concept to a rack and pinion, but without the teeth. For such a design, slippage must be prevented.

While these concepts showed promise, another concept was pursued with greater interest due to its potential to provide a very low cost alternative while simultaneously achieving large efficiency gains. This novel actuator will be introduced in the next section and is also the focus of the rest of this work.

3.2. Mockup rolling bearing and actuator

In a second concept, the functionality of the bearing and actuator were combined. Figure 3-2 shows the first mockup of a rolling bearing and actuator concept. Four cables are used to provide stiffness in pitch and actuate the glass, while the window run provides stiffness in roll and yaw. A threaded spool controls the wind of the cables so they do not interfere with each other. The spool should also be constrained to the rail to prevent motion in the x direction, this is accomplished easily with a raised portion of the spool that rides in a slot in the rail.

How the cables are wound warrants a careful explanation. Note that in the picture, two nearly collinear cables (wires) approach the spool from each end of the rail. They are offset by one lead of the screw, or in the case of the mockup, one diameter of the wire. The spool is threaded with the same handed thread all the way across; in the case of the picture below, the lower string approaching the spool to the right of the upper string implies a right hand winding direction. When the wire meets the spool, it spirals away from the other wire. The wire then terminates to the spool. This configuration is repeated a few inches over to provide pitch stiffness. While this concept is stiff in the necessary directions, it is compliant in the y and yaw directions; this is very desirable to prevent overconstraint.

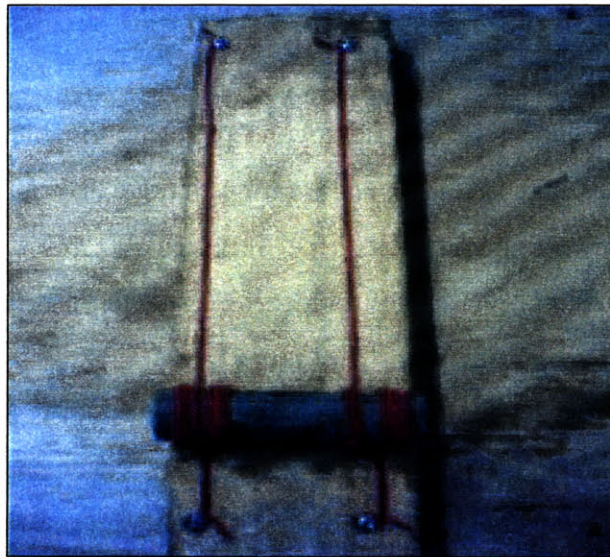


Figure 3-2: Mockup of rolling bearing and actuator

A motor and gearbox (not shown) attached to the spool would actuate the mechanism. In this configuration, the motor would travel with the glass. In the simplest configuration, the reaction torque from the motor must be counteracted by the glass. Alternatively, some means of constraining the motor to the rail that allows translation while preventing rotation could be used. The only major reason to require the motor to be stationary is to eliminate flexing of wires. Should energy harvesting be feasible, the window could carry its own power source. Alternatively, several robust ways of delivering power to the motor exist.

An alternate design where the cable approaches the spool, wraps a few times around, and then continues on to the other end of the rail was briefly considered. Worries of slip creating an angular pitch error prevented this from receiving strong consideration. However, in the case where some other kind of bearing is used to constrain the pitch of the window, a single cable could be used as part of the actuating scheme. Using a single cable and a traveling motor on a rail similar to existing technology might give many of the efficiency benefits of the proposed system, and could result in fewer changes to tooling and production techniques. Additionally, the motor and spool could be oriented such that the motor torque acts in the pitch direction, and would be accommodated by the same bearings that prevent rotation of the glass.

3.3. Concept Selection

To decide which concept to pursue, a concept selection table was created. Table 3-1 lists desired qualities in an actuator and a rating of +, -, or 0 for each concept. The baseline “existing slider bearing and cable drive” received all zeros. The desired qualities were low cost, environmental resistance, robustness (expected lifespan), efficiency, and manufacturability.

For cost, only the rolling cable concept received a “+”. It is expected that due to efficiency, a much smaller motor and gearbox can be used. The rest of the system should be quite familiar to manufacturers, involving steel stampings, plastic moldings, and cables. Ballscrews and high helix screws were given a “-” on cost. Even in large quantities, two manufacturers quoted prices for the screws exceeding the cost of the existing window unit by several times (the actual cost of the existing unit is proprietary

and not to be disclosed). In an automotive manufacturing setting, these prices would surely come down, but probably not as low as what is currently in use.

Table 3-1: Concept Selection

	Cost	Environmental Resistance	Robustness	Efficiency	Manufacturability	Total
Existing slider bearing and cable drive	0	0	0	0	0	0
Ballscrew and rolling element bearing	-	-	+	+	+	1
High helix leadscrew and roller bearings	-	0	+	+	+	2
Rolling cable actuator and bearing	+	0	+	+	0	3

Ratings for environmental resistance were based on how the system might perform covered in water and dirt. Existing units do wear out fairly quickly—the cables get dirty, lubricant washes off, zinc plating on the cables sacrifices itself to prevent rust, and the constant sliding eventually wears through some cable strands. A common failure mode is for the cable to slip off the edge of the spool resulting in failure of the unit. It is suspected that increased friction may be responsible for loss of preload on the cable, allowing it to fall out of the groove in the spool.

The rolling cable concept received a “+” in this category, since the rolling cable should need no lubrication. Additionally, it never slides, preventing strands from getting worn through in a dirty environment. High helix screws were given a “0” and ballscrews were given a “-.” Ballscrews would require seals, adding to cost and requiring a good surface finish on the screw. The low-cost rolled ballscrews used in ABS applications are very rough, which would quickly damage seals.

“+”s were given all around for robustness of all of the concepts—assuming proper sealing of the ballscrew from the environment. All concepts also received a “+” for efficiency—it is certainly not hard to consume less power than the existing design.

The only category in which the rolling cable actuator did not improve on the existing unit was ease of manufacturing. Winding four cables at once can be tricky, especially when prototyping. In production, jigs would make this more straightforward. The added complexity of winding cables was deemed to offset the lack of cable sheath and separate motor housing. Therefore, both the existing unit and the rolling cable

concept were given a “0.” Even without a “+” in this category, the rolling cable design earned the highest rank for further development.

Another concept, generated during development of the rolling cable, didn’t make it onto the concept selection table. This will be described in Chapter 7.

3.4. Anti-backdrive clutch

There was one missing element before further design details could be worked out. Some kind of anti-backdrive clutch is necessary in any efficient design. One option that was considered was a magnetic brake. However, because it would require electricity to operate, a purely mechanical solution was desired. The familiar overrunning clutch from bicycles and automatic transmissions is not viable because it would have to be disengaged for the window to be lowered. A simple solution to the backdriveability problem was found in a cordless screwdriver manufactured by Black and Decker®. This unit contains a “shaft lock” that cannot be backdriven in either direction, yet the device is completely passive. This type of clutch is shown in Figure 3-3.

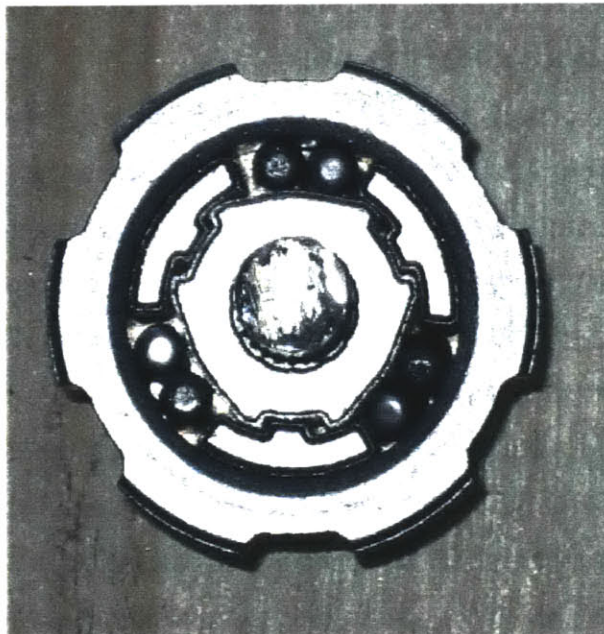


Figure 3-3: Cordless screwdriver clutch

Operation of the clutch is similar to an overrunning (“sprag”) clutch, but there are two sets of rollers; of six rollers, only three engage at a time. Displacement of the output shaft relative to the input shaft causes one set of rollers to engage. Rotation of the output

shaft in the opposite direction causes the other set of rollers to engage. Rotation of the input shaft drives the output directly through a spline that has some backlash. This slop is critical to making the device work. Unfortunately, the output shaft can move a small amount in each direction before the rollers engage. The angle between locking positions is known as the “free angle.” A small spring forcing the rollers apart (not found on the Black and Decker® screwdriver) can accomplish reduction of this angle significantly, effectively limiting the free angle to the backlash in the spline between the input and output shafts. This angle was measured to be around 8°. This angle corresponds to about 2.8 mm of travel with a 40 mm diameter spool.

U.S. patent 3,243,023 describes the operation of this type of clutch in further detail. This device was initially used to eliminate steering feedback on farm equipment. The patent has since expired, making the use of this clutch viable. The patent also mentions the use of a spring to force the rollers apart.

Other manufacturers also offer screwdrivers and drills with shaft locks that are conceptually similar. Screwdrivers often have this feature so the user can drive screws even with a dead battery, or finish tightening a fastener that requires more torque than the screwdriver can provide. Some drills also have this feature so keyless chucks can be loosened or tightened without backdriving the motor.

Skil® offers such a clutch that uses only two rollers, described in U.S. patent 5,016,501. It appears that this patent has been invalidated. Milwaukee® holds two patents on a more sophisticated shaft lock. U.S. patents 7,063,201 and 6,702,090 describe a shaft lock that contains a spring to decelerate the load. Drills present more of a challenge than screwdrivers; when a drill is turned off, a large tool such as a hole saw will try to backdrive the motor and actuate the clutch. This tends to cause unsettling noises for the user and can also damage components. Due to the low speeds involved, it is not expected that a window actuator would require the added features of these shaft locks.

A final example of a shaft lock can also be found in a Black and Decker® screwdriver. Some models have a manual shaft lock consisting of two large gears, one externally toothed, and the other internally and externally toothed. A picture of this shaft lock is shown in figure Figure 3-4. The white gear with two sets of teeth is displaced axially to disengage the lock. The outside teeth engage the ring gear formed into the

black plastic case. Moving this element locks the metal gear (center of photo) to the middle to the plastic case. Actuation of the white gear manually is not necessary; it could be accomplished magnetically, or possibly with some kind of automatic mechanical device. Note that the functionality of the planet carrier for the 2nd stage gear reduction is combined with the inside gear of the clutch.

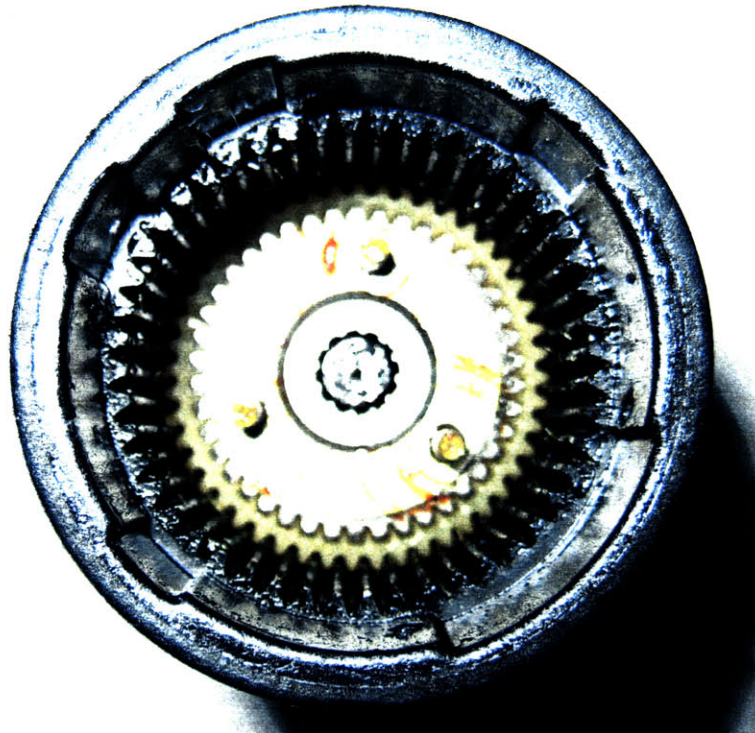


Figure 3-4: A non-automatic shaft lock from a Black and Decker® screwdriver

Chapter 4

Proof of Concept Prototype

Before a lengthy detailed design phase began, a proof of concept was constructed and installed in a door supplied by the sponsor. Using rolling cables to actuate and guide a window in place of sliding bearings on a rail is a radical departure from standard automotive practice, so it was desired to work out as many issues as possible and evaluate the concept before investing much time in the details of implementation. For this first attempt, it was desired to minimize difficult machining operations or rapid prototyping costs. The idea was to construct a simple actuator based on off-the-shelf parts, modified original window actuator parts, and a few one-off parts designed to be simple to machine.

