

ACTUATED CONTINUOUSLY VARIABLE TRANSMISSION
FOR SMALL VEHICLES

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ACTUATED CONTINUOUSLY VARIABLE TRANSMISSION
FOR SMALL VEHICLES

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Thesis

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ABSTRACT

This thesis is to research the possibility of improving the performance of continuously variable transmission (CVT) used in small vehicles (such as ATVs, snowmobiles, golf carts, and SAE Baja cars) via controlled actuation of the CVT. These small vehicles commonly use a purely mechanically controlled CVT that relies on weight springs and cams actuated based on the rotational speed on the unit. Although these mechanically actuated CVTs offer an advantage over manual transmissions in performance via automatic operation, they can't reach the full potential of a CVT. A hydraulically actuated CVT can be precisely controlled in order to maximize the performance and fuel efficiency but a price has to be paid for the energy required to constantly operate the hydraulics. Unfortunately, for small vehicles the power required to run a hydraulically actuated CVT may outweigh the gain in CVT efficiency. The goal of this thesis is to show how an electromechanically actuated CVT can maximize the efficiency of a CVT transmission without requiring too much power to operate.

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CHAPTER I

INTRODUCTION

What is a CVT?

A continuously variable transmission (CVT) is a transmission which can gradually shift to any effective gear ratios between a set upper and lower limit. In contrast, most transmissions equipped on production cars have only 4-6 specific gear ratios that can be selected. The almost infinite variability of a CVT allows the engine to maintain a constant speed while the vehicle increases in velocity. This can result in better vehicle performance if the CVT is shifted such that the engine is held at the RPM that it runs most efficiently at and/or produces the most power.

Because there are no steps between effective gear ratios, CVTs operate smoothly with no sudden jerks commonly experienced when a typical transmission is shifted to a different gear. This apparent advantage has ironically been one of the major factors to why they haven't been used more in production vehicles today. Since drivers expect a car to jerk or the engine sound to change as they press the accelerator pedal further, it is very confusing for them when the car smoothly accelerates without the engine revving faster. Drivers have unfortunately perceived this as the car lacking power which is causing a marketing problem for the transmissions.

Physical limitations of strength and friction have in the past restricted the CVT transmission torque handling capabilities to light-duty applications such as lawn mowers, ATVs, and snowmobiles. There was very little desire to develop them to their full potential. However, a renewed public outcry for improved vehicle efficiency combined with advancements in lubricants and materials have sparked new interests in CVTs. They have now been proven to support the torque requirements for production vehicles, buses, heavy trucks, and earth-moving equipment.

Friction CVTs

The most common types of CVTs are based on friction between two or more rotating components in which the radius for the point of contact can be varied. This is typically achieved with a variable-diameter pulley (VDP) (see figure #1) also known as a Reeves drive. In this system there are two V-belt pulleys comprised of a stationary and movable sheave separated perpendicular to their axes of rotation, with a V-belt running between them. The effective gear ratio is changed by moving the sheaves of one pulley closer together while farther separating the other. Due to the V-shaped cross section of the belt, it will be forced to ride the pulleys at a different radius than before, thus the effective diameters of the pulleys changes. Since the center of rotation for each pulley and length of the belt doesn't change in the process, the movable sheaves of both pulleys must adjust simultaneously in opposite directions in order to maintain the proper belt tension.

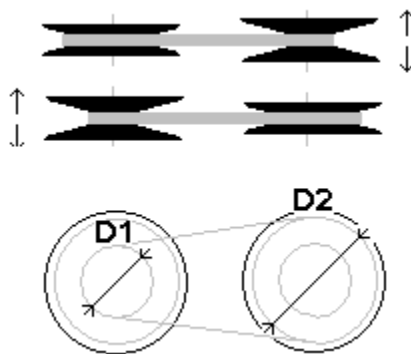


Figure 1. Variable-Diameter Pulley CVT

Hydrostatic CVT

Hydrostatic CVTs (see figure #2) use a variable displacement pump to vary the fluid flow into a hydrostatic motor. These types can generally transmit more torque since hydraulic fluid is not limited by tensile strength, but can be sensitive to contamination. Typically the hydraulic motor(s) are mounted directly to the wheel hub. Doing this eliminates the efficiency losses from friction in the drive train components. No additional gear boxes are required after this transmission to archive a reverse gear since flow to the hydraulic motors can easily be reversed using valves. Do to the heat generated by the flowing hydraulic fluid, hydrostatic CVTs are generally not used for extended duration high torque applications.

