Design of a Clutch/Brake Mechanism for a Walking Robot

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July 17, 2006

Abstract

This paper details the creation of a prototype clutch/brake mechanism. Conceptual design is tailored to meet system needs; detailed design implements conceptual design. Fabrication is executed to produce a working model; the model is tested to answer questions and investigate concerns stemming from the design. The final product is evaluated against the original system requirements and suggestions for improvement are discussed based on the experience. A revision iteration of these steps is expected to yield a satisfactory product for use in a walking robot.

1 Introduction

There are several approaches toward creating a walking machine. One of the most popular is to model the geometry of the machine after that of the human body and precisely control joint angles in time to mimic the human walking motions. While these machines are versatile and robust, they are extremely inefficient, and as a result support only short runtimes with current battery technology. The passive dynamic approach, on the other hand, is to model dynamic properties of the machine after those of the human body but offer no control or power other than that of gravity. These machines are extremely efficient, but not at all versatile or robust, which eliminates practical use. An approach in between these extremes seeks to apply passive dynamic principles for efficiency, but also provide powered control when necessary to achieve greater versatility and robustness. The robotic subject of [3] achieves a walking cycle through this approach and thus has the potential to reach new levels of efficiency and reliability.

One of the greatest sources of inefficiency in that robot, however, was power spent locking joints by stalling motors, or the opposite, driving motors when a free pendulum motion would suffice. For example, Figure 1 illustrates three positions of the hip joint in a walking cycle. When the front heel of the robot strikes the ground, the rear foot pushes off, then retracts. At this point the rear leg must swing through to complete the next step. At the beginning of the swing, it is often desired to drive the hip with a motor, to accelerate it more quickly. Once it is moving at the desired rate, it would suffice to let the leg swing freely as a pendulum. At this point a clutch mechanism should disengage the hip from motor input such that electrical power is not wasted in driving the motor to follow along nor is mechanical energy lost by the leg in backdriving the gearhead of the motor against friction. Then, when the front leg reaches the desired angle with respect to the ground/rear leg, it should lock in this position to ensure that it does not move further forward or begin to swing backwards before the heel strikes the ground. Furthermore, a brake mechanism should lock the hip joint rather than wasting power by re-engaging and stalling the motor.

This paper serves as documentation for the creation of such a clutch/brake mechanism, which accomplishes these two tasks for any joint in a convenient, easily retrofitted or originally implemented package, thus increasing the potential efficiency of the target robot. Section 2 describes the conceptual design from the generation of a problem statement, needs, and requirements, through the completion of a general design concept in progressively increasing detail. Section 3 describes how the mechanism works in greater detail, discusses some of the less obvious elements of the design, and outlines relevant analysis. Section 4 comments on drafting, part and material selection, conventional and advanced machining, and assembly. Section 5 records the procedures and results of tests performed on the prototype, the results of which will guide future revisions of the concept. Section 6 comments on other factors taken into the design, including economics, safety, reliability and maintainability, and aesthetics. Section 7 reflects on the success of the project as a whole and considers the direction of future extensions.

2 Conceptual Design

2.1 Defining the Problem

2.1.1 Problem Statement

Design a mechanism that will allow a joint to be driven by an actuator, disengaged from the actuator to permit free rotation, or be rigidly fixed at a desired angle.

2.1.2 Objectives

The objective of the project is to meet the following system level needs [1, 2.13.06]:

- Must be small and light as required by overall robot size
- Must be robust and reliable
- Must be easy to service/repair/replace
- Must be inexpensive to fabricate
- Must not allow damage to gearhead or actuated compenents due to over-torque
- Must be scalable to different sized actuators/actuated compenents and adaptable to different actuator/actuated component interfaces

In addition, the clutch and brake mechanisms must meet the needs summarized in in Table 1 [1, 2.13.06]:

2.2 Evaluation Criteria

From the qualitative needs above, the product can be evaluated according to how it meets the following specifications given in Table 2. Most of the design will be driven, however, by the need statements, while evaluation can be accomplished by comparing system parameter values to the relevant specifications.

Specification	Target Value
Volume	$\approx$ 1.5” $\times$ 1.5” $\times$ 1.5”
Mass	$<$ 200g
Cost	$<$ \$100
Power (max)	$<$ 1W
Power (rms)	$<$ .1W
Switch Time	$<$ 10ms
Slip Torque	$< \tau_{max,gearhead}$

Table 2: Specifications

2.3 Concept Generation

The clutch/brake can be broken down in many ways, but the following components are convenient for concept generation purposes [1, 2.17.06]:

- Actuator Input
- Brake Mechanism
- Brake Actuator
- Clutch Mechanism
- Clutch Actuator
- Mechanism Output

2.3.1 Clutch and Brake Mechanisms

The first of these to be considered are the clutch and brake *mechanisms* - the means by which the desired effect is accomplished. The central problem in each is to be able to couple and decouple two components at will, so a brake can be considered as a special case of the more general clutch. Therefore, for simplicity and convenience of design, the two mechanisms should be considered together.

The article by Power Transmission and Design [2] was very useful in generating the following list of mechanisms:

Friction Mechanisms

- Disc - two discs meet face to face to form contact
- Drum - One of two concentric cylinders expands or contracts to make contact