4.1. The Motor and Gearbox

While investigating clutches, it was also found that a typical cordless screwdriver has roughly the power output required (~15 W) and approximately the speed required (~150 RPM) to actuate a prototype actuator. This meant that the entire drivetrain from a cordless screwdriver could be used for mocking up a prototype actuator. In fact, the first prototype incorporated the entire front half of a Black and Decker® model 9078 screwdriver, from motor to output shaft. The batteries and switch were removed to make the unit short enough to fit in the door, and also to allow the motor to be controlled from outside the door. Wires were soldered directly to the motor leads.

The 9078 is one of few screwdrivers with a single speed gearbox and an automatic shaft lock. It also has a powdered metal ring gear in the planetary set instead of teeth molded into the plastic screwdriver housing. These features became important during the detailed design phase of this project. Several common Black and Decker®

screwdrivers and their features are listed in the next chapter in Table 5-1. The next chapter will discuss in more detail why certain properties are desirable in the prime mover. At the time the proof of concept was constructed, the desirable features had not been identified. Only by coincidence, parts for the same model of screwdriver were used in both the proof of concept and the final design.

4.2. Prototype Spool and shaft

The spool was the centerpiece of the proof of concept. It had to meet the following functional requirements:

- Four cables must be terminated to the spool.
- The spool must be threaded to control the wind of the cables.
- The spool must be supported by a bracket attached to the glass.
- The spool must be able to be driven by a cordless screwdriver.

The spool was made from Delrin® bar stock on a manual lathe. Threads were cut into the spool to maintain the cable location. To match the desired speed of the window to the expected speed of the screwdriver, the diameter was set at 30 mm.

Four cables needed to be terminated to the spool in such a way that there would be no high spot at the termination that would interfere with the rail. Electrical ring terminals were crimped on to the ends of the cables. From a strength standpoint, this was a poor solution. However, the force-multiplying nature of the capstan effect allowed the terminations to work temporarily.

Reliefs for the cable terminations and screw heads were cut by chucking the spool eccentrically in a four-jaw independent chuck. The relief was then drilled and tapped for a #10-32 screw to hold the ring terminal on.

A means to connect the output of the screwdriver to the spool was needed. To accomplish this, a steel shaft was welded to the output shaft of a spare screwdriver. This shaft was pinned to the spool, and was supported on two Nyliner bearings installed in brackets described below. To allow for some angular and axial error between the spool and the screwdriver, two ball-end hex keys were cut in half and rejoined in the middle.

This created a coupling that would allow for angular and axial misalignment. It had the added advantages of low cost and use of off the shelf parts. This coupling also did not require modification of the screwdriver.

Figure 4-1 shows the completed spool and coupling, as well as the bracket described below.

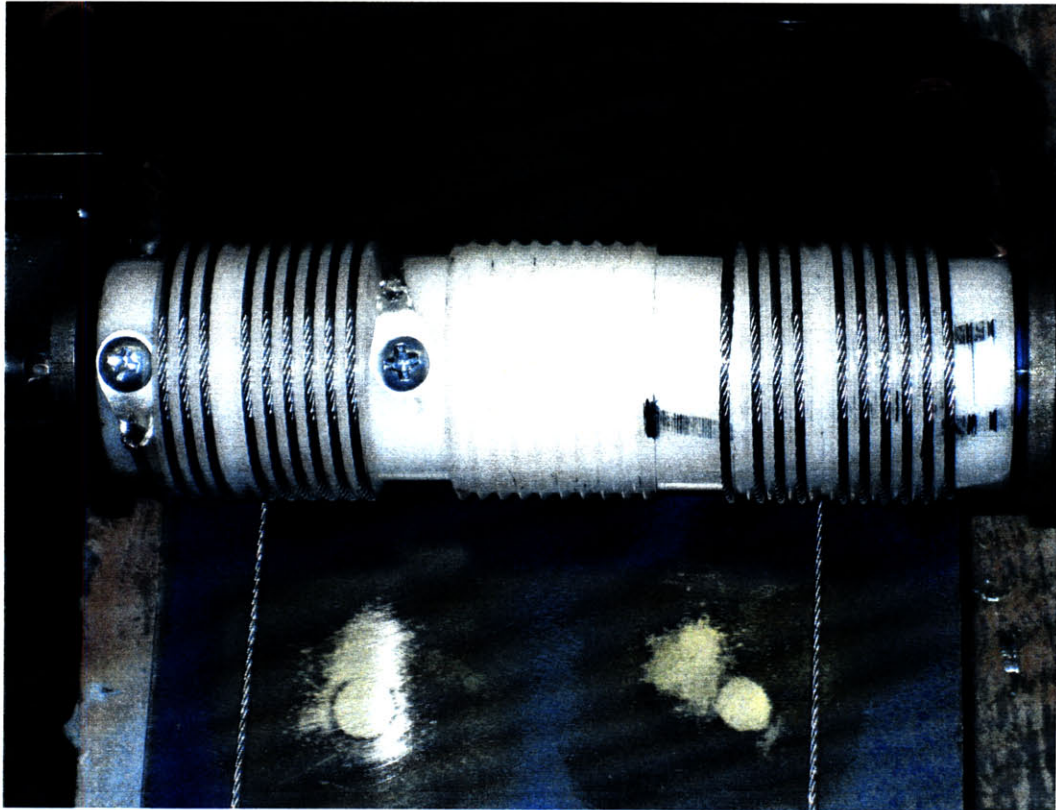


Figure 4-1: Proof of concept spool

4.3. Brackets

Sheet steel brackets were designed to match the existing bolt pattern of the stock window glass in the door supplied by the sponsor. Components were waterjetted from 1/8" stock and welded together. A 1/16" sheet steel cap was welded to the stock rail, allowing the unit to be bolted into the stock door.

A piece of angle section steel was welded to the bracket to support the screwdriver. The plastic case of the screwdriver was clamped to the bracket that carries the glass with a hose clamp, a solution that was not aesthetically pleasing, but simple and effective.

4.4. The Completed Prototype

Figure 4-2 shows the completed prototype system, including spool, brackets, and screwdriver. In this photograph, the cables can be seen running at an angle from top to bottom. The thread on the spool follows this path, effectively eliminating sliding of the cable. The cables in the photo can be seen running from bottom left to top right on the rail, yet the cable appears to be running the opposite direction on the spool. Despite looking odd, this is the correct orientation for elimination of sliding. It is important to remember that the cable is running from the bottom left to the top right of the photo on the side of the spool facing the rail.

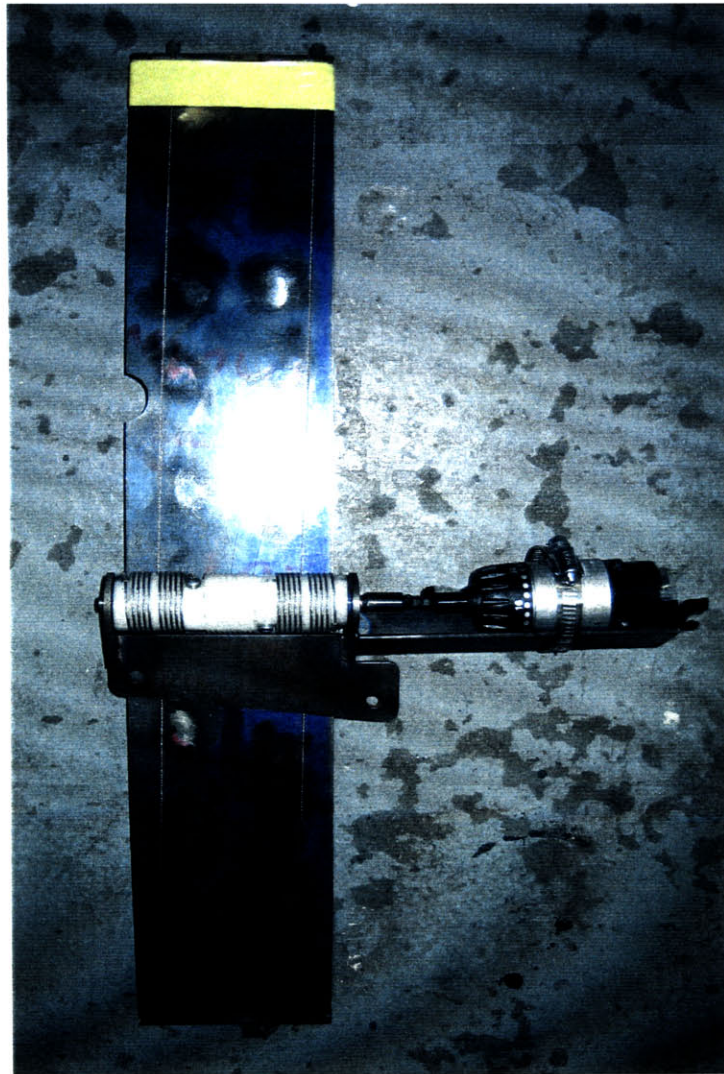


Figure 4-2: Complete proof of concept prototype

4.5. Shortcomings of the Proof of Concept

Several issues arose during evaluation of the prototype. Every one of these issues stemmed from the compromises made to simplify construction—none were the direct result of a flawed concept.

First, the ring terminals used to attach the cable to the spool lacked the strength needed to properly tension the cables. The capstan effect from the cable wrapped around the spool allowed the wire terminals to perform well enough to demonstrate the concept, but the terminations eventually slipped, resulting in loss of cable tension and thus pitch stiffness.

The second flaw was that the motor was somewhat undersized and designed for 3.6 volt operation. Besides the voltage being incompatible with automotive electrical systems, low voltage motors are also inefficient due to the high currents involved. As the prototype did not have a counterbalance spring, the motor was able to lower, but not raise, the window. This led to the conclusion that a counterbalance spring is necessary, a conclusion that will be challenged in the closing comments.

Third, it was initially thought that the window run might provide adequate constraint in the x direction. This turned out to not be true—the OEM version of the window actuator used the actuator to control this motion, and the window run was left with enough clearance to prevent overconstraint.

The proof of concept was modified to add a constraint in the x direction. This was done by bolting a strip of aluminum along the center of the rail, as shown in Figure 4-3. In production, this would best be done by forming a step into the sheetmetal rail. It was realized at this time that by making the step a depression instead, a constant force spring (to act as a counterbalance) could be packaged in the groove. Because the spool would be rotating with the constant force spring, the relative velocities would be minimal, maintaining the goal of high efficiency. With this revelation, work on a beta prototype design began.