Slow moving agricultural equipment is a typical application for a hydrostatic CVT, such as tractors, combines, and foragers. They can be commonly found in many riding lawn mowers and garden tractors designed to pull equipment like a reverse tine tiller or bladed plow. There is even a variety of heavy earth-moving equipment manufactured by Caterpillar Inc. which currently uses hydrostatic CVT transmissions.

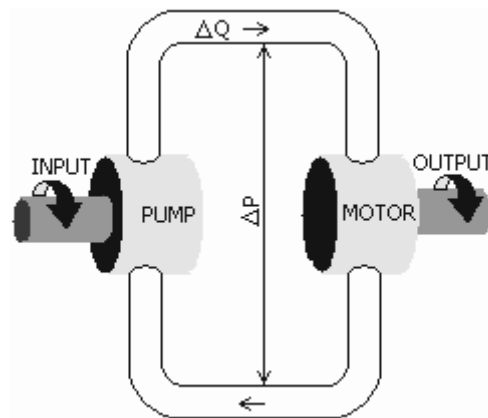


Figure 2. Hydrostatic CVT

Ratcheting CVT

The ratcheting CVT (see figure #3) is based on a set of elements that repeatedly engage and disengage based on the variable stroke of a reciprocating component connected to a one-way clutch or ratchet. The ratchet only allows the work to be transmitted in the forward motion when it is locked into static friction such that the driving and driven surfaces momentarily move together without slipping. The effective gear ratio is adjusted by changing linkage geometry of the oscillating elements.

Ratcheting CVTs can transfer lots of torque because they are based on static friction which increases relative to the transmitted torque. In a properly designed system slippage is impossible. Efficiency is generally higher than friction CVTs, such as VDP, which are based on dynamic friction that wastes energy through slippage of twisting surfaces. The major drawbacks to ratcheting CVTs are their complexity and the vibration caused by the oscillating elements.

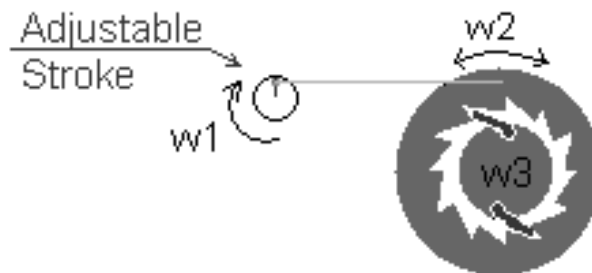


Figure 3. Ratcheting CVT

CVT applications for small vehicles

The transmission setup of a small vehicle is very critical for determining the performance. There are two main choices, a manual transmission or a CVT. Many small vehicle designers have chosen to use a CVT over a manual transmission because it automatically shifts its effective gear ratio to what is hopefully an optimum effective gear ratio thus improving performance, efficiency, and drivability.

CVTs can be deceptively simple mechanisms. As previously discussed a VDP CVT (see figure #1) is comprised of a primary drive pulley and secondary drive pulley transmitting power through a belt. Each pulley has a set of circular sheaves that forms a V shaped groove where the belt wraps around. Depending on the position of the sheaves, the belt will ride the spinning pulleys at a certain diameter to create the effective gear ratio. Controlling the position of the sheaves in mechanical actuated VDP CVTs is a system of weights, springs, and cams forcing the sheaves of the pulleys to slide in or out depending on the rotational speed of the unit.

In an ideal situation the weight, spring, and cam system will control the CVT effective gear ratio such that the engine is always running at a desired RPM for peak power and/or efficiency. In reality it is very likely the CVT doesn't maintain the best possible effective gear ratio at all times because the car has to perform on various surfaces, various inclines, and numerous other situations that likely require a completely different setup for optimum performance. Finding the optimum CVT setup can be difficult.

One way to improve the CVT is to replace the clumsy springs, weights, and cams with something that can precisely control the CVT in any situation. A computer controlled actuation system for shifting the CVT's ratio based on input parameters should be able to ensure the transmission is close to the ideal effective gear ratio in any situation. This could improve the performance and efficiency of the CVT for small vehicle use.

