CHAPTER IV

Dynacorp Brake Redesign

4.1 Design Goal

Primary goal for redesigning the Dynacorp brakes is twofold. First, the new units must incorporate means of measuring applied torque for use with feedback control. This can be accomplished through utilizing Hook's law with a compliant component. If a component is allowed to deflect under applied torque, resulting strain can be measured. Assuming linear material properties this recorded strain can be directly related back to applied torque.

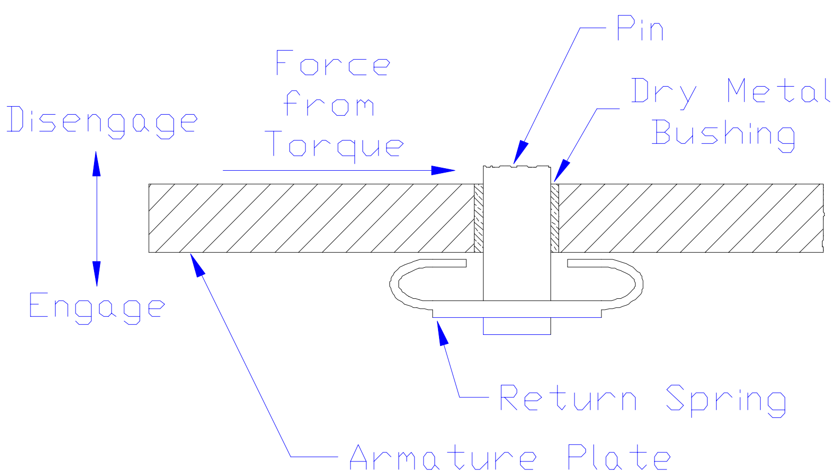

Figure 4.1: Pin / Spring Mechanism

The second goal is to eliminate as many non-repeatable and non-predictable characteristics. In its original configuration the armature plate slides over pins and against a return spring during clutch engagement. To prevent excessive sliding friction Dynacorp mounted dry soft metal bushings in the armature plate. (See Figure 4.1) Even with these bushings it can be speculated that forces due to torque transmission can cause binding as the armature plate attempts to slide along the pins. For these reasons it was concluded that the spring / pin mechanism for locating the armature plate is far from optimal. It may be improved through use of linear bearings instead of bushings, but a simpler mechanism that both locates and provides spring back force is preferable.

4.2 Design Constraints

It is desirable to stay away from modifications to PTER's other components. Therefore it was decided that the modified clutches must be compatible with the original clutches mounting configuration. This allows a total height of 3.8125 inches and a maximum diameter that must be under 12 inches. Will Stone found that the maximum available torques from the existing clutches were 15 ft-lbf, 45 ft-lbf, 45 ft-lbf, and 125 ft-lbf. Simulations of PTER only command a maximum torque of 500 in-lbf (≈42 ft-lbf). It was decided that the modified units should be able to endure an applied torque of 50 ft-lbf.

The coils are designed for a steady state 24 volts with a current of approximately 2.4 amps. The limiting factor is how much power the electromagnetic coil can dissipate for how long before burning up. According to Dynacorp, the brakes can be overexcited at higher voltages for short periods of time. This will open some options to the amount of control current that is applied; it may be possible to use short bursts of higher control currents and voltages.

The amount of force the electromagnet can apply to the armature plate is critical. This force must both deflect the armature plate's spring back mechanism and provide normal force against the friction material. In order to determine available normal force, two methods were employed. The first method consists of using general equations for brake torque given a brake's dimensions while the other consists of developing an initial model for the electromagnet from first principles and crude experiments.

4.2.1 Modeling Based on Brake Equations

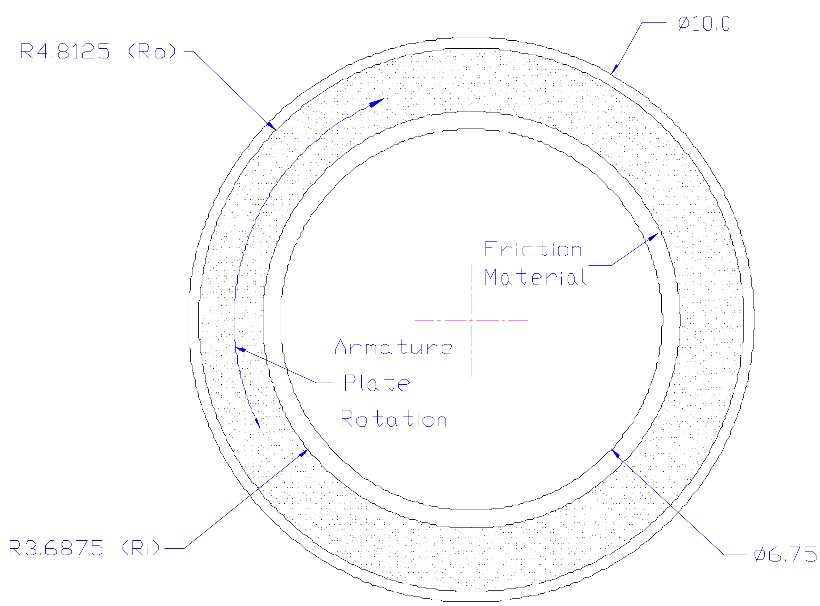

Figure 4.2: Field Magnet Assembly with Friction Material

Resulting torque and required force for friction brakes are based on one of two assumptions, either uniform wear of the friction material or uniform pressure across the friction material. In short, friction material wear is claimed to be proportional to both slip speed and applied pressure ($wear = k \cdot \Pr es \cdot Vel$). With the slip speed across the friction material greater at the outside edge then the inside edge, pressure must be less at the outer radius then inner radius to maintain uniform wear. On the other hand if uniform pressure is assumed, friction material must wear faster at the outer radius. For this to occur components must be flexible enough to allow deformation and maintain uniform pressure as friction material wears unevenly. Equations 4.1 and 4.2 are the resulting torque and required axial force respectively for uniform wear. Equations 4.3 and 4.4 are the equivalent equations for uniform pressure. These equations are easily derived based on assumptions stated above and the derivation can be viewed in "Clutches and Brakes Design and Selection". [Orthwein, 1986]