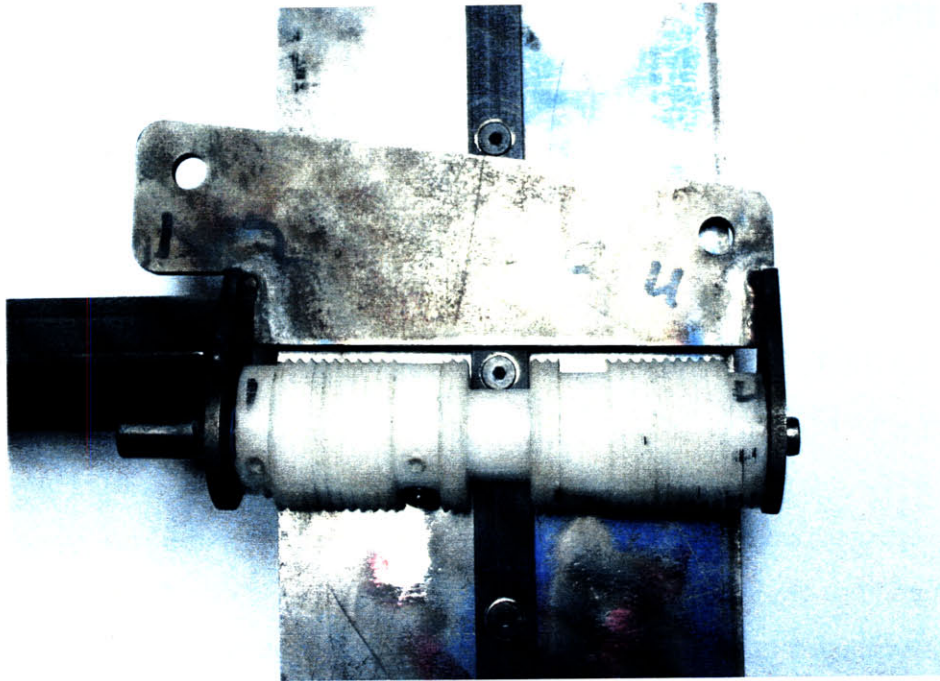


Figure 4-3: Rail modified to add a constraint in the x direction

Chapter 5

Detailed Design of the Rolling Cable

Actuator

Based on issues found with the proof of concept, the design goals for the new actuator were set as follows:

- incorporate a lower speed motor with higher torque
- make unit shorter (i.e., package the drivetrain inside the spool)
- address the cable terminations in a more robust manner
- design a restraint in fore and aft direction (x direction)
- design the unit with mass production in mind

5.1. Conceptual Design of the Spool

To keep the actuator as small as possible, one of the design goals was to put all of the drive components within the spool, including the motor, the planetary reduction, and the shaft lock. Further in the development stage, it was found that putting a motor in the spool was not feasible due to space constraints. The decision was made to leave the motor outside the spool, but place the other power transmission components inside.

Placing the components in the spool required some method of loading the parts during manufacture. The relatively large diameter of the planetary reduction best fit in the center, where the spool was expected to be larger to prevent movement fore and aft. This concept is shown in Figure 5-1. As mentioned in the previous chapter, the raised portion also makes an excellent location for mounting a constant force spring to act as a

counterbalance. Because the gearbox would likely be larger than the desired diameter of the threaded portion of the spool, loading parts from the end was not practical. Preferably, a method of opening the spool for assembly could be found. If no reasonable solution existed, a way would have to be found to allow the parts to be loaded from the end, such as using smaller components.

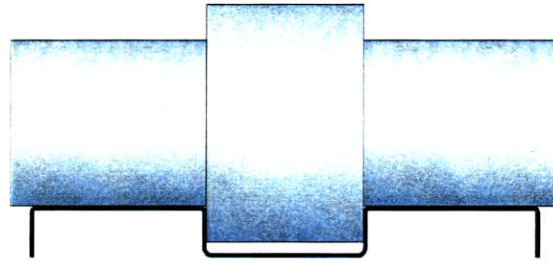


Figure 5-1: Concept spool and rail profile

Two methods of splitting the spool were considered. Splitting the spool as shown in Figure 5-2 (a), would be difficult (and consequently, expensive) due to bending moments created by the forces applied by the cables and the glass. Splitting the spool lengthwise as shown in Figure 5-2 (b) would make the spool much stronger in bending.

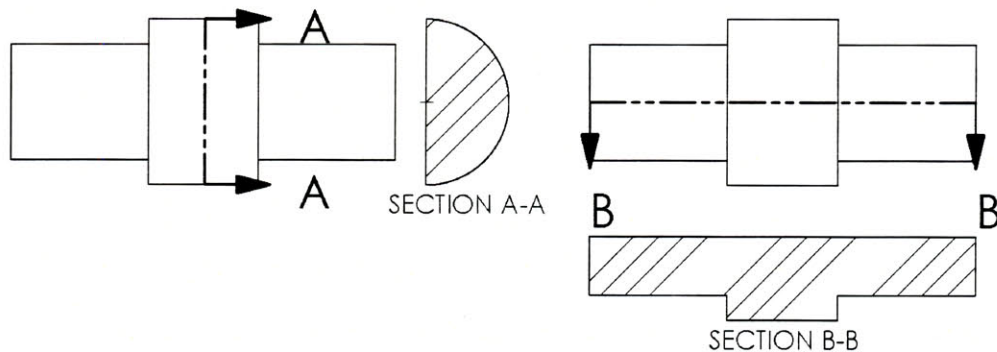


Figure 5-2: Two proposed methods of splitting the spool

It also appeared that the second approach would make it possible to design a spool such that no glue or fasteners would need to be used during assembly. The spool, when assembled into a complete actuator, would have several (~15) wraps of cable around it at all times. As the spool rolls, one cable winds as the other unwinds. These provide

clamping force distributed fairly evenly over the length of the spool. Locating features could be designed to provide mechanical keying and shear strength at the interface between the two spool halves. These locating features are described in detail in Section 5.4

The merits of splitting the spool lengthwise made the decision to use this method straightforward. Furthermore, it was hypothesized that the spool could be manufactured in two identical halves. While this would not be required in production quantities, exploiting symmetry in this manner saves greatly on prototyping costs. Additionally, the effort spent on making the halves identical led to the discovery of a very elegant solution for terminating the cables to the spool. This solution is discussed towards the end of the next section.

5.2. Symmetry Considerations

A thread traverses one lead in one rotation. In a “normal” single start thread, it is not possible for threads to line up with themselves when rotated by one half turn; a peak would meet a valley. There are at least two ways to accomplish the mating of two identical threaded pieces.

The first way is to flip the mating half spool end for end, as long as the part is designed with the correct thread geometry. For the two sets of cables to be parallel, the same hand thread is used on the entire spool. Flipping end to end does not change handedness of the threads, and the peaks and valleys can line up correctly. The necessary mechanical keying can then be implemented by putting one set of features on one side of the midplane, and their mating features on the other side of the midplane.

A second method of mating identical threaded pieces is to use multiple start threads. Double start threads would allow for a spool to be created with identical halves rotated around the long axis. Consider a spool threaded twice, with the second thread cut in the middle of a peak of the first thread. Rotation by half a turn means that the peaks of the first thread will meet the peaks of the second thread.

Using the second method tends to create excessively large leads; the lead must double to make room for the second thread. Packaging considerations ruled double start threads out; the spool would have to be much longer to fit the same amount of cable as

the “flipping” method. Given more flexibility to use a larger diameter or longer spool, or for applications with a shorter actuator stroke, this kind of design could give more flexibility in the design of the internals of the spool, since these features would no longer have to be mirrored.

Splitting the spool in the first manner also provided an ideal solution to the cable termination problem experienced on the proof of concept. By spiraling the last half turn of each thread towards the long axis of the spool, the cable could be terminated in a very robust and sleek manner. The terminal would be completely enclosed in the spool, leaving nothing to interfere with the rail. A pocket created at the interface between the two spool halves could house a swaged cable fitting. This type of fitting typically allows the cable to develop its full strength before breaking. Figure 5-3 shows the proposed thread and pocket in detail. Note that the cables can be installed with the termination in place, a major convenience for assembly.

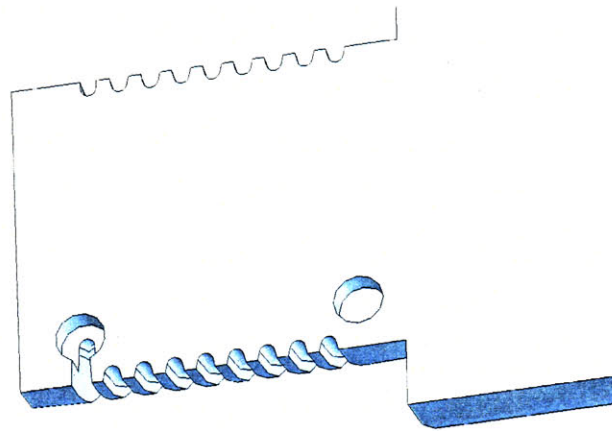


Figure 5-3: End thread and pocket detail (for cable termination)

Making cables of identical length is easier than making cables with an exact difference in length. The pitch angle of the spool is quite sensitive to cable length, so it was desired to keep each set of upper or lower cables the same. This requirement means that each set of upper or lower cables must be terminated on the same half of the spool. Fortunately, this does not violate any of the conditions for a symmetric spool.

5.3. Solid modeling of the spool

The above arguments were enough to begin detailed solid modeling of a workable symmetric spool. A few subtle points concerning symmetry were found along the way.

No matter how a threaded component is manipulated in space, the handedness of the thread never changes. Mirroring of a right hand thread, however, will result in a left hand thread. As a result, mirroring of the threaded features in the solid model of the spool does not yield the desired results.

Instead, the threaded features and the inward spiral for cable termination were modeled as a linear pattern in SolidWorks. Each thread begins with a cable termination, traverses some distance on the order of a few cm, and ends with another cable termination. One end of the thread terminates a top cable, while the other terminates the bottom cable. The threaded feature then repeats some distance over. The distance that the features are shifted over in the second instance is the distance between the cables. This distance is determined by packaging constraints and the desired pitch stiffness. This distance was set to maintain the greatest cable spacing possible while maintaining a spool length of less than 140 mm. The actual cable spacing ended up being 88 mm.

Because four cables must attach to the complete spool, each spool half needs two cable terminations. At the other end of the thread, the thread simply stops one half turn short of an integer number of turns. The thread then continues on the *other* spool half for one half of a rotation before reaching a terminus. If a single cable were wrapped around the thread from one terminus to the other, it would make an integer number of turns—in this case, the number of turns is nine. The number of turns should be chosen to allow the full stroke of the actuator without unwinding the last wrap of cable from either side. This helps ensure the integrity of the terminations.

Figure 5-4 is a transparent view of the inside of the spool; threads show as dotted lines. The circles represent reliefs cut into the spool to make room for swaged cable fittings. With two cables terminated to each spool, the distance between them is an integer number of turns multiplied by the lead. With a 4 mm lead, the circles are 36 mm apart. If the distance between thread starts is correct, the circles will be symmetrically placed such that when the spool is flipped end to end and mated with itself, the circles are concentric.