Some of today's most advanced vehicles are exploring the benefits of precisely controlled hydraulically actuated VDP CVTs (see figure #9) to improve fuel efficiency and performance by using an optimum ratio at all times. These vehicles use a dedicated pump connected to the engine to constantly pressurize and actuated the VDP CVT. This system can

require several horsepower to operate. In a large vehicle the increase in the efficiency of the VDP CVT can out weigh the power requirements of the pump (see figure #4).

For a small vehicle with a small motor, it is not practical to divert several horsepower to the operation of the transmission. This would greatly reduce the overall efficiency of the system. This thesis proposes that an electromechanical actuation system may be able to deliver the performance gains for small vehicles with an acceptable amount of engine power dedicated to operation of the transmission.

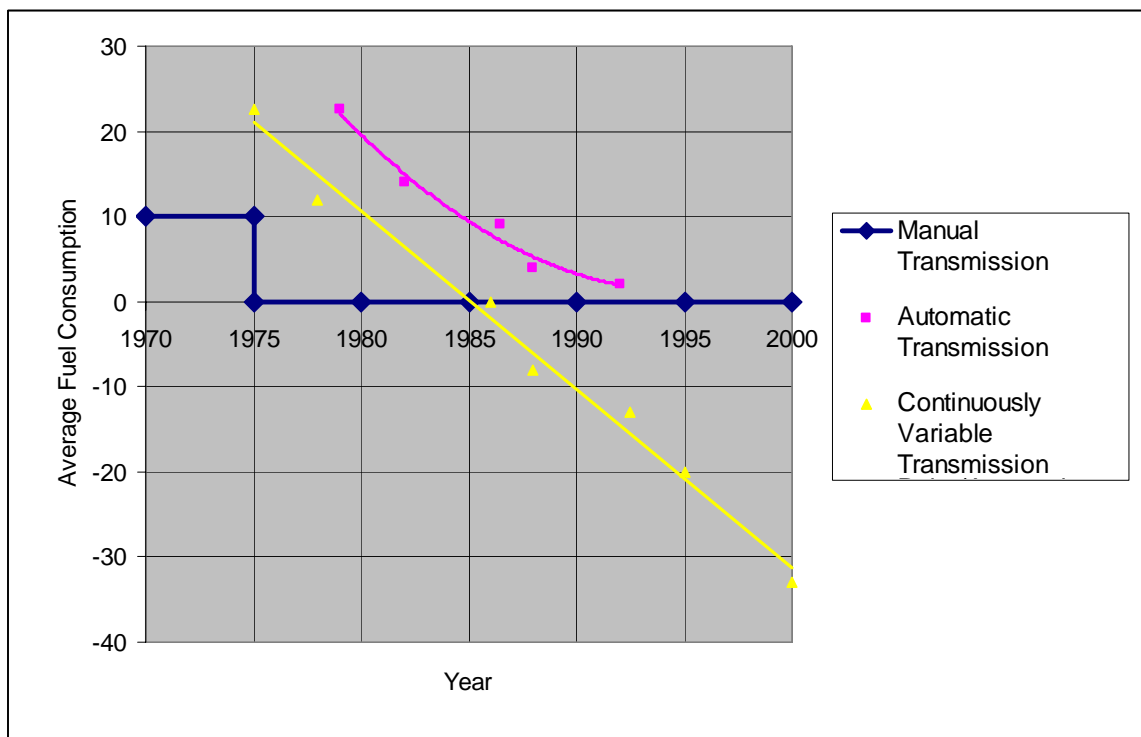


Figure 4. VDP CVT Performance Improvements

CHAPTER II

MATHEMATICAL MODELS OF VDP CVTS

Basic VDP CVT Equations

The following basic VDP CVT equations were found in a journal titled “Control of a hydraulically actuated continuously variable transmission”, see bibliography for more information. A basic mechanically actuated VDP CVT (see figure #7) is comprised of a primary drive pulley at the engine side and secondary driven pulley at the wheel side. Each pulley has a set of circular plates that forms a V shaped groove where a belt wraps around to transmit power from the primary to the secondary pulley. The CVT's transmission effective gear ratio r_{CVT} is defined as the ratio of the primary pulley speed ω_p over the secondary pulley speed ω_s .