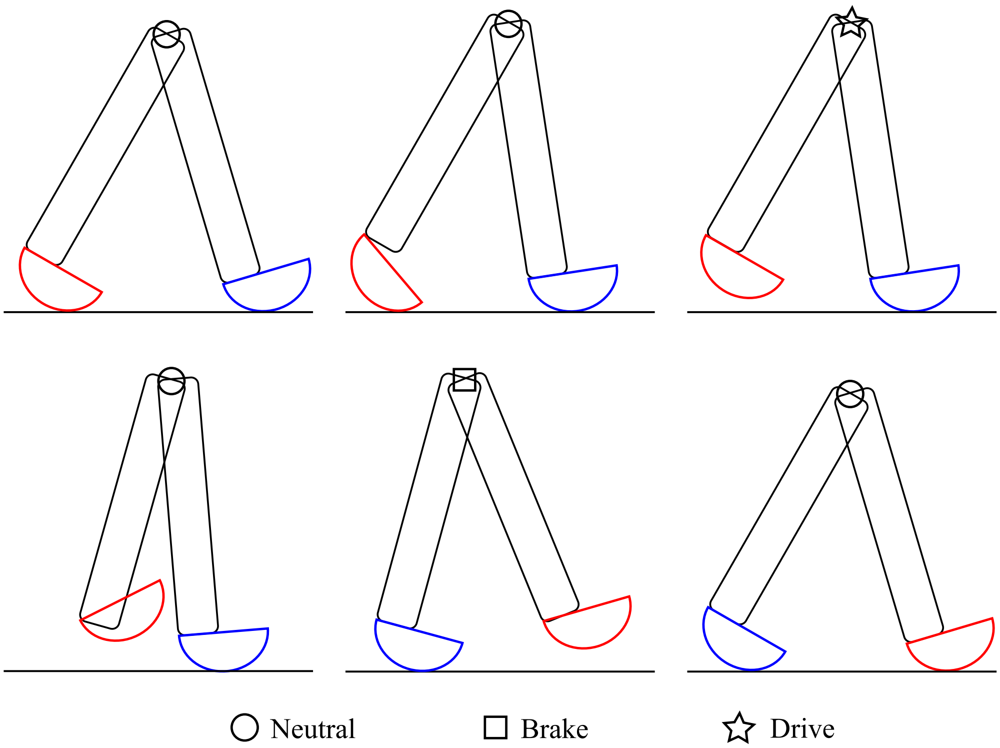


Figure 1: Drive, Neutral, and Brake Requirements of a Hip Joint

Clutch	Brake
Must not require power to maintain state	Must not require power to maintain state
Must use little power to change state	Must use little power to change state
Must allow state to change at any time	Must allow state to change at any time
Must completely decouple joint from gearhead	Must rigidly fix join
Must completely couple joint to gearhead	Must completely release joint
Must engage/disengage quickly	Must lock/unlock quickly

Table 1: Clutch/Brake Needs

- Cone - Concentric and coaxial taperered cones meet to form contact

Mechanical Lockup Mechanisms

- Square jaw - mating teeth engage
- Spiral jaw - similar to square jaw, but with sloping enagement surfaces
- Sprag - Sprags wedge between concentric cylinders when rotation is in particular direction
- Wrap Spring - Spring tightens to engage coaxial shafts
- Roller Ramp - Rollers wedge between concentric cylinders

Electromagnetic Mechanisms
Oil Shear Mechanisms

It is best to reduce this list before continuing. Electromagnetic mechanisms are immediately removed from consideration because they inherently require power to maintain state. The oil shear mechanism is also removed due to inherent complexity and the fact that it is better suited to designs that require variable slipping rather than complete linkage. Mechanical lockup mechanisms were initially considered, but then eliminated as they would not permit slippage at over-torque - a safety feature required to protect the gearbox. The drum concept is best suited for brakes rather than the more general clutch, and the cone mechanism would be too difficult to fabricate. This leaves the disc mechanism [1, 2.21.06] - a reasonable choice, as it is the preferred method for both clutching and braking in the automotive industry.

The essential components of a disc clutch mechanisms are shown in Figure 2. Two friction discs are concentrically aligned on a shaft but may translate with respect to one another. To engage, one disc translates to meet the other. A preload is created between the two faces, locking the two together by friction. To disengage, the disc translates in the opposite direction, allowing the discs to rotate independently. The required actuation is linear, and it should be noted that the actuator must provide for a preload to be held constant without wasting power.

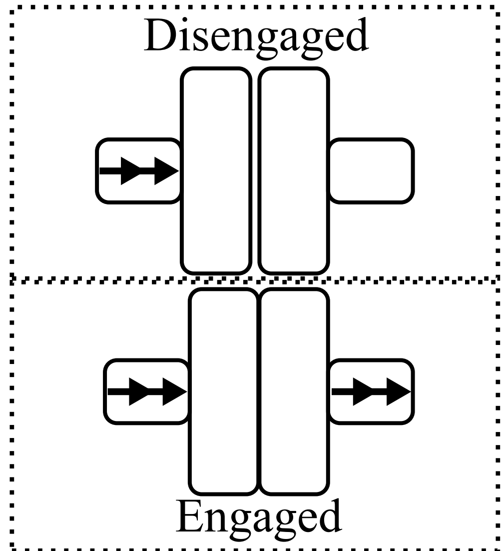


Figure 2: A Simple Disc Clutch

2.3.2 Clutch and Brake Actuation

Next the concepts for actuation of the clutch and brake mechanisms should be considered. Again, because the clutch and brake mechanisms are so similar, their actuation can be considered together. Here the list [1, 2.13.06] is reduced because the choice of a disc clutch has already been made:

- Mechanical - Controlled mechanicaly by a human
- Electric - Solenoid or Motor
- Pneumatic - Piston
- Hydraulic - Piston
- Self-Actuating - relies on motion or position of part to engage or disengage

Clearly self-actuation can be ruled out because it is desired to be able to change state at any given time. Human mechanical control would be suitable for a display model, but not an autonomous robot. Pneumatic actuation is chosen for convenience for the clutch/brake prototype as a piston provides the linear actuation required by the disc clutch and brake mechanisms and does not require energy to maintain its state. For the final robot version of the clutch/brake, electric actuation would be preferable, but this would require some sort of toggle mechanism to yield the desired linear motion and maintain preload between the discs without wasting power. Making this decision requires the *mechanism* to be independent of its particular *actuator* in the design, so the final product can be used with any particular actuator. The mechanism may impose requirements on the nature of the force/torque required to change and hold state, but should not make demands on the method by which this actuation force/torque is produced.

2.3.3 Input and Output

The following coupling methods are possible in transmitting torque from the motor to the clutch/brake (input) and in transmitting torque from the clutch/brake to the joint:

- Direct Drive (Slot and Key, Setscrew and Flat, etc...)
- Flexible Drive (Belt, Chain, etc...)
- Gear Drive

These components should not drive the design as they are all effective and well established, rather the specific technology will be chosen as needed once the configuration of the overall mechanism is fully determined.