Uniform Wear $$\tau = \mu P_{\max} \frac{\theta}{2} R_i (R_o^2 - R_i^2) \qquad \textbf{(4.1)}$$

$$F = P_{\max} \theta R_i (R_o - R_i) \qquad \textbf{(4.2)}$$

Uniform Pressure $$\tau = \mu P \frac{\theta}{3} (R_o^3 - R_2^3) \qquad \textbf{(4.3)}$$

$$F = P \frac{\theta}{2} (R_o^2 - R_i^2) \qquad \textbf{(4.4)}$$

P_{max}	Maximum Pressure
P	Uniform Pressure
θ	Angle the friction material sweeps
R_o	Outside Radius (See Figure 4.2)
R_i	Inside Radius (See Figure 4.2)
μ	Friction Coefficient

Using the values in Figure 4.2, an angle of 2π, and the brake's a coefficient of friction (0.45), the above equations can be used with maximum delivered torques found by Will Stone to calculate available normal force. [Stone, 1995]

Table 4.1: Maximum Torque and Normal Force Calculations

		Uniform Wear		Uniform Pressure	
Brake	τ (in-lbf)	P_{max} (lbf / in^2)	F (lbf)	P (lbf / in^2)	F (lbf)
1	180	3.61	94	3.11	94
2	540	10.83	282	9.34	281
3	540	10.83	282	9.34	281
4	1500	30.09	783	25.96	780

It is clear that uniform wear and uniform pressure provide almost equivalent results. More uncertainty is in the values used for calculation then the difference between these two methods. It also becomes evident that a large diversity exists between normal forces available from each brake. Again some variation is due to power supply differences, but not all. To better understand how air gap factors and deficient power supplies can affect the available torque, a better model is needed for the electronic magnet assembly.

4.2.1 Electromagnet Force Model

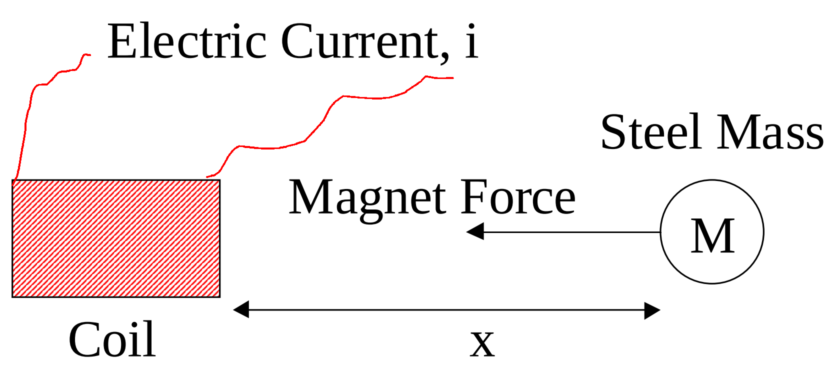

Figure 4.3: Electronic Magnet

A electromagnet makes up a resistor-inductor electronic circuit with inductance inversely dependent to the distance of the attracted steel object from the magnet. The attractive force from the electronic magnet can be derived using principles of virtual work. In short, magnetic force times virtual displacement is equal to virtual work performed or change in energy ($\delta Energy = Force \cdot \delta x$). The energy term for a resistor-inductor circuit can be derived through integrating the expression for electrical power with respect to time. (See Equations 4.5 through 4.7) Derivation of magnetic force is through differentiation of this energy expression with respect to *x*.

$$Power = vi = \frac{d(Energy)}{dt} = Li\frac{di}{dt} \tag{4.5}$$

$$L = L_1 + \frac{L_0}{1 + x/a} \qquad \text{(Assumed Inductance)} \tag{4.6}$$

$$Energy = \frac{1}{2} Li^2 = \frac{1}{2}\left(L_1 + \frac{L_0}{1 + x/a} \right) i^2 \tag{4.7}$$

$$Force = \frac{\delta Energy}{\delta x} \approx \frac{d(E)}{dx} = -\frac{1}{2}\frac{L_0}{a}\frac{1}{(1 + x/a)^2} i^2 \tag{4.8}$$

In the above expressions "L_1", "L_0", and "a" are modeling constants to be determined for a specific system. They are dependent on the magnetic properties of both the magnet core and attracted object. Variables "x", "i", and "v" are distance of the object from the magnet, applied current to the coil, and applied voltage to the coil. (See Figure 4.3) Inspecting Equation 4.8 shows magnetic force is proportional to the square of applied current and inversely proportional to the square of distance from the magnet, making it a highly nonlinear system.

To understand roughly what force is available from the magnet a crude test was performed to estimate parameters "L_0" and "a". The test consisted of picking up a weight positioned a set distance from the magnet (creating a given air gap). Current was adjusted until the magnet could no longer support the weight from falling. Air gap distance, weight, and current were recorded as viewed in Table 4.2.