$$r_{CVT} = \frac{\omega_p}{\omega_s}$$

Equation 1

Assuming that the pulleys are rigid and perfectly aligned. The belt is assumed to not stretch and run in perfect circles on the pulleys. Pulley clamping forces are also assumed to be large enough to prevent belt slip. Then assume the power transmission between the belt and the pulleys can be modeled as Coulomb friction. Using these assumptions, the running radii of the belt in relation to the primary R_p and secondary R_s pulleys are functions of the ratio r_{CVT} only.

$$R_s = r_{cvt} \cdot R_p$$

Equation 2

The axial position of the movable pulley sheave s_α of pulley α ($\alpha = p$ for the primary pulley, $\alpha = s$ for the secondary pulley) is determined by the running radii of the belt R_α and the

taper angle of the conical sheaves φ (see figure #5). Subscript 'max' (or 'min') implies the maximum (or minimum) value possible unless stated otherwise. Differentiation with respect to time yields the axial velocity $s'_\alpha(t)$ of the moveable sheave of pulley α . The factor 2 appears, as there are two angled sheaves. φ

$$s_\alpha = 2 \cdot \tan(\varphi) \cdot [R_\alpha - R_{\alpha,\min}]$$

Equation 3

$$s'_\alpha(t) = 2 \cdot \tan(\varphi) \cdot R'_\alpha(r'_{cvt}(t))$$

Equation 4

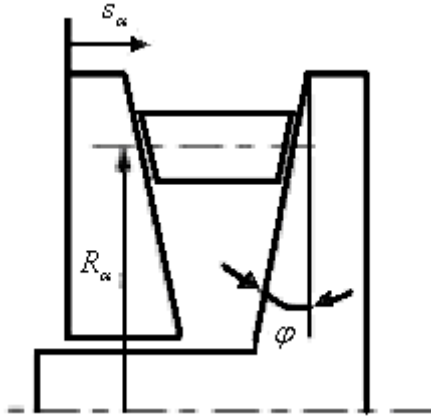


Figure 5. Pulley Sheave Definition

Radial force between the belt and pulleys is greatly reduced due to centripetal and coriolis forces. Assuming that the radial friction force component between the pulley and the belt is zero, the critical pulley clamping force to prevent slippage of the belt is given by $F_{\alpha,crit}$. Where T_α is the torque to be transmitted and μ is the coulomb friction coefficient between pulleys and belt. To find $F_{\alpha,crit}$ a worst-case approach should be applied by choosing the maximum value of μ , which is the point of transition to belt slip. The factor 2 appears, as there are two friction surfaces between pulley and belt.

$$F_{\alpha,crit} = \frac{\cos(\varphi) \cdot |T_\alpha|}{2 \cdot \mu \cdot R_\alpha}$$

Equation 5

The torque ratio τ_α is the ratio of transmitted torque to maximum transmittable torque

$T_{\alpha,\max}$ without belt slip for pulley α determined by the actual pulley clamping force F_α .

$$\tau_\alpha = \frac{T_\alpha}{T_{\alpha,\max}} = \frac{\cos(\varphi) \cdot T_\alpha}{2 \cdot \mu \cdot R_\alpha \cdot F_\alpha}$$

Equation 6

Because a good estimate of the torques acting on the secondary pulley is not available, the following estimated torque ratio $\tau_{S,estimate}$ is true assuming very little loss of power $P_p \approx P_s$ occurs during the transfer from one pulley to the other.

$$\tau_{S,estimate} \approx \frac{\cos(\varphi) \cdot T_p}{2 \cdot \mu \cdot R_p \cdot F_s}$$

Equation 7

$$\tau_{S,estimate} \approx \frac{P_p}{P_s} \cdot \tau_s \quad P_p = T_\alpha \cdot \omega_\alpha$$

Equation 8

Using equation 9 we can find the required steady state primary pulley clamping force F_p that is required to maintain a given ratio r_{CVT} with a given secondary pulley clamping force F_s and an estimated secondary pulley torque ratio $\tau_{S,estimate}$. The quantity κ in equation 9 is called the thrust ratio, which is a manifestation of the difference in traction between the belt and the two pulleys due to the difference in contact areas. The value of κ is based on a highly non-linear function of the CVT ratio r_{CVT} and the estimated secondary pulley torque ratio $\tau_{S,estimate}$. Values for κ can be found in figure # 6.