2.4 Configuration

2.4.1 Number of Axes

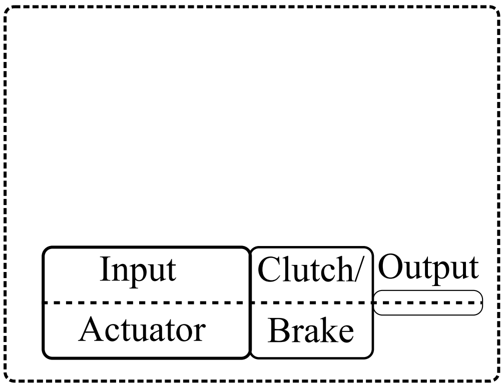
The technology used for input and output can only be chosen once the number of axes over which the input-clutch/brake-output power train will take place is determined [1, 2.22.06]. Some constraints to keep in mind are the following:

1. Motor itself must not translate (unnecessary moving mass)
2. Final output shaft must not translate (the mechanism may rotate the joint only)
3. Motor must be protected against axial loads
4. Preferable to translate entire shaft, with disc constrained axially, rather than translating disc along shaft.

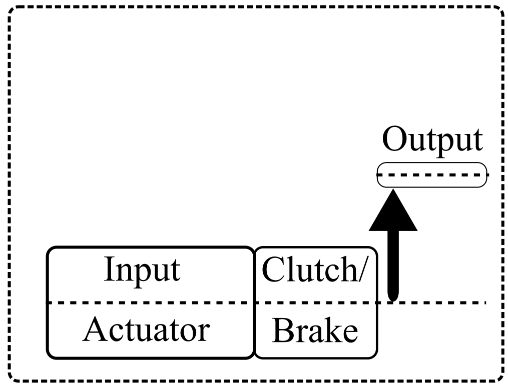
An inline configuration is shown in Figure 3(a). Here the input motor, clutch/brake, and output are arranged on one axis. A disadvantage to this design is the effectively increased axial length of the drive actuator assembly and that, according to the first two constraints, the input disc must be fixed angularly to the motor shaft but must translate axially along it - indicating the need for a spline drive, which is undesireable according to constraint 4. Figure 3(b) and 3(c) show the two dual-axis configurations. Power can be transferred between the axes by belt or chain. The advantage of both dual-axis configurations is that the need for spline drive is eliminated because the whole shaft can translate, provided that small angular misalignment of the belt/chain drive is permissible. The first of these configurations is beneficial because the output shaft does not have to support axial loads. The second is beneficial because the axial length of the drive actuator assembly is less than for the inline configuration. The three-axis configuration shown in Figure 3(d) essentially combines the two dual-axis configurations, reaping the benefits associated with each. It is very compact (axially) and neatly isolates the input, clutch/brake, and output parts of the system from one another for design convenience.

2.4.2 Number of Actuators

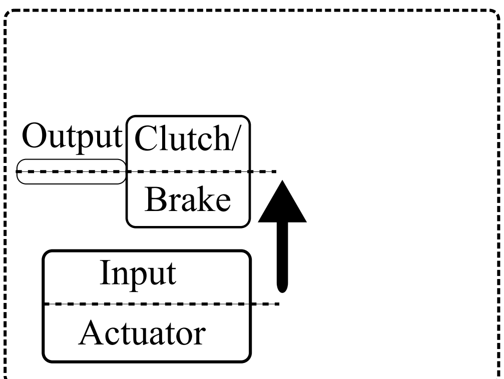
Although it is necessary for the mechanism to support three states - drive, neutral, and brake - it is not necessary for the mechanism to support multiple states simultaneously - for example, it is not necessary to engage the output with the drive motor and brake simultaneously [1, 2.28.06]. Therefore, while one actuator could be used to activate the clutch and another be used to activate the brake, it is also conceivable that only one actuator is needed to switch among the three states [1, 2.28.06]. A single-actuator method would certainly be preferable, then, as it lends itself to more compact and lighter designs. Figure 4(a) shows one such configuration, the ‘Neutral-Drive-Brake’ (NDB) configuration where the input side of the clutch can translate a certain distance to engage with the output clutch and translate even further to en-



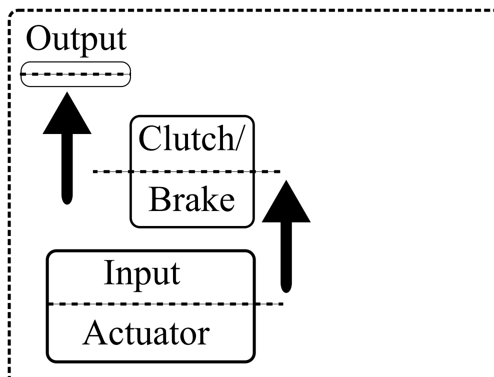
(a) Inline



(b) Dual Axis



(c) Another Dual Axis



(d) Three Axis

Figure 3: Axis Configurations

gage with the brake. This would require that there be axial flexibility in the connection between the two clutch discs, which is a prohibitive complication in the design. Figure 4(b) shows the 'Drive-Neutral-Brake' (DNB) configuration where an output disc is translated between the clutch and the brake to select the state. The disadvantage here is that when switching between drive and brake there is a period where the output can rotate freely - a disadvantage *not* present in the NDB configuration. It was ultimately decided, though, that this effect could be minimized by very low switching times, and so the latter configuration was chosen on the basis of simplicity.

3 Detailed Design

From here, a detailed design implementing the chosen mechanisms and configurations was generated. Once the inital CAD model was created, little changed conceptually over the course of four revisions (presented in Appendix A).