Equations 5-1 and 5-2 can be used to guide the design of similar actuators. It is assumed that the distance between pairs of cables, D , is a driving variable and that L , the length of the spool, is desired. The spool length then depends on the number of turns N , the lead l , and the distance between the end of the spool and the start of the thread, d . Equation 5-2 sets a condition on N such that the last wrap of cable is never unwound when traversing a stroke length S with a spool radius r . One additional turn is then added because one thread goes unutilized due to the way the cables must be wound; where the two cables approach the spool, one from each side, one thread has no wrap of cable around it. The position of the unutilized thread changes as the spool rotates, but the amount of unused thread is constant. Figure 5-5 shows two identical halves mated together, verifying that a spool meeting the above conditions can indeed mate with itself.

$$L = N \cdot l + D + 2 \cdot d \quad (5-1)$$

$$N \geq \frac{S}{2 \cdot \pi \cdot r} + 3 \quad (5-2)$$

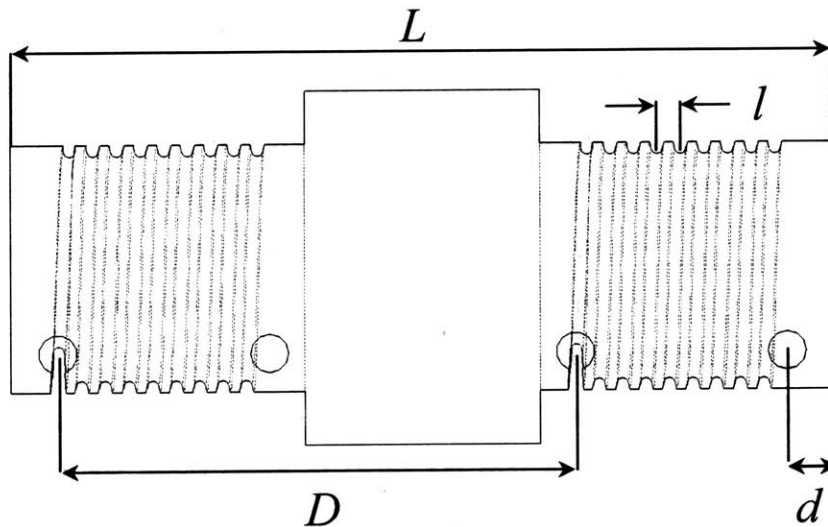


Figure 5-4: Transparent view of spool, showing threads and terminations

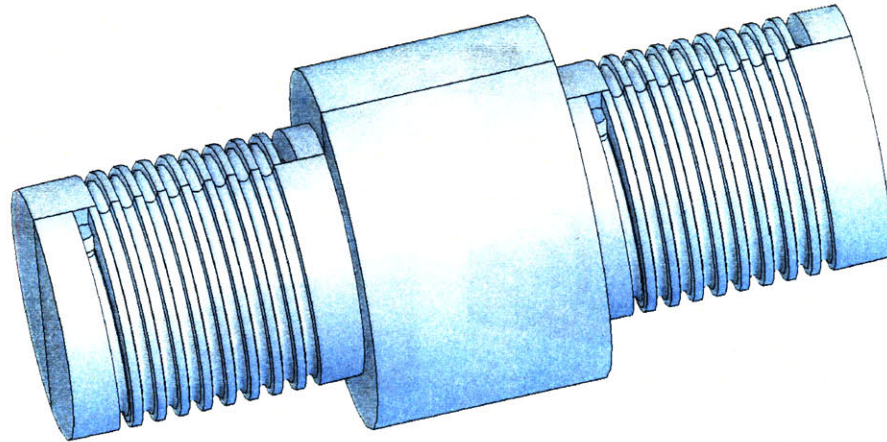


Figure 5-5: Two identical spool halves mated together

With the threading and cable terminations modeled, and with a workable split spool concept sketched out, the focus shifted to detailing the design into a workable actuator. Compliant features that locate the spool halves relative to each other were analyzed and modeled; this is described in the next section. The inside was also detailed to receive drivetrain components; this is described in section 5.5

5.4. Compliant locating features and analysis

Two interlocking mating features were designed to keep the two spool halves in alignment. The male feature is essentially a thinwall tube. The female feature consists of a round hole smaller than the male feature. At four locations, the wall is scalloped. Because the material is basically incompressible, it must be able to go somewhere when the features are mated. When the male feature is inserted, it is forced to assume a more square shape. Figure 5-6 shows an exaggerated cross section of each mating feature.

To determine the proper amount of interference, the male feature was modeled by itself in SolidWorks®. The female feature was considered to be rigid, an assumption justified by the large difference in wall thickness between the male and female features. 3° of draft was applied to the male feature (inside and out) before FEA, as the thicker base would have a significant impact on results. Since spools would likely be molded from an engineering plastic in production, a Nylon 6/10 material was specified for the analysis.

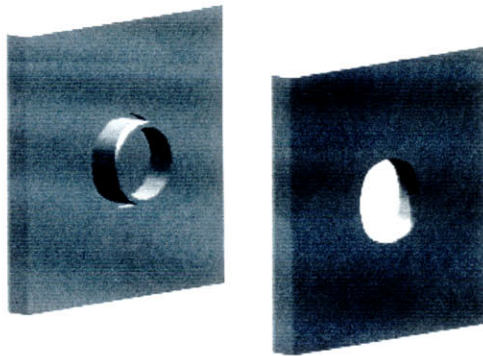


Figure 5-6: Exaggerated shape of mating features

A 1 N force was applied to a small protrusion from the mating feature and the displacement measured. Assuming linearity, the amount of interference to bring the assembly to yield was then calculated. The thickness was then adjusted such that the amount of interference could be molded with reasonable accuracy. It was found that a 0.75 mm wall thickness (at the small end) would cause the material to yield at approximately 80 N. Displacement at this force would be about 25 microns, resulting in the mating components having a difference in diameter of about 50 microns. Figure 5-7 shows the stress in the compliant feature with 1 N applied to each protrusion.

Thinner walls mean that more interference is acceptable, allowing more variation in part size. This comes at the cost of lower shear strength of the interface. The 0.75 mm wall thickness was deemed to give good balance between strength and required accuracy.

The features were modeled on the spool just as in the analysis; straight features were extruded, and then draft was applied. By selecting the neutral plane to be the same for both the male and female features, approximately the same interference was maintained.

A sketch driven pattern was used to place the compliant features and their mates. This ensured that a change in a mating feature would appear in all instances of that feature. Also, the points that drive this pattern were mirrored. This ensures that the mating features would always be in alignment when the halves were mated.

Three features of each type were placed on the solid model of the spool. The mating features were placed near the edges to ensure enough room to place the drive components in the center.

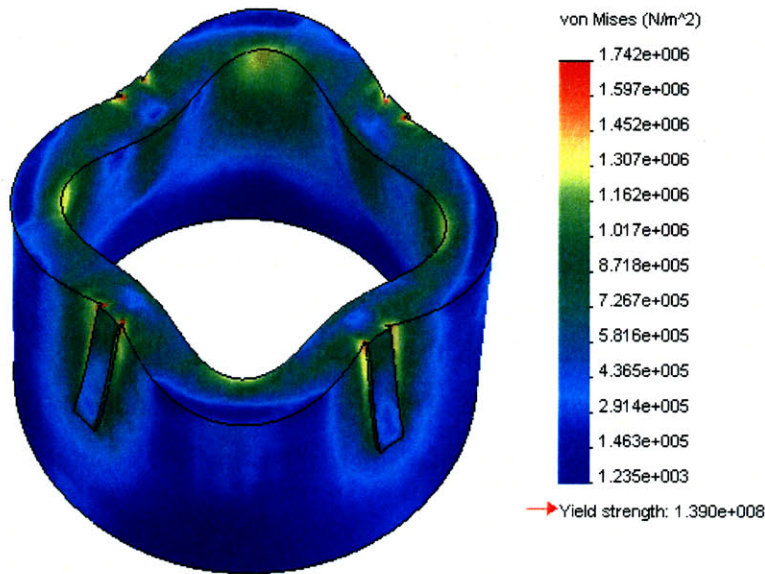


Figure 5-7: FEA results of interference fit on male mating feature

5.5. Selection and fitting of drive components

To address the need for more torque, several Black and Decker® cordless screwdrivers were investigated (meaning taken apart) to evaluate their use as parts donors. Certain features were deemed desirable—of course, an automatic shaft lock was necessary to prevent backdriving. A clutch (normally used to prevent overtightening of screws) was also found to be necessary, but for more subtle reasons. Screwdriver models investigated that didn't have this feature had gear teeth molded into the plastic body of the tool. Models with this feature have a gearbox that can rotate inside the tool. This gearbox is manufactured from powdered metal (good for strength) and conveniently packages a two stage planetary gear set into a small, readily adaptable housing. The clutch itself is of no use to a window mechanism; fortunately, it is very easy to disable. Two speed gearboxes were considered undesirable due to a desire to not use a component that would have unimplemented features. The best solutions usually involve the simplest

parts. The features of several Black and Decker® screwdrivers are summarized in Table 5-1. Desirable features are colored green, and undesirable features are colored red.

Table 5-1: Properties of Black and Decker® Cordless Screwdrivers⁴

	Shaft lock	Slipper clutch	Gearbox type
9078	auto	Yes	1 speed
VP800	auto	Yes	1 speed
PD600	auto	Yes	2 speed
VP750	auto	No	1 speed
9074CTN A	auto	No	1 speed
9072CTN A	manual	No	1 speed
AS600	manual	No	1 speed

At the time of the initial search for a gearbox, only the 9078 had the desired features. As of the time of writing, the VP800 is a new contender in the screwdriver market. It likely has the same or very similar internals as the 9078, but this claim has not been verified.

The inside of the spool was detailed to accept the planetary reduction and clutch components from the 9078 screwdriver. A photo of these components is shown in Figure 5-8. The clutch is not shown in the photo; it consists of dowel pins that engage ramps on one face of the cylindrical ring gear. These pins are spring loaded, and the spring force can be adjusted by the user. Once the ring gear is removed from the case, the slipper clutch is disabled.

The planetary reduction fits nicely in the center of the spool. For prototyping, this area of the spool was left cylindrical—because the ring gear must rotate when the clutch slips, the ring gear did not have a nice way to key it mechanically to the spool. Instead, it was decided to bond the ring gear to the spool while building prototypes. In production, a key or spline would be a more robust solution for preventing rotation of the ring gear relative to the spool. For prototyping, it was decided to gouge both the ring gear and the spool with a carbide rotary file. The adhesive then forms a mechanical key after curing.

⁴ Black and Decker®. <<http://www.blackanddecker.com>> (2007)

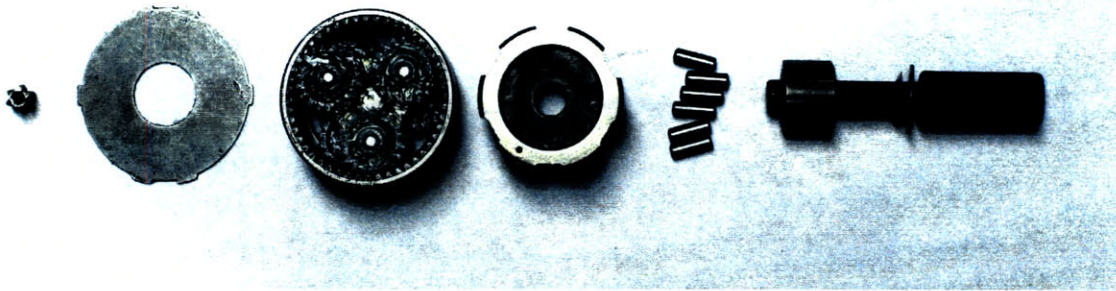


Figure 5-8: Internals of the Black and Decker® 9078 screwdriver

The shaft lock is much easier to mount, as it is splined where it would normally mate to the screwdriver case. The spool was designed to have mating splines to prevent rotation of the clutch. These splines appear in two places on each spool half, since the part must be flipped to mate with itself. Only one set of splines is used on each half. The internal details of the spool are shown in Figure 5-9.

5.6. Final detailing of the spool

Several other details also stand out in Figure 5-9, the final version of the spool. Note that the middle has a relief cut in it for a constant force spring. The width of this was set at 26 mm to allow for a spring 25.4 mm wide, a commonly available size. Also visible are cuts made in each end to allow for the installation of seals. Finally, note that there are two flats at each end of the spool. The idea with these is that a motor could be mounted externally to the spool but rotate with the spool. This decision is complicated by the fact that allowing the motor to rotate with the spool could allow the cables that wrap around the spool to carry power to the motor, eliminating the need for the motor to drag a power cord up and down. These flats do not get in the way of any internal or external component, so the option to drive something with the spool was included in the prototype. Lastly, two semi-cylindrical areas are provided, one on each end. These are 15 mm in diameter when the spool is put together. Two pins attach the spool to a bracket attached to the glass. One pin has a hex broached in it to connect the output shaft of the clutch and gearbox assembly to the bracket.