$$F_p = \kappa(r_{cvt}, \tau_{S,estimate}) \cdot F_s$$

Equation 9

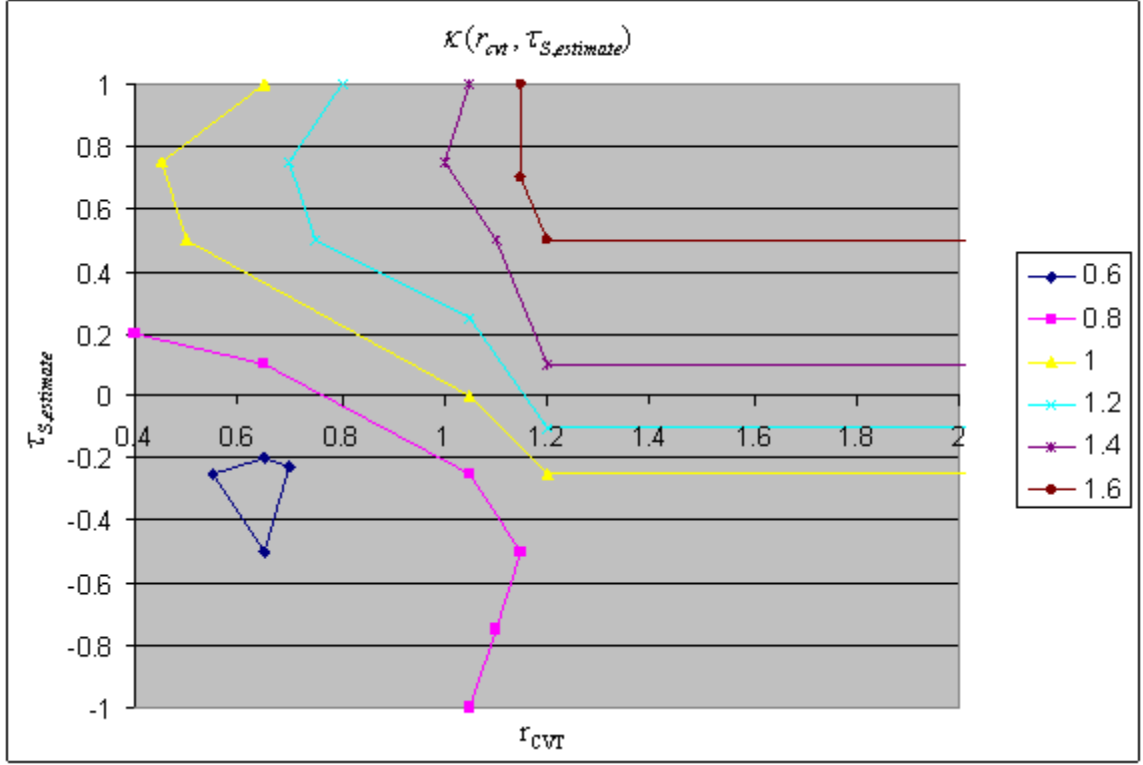


Figure 6. Contour Plot of $\kappa(r_{cvt}, \tau_{S,estimate})$

For VDP CVTs shifting is caused when equation 9 is not balanced. The difference between F_p and F_s weighted by the thrust ratio κ is the shift force F_{shift} . The rate of ratio change $r'_{cvt}(t)$ is a function of the ratio r_{cvt} , primary pulley speed ω_p , and shift force F_{shift} . Rotational speed ω_p needs to be taken into account because more length of the belt enters the pulleys per second at increasing speeds resulting in decreasing required shift force to obtain the same rate of ratio change. A non-linear function $f(r_{cvt})$ is dependent on the ratio r_{cvt} and has to be obtained experimentally for a particular CVT.

$$F_{shift} = F_p - \kappa(r_{cvt}, \tau_{S,estimate}) \cdot F_s$$

Equation 10

$$r'_{cvt}(t) = f(r_{cvt}) \cdot |\omega_p| \cdot F_{shift};$$

Equation 11

Mechanically Actuated VDP CVT Model

For a purely mechanically actuated CVT, the speed sensitive system (see figure #7) of weights, springs, and cams must be designed such as the forces of F_p and F_s are balanced at a particular ratio r_{CVT} for a particular rotational speed ω_p . Otherwise the CVT will shift until the forces are balanced. Equation 12 governs the clamping force of the primary pulley. The total of the cam weight and lever mass is M_{weight} . The distance from the center of pulley rotation to the center of mass for M_{weight} is R_{weight} . A torsion spring is used to hold the lever down until the engine reaches engagement speed and has a spring constant $k_{spr,P}$ with a pre-tension setting of $F_{0,P}$. The lever arm has a length L_{arm} from its pivot to the center of mass for M_{weight} . The distance from the center of pulley rotation to the lever arm pivot is R_{pivot} .