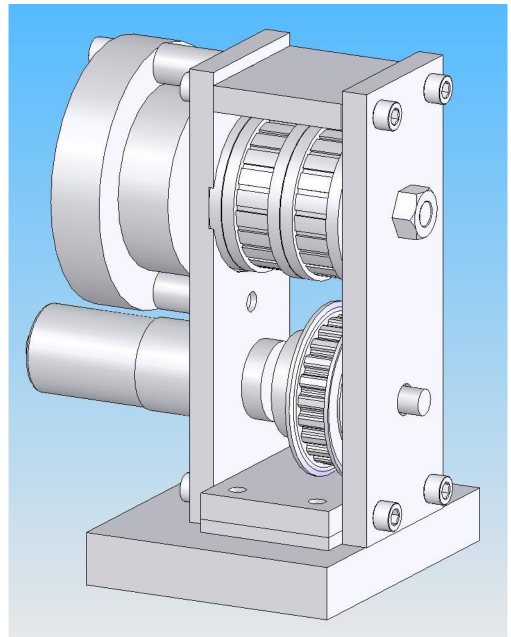


Figure 5: Mechanism Front View

3.1 Functionality Overview

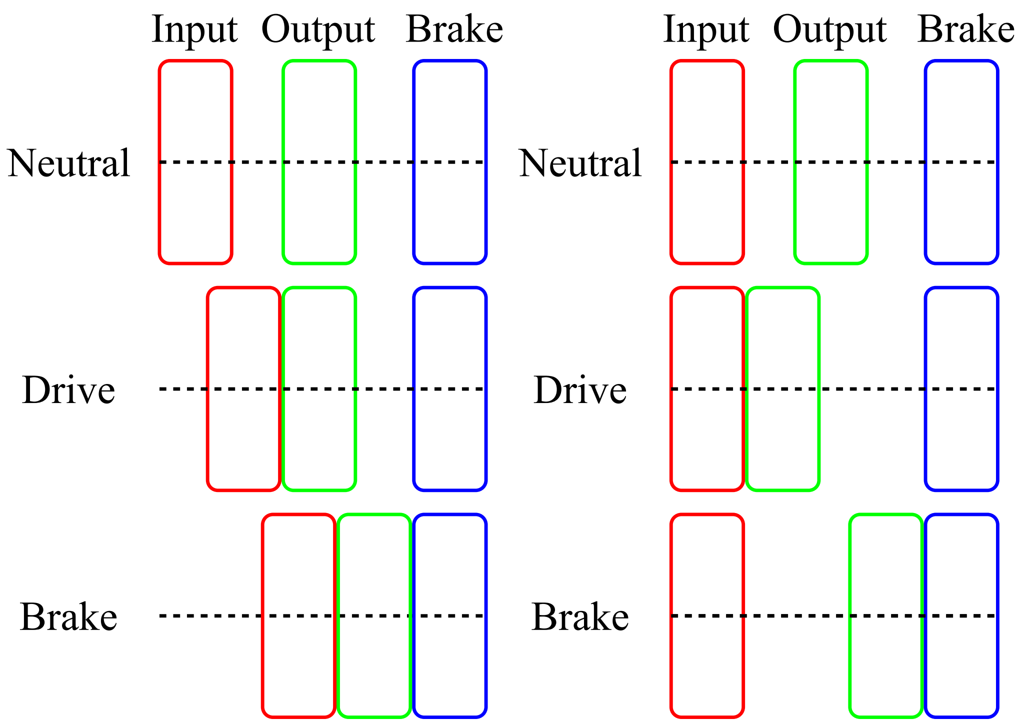
Figure 5 highlights the important parts of the design. The 'Drive Gearmotor' is mounted to the left support and is coupled to the 'Drive Sprocket' through the 'Shaft Adapter', which is necessary to increase the shaft size of the motor (4mm) to the hub size of the sprocket (.25"). The 'Input Pulley' spins freely on a 'Protrusion' from the 'Right Support' but is fixed axially. The 'Output Pulley' spins freely on the 'Central Shaft' but is fixed axially on that shaft. The 'Pancake Cylinder' is mounted to the left support above the motor and is coupled directly to the 'Central Shaft'. The 'Brake Pad' is fixed to the 'Left Support'.

In operation, power from the Drive Pulley is transmitted to the Input Pulley by a timing belt. When the Pancake Cylinder is extended, the central shaft pushes the Output Pulley toward the Input Pulley. Friction generated by the preload between the friction discs forces the Output Pulley to rotate with the Input Pulley. This is the 'Drive' state. When the Pancake Cylinder is retracted, the central shaft pulls the Output Pulley towards the Brake Pad. Here friction forces the Output Pulley to lock rigidly, as the Brake Pad is fixed to the Left Support. This is the 'Brake' state. When the Pancake Cylinder is vented, the Central Shaft does not force the Output Pulley in either direction. There is no preload between the friction discs of the Output Pulley and the Brake Pad or Input Pulley, so the Output Pulley spins freely on the Central Shaft - the 'Neutral' state. The Output Pulley transmits power to the desired output joint via a second timing belt.

3.2 Design Overview

Figure 6 shows a cross-section view of the entire mechanism. The Drive Gearmotor is seen to fit inside the Shaft Adapter; a setscrew transmits torque between the two. The Shaft Adapter fits through the Drive Pulley, another setscrew transmits torque between these two. The far end of the Shaft Adapter fits loosely in a rulon insert in the Right Support to handle loads which would otherwise create cross-axis moments on a cantilevered motor shaft.

The Central Shaft screws into the female threads of the Pancake Cylinder. It is free to translate axially, and radial loads are supported on either end by rulon inserts. The friction discs are cut out of a brake lining material and