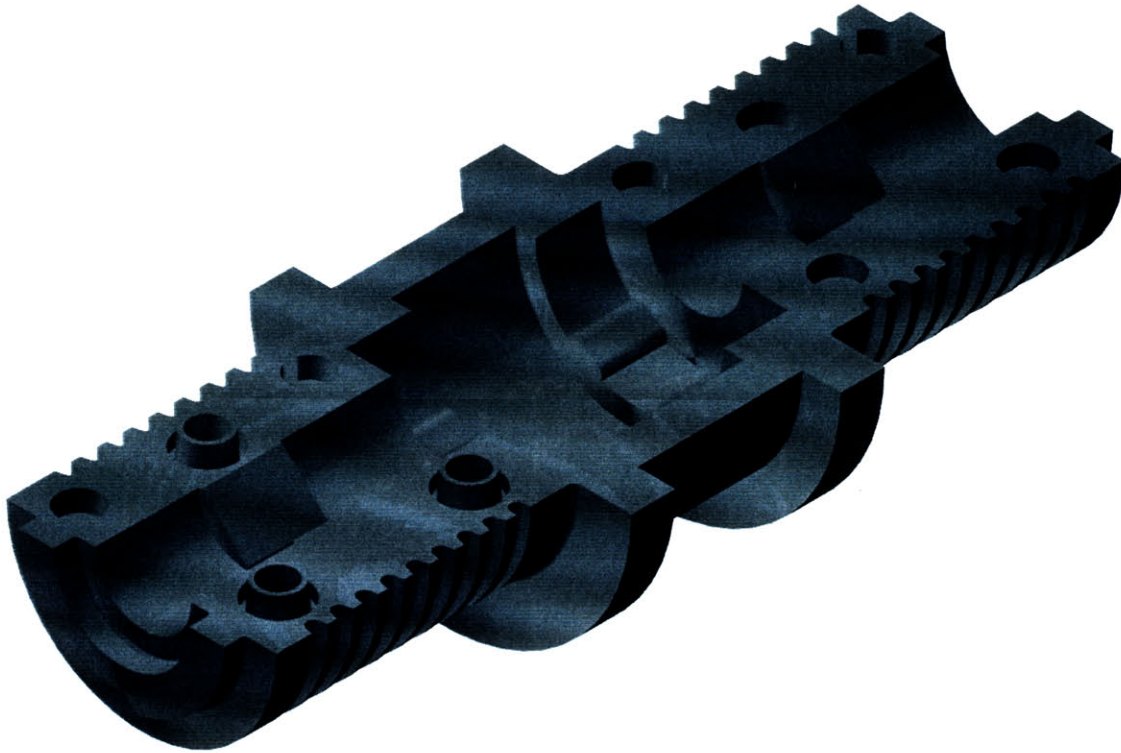


Figure 5-9: Rendering of final spool design

5.7. Counterbalance

A constant force spring provides a nearly ideal counterbalance for the window mechanism. They are simple, light, cheap, and as implied by the name, deliver a nearly constant force throughout their travel.

- The spring should produce a force approximately equal to the weight of the moving window components, which is approximately 40 N
- The spring should be 25.4 mm wide, should fit on a spool approximately 40 mm in diameter, and be able to extend 0.5 m.
- The spring must be able to withstand at least 10000 cycles. This number was chosen to allow four complete window cycles every day for about seven years.
- Because the spring is exposed, it should be able to resist corrosion from saltwater.

Two off-the-shelf constant force springs were found that met the above requirements. Both were available from Stock Drive Products, but appeared to be

manufactured by Vulcan Spring. Vulcan spring also offered to make custom springs meeting the specs, but at a higher price than the off-the-shelf components. Both springs are made of 301 stainless steel, which resists corrosion from saltwater. The spring with higher cycle life is made from thinner stock, which naturally results in some loss of force.

Table 5-2: Standard constant force spring specifications

Part number	Force (lbs)	Free ID (in)	Max deflection (in)	Width (in)	Cycles
E 3X50-SK18P49	5.69	2	32	1	13000
E 3X50-SL15P46	3.5	1.98	30	1	20000

The free ID given in the table above is the nominal ID of a spring that is not installed on a drum. In a typical application, the spring would fit on a drum somewhat larger than the free ID, such that the spring winds tightly around the drum. The manufacturer recommends a spool approximately 2.4 inches in diameter for both of the above springs. See Figure 5-10 for a sketch of the manufacturer’s suggested mounting arrangement.

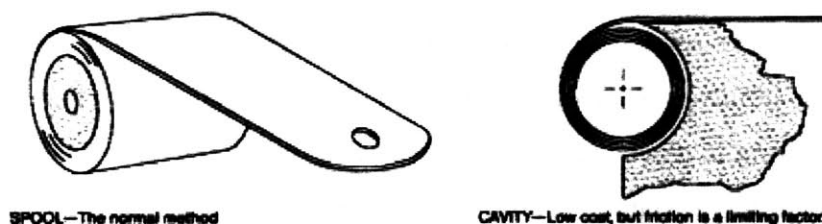


Figure 5-10: Suggested constant force spring mounting arrangements⁵

When these springs fit tightly on a drum, they behave as a capstan and grip the drum. Therefore, the spool must be able to rotate freely, or the spring cannot extend. Note that neither method suggested by the manufacturer involves letting the spring rotate on a fixed drum *inside* the spring—because it often results in binding. It can be done, but the drum must be significantly smaller than the free ID of the spring.

⁵ Vulcan Spring Company®. *Constant Force Spring Mounting Methods*. <<http://www.vulcanspring.com/images/mount.gif>> (2007)

There is little space in a car door to make the assembly larger in diameter, so it was desired that the spring would ride directly on the spool rather than on a sleeve. Additionally, installing a sleeve over the spool complicated the construction of the spool.

The desire to go against the manufacturer's mounting suggestions posed some challenges for the use of a constant force spring. These issues have not been completely resolved, but there are multiple good avenues to pursue.

One challenge of this type of spring is that the free ID of the spring can vary greatly. The Vulcan springs can vary up to 10% from nominal and still be in spec. Additionally, the friction between wraps of the spring is high enough that a single spring can assume nearly any diameter in the specified range. If the end of the spring gets snagged, the ID can shrink until it binds on the drum.

On the prototype actuator, the innermost wrap of the constant force spring was bonded to itself when the prototype window mechanism was assembled. Better solutions certainly exist. Automotive production quantities are high enough to grab the attention of nearly any supplier. The ID of the spring could likely be permanently fixed with some kind of fastener either welded or mechanically attached to the spring. For example, a pin spot welded to the spring could fix the ID by dropping into a hole elsewhere on the spring, or a rivet could be set that would accomplish the same function.

Another possibility is to modify the spool so that the raised areas that engage the rail and keep the spring in place could be replaced with snap rings. This would allow the installation of a thin ring to support the ID of the spring. Another benefit is that the snap rings would hold the spool halves together during assembly.

Yet another good solution may be to eliminate the spring entirely. A part that does not exist weighs nothing, costs nothing, and never breaks. Some results found during testing strongly suggest that elimination of the spring is the best solution. This will be revisited in the next chapter.

One important note is that the sum of the weight of the window and the counterbalance force should never approach the drag force from the window run. This would result in chatter, unless there is no backlash in the geartrain and clutch system.

5.8. Force, Moment, and Stiffness Issues with Cables

Designing the window actuator requires some basic knowledge of the forces and moments that will be experienced by the window being actuated. Cables are very strong in tension; the 1.6 mm cable used on the stock actuator has a breaking strength of over 2 kN. Because the actuator must not develop enough force to hurt passengers of the car, failing a cable from a load applied directly to the actuator is extremely unlikely.

However, if a force that is not inline with actuator is applied to the glass, the moment must be counteracted by forces in the cables. These loads can be considerably higher than from centrally applied loads.

Equation 5-4 estimates the moment that the cables can withstand. This equation assumes that failure occurs when one upper cable and one lower cable are at their breaking load limit and that the other two cables are slack. Equation 5-5 estimates pitch stiffness of the assembly using the same assumptions. Figure 5-11 helps explain the origins of Equation 5-5. Note that the small angle approximation for sine is used in the derivation. Also note that the angular units of K_{pitch} are radians, not degrees.

$$M_{break} = D \cdot T_{break} \quad (5-4)$$

$$K_{pitch} = \frac{M}{\phi} = \frac{K_{cable} \cdot D^2}{2} \quad (5-5)$$

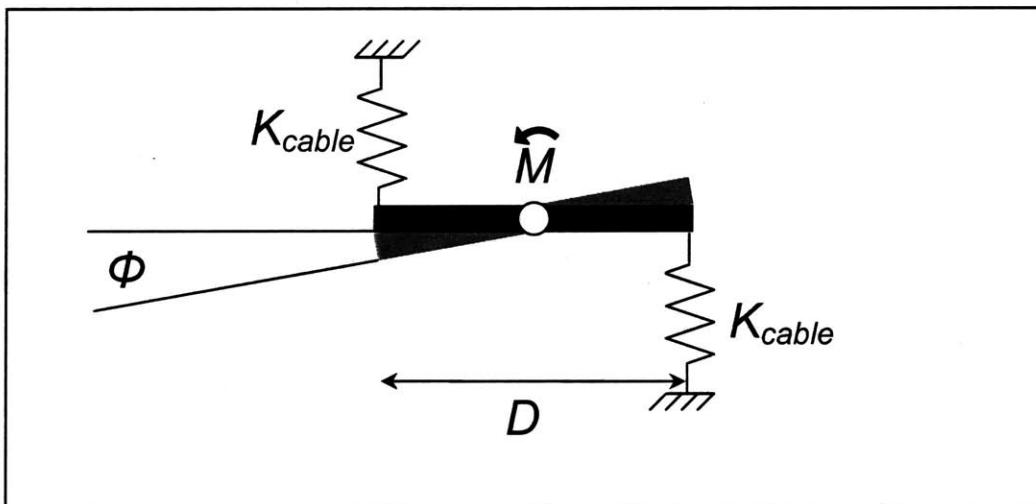


Figure 5-11: Determining pitch stiffness of assembly

The value of K_{cable} needs to be known to complete this part of the analysis. Cables become considerably stiffer after removal of “structural stretch” by prestretching⁶. This stretch is the result of wires adjusting themselves; the amount of structural stretch is quoted by Loos & Co. as less than 1% of the length of the cable that they manufacture.

At higher loads, the cable experiences “elastic stretch.” Loos & Co.® supplies the equation (in imperial units):

$$E = W / D_{cable}^2 \cdot G \quad (5-6)$$

E is percent stretch of the cable, W is the tension in lbs., D_{cable} is the diameter of the cable in inches, and G is a factor based on the construction of the cable. For a 7x19 galvanized cable $G=1.4 \times 10^{-5}$. A 7x19 stainless cable is somewhat less stiff, with $G=1.62 \times 10^{-5}$.

Modifying this equation to use metric units and provide stiffness instead of elongation results in

$$K_{cable} = \frac{D_{cable}^2}{1.448 \cdot G \cdot L} \quad (5-7)$$

W has units of N, and L and D are in mm. k has units of N/mm. G is left unchanged; instead the correction factor of 1.448 is introduced. A 1.6 mm diameter cable with a length of 300 mm gives a stiffness of 420 N/mm. Using this result with Equation 5.5 (and converted into more familiar units) results in a stiffness of 28 N-m/deg for the proposed actuator with 300 mm long 1.6 mm diameter cables spaced 88 mm apart. Also note that the stiffness is essentially constant throughout the actuator stroke. If instead of both cables being 300 mm long, one is 0 mm long and the other one is 600 mm long, the stiffness is the same. This result makes intuitive sense—as the actuator moves, the total length of cable in tension is left unchanged. A unit moment load will result in that length of cable lengthening by a certain amount. Whether the top cable or the bottom cable stretches more is irrelevant, as the angular displacement will be the same.

⁶ Loos & Co.® *Stretch: Structural and Elastic*. <<http://www.loosco.com/Pages/page108.htm#stretch>> (2007)

A more in depth analysis of the pitch stiffness involves adding preload and modeling all four cables as springs. This approach is made complicated by the stiffness of each cable decreasing to zero as it goes slack. Preload will increase the pitch stiffness of the assembly. The amount of preload depends on the goals of the designer.

Light preload, on the order of a few percent of the breaking load of the cable, basically just removes the structural stretch. Equations 5-4 and 5-5 apply.

Moderate preload, up to 50% of the breaking load of the cable, doubles the moment stiffness over cables with no preload, until a cable goes slack. At this point, the incremental stiffness goes back to the value obtained with Equation 5-5. With preload at half of breaking load and the cables of equal length, two cables go slack and the breaking limit of the cable is reached simultaneously. Equation 5-4 still applies, and K_{pitch} is approximately double that given from Equation 5-5. If stiffness is a major concern, this type of preload should be used. Otherwise, less preload means less stress on the components, more compliance in the y direction, less creep, and potentially longer life.