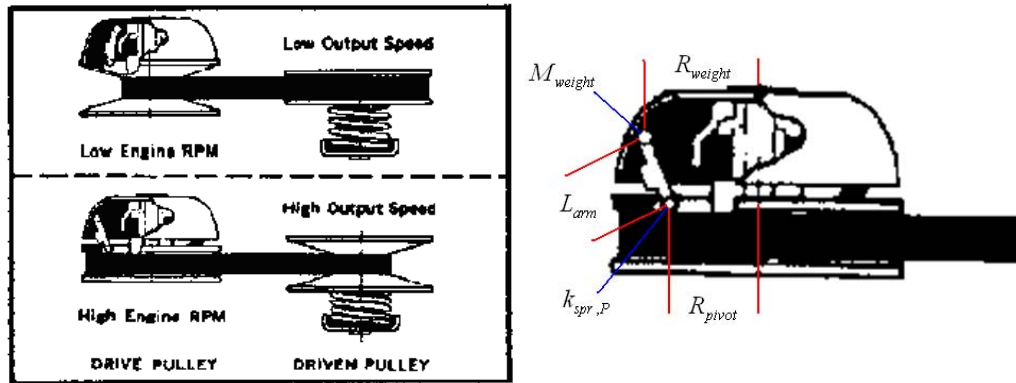


Figure 7. Speed Sensitive CVT

$$F_p(r_{CVT}, \omega_p) = [M_{weight} \cdot \omega_p^2 \cdot R_{weight} - k_{spr,P} \cdot \left\{ \frac{\pi}{2} - \cos^{-1} \left(\frac{R_{weight} - R_{pivot}}{L_{arm}} \right) \right\} - F_{0,P}] \cdot \tan \left(\cos^{-1} \left(\frac{R_{weight} - R_{pivot}}{L_{arm}} \right) \right)$$

Equation 12

$$R_{weight} = \text{function_of_} r_{CVT}$$

Equation 13

$$\omega_p = \frac{RPM_{engine}}{60} \cdot 2 \cdot \pi$$

Equation 14

Equation 15 governs the clamping force F_S for the secondary pulley in a purely speed sensitive design. It should be noted that for a purely speed sensitive system the secondary pulley can be completely linear since the cam on the primary pulley controls the non-linear aspects of determining the correct effective gear ratio r_{CVT} for a particular speed. The linear spring produces the clamping force F_S based on the product of the spring constant $k_{spr,S}$ and displacement s_s plus the pre-tension $F_{0,S}$.

$$F_S = k_{spr,S} \cdot s_s(r_{cvt}) + F_{0,S}$$

Equation 15

A purely speed sensitive system cannot compensate for a sudden increase in the demand for torque, such as driving uphill or increased weight to the vehicle. In order to compensate for these increased demands in torque designers have developed a torque sensitive secondary pulley (see figure #8) by adding a cam to the secondary pulley. Equation 16 describes the clamping for this secondary pulley design. In which $R_{CAM,S}$ represents the radius of the cam and $\phi_{CAM,S}$ represents its incline.

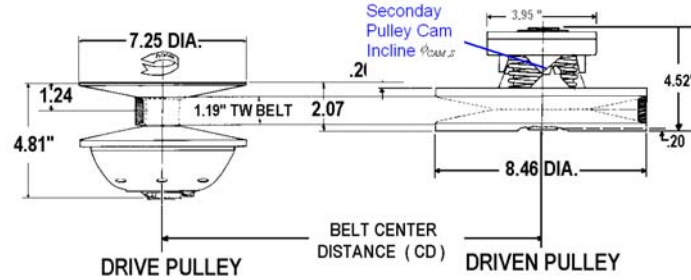


Figure 8. Torque Sensitive CVT

$$F_S = k_{spr,S} \cdot s_s(r_{cvt}) + F_{