(a) NDB Configuration

(b) DNB Configuration

Figure 4: Single-Actuator Configurations

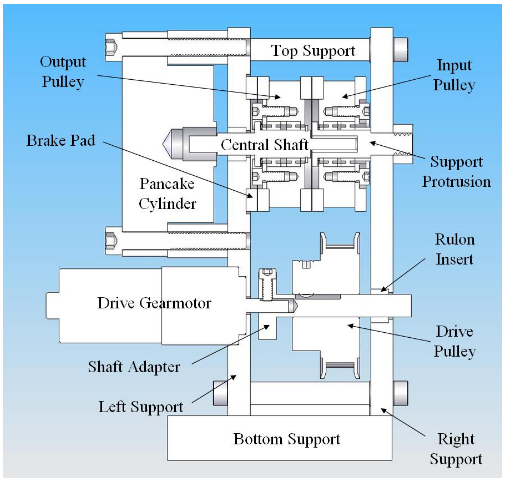


Figure 6: Mechanism Cross-Section View

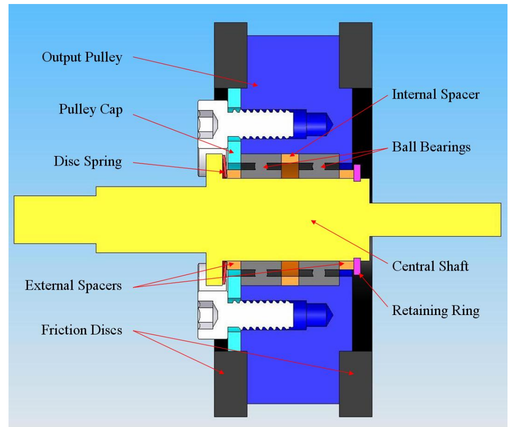


Figure 7: Bearing Preload Close-Up Cross-Section

are bonded to the pulleys and left support; they serve to couple the Output Pulley to the Brake Pad or Input Pulley according to the applied preload and also as shallow flanges for the timing belt.

Perhaps the most confusing part of the design is the preloaded bearing pairs. A single deep-groove ball bearing is only designed to support radial and axial loads; it cannot support cross axis moments. A pair of ball bearings can supports moment loads in addition to force loads, and does so best when the radial play of the bearing is removed by 'preloading' the balls against the races. Back-to-Back (DB) preloading (fixing the outer races apart and forcing the inner races together) is preferable for handling moment loads because the points at which the load lines cross the axis are far away from each other [4].

Preloading is accomplished in the Output Pulley of Figure 7 as follows:

- Separating the outer races with a spacer
- Fixing the outer races axially between a flange machined into the pulley and a cap screwed onto the pulley.
- Fixing the inner race of the right bearing against a retaining ring
- Forcing the inner race of the left bearing toward the right with a belleville spring, which is supported by the flange on the central shaft.

Assembly is managed by first fitting the outer races into the output pulley and retaining them with the cap, then pushing the inner races onto the central shaft from the right to the left and, while under load against the belleville spring, snapping the retaining ring into place. Preloading is accomplished similarly in the Input Pulley except that in place of the retaining ring, a spacer supported by the Right Support holds the inner race of the right bearing fixed. As for assembly, after fitting the sub-assembly over the protrusion, the right end is fit through a hole in the Right Support and tightened in place with a nut until the flange of the Protrusion contacts the face of the Right Support.

3.3 Analysis

The loads are small for the part sizes concerned, so little material analysis is necessary. Two areas requiring analysis for sizing purposes, though, are the friction discs [1, 3.10.06] and timing belts [1, 4.16.06].

3.3.1 Friction Discs

The contact patch of the friction discs is an annulus with inner radius r_1, outer radius r_2, and thickness given by $t = r_2 - r_1$. To be conservative and simplify the problem of calculating the relationship between slipping torque T_r and preload F_N, we can assume that the friction force acts only at radius r_1 and that friction coefficient μ is a constant. The relationship is then given by

$$F_N = \frac{T_r}{\mu r_1} \tag{1}$$

Taking the stress to be constant over a contact area of $2\pi r_1 t$ also simplifies the problem while yielding conservative results. The contact stress σ_c is

$$\sigma_c = \frac{F_N}{2\pi r_1 t} = \frac{T_r}{2\pi \mu r_1^2 t} \tag{2}$$

Friction material is readily available with a friction coefficient of $\mu = .55$ and max contact stress of 500 psi at 5000 rpm. It can be verified that an annulus of $r_1 = 1.4$" and $r_2 = 1.5$" will support a torque of 1 Nm with only 25 lb preload and simultaneously satisfy a safety factor of two on contact stress.

3.3.2 Timing Belt

According to [5], timing belt sizing is based on three factors:

- Tooth Shear Strength
- Belt Tensile Strength
- Belt Flexibility

If the conditions of all three of these factors are met under the given situation, then the belt should have serve a long, maintenance-free life.

Tooth Shear Strength: The following equations must be satisfied to ensure that belt teeth do not break in shear:

Peripheral Force:

$$F_u = \frac{2 \times 10^3 \cdot M}{d_0} = F_{u,spez} Z_e b \tag{3}$$

where F_u is the shear force, M is the torque (Nm), d_0 is the pitch circle diameter (mm), $F_{u,spez}$ is the specific tooth shear strength (N/cm-width, as tabulated), b is the belt width, and z_e is the number of teeth in the mesh given by

$$z_e = \frac{Z_1}{180} \cos^{-1} \frac{(z_2 - z_1)t}{2\pi a} \tag{4}$$

where z_1 and z_2 are the number of teeth on the small and large pulley, respectively, t is the pitch (mm), and a is the center to center distance of the pulleys (mm).

Torque:

$$M = \frac{M_{spez} z_1 z_e b}{100} \tag{5}$$

where M_{spez} is the specific torque (Ncm/cm-width, as tabulated).

Power:

$$P = \frac{P_{spez} z_1 z_e b}{1000} = \frac{Mn}{9.55 \times 10^3} \tag{6}$$

where P_{spez} is the specific power (W/cm-width, as tabulated) and n is the rotational velocity (RPM).

These three equations are satisfied well by a 1/4" wide XL timing belt around equal 1.4" pitch diameter pulleys.

Tensile Strength The weakest of BrecoFlex's 1/4" XL belt satisfies the tensile strength imparted under the situation above according to tabulated ratings.

Flexibility BrecoFlex XL belt can wrap around a pulley with a minimum of 10 teeth. The 1.4" pulleys quoted above have 22 teeth, so flexibility is not a concern. 1/4" XL belt satisfies all requirements.

3.3.3 Other Parts

Detailed analysis does not need to be shown for other parts, but a few remarks should be made:

- The only torque bearing shaft is the shaft adapter. It is to be made of 7075 aluminum because it is to be threaded, so stresses will be safely below yield for this material.
- The only significant radial forces on the pulleys are due to pretension in the belts, which should be low, so the cantilevered Protrusion need not be analyzed. It, too, is to be made of 7075 aluminum due to threadings.