Table 4.2: Recorded Data from Magnet Modeling Test

Object	Weight (lb)	Air Gap (in)	Voltage	Current (A)
Armature Plate	6.809	0.0188	3.37	0.3337
		0.1438	16.3	1.6139
Magnet Assembly	11.6005	0.0188	3.8	0.3762
		0.1438	21	2.0792

Equation 4.8 can be rearranged to form equation 4.9, allowing test data to be arranged in a linear equation so that a least square pseudo-inverse can be used to calculate the linear coefficients. Coil's parameters can be calculated from the linear coefficients using Equations 4.10, while implementation of these calculations can be reviewed in Appendix A. Values for "a" and "L_0" were found to be approximately 0.0113 and 11.298 respectively. These values were substituted back into Equation 4.8 to calculate estimated magnetic force for a given combination of current and distance from the magnet; the resulting table can be viewed in Appendix A. Only one magnet was tested, but it is assumed that the three remaining magnets are equivalent, less manufacturing tolerances. It should be noted that these derived constants are for Dynacorp's armature plate. If a steel object is magnetically hard, resist change in magnetism, it will see less magnetic force than an object made from magnetically soft steel. Therefore the clutch's electromagnet modeling constants depend on the properties of the armature plate's steel.

$$\frac{i}{\sqrt{F}} = \left(\frac{1}{a}\sqrt{\frac{2a}{L_0}}\right)x + \sqrt{\frac{2a}{L_0}} \Rightarrow y = mx + b \qquad \textbf{(4.9)}$$

$$a = \frac{b}{m} \text{ , } \; L_0 = \frac{2}{bm} \qquad \textbf{(4.10)}$$

From the table in Appendix A it can be seen that magnetic force drops rapidly with distance from the magnet, confirming that slight variations in air gap will have drastic affects on amount of available torque. It was noted earlier that two of the power supplies only delivered a maximum of 1.5 amps. When compared to the power supply capable of delivering 3 amps, the smaller units limit maximum torque to one fourth that available from the larger unit. Assuming the existing friction material is kept, to achieve 50 Lb-ft the magnet must be able to apply a normal force of approximately 310 Lbf in addition to any force required to deflect the spring back mechanism. It is also assumed that though the clutches were rated for a steady state 24 volts (approximately 2.4 amps) they could survive higher current for short periods of time. If an air gap of 0.0188 in. (approximate friction material thickness of tested unit) can be maintained, this will allow approximately 95 Lbf of available excess magnetic force to deflect the spring back mechanism if 2.4 amps of current were applied and approximately 325 Lbf excess force for 3 amps. If friction material was machined down and a smaller air gap could be utilized, more magnetic force would be available. For example, if the friction material were shaved to $^{1}/_{64}$ in. the excess force at 2.4 and 3 amps would be approximately 200 and 480 Lbf respectively.

4.3 General Design Layout

The new clutch design is based on connecting the hub to the armature plate via compliant spokes, which transmit torque to the armature plate when it is in contact with the friction material; this can be better viewed in Figure 4.4 & 4.5. As the armature plate engages and resists motion of the hub with respect to the friction surface, a bending force normal to the page (figure 4.5) is transmitted on the spokes. The new spokes are equipped with strain gauges at the hub side to measure resulting bending strain from applied torque. It can also be seen in Figure 4.5 that the armature plate is suspended below the hub and held in place only by the spokes. As the magnet attracts the armature plate the spokes must deflect. For this reason the center section of the spokes are designed to be compliant in the vertical direction, but stiff in the torque direction (normal to the page). In contrast, the outer edges of the spoke are designed opposite, to be stiff in the vertical direction but compliant in the torque direction. What dimensions should the spokes be to promote sensitivity for torque measuring, strength for durability, and compliance for armature plate engagement? This will be addressed in the next section.