Heavy preload of over 50% the breaking strength of the cable maintains moment stiffness throughout the load range, but actually decreases the moment at which the assembly will fail. There is no reason to use such heavy preload in an assembly such as this.

In making the decision of how stiff a cable to use and how much to preload it, it is useful to find the maximum moment that is likely to ever be applied to the glass. For safety reasons, the window controller must stop the motor when an obstruction applies a force F_{stop} . If F_{stop} is 100 N, and the force is applied 0.4 m from the actuator, then $M_{applied}$ is 40 N-m. To satisfy equilibrium, the applied force must also be counteracted by the cables. Ignoring effects of preload, the tension in one upper and one lower cable will increase to create a force couple equal to the applied moment. The upper cable will also see an additional tension of $F_{stop}/2$. and the tension in the other cable will decrease by the same amount. Equation 5-8 shows that the more highly stressed cable must withstand an additional 500 N over the preload tension.

$$\Delta T = \frac{M_{applied}}{D} + \frac{F_{stop}}{2} \quad (5-8)$$

This suggests that the assembly should have around half of ΔT , or 250 N, of cable preload. This means that should the worst case moment load be applied, full pitch stiffness is maintained. Two cables will go slack just as the limit moment is applied. Therefore, the cables and their connections to the rail should be able to withstand 750 N. The 1.6 mm cable selected has a breaking strength of 2100 N, an appropriate strength when a reasonable safety factor is figured in.

Because the cables control the pitch stiffness of the window, they must be kept tensioned throughout the life of the actuator. Creep, overload, and environmental effects may all contribute to cables becoming slack with time.

The capstan effect can be used to maintain stiffness throughout the life of the car. One end of the rail should simply have a drum that the cables wrap around before terminating to a spring—this point will be illustrated with figures in the next section. The capstan equation can be applied once again to give the amount of force that the cable can take before slipping. Typical values of 0.15 for μ (lubricated steel on steel) and a 540° wrap angle gives an increase in force of around 4:1. A cable with a breaking strength of 2100 N, for example, would need only about 530 N of tension to ensure that the cable would break before the connection would slip.

Breaking the cable, however, is not a desirable failure mode. Instead, in the case of an extreme overload, the cable should slip, displacing the preload spring (discussed below) and allowing more displacement. The 530 N preload tension should therefore be considered an upper bound.

The 250 N preload arrived at from the limit moment method will result in up to 1000 N of force before the cable slips. If a greater force is desired, then the wrap angle can be increased, the friction of the cable on the drum can be modified with lubrication or lack thereof, or the tension could be increased.

It is expected that time, lubrication, vibration, microslip, etc. will cause the preload to nearly equalize across the capstan. During assembly, however, the same 4:1 force ratio provided by the capstan can prevent full tensioning of the cable. This should also be considered when deciding how much preload to apply; it is desirable to remove all slack from the cable and stretch it at least beyond the structural stretch range.

Alternatively, the cable can be seasoned (stretched to near failure limit and cycled a few times) before installation.

To aid in assembly and to maintain tension throughout the life of the actuator, there should exist a preload spring that can maintain tension over a relatively long length of travel compared to the expected stretch of the cable. In the case of overload, cable slippage should result in a hard limit being reached. Both goals can be met easily by terminating the cable such that it compresses a long spring or a stack of Belleville washers.

5.9. Rails and endcaps

The guidance rail was designed to accomplish these four functional requirements:

- Keep the spool centered on the rail (constraint in x direction)
- Force the cables and the spool to follow the curved path of the window
- Accommodate the constant force spring (counterbalance)
- Allow attachment and tensioning of the cables

To keep manufacturing costs low, the rail was designed to be made from rolled or stamped steel. A cross section is shown in Figure 5-12. Due to the difficulty in prototyping this part, it was initially going to be made from two curved aluminum strips bolted to a steel backing sheet. Instead, the sponsor provided steel prototypes of the specified cross section.

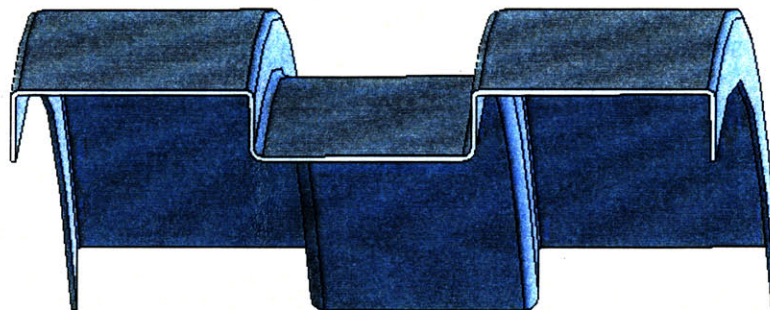


Figure 5-12: End view of rail

A method of terminating the cables to the rail was necessary. Each end had unique requirements, so first the focus will be on a fixed termination used at the bottom of the rail. It was necessary that one end of the rail would have a method for preloading the cable to remove slack. It was further desired that the preloading mechanism would offer enough compliance to keep the cable taught as it aged.

Both brackets are complicated by the path the cables must follow to enforce a rolling without sliding condition. While the cables do remain parallel over the length of the rail, the path they follow is slightly angled with respect to the actuator's direction of travel..

Figure 5-13 shows the bracket that terminates the lower set of cables to the rail. The "x"s are symmetrically placed D apart about the centerline of the bracket; they represent where the cables would attach if no offset was necessary. The slots that hold the cable are spaced d_{offset} away. For a right hand threaded spool, the lower bracket should offset the cables to the left when facing the spool, i.e. when viewing an actuator from the outside of the vehicle. The angled path of the cables can be seen in Figure 4-2, in the "Proof of Concept Prototype" section.

The necessary offset of the cable terminations can be calculated from Equation 5-9. The expression in parenthesis represents the number of rotations of the drum in traversing from one end of the cable to the other, plus one turn that must be added due to the one lead offset between the top and bottom cables. This equation assumes that the spool is rotated such that the terminations are closest to the rail; this equalizes the length of the cables. It is also assumed that the spool is situated in the middle of its travel, so that the offset on the top and bottom brackets are equal. L is the path length of the cables, which must be somewhat larger than the previously introduced S . Setting L to the sum of the spool diameter $2r$ and the stroke of the actuator S is a reasonable starting point

$$d_{offset} = \left(\frac{L}{\pi \cdot d} + 1 \right) * \frac{l}{2} \quad (5-9)$$

The termination for the upper cables is a capstan, as discussed in section 5.8. The wrap angle is 540° , this is the result of the cable wrapping around the capstan 1.5 times

and terminating to a spring on the backside of the rail. Packaging constraints suggested that this location was best for terminating the cable.

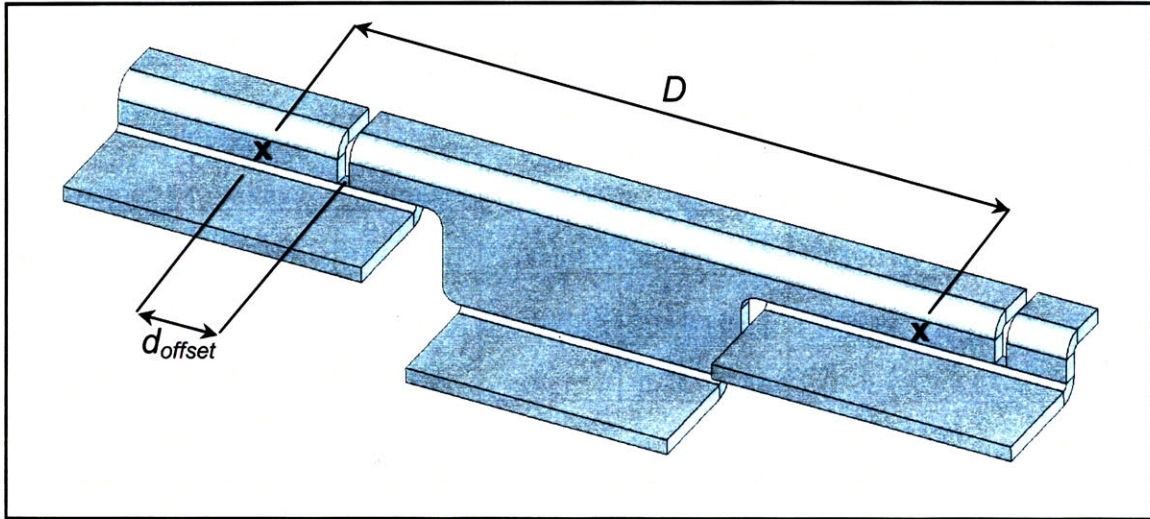


Figure 5-13: Lower cable bracket

5.10. The Motor

Sizing the motor requires knowledge about the expected speed and load. It is desired that the window travel at about 100 mm/s, based on the window opening in 4 seconds. Equation 5-10 can be used to determine the required motor RPM under load. In this equation, v is the velocity of the window, r is the radius of the spool (in units consistent with v), and N is the gear ratio, which is 81:1 for the Black and Decker® gearbox.

$$RPM = 60 \cdot N \cdot \frac{v}{2 \cdot \pi \cdot r} \quad (5-11)$$

The spool has a diameter of approximately 40 mm (measured at the center of the cable), resulting in the motor needing to deliver about 4000 RPM. The power budget provides an additional constraint on the motor—approximately 15 W of power is needed at this speed. These two pieces of information are enough to select the motor.

Three low-cost motor that met these basic criteria were found. All three were manufactured by Mabuchi®, and all were from their “555” motor series. The published characteristics of these motors are summarized in Table 5-3. Note that the data in this

table came from two different sources. The data for the first motor came from a datasheet with conservative specifications; minimum and maximums are provided rather than nominal values. Specifications for the other two motors came from a sheet that provided only nominal values.

Table 5-3: Characteristics of three Mabuchi® motors (brushed DC type)⁷

Motor	Volts	rated speed	rated torque	rated current	no load current	stall current	stall torque
555-PH-3255	12	3950	37.3 mN-m	2.45 A (max)	360 mA (max)	11.8 A	133 mN-m
555-PC-3550	12	4240	25.6 mN-m	1.3	170 mA	9.9 A	221 mN-m
555-VC-3754	12	3610	34.8 mN-m	1.48	200 mA	10.9 A	292 mN-m

DC motors deliver maximum power at approximately half the no load speed. At this point, torque is half of stall torque, and the product of torque and speed is at the maximum of a parabola. Many motors for intermittent use are sized this way, but this method says nothing about the efficiency of the system. The maximum efficiency of a motor is generally found at a lower load than maximum power output.

The current at which efficiency is maximized is the geometric average of the no load and stall current⁸. This desired current can easily be related to the torque by the motor constant:

$$\Gamma = K_t \sqrt{I_{no\ load} \cdot I_{stall}} \quad (5-11)$$

It is known that the motor needs to deliver 15 W at 4000 RPM (420 rad/sec). It is desired to maximize efficiency while operating at this point. This information can be used to find the motor torque desired at maximum efficiency:

$$\Gamma = \frac{P}{\omega} \quad (5-12)$$

Comparing the torques from Equations 5-11 and 5-12 will give an idea of how suitable a motor is for driving a load at maximum efficiency. In the case of the Mabuchi motors, using Equation 5-11 isn't necessary; the rated torque and speed provided by the

⁷ Mabuchi Motor Co., Ltd.® <http://www.mabuchi-motor.co.jp/en_US/index.html> (2006)

⁸ Aveox, Inc.® <<http://www.aveox.com/technical/dc.html>> (2007)

manufacturer is given at maximum efficiency. Maximum efficiency published by the manufacturer is around 70% for both the 555-PC and 555-VC motors. Efficiency was not published for the 555-PH motor, but the value is thought to be consistent with the other two motors.

Equation 5-12 results in a desired torque of 35 mN-m. The 555-VC and the 555-PH are both good matches for required torque at the maximum efficiency. The 555-PH is a better match than the 555-VC for speed. It is also readily available from several surplus stores, so this motor was selected for constructing prototypes.