- The ball bearings used have a static thrust load rating of over 50 lb when only $\approx$ 25 lb is the necessary preload between the friction plates, and preload due to the belleville springs is only 8 lb. Radial loads supported over the two bearings are only due to pretension in the belts and thus very small compared to both dynamic and static ratings. Moment loads supported by the pair are incidental and small.

- The load on the caps that retain the ball bearings within the pulleys is limited by the maximum force exerted by the piston. The eight #4 screws retaining these caps are more than sufficient.

- rulon inserts support radial loads transmitted through the central shaft but permit axial translation. Each of the rulon inserts is long, the separation between them is high, and rulon has a very low coefficient of friction against metal - it is expected that these will be a good space-saving substitute for items sold as linear bearings.

4 Fabrication

4.1 Drawings

Once the models were complete, drawings were generated for each part. Critical dimensions were generally labelled from the datum off which the machine would be 'zeroed'. Notes regarding hole tapping, shaft threading, and fits were included. As a test of the ease of fabrication, specific tolerances were *not* included on the drawings - it was hoped that reasonable care in machining would be sufficient to produce a satisfactory product. Many copies of this mechanism will be necessary for a robot and thus it would be undesireable to spend time adhering to strict tolerances. The degree of success of this approach will be noted later in the report. Machining drawings are included in Appendix B.

4.2 Stock Parts

Several parts were ordered from vendors, most notably McMaster-Carr, as it was desired that parts be inexpensive and readily available. A complete list is included in Appendix C.

4.3 Materials

Raw materials were ordered from McMaster-Carr when not immediately available in the laboratory or machine shop. The metal of choice was generally aluminum 6061-T6 for its good machinability, low cost, reasonably high strength, and relatively low density. For externally threaded parts, however, aluminum 7075 was chosen due to its higher strength and thus greater robustness. Only the output pulley cap was machined from steel (unknown alloy) due to its necessarily low thickness - this was more of a precaution than a definite necessity, as only the most rudimentary analysis would have been possible analytically. Rulon was designated to replace linear ball bearings because it is the industry standard for this use.

4.4 Conventional Machining

Most parts were machined from raw material on conventional mills and lathes. Specific steps for machining were not formally planned due to the level of experience of the machinist. All operations on the mill required only a standard vise to hold the part - a rotary table was not required. Round parts were always held with appropriate collets; the collets were mounted to blocks which could be held in the vise. On the lathe, parts were also held by collets, with the exception of the pulley stock. This was too large for the collets compatible with the lathes in the Emerson machine shop, so the pulley stock was wrapped with shim stock for protection and centered with a test indicator in the four-jaw chuck. Drill chucks were useful on both machines for using drill bits with no corresponding collet. Only one special cutting tool was required on the lathe - the tool grinder was used to reduce the width of a groove cutter to .025" as required by the retaining ring. Other operations included linear cutting (horizontal and vertical bandsaw and hacksaw), external threading (die set), and internal threading (tap set).

4.5 Advanced Machining

While the clutch-brake was designed such that all parts could be completed using only conventional equipment, several parts were machined with advanced equipment for convenience.

4.5.1 NC Machining

The ‘Left Support’ features an annular pocket in which the brake pad is seated. This could be cut in a number of ways with conventional equipment - for instance, using a rotary table on a manual mill. However, it was decided that this was a perfect job for the NC mill in the Emerson machine shop. It should be noted that although the dry run of the very simple program was executed by the machine perfectly, when the machine actually began to cut the material, it deviated significantly from its programming on the third pass when it made a sharp turn from the circular path and began to cut a roughly linear groove. The emergency stop was pressed; no reason for the malfunction could be determined. The same program was tested again, first with a dry run, and then re-running the cut. Both times the NC machine performed as expected. While the part did not come out as planned due to the malfunction, the pocket was still functional so the part did not have to be re-made.

4.5.2 Waterjet

The friction discs could be conventionally machined, for instance, with properly sized hole saws, however the dust that would have been produced by the operation would be a lung hazard. Instead of risking a very big mess, the friction discs were cut using Thomas Besemer’s waterjet machine. Also, the pulley caps could have been machined by starting from round stock, drilling the required hole patterns, and cutting the pieces to the desired thickness. Instead, though, these pieces originated from rectangular stock due to limited lathe access. The circular discs containing the hole paterns were then cut from the rectangular blanks using the waterjet.

4.6 Assembly

Once all parts were machined and cleaned they could be assembled. As in the CAD model, the overall assembly could be broken down into several subassemblies. Specific procedures were not documented, but rather can be inferred from the CAD model. Contrary to expectations, assembly was relatively easy and took very little time. Unfortunately, though, the final assembly was not fully complete as the rulon stock and drive motor were not received in time and the pneumatic equipment was never ordered due to its cost.

5 Testing

The clutch-brake prototype was intended to support the investigation of several questions, the results of which would permit the design of a final model for the walking robot:

- Coefficient of friction of friction material
- Actual relationship between friction disc preload and slipping torque
- Axial play necessary to ensure complete disengagement of friction discs

5.1 Coefficient of Friction

The coefficient of friction of the friction material was measured by setting spare friction discs on the remaining friction material sheet stock and increasing the angle of the sheet with respect to horizontal. The static coefficient of friction could be determined by finding the angle at which as static disc began to slide; the dynamic coefficient of friction could be determined by finding the angle at which a sliding disc continued to slide at a constant rate. The relationship between plane angle θ and coefficient of friction μ is $\mu = \tan\theta$ where θ. According to the coulombic model, the *coefficient* of fricion should be constant; it should not depend on the preload between the sliding surfaces. In other words, the the friction force should be linearly related to the preload with a slope equal to the constant coefficient of friction. To test this assumption, the preload between the friction disc and friction sheet was increased by attaching a mass to the friction disc. The results of the tests are summarized in Table 3. The static coefficient of friction is clearly greater than the dynamic coefficient of friction in both preload cases, as expected. Also, the dynamic coefficient of friction under increased preload is approximately the rated value. However, there is a sharp decrease in the measured coefficient of friction when the preload was increased, indicating that the coulombic model does not hold. If this curve can be extrapolated, then under even higher preloads, such as those

Preload	Static	Dynamic
Low	1	.8
High	.6	.4

Table 3: Measured Coefficients of Friction for Friction Discs

present in the clutch/brake, the coefficient of friction will be lower than the rated value and thus the slipping torque will be lower than expected.