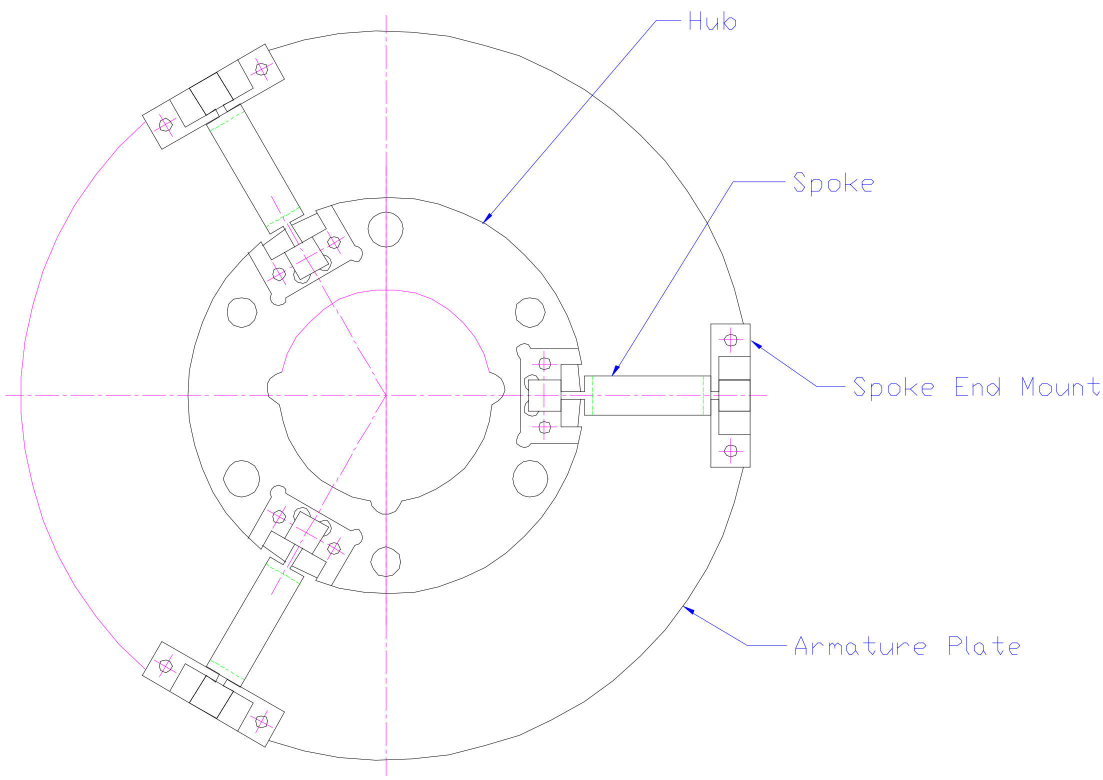

Figure 4.4: Top View of New Clutch Design

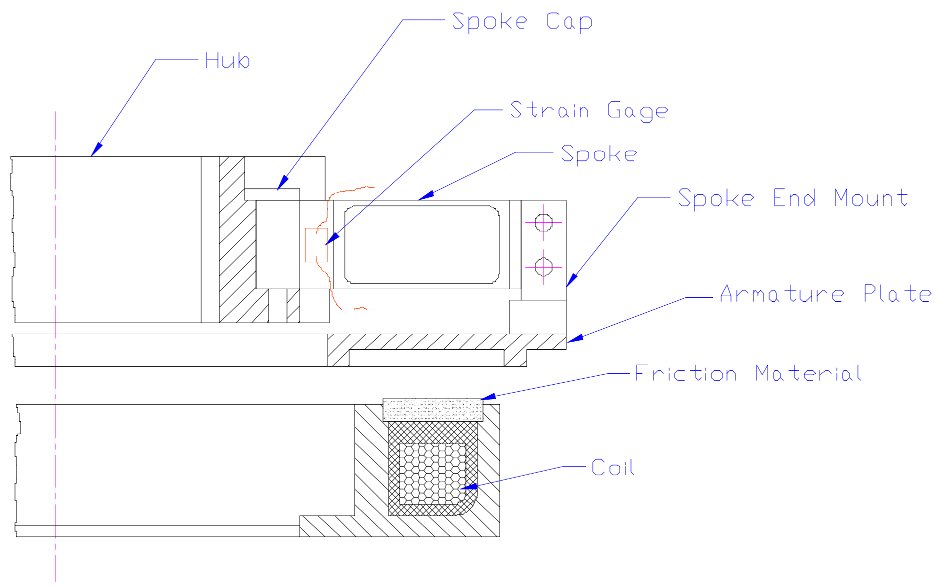

Figure 4.5: Section Side View of Half the New Clutch Layout

4.4 Spoke Design

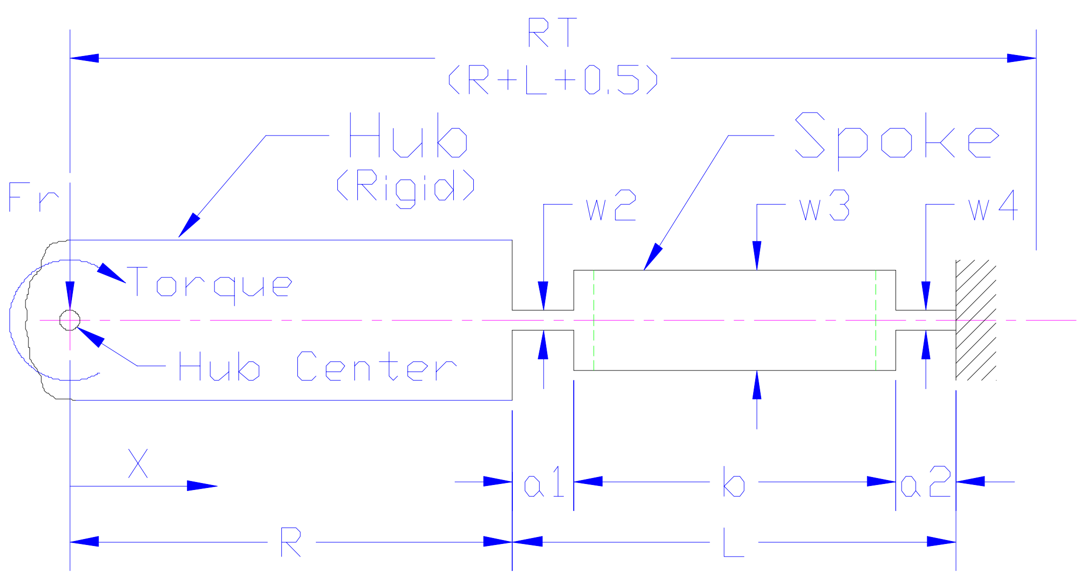

Figure 4.6: Mechanics of Materials Model for Spoke

What dimensions should the spoke be built for both strength and sensitivity? What material should be used? To answer these questions a mechanics of materials model was created. Symmetry was utilized so that only one spoke was modeled as supporting a third of the torque. The basic model and free body diagram can be viewed in Figure 4.6. The model is split into four sections. The first consists of the hub that was assumed to be rigid and length "R"; the remaining three sections are from the spoke (total length "L") with lengths "a1", "b", and "a2". The assembly length is "RT" consisting of the hub, spoke, and 0.5 inches for a spoke to armature plate mount. From one of the physical design constraints "RT" must be under 6 in, therefore "RT" was limited to a maximum length of 5.75 in. The right end of the spoke is assumed fixed and cantilevered while the hub is allowed to rotate around its center but not deflect. Both a torque and reaction force are applied at the hub's center to simulate both the clutches torque and required reaction force to constrain the hub's center from deflecting. Boundary conditions on the cantilevered outside (radial) end of the spoke require both slope and deflection to be zero.

For modeling axial deflection, only the center section of the spoke was considered. The other two sections of the spoke are orders of magnitude stiffer in the axial direction then the center section, effectively behaving as stiff members. Figure 4.7 shows assumed deflection shape of the center section under engagement. This deflection shape was assumed from initial analysis of equivalent shaped members with a finite element package. Both the upper and lower legs of the center section are identical, again allowing symmetry to be utilized and only one leg modeled as taking half the force. The resulting moment "Mr" is required to guarantee the constraint of zero slope at the end.