The downside of these motors is that it is brushed. Due to the limited number of cycles that a power window must endure, it is hypothesized that brushed motor construction will be adequate to outlast the rest of the mechanism. Should a brushless motor be necessary, a Nidec® 22H can also meet the speed and torque criteria. This motor has an advantage over some other brushless motors in that the commutation is done internally, making it a drop-in replacement for brushed motors.

5.11. Final assembly details

The previous sections did not cover some of the more mundane design details. These are mentioned briefly here for thoroughness.

- The cables needed some kind of termination at each end; Nicopress® stop sleeves were selected. This allowed termination of cables with relatively low cost equipment; a roller swaging tool would have been prohibitively expensive. The Nicopress stop sleeves have a somewhat lower strength than the cable, while roller swaged connections would develop the full strength of the rope.
- A flexible coupling was needed to connect the motor to the gearbox. A 1/8" welding rod was ground to mate with the sun gear from the cordless screwdriver, cut to length, and a flat for a setscrew was ground on the opposite end. A flexible coupling intended for 1/8" shaft was used. The motor already had this size shaft and a flat for a setscrew. Note that a setscrew must not be used to secure the shaft on a production version of the mechanism; it would not be robust enough.

- Brackets were designed to attach the spool to the glass and hold the motor in alignment with the spool. These were only slightly modified versions of the brackets used in the proof of concept—a larger bolt circle was necessary for the bushings at each end. These bushings bolt into the bracket and engage the ends of the spool. One bushing is drilled through for the motor shaft to pass through. The other bushing was bored and broached with a ¼” hex. This allowed an Allen key to be used to connect the output of the gearbox to the bracket. A picture of the brackets and bushings is provided in Figure 5-14.
- A motor mount was designed to attach the motor to the bracket. It uses the same bolt holes as the end bushings. The motor is bolted to the mount from the inside, as space did not permit any other approach. The mount is then attached to the bracket during final assembly.



Figure 5-14: Brackets and endcaps

Chapter 6

Construction and Testing of the Actuator

A test system to cycle the rolling cable actuator was designed and built by an undergraduate researcher (UROP), Edward Summers. The test setup replaced the bracket that mounts the spool to the glass with a similar, but larger, bracket. Ball plungers were installed into the bracket; these contact the backside of the rail. These also provide the necessary constraint in the roll direction; this would be normally be provided by the glass. The plungers also add friction, simulating friction from the window run. Adjusting the displacement of the plunger adjusts the friction force.

Two compromises were made to speed the construction of the first version of the test apparatus. First, it had a rapid prototyped spool. Also, the capstan cable tensioning mechanism was not implemented in this version. The next version of the test apparatus had spools made from cast polyurethane, the construction of which was contracted to a local supplier. The mechanical properties of polyurethane are superior to that of the rapid prototype material.

The gearbox was glued to one spool half to allow later disassembly. A toughened superglue (Loctite® 380) was used on the first prototype. Though the superglue never failed, a thickened structural bonding epoxy was used on later prototypes, as it was felt that it had much better gap filling abilities. Note that a production version must not rely on adhesive. For the first prototype, the ring gear was scribed with a sharp carbide tool. The ring gears and spools on subsequent assemblies were heavily gouged with a carbide rotary file, allowing the epoxy to form a mechanical key between the components. Also note that the major gearbox components and the splined outer part of the clutch must be installed before gluing. The sun gear, clutch pins, and clutch output shaft can be installed

after the glue cures. Figure 6-1 shows the rapid prototyped spool with the gearbox and clutch components bonded in.

Figure 6-2 shows an early version of the assembled test apparatus. Figure 6-3 shows the spool and related components in more detail.

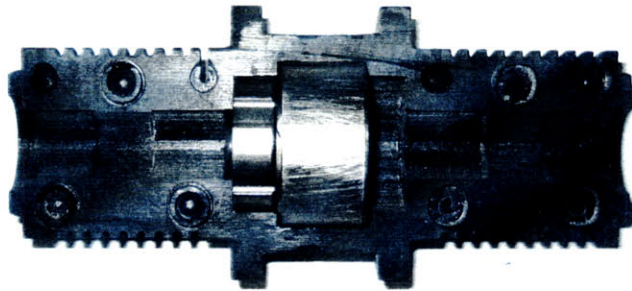


Figure 6-1: SLA spool with components bonded in

In the lower right corner of Figure 6-2, a circuit board can be seen. This board contains the motor controller to cycle the actuator. Many integrated H-bridges and H-bridge drivers exist. However, a heavy duty H-bridge and driver circuit had been constructed from discrete components by the author for another project. This board design accommodated all of the necessary MOSFETs, drivers, freewheeling diodes, and a microcontroller.

Limit switches were installed such that they one would be triggered at each end of the actuator stroke. The microcontroller was programmed to reverse the motor when a limit switch was actuated.

The heavier of the two available springs was used in the cycle test, as it matched the weight of the glass better. The compromise was a shorter rated cycle life, but this turned out not to be the limiting factor. Very shortly after cycle tests began, the adhesive supporting the ID of the constant force spring let go. The sharp end of the spring would catch on the down stroke of the actuator, causing the spring diameter to grow. On the up stroke, friction would cause the spring to collapse. At the very top of the stroke, the

spring would bind, overloading the assembly and causing unsettling noises. It is unknown at what point exactly the glued joint failed. After the spring had experienced 3396 cycles, the spring was removed and tests were continued without it. The rest of the apparatus continued to run for 62109 cycles, and would have continued beyond that. At this point, the test was stopped to start work on an improved test apparatus. The cables demonstrated a few failed strands, but were far from catastrophic failure.

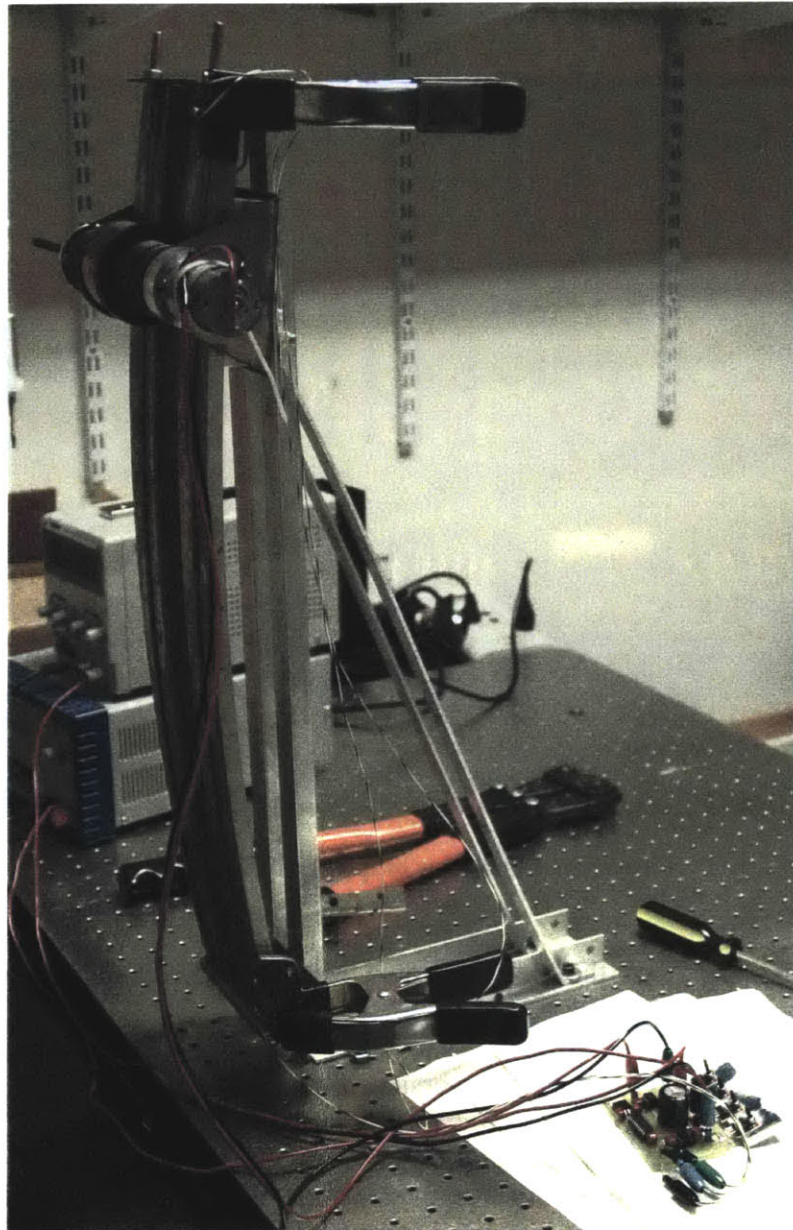


Figure 6-2: First version of the test apparatus

The failure of the glue joint in the spring led to questioning the desirability of the spring in the first place. The function of the counterbalance is to even out motor load between lowering and raising the glass. This is desirable in an efficient actuator because the motor operates at peak efficiency at only one load. However, if the designer is willing to accept a slight decrease in efficiency, then eliminating the counterbalance removes a likely failure point. It is important to note that motor efficiency is parabolic, so small deviations from the most efficient operating point result in small changes in efficiency.

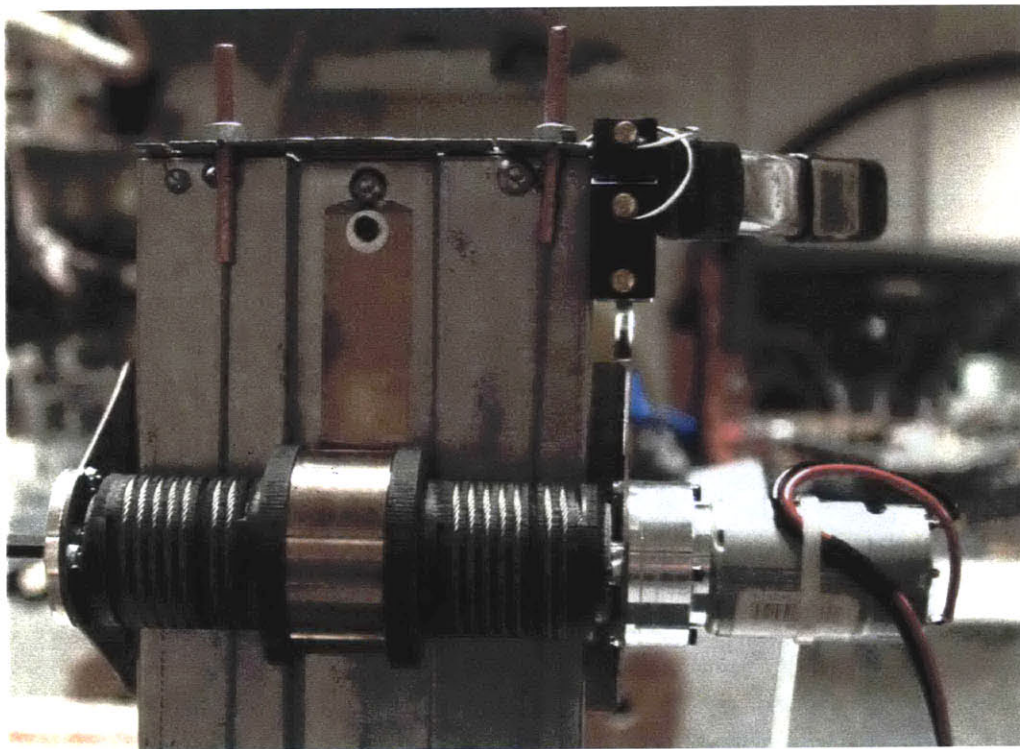


Figure 6-3: Detail view of spool and bracket

A second version of the test apparatus was constructed that included a cast polyurethane spool and a capstan cable termination at the top. The counterbalance spring was not installed. Figure 6-4 shows this version of the apparatus. Figure 6-5 shows the capstan cable terminations in detail.