5.2 Preload vs Slipping Torque

Testing the relationship between friction disc preload and slipping torque would have been much easier had the drive motor and pneumatic system been present in the final assembly - it would have been simple to regulate the pressure of the air driving the cylinder to preload the output pulley against the input pulley, lock the output pulley, and varying the voltage applied to the drive motor until slipping between the input and output pulleys occured; the slipping torque would then be the motor current multiplied by the motor's torque constant. Without the desired actuators, however, an elaborate system of masses and pulleys was devised to apply a constant preload betweent the output pulley and the brake disc and apply an increasing load at a known radius from the output pulley until the discs began to slip. Precise results could not be obtained because the weight of the mass enforcing the preload between the friction discs could not be determined, but it was estimated that the slipping torque was between one quarter and one half of the predicted value. This was consistent with general observation of the behavior of the mechanism - it did not feel like the friction between the discs would be sufficient for desired operation - and with the trend that coefficient of friction decreased with preload.

5.3 Required Axial Play

The axial play required to ensure complete disc disengagement depends strongly on the precision to which the clutch/brake is manufactured. It is to be assumed, however, that the final versions of the clutch/brake would be no more precise than this prototype model unless the prototype testing reveals that such precision is necessary, so it is important to determine the smallest play such that the device functions as intended. This could be accomplished by adjusting the number of washers between the right support and top and bottom supports and testing the operation of the mechanism - the minimum play required can be measured directly in this way. Without the drive motor and pneumatic system, however, this test was too difficult to complete, but general observation suggests that the nominal play designed into this prototype - a few thousandths of an inch - would be sufficient.

6 Evaluation

While the design will not be measured metrically against the specifications, it is important to discuss the consideration of certain needs and additional constraints and the extent to which these needs were met.

6.1 Cost

The cost of the mechanism was not surprisingly low, however, almost all of the required components were readily available from the (generally) economical vendor McMaster-Carr - no custom or special order parts were necessary. Some raw materials, such as the friction material, rulon, and pulley stock, had high purchase costs - however, only a small portion of these materials would be used in each clutch/brake, so the unit cost of these raw materials was low. It is not undesireable that these materials could only be ordered in large quantities, because many copies of the clutch/brake would be fabricated for use in a walking robot. The amount of other raw materials out of which any conceivable clutch/brake would have been machined - namely aluminum - was not excessive, and thus this aspect of the cost is reasonable. One item of raw material - the .5" OD, .43" ID aluminum tube - was not justifyable economically - it would have been almost as convenient to use the least expensive of round aluminum stock, and it would take dozens of clutch/brakes to use the full 6' of raw material that was ordered. The most expensive off-the shelf components were the ball bearings. It is difficult to find bearings cheaper than those used, and it is completely reasonable to use four ball bearings in a mechanism which, inherently, must support two independently rotating shafts. In total, the cost of all ordered materials and parts was $294.51. The cost of all ordered materials and parts used in the core clutch/brake mechanism (excluding actuators, which are not the components of concern in this design project) is estimated at $71.96, but should be rounded up to $75 due to the uncounted value of scrap material. As this is comparable to or less than the cost of the actuator it modifies, the design is certainly cost-effective. The total cost per unit can be calculated by dividing the total cost of all ordered components by the total number of units produced, but it is uncertain at this time how many of these mechanisms will be needed on the robot. It is expected that with careful ordering, the unit cost could be kept below the $75 quoted above due to large quantity discounts.

6.2 Safety

Other than inherent dangers associated with powered actuators, the mechanism itself poses little added safety threat. One possible precaution that should be taken when operating the mechanism is that of adequate ventilation, as the brake lining material used for the friction discs, while asbestos-free, may produce fine dusts which could irritate the lungs. This, however, is the case with most friction brake designs, and the health threat is not particulary severe or difficult to prevent. One benefit over other possible mechanisms is the use of belts, which arguably pose less of a safety threat than gears. The pressure in the pneumatic system is less than that used to pump bicycle tires, and the voltage used by the motor is ten times lower than that required by househould appliances - so the safety threats posed by these forms of potential are small.

6.3 Reliability and Maintainabilty

No significant stresses are imposed on the mechanism at any point in its operation, and so no structural component should ever fail. All bearings have a rated life under load, but since the most significant of the loads on the bearings are constant preloads small relative to the 10 million cycle dynamic load rating, they should serve the mechanism smoothly for the operating life of the robot. One of the most likely components to wear are the rulon inserts used as linear bearings, but these can be replaced simply by slipping the used ones out and sliding the new ones in. This will require minimal disassebly of the rest of the mechanism, at most requiring a few relevant sub-assemblies to be removed from the overall assembly. The shortest-lived component of the mechanism, though, will probably be the friction discs. Their rate of wear is unknown, but it is certain that material will wear away during normal operation; it is a problem inherent in all friction mechanisms. Like the rulon inserts, though, the brake pads are held in place by the close fit with the structure and are easily accessibly by removing a few sub-assemblies, and so should not be difficult to replace.

6.4 Aesthetics

The prototype clutch/brake is not ugly nor is it beautiful. Aesthetics can certainly be improved significantly by enclosing the mechanism in an attractive case. This, of course, would have been counterproductive for the prototype, as it is necessary to be able to inspect the internals during testing. Also, several parts will be given functional chamfers and fillets to facilitate assembly, which will also make the parts appear more attractive. Finally, anodization could protect the running shafts against wear, and certainly can be applied to stationary components as well to make them look more professional.

7 Discussion

All of the objectives stated in 2.1.2 were met to some extent by the design and also the final realization of the prototype clutch/brake mechanism, so the project can be judged a success. More important than assessing the level of success, however, is making suggestions for improving the design and fabrication of the clutch/brake.