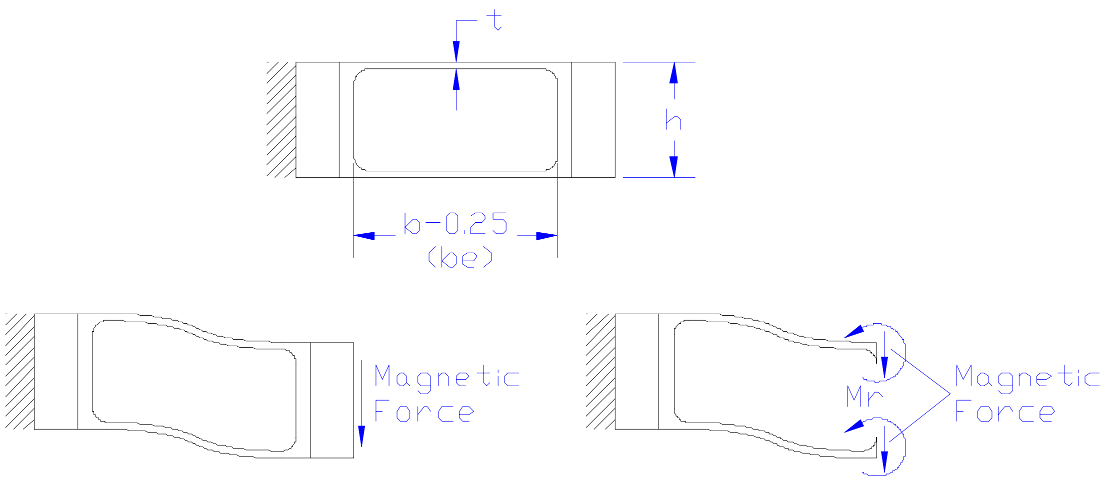

Figure 4.7: Axial Deflection for Engagement

Stiffness of a uniform beam in bending is inversely proportional to its length. With this in mind one can deduce that a longer spoke (maximum "L") will produce more sensitive torque measuring capabilities. In addition the longer the middle section of the spoke ("b"), the more compliant it will be to axial deflection resulting in less required axial force for a given deflection distance. Because "RT" is limited to a maximum distance, the only way to increase spoke length is to minimize the length "R". For this reason the spoke will be recessed into the hub. After investigating other existing features of the hub it was determined that the minimum distance for "R" would be set to 2.75 inches, allowing a maximum spoke length of 2.5 inches.

4.4.1 Mechanics of Materials Model

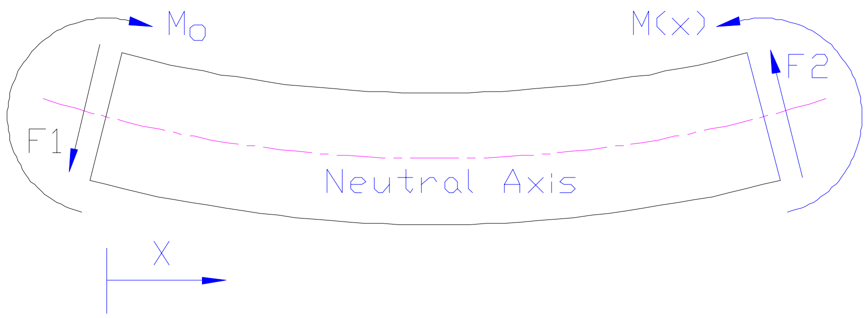

Figure 4.8: Typical Element for Simple Beam Theory

Simple beam theory was used for analyzing the deflection, stress, and strain in the spoke when subjected to both axial force and applied clutch torque. By splitting up the spoke and analyzing each section separately, the model's strain and deflection due to torque transmission becomes fairly simple to solve. One restriction is boundary conditions for connecting sections must be equivalent. For example the angle and deflection at the hub - spoke interface must be the same for both the hub and spoke. Simple beam theory relates internal moment to the second derivative of deflection (δ'') through geometric constant I (area moment of inertia) and material property E (young's modulus). This relationship can be integrated twice to form the slope (δ') and deflection (δ) equation, while integration constants C and D are used to satisfy boundary conditions. The generic expressions can be reviewed in Equations 4.11 through 4.13. In these equations *x* in figure 4.8 is replaced by λ.

$$EI\delta''(\lambda) = M(\lambda) = M_0 - F\lambda \qquad \text{(Moment)} \qquad \textbf{(4.11)}$$

$$EI\delta'(\lambda) = EI\theta(\lambda) = M_0\lambda - F\frac{\lambda^2}{2} + C \qquad \text{(Slope)} \qquad \textbf{(4.12)}$$

$$EI\delta(\lambda) = M_0\frac{\lambda^2}{2} - F\frac{\lambda^3}{3} + C\lambda + D \qquad \text{(Deflection)} \qquad \textbf{(4.13)}$$

Splitting the spoke into three sections results in 6 integration constants, with the required reaction force at the hub's center acting as a seventh variable. Because the hub is considered rigid, the deflection at the end of the hub is taken to be simply the length of the hub multiplied by the slope of the hub ($\delta(R) = R\theta_{hub}$). Equating connecting boundary conditions for neighboring sections and enforcing the end point boundary conditions gives 7 simultaneous linear equations that can be solved for the integration constants and resultant reaction force. Once these constants are determined they may be substituted back into the original equations for calculating moment, strain, stress, slope, or deflection at any point along the spoke. Symbolic manipulation of the simple beam theory equations for a closed form solution can be reviewed in Appendix B. Similarly, simple beam theory equations can be used to develop a closed form solution to required axial force and stress for a given deflection. Again, symbolic formulation can be reviewed in Appendix B.

Because this analytical model is assumed to be a design tool and not intended for in depth analysis of the spoke, several simplifications and assumptions were made. The first simplification was with regards to bending from torque transmission. When inspecting the top section (Figure 4.6) it appears the three spoke sections directly connect with immediate change in geometry. Immediate changes in geometry are not practical; therefore to facilitate transition between geometry configurations a rigid portion 0.125 inches in length was required at each joining spoke section. Again this rigid section was not modeled in the analytical solution and is hypothesized to only add stiffness to the already relatively stiff center section. In addition, fillets in the center section were not modeled. These fillets will add stiffness to both bending from torque transmission and axial deflection. It is assumed that resulting stiffness from this added feature is minimal and does not warrant complex modeling of the transitioning cross section. Material property is assumed linear, allowing superposition of both axial and torque models for calculating total stress at a given location. Verification of this assumption will be made through inspection of stress and determining if it exceeds the materials yield stress. In addition, stress concentration factors were not used for joining sections.

4.4.2 Cyclic Fatigue

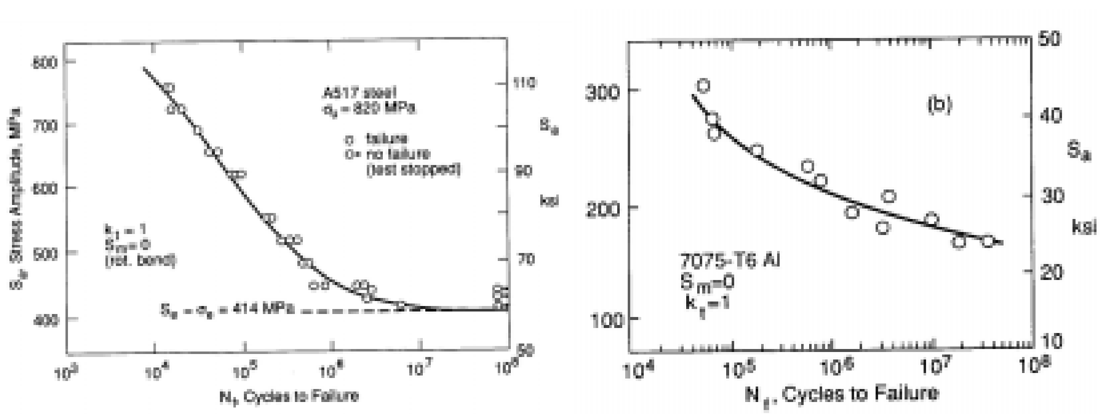

Figure 4.9: S-N Curves for Steel and Aluminum [Dowling, 1993]

Fatigue is an important consideration in designing the spoke. The spoke will be subject to cyclic loading as torque transmission changes direction and the armature plate engages and disengages. There are three possible ways to look at cyclic loading. The first is as a fully reversed loading with zero mean stress. This applies to stress caused by torque bending in spoke sections one and three. The center section is a little more complicated because it sees stress from both torque transmission and axial deflection. Treating axial deflection as a mean stress and torque transmission as a cyclic load is one method of combining both models (method σ_{ar1} in Appendix B). Another method is to directly combine the models and treat total stress as a cyclic load from zero to the combined stress. Mean stress and amplitude stress would then be equal and half the calculated stress (method σ_{ar2} in Appendix B). Either method utilizes Equations 4.14 and 4.15 to determine equivalent fully reversed cyclic loading for use with a specific materials S-N data. In the equations σ_a and σ_m are the amplitude and mean stress of the applied cyclic load while σ_u and σ_{ar} are ultimate strength of the material and equivalent fully reversed load to be used with S-N data. Some materials have an endurance limit; a stress limit that if not exceeded the material is not expected to fail from cyclic fatigue. This is illustrated in the S-N plot for steel in Figure 4.9. It can be seen that the curve reaches a flat spot as fully reversed stress is decreased, signifying that lower stress levels do not correspond to a cycle life. In contrast some materials like aluminum do not have an endurance limit as illustrated by a curve that does not flatten off, but continues to slope down.

$$\frac{\sigma_a}{\sigma_{ar}} + \frac{\sigma_m}{\sigma_u} = 1 \qquad \textbf{(4.14)}$$

$$\sigma_{ar} = \frac{\sigma_a \sigma_u}{\sigma_u - \sigma_m} \qquad \textbf{(4.15)}$$

4.4.3 Design Iteration & Material Selection

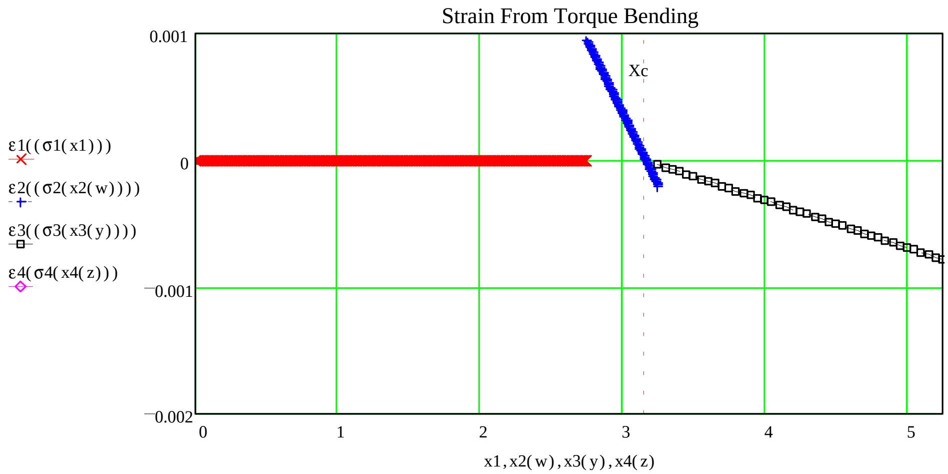

Figure 4.10: Section 2 Stiff and Section 3 Eliminated

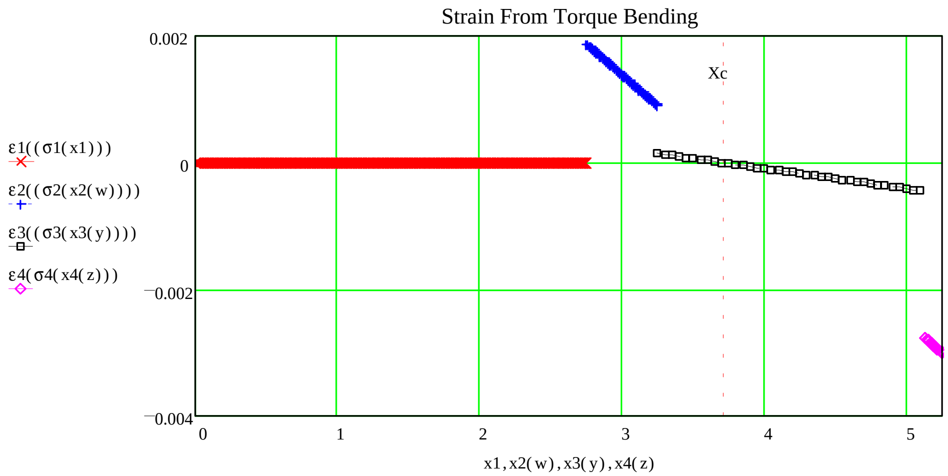

Figure 4.11: Section 2 Stiff and Section 3 Not Eliminated

Before reviewing material selection, the iteration of section lengths will first be addressed. It is beneficial to have the spoke's second section as long as possible to minimize required axial force. One option is to eliminate the spoke's third section entirely. Because a strain gauge is to be mounted in section one of the spoke, it is best to not have an inflection point or zero resulting stress-strain anywhere in or near section one. As can be seen in Figures 4.10 and 4.11 if the third section were eliminated, the inflection point (vertical line marked Xc) moves towards section one (middle line in Figure 4.10). Furthermore, the stiffer section two (right most line in figure 4.10) the closer the inflection point moves towards section one. By having section three (right most line in Figure 4.11) in the spoke design it helps pull the inflection point away from section one. It is assumed section two will actually be stiffer than determined in the model because of simplifications made, therefore it was decided to minimize the length of section three to 0.125 (a2=0.125 in.) inches but not eliminate it. Section one must be large enough for mounting strain gauges. After reviewing available strain gauges from commercial vendors it was opted to make section one 0.375 inches long (a1=0.375 in.). This leaves the center section 2.0 inches long (b= 2.0 in.).

Properties for various materials can be viewed in Appendix C. This list was narrowed to three basic materials; Steel ASTM-A514, Aluminum 7075-T651, and Delrin 100P (Dupont). Spoke configurations were generated based on trial and error for each material. Both dimensions and resulting stresses & forces can be viewed in Appendix C for each material. Figure 4.12 shows required axial force from the electromagnet for each of the three materials given a specific axial deflection.

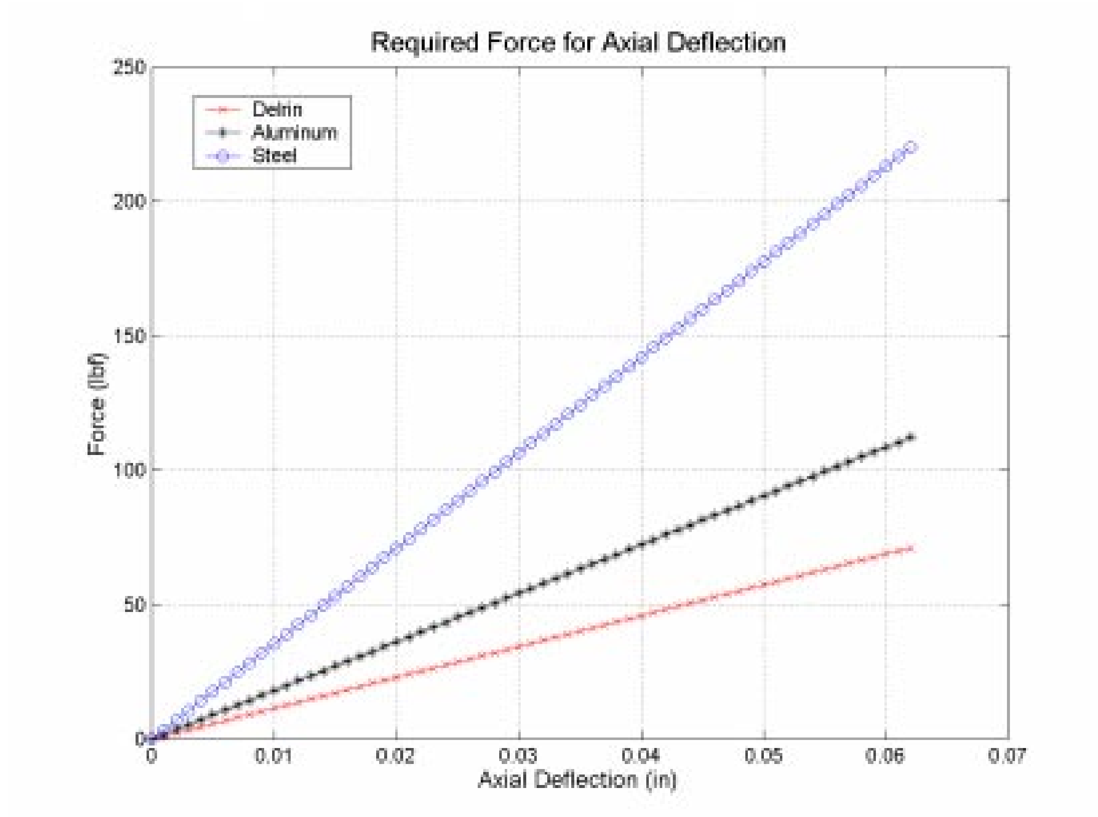

Figure 4.12: Required Axial Force vs Axial Deflection

Advantages of Delrin are that it is very flexible and easy to machine. Having a flexible material will increase sensitivity and decrease required force for axial deflection. Resulting strain in section one of the spoke from torque transmission is roughly three times greater than that from the Aluminum model and five times greater than the steel model. Required axial force is roughly 1.5 times less then Aluminum and 3.5 times less then the steel model. (See Figure 4.12) Though Delrin is a plastic, Dupont claims Delrin 100P has an endurance limit of 4.7 ksi; so this was used as the critical stress in the model. Inspection of the model in Appendix C show that the Delrin configuration looks good with stress levels below critical stress for all methods of combining cyclic loading. Disadvantage of Delrin is that it is subject to creep deformation over time and the transition from linear elastic material behavior to nonlinear and plastic behavior is not distinct. Recovery properties from a prolonged load are not as predictable as metals. Other problems are strain gauge bonding to plastics is very difficult and plastics are not very good at heat dissipation. As the grid on a strain gauge heats up from electrical excitation it relies on the material it’s bonded to for heat dissipation. Heat will not dissipate if a gauge is bonded to a plastic, causing the strain gauge to suffer from thermal errors. Because some of its material characteristics are difficult to model and predict, along with difficulties of using strain gauges, Delrin was not chosen as the construction material.

ASTM-A514 steel has an impressive yield & ultimate strength and a high distinct endurance limit of approximately 60 ksi (See Figure 4.9); which was used as the critical stress in the model. Another benefit of steel is practices for bonding strain gauges are readily established. Steel is the stiffest of the three materials, evident by a high modulus of elasticity; causing both resulting strain from torque transmission to be low and required axial force for a given deflection to be high. (See Figure 4.12 and Model in Appendix C) Having a low strain decreases sensitivity and high axial force requirement takes away from available normal force for torque transmission. For these reasons, Steel ASTM-A514 was not chosen as the spoke’s structural material.

Aluminum 7075-T651 was chosen because it is both strong and flexible (lower modulus of elasticity). This allows for a balance between sensitivity and strength.

Required axial force is slightly greater than the Delrin 100P model, but much less than the steel model. The model in Appendix C predicts a spring constant of 1.777×10^3 lbf/in in the axial direction and 2,217 in-lbf/deg in the radial (torque) direction. The extra axial force of 95 lbf determined from the magnet model will deflect the spoke just over 0.0535 inches. Though Aluminum does not have an endurance limit (See figure 4.9), a fatigue limit for such materials is often arbitrarily chosen to correspond with a long life of 10^7 or 10^8 cycles. [Dowling, 1993] Therefore a conservative critical stress of 20 ksi was used for the aluminum model. Aluminum has good heat dissipation characteristics and practices for bonding strain gauges are established. The final model can be viewed in Appendix C and drawings of the final design can be viewed in Appendix D.

4.5 Other Design Considerations

A few more key points to the overall design should be mentioned. First, a new armature plate is required. This new armature plate was designed differently then the original unit. The original armature plate supplied by Dynacorp was designed flat and grooved. (See Figure 4.13) The grooves were intended to act as a fan under high slip speeds for increased cooling of the electromagnet. Because PTER does not turn the armature plate at high rotational speeds these fins are useless. Therefore the new armature plate was designed smooth, without these grooves for increased contact area. In addition the new armature plate was designed to seat around the magnet (see Figure 4.5 in comparison to Figure 3.4) when in contact with the friction material. This was done in hopes to increase magnetic coupling between the electronic magnet and armature plate through providing more area for the magnetic circuit. A picture of the new armature plate next to the original unit can be viewed in Figure 4.13. As mentioned above, the magnetic force applied to the armature plate is dependent on the material it is made from. According to Dynacorp engineers the original unit is fabricated from magnetically soft low carbon steel (under 1010). The new unit was machined from 1008-1010 steel to minimize carbon content and maximize available force.

Figure 4.13: Original Dynacorp Armature Plate & New Armature Plate

For mounting the spokes a modified version of the original Aluminum hub is used. The hub is modified in such a way as to not compromise its abilities of being used in its original configuration with the original armature plate. The spokes are attached to the armature plate via mounts machined from aluminum. Machine drawings for each component can be viewed in Appendix D.

Two of the spokes will be fitted with strain gauges (two per spoke). This allows use of a full wheatstone bridge, maximizing sensitivity of the measuring system and minimizing adverse affects from temperature changes. Gauges are bonded with an epoxy-based adhesive for increased durability.

Some consideration was given to changing friction material. For example Delrin is rated at having a lower dynamic coefficient of friction (0.35) then static (0.20) when used with steel. [Dupont] Theoretically a lower dynamic friction would eliminate stick slip characteristics. Some in house testing has found this not to be true for another brake configuration, but the only way to determine how Delrin will act is to subject it to the exact application because friction material properties are dependent on application. At this time it was determined to proceed with the original Dynacorp material. Though not very difficult, machining the old material and fitting a properly sized Delrin piece will be reserved for after the new brake configuration is tested, modeled, and better understood.

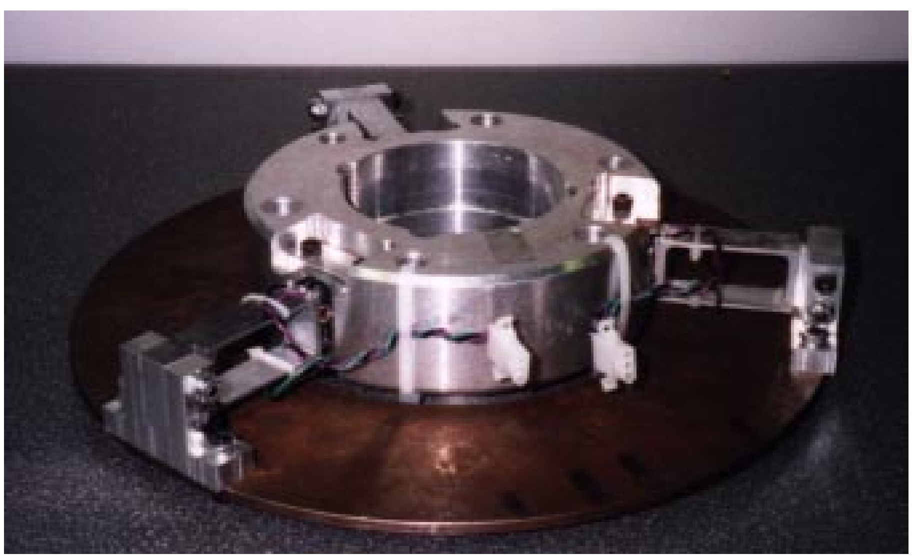

Figure 4.14: New Clutch with Instrumented Spokes