Another fault found in testing was that the cables could rotate and lose tension. On the 2nd test rig, the lower cable terminations were flattened in a vice. Contact with a flat on the lower bracket prevented rotation. Nothing was done to prevent the top cables

from rotating, but the capstan and friction from the bolts that tension the cable appeared to be adequate to prevent rotation. The cables were never retensioned on the second test setup, but required constant maintenance on the first version.

Motor current was measured during the operation of the test apparatus. It was verified that the actuator does indeed consume between 12 and 20 W, depending on direction of stroke and applied load. This is very near the prediction made from the power budget.

Some problems with the spools were discovered on both versions of the prototype. Unfortunately, the cast polyurethane spools were slightly out of round when assembled; each half was slightly too thin. The rapid prototype spool was also out of round, but had been shimmed with a 0.4 mm shim laser cut from PVC. The eccentricity of the cast spools led to large variations in the drag force from the ball plungers. The gearboxes failed from overload after a few thousand cycles. The element that failed first was either the 2nd stage sun gear or a 2nd stage planet. A likely scenario is that a single tooth sheared from one of the plastic planet gears, allowing the sun gear to climb out of contact with the other two planets. The teeth on the aluminum sun gear quickly wore down and would no longer engage the planets.

Note that the issue with load variation is an artifact of testing, not an issue with the mechanical design of the actuator.

Both the rapid prototype spool and the cast polyurethane spools arrived with some excess curvature. When the halves were mated, the long edges would touch, but a gap would be left in the center. It is suspected that the curvature and the thickness issues are a limitation of the rapid prototyping process used. In both cases, the spools were fixed by clamping the halves together with multiple hose clamps and heating in an oven for a short time. The ring gear from the planetary gear reduction was installed to prevent collapse of the wall from the hose clamps. The rapid prototype spool was heated to about 50 °C for about 10 minutes, and the cast spools were heated to about 90 °C for up to 30 minutes; the cast polyurethane material exhibits a much higher heat distortion temperature than the rapid prototype material.

It was hoped that more spools would have been cycle tested as of the time of writing, but the gearbox failure was a major setback. It is hoped that rounder spools will

allow the gearboxes to survive life tests; in a production design, a stronger gear reduction would also be desirable. Work will continue on cycle testing of the actuator. The spools will have to be shimmed round, or the friction applied in a way that is less sensitive to the diameter of the spool. One option is to place the ball plungers on the side of the rail instead of behind it. This way, the forces from the plungers oppose each other, keeping the bearings at the ends of the spool from seeing a greatly elevated load that only exists in the test apparatus, better simulating conditions in an actual automobile door.

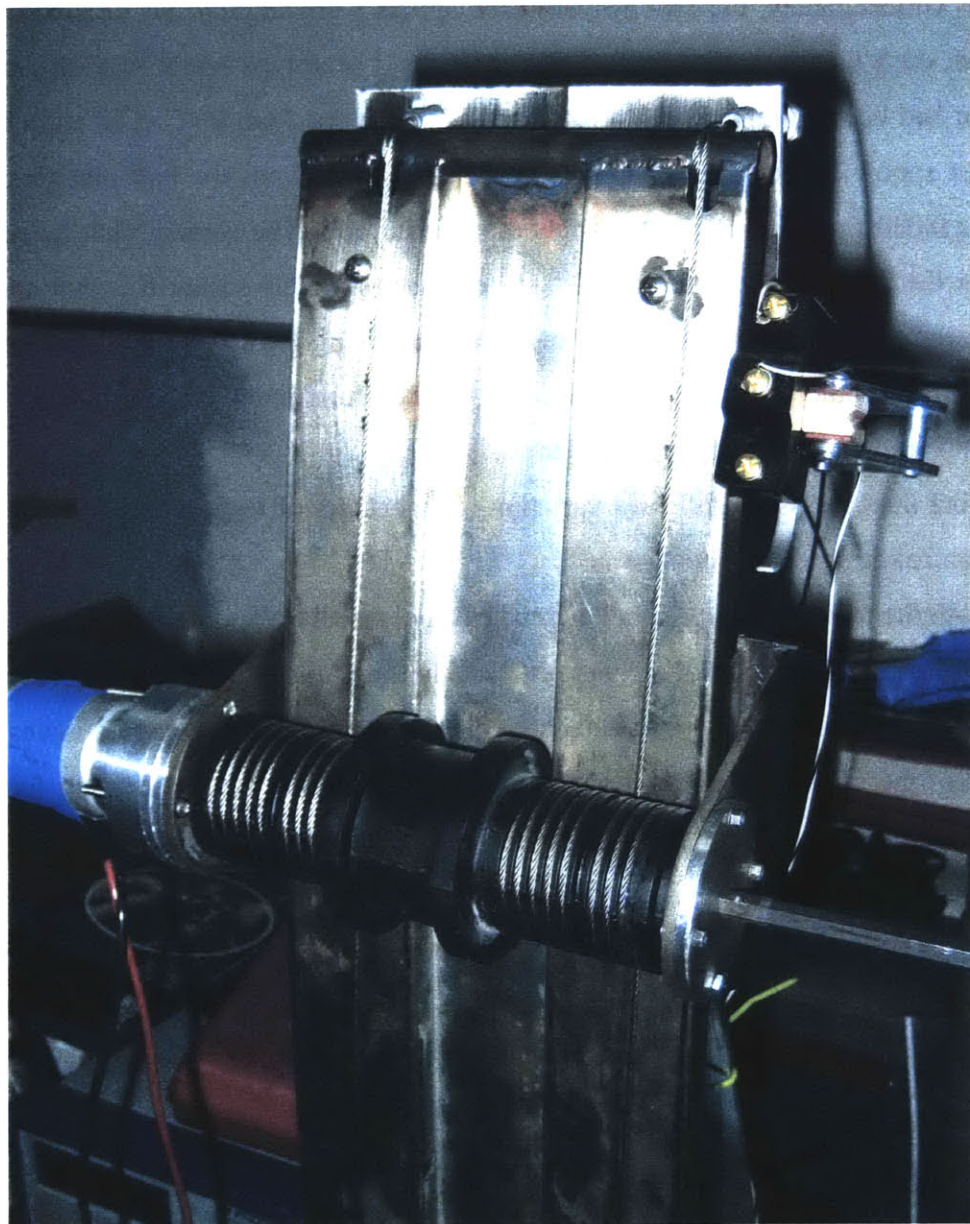


Figure 6-4: 2nd version of the test apparatus



Figure 6-5: Capstan cable tensioner

Chapter 7

Conclusion and Future Work

In summary, the rolling cable actuator can triple the efficiency of a typical automobile power window while using simple and inexpensive parts. The concepts presented in this work can be applied to the design of many actuators, automotive and otherwise. Anywhere where approximately linear motion is desired, rolling cables can offer efficiency and low cost. Sliding minivan doors, power seats, and possibly even steering could benefit from low cost rolling elements. With such actuators in a variety of locations on an automobile, one can envision significant reductions in weight and cost and increases in efficiency.

There is still room for improvement. A more robust way of bringing electricity to the spool would eliminate a major issue with this type of actuator. The elimination of the constant force spring of the device presented in Chapters 5 and 6 may easily allow for a more robust way of attaching electrical wires. A printed flex circuit, similar to that used in printers, could be wrapped around the area provided for the spring to allow power and data transmission from the rail to the spool. With the ability to add an arbitrary number of signal lines, an encoder could be used to detect the position of the spool. This information could be used to limit travel of the actuator, preventing stall conditions at the end of travel from consuming excess energy. Another simple solution may be to use the cables that support the spool as conductors.

Recent work on the energy harvesting concept shows that it is unlikely to generate enough electricity to power large actuators such as window mechanisms, and the power consumption of power door locks and power mirrors has yet to be addressed. Automakers can still benefit from this work, however. An efficient actuator allows the

use of much smaller motors and power transmission components, decreasing cost significantly. The wholesale price of the existing window actuator components is confidential to a vendor consulted, but it can be said that 75% of the cost of the actuator is the motor and gearbox. The rail, bearings, cables, spool, and carriage constitute the other 25%. Obviously, using much smaller motors would be very attractive to automakers.

If cost minimization is emphasized instead of efficiency maximization, the final solution might be somewhat different. It is proposed that a crossbreed of the rolling cable actuator and the existing window actuator be pursued for development of a low cost actuator. Such an actuator could use a rail, bearing, and carriage very similar to the existing technology. A traveling motor, gearbox, and clutch could then be used to gain efficiency and cost benefits. A schematic of the proposed actuator is presented in Figure 7-1.

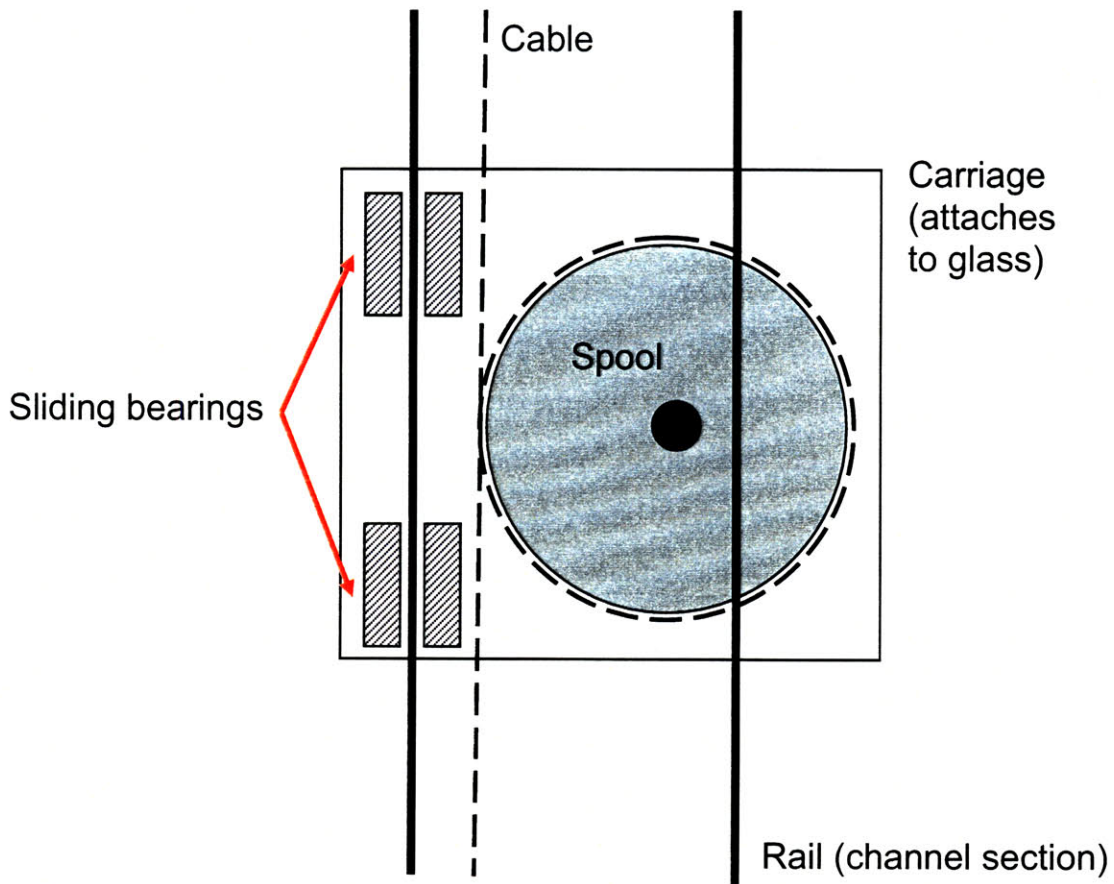


Figure 7-1: Proposed low cost traveling motor actuator

One benefit from this approach is that the glass would no longer have to be used as a structural element. The spool can be oriented such that it operates in the plane of the glass. A cable would be attached at the top and bottom of the rail; the motor, drivetrain, and carriage would climb the cable. Because the bearings are required to be stiff in pitch, torques in the plane of the glass would be resisted by the same bearings that prevent the glass from rotating. Conventional sliding bearings would likely be used, just like in the existing window unit. For more efficiency, a rolling element bearing, as presented in Chapter 3, could also be used.