7.1 Conceptual Design

The conceptual design of the clutch/brake has proven itself to be a reasonable choice as it theoretically *and in reality* meets design objectives. Specifically, the three axis DNB configuration of a friction-type clutch does permit a joint to be driven by a motor, disengaged from the motor and its gearbox to allow free rotation, or be rigidly fixed at a desired angle in a small, light, robust, and easily implemented package. While it is undesirable that the mechanism must pass through the neutral state when switching between drive and brake in theory, the prototype fabricated in this project suggests that the simplicity of the current design outweighs this inconvenience. Also, while the choice of linear actuation remains open - and an electric system would be preferable over a pneumatic system - it is clear that a single linear actuator is sufficient for the system's needs.

7.2 Detailed Design

The detailed design of the mechanism has proven sufficient for meeting the needs of the system, practical to manufacture, and simple to assemble. One area of improvement to be considered is the retention of the rulon inserts. Currently the inserts are all free to shift axially in their respective cylindrical holes, but a cap of some sort should be designed to prevent them from slipping out of place. Also, while preloading the ball bearing pairs was a good idea, the implementation can be improved. The bearing pairs are indeed stiffer under moment loads as desired - but they are also significantly stiffer under *torque* loads - some resistance can be felt in rotation. It seems that this is due to errors in fabrication. If the parts had fit together exactly as planned, it is likely that the preloading would not have stiffened the bearings significantly in rotation. As fabricated, however, it seems that the bearings are being preloaded with excessive force - that the disc springs are being fully compressed rather than limiting the preload on the bearings. A revision might include stacked disc or wave springs to allow for looser tolerances in construction. A remaining concern with this 'soft' (spring) preloading method, though, is that the bearings will become unloaded under certain conditions if the applied axial load is greater than the preload. Remedies exist, however, each has its downside:

- Increase the preload to a value greater than the maximum possible applied axial load. This is undesirable because it will make the bearings stiffer in rotation even under no load, and it will reduce the life of the bearings.
- Change from the DNB configuration to the NDB configuration so that that applied axial loads always acts in the same direction, and have the wave or disc spring preload the bearings in that direction. Another benefit of this change is mentioned in subsection 7.2, but this would require a conceptual design change and would complicate the design, as considered in 2.4.2.
- Change from a 'soft' preloading scheme to a 'hard' preloading scheme - have the inner races of the bearings precision ground such that they only meet when a specified preload is applied. This, however, would increase the cost of each bearing pair significantly.

The proper solution must be considered carefully in the revision of the prototype design.

As a general recommendation, each part should be reviewed and chamfers and fillets should be added where appropriate to enhance functionality and also where aesthetics could be improved. For instance, the steps of the support protrusion and central shaft and the bore of the pulleys should be chamfered to assist in the installation of the ball bearings. Finally, a better solution for fixing the brake pads to the pulleys should be devised. Originally, they were to be fixed using an adhesive; currently they are held in place by material deformation/friction. The final solution should allow for easy replacement, as the pads will wear over time, but retain the pads during operation more reliably than a friction fit.

7.3 Fabrication

Before machining the revision, the tolerances associated with each part's machining to ensure functionality must be calculated and adhered to. If required tolerances would be too tight should each part be machined irrespective of the actual dimensions of previously fabricated parts, then each part should be inspected after machining and the nominal dimensions of other parts should be altered accordingly. For example, after determining the *actual* depth of the bore in each pulley, the relevant lengths of the shafts, etc..., the spacers can be cut to the proper length to ensure proper bearing preload.

All of the stock components found at McMaster were inexpensive and well suited to their purposes, especially the ball bearings, but some of the materials should be reconsidered. The aluminum tube used for spacers, for instance, should not be ordered; rather the spacers can be machined from the same aluminum used for the central shaft and support protrusion. Also, the pulley caps should be machined from round stock instead of from rectangular blanks. For machining, more use should be made of the NC milling machine, as it could quickly make some of the simpler parts. For instance, when machining the left and right supports by hand, each hole was started with a center drill and completed with a drill bit; the tables were locked at each position and the drill bits were changed to ensure that the drill bit was exactly centered at the position of the center drill. The NC mill is not generally prone to the errors of a human machinist, and can instead be used to quickly create the hole patterns with the center drill and run a second pass to complete the holes with a drill bit. A third pass could chamfer each hole to remove burrs - a useful procedure neglected in the original fabrication. Using the NC mill, it is expected that these simple blanks could be completed in a matter of minutes rather than a matter of hours. Before assembly, all parts should be cleaned and degreased to ensure that mating parts fit and can be disassembled without difficulty, and to keep chips out of moving parts to prevent wear and binding.

7.4 Testing

The testing procedures discussed in section 5 were rudimentary and yielded weak quantitative results which should only be trusted for their qualitative implications. More sophisticated test procedures could be executed with the installation of the designated drive motor and pneumatic actuator, especially the experimental determination of the relationship between friction disc preload and slipping torque. One test not attempted with the current system is that of component-wise and overall system efficiency. Under constant electric power input, the power output at the motor pulley, input pulley, output pulley, and a third axis pulley should be measured with a dynamometer. This will determine whether the efficiency of transmitting power between the axes by belt is acceptable and whether the preload on the bearings is significantly reducing power output. Another test that should be conducted is that of state switch time, and the extent to which passing through neutral will affect the output of the system. This will determine whether the DNB configuration is acceptable or if a switch to a NDB configuration will be necessary for the final product.

8 Conclusion

This prototype of the clutch/brake mechanism has given a firm foundation upon which future revisions can build. The concept described in the paper is solid. While the details of the design discussed are generally good; a few small changes can yield significantly better results. The fabrication techniques used can be optimized in some areas to increase the speed and ease of manufacture. Testing shows that the basic goals have been realized by the current design, further testing can more precisely characterize the extent to which this design meets the criteria for a walking robot. It is expected that implementation of the suggestions provided in this paper could result in the production of a second generation prototype from which a minimally altered final product could be derived. This final clutch/brake will serve as a critical component in an efficient bipedal walking robot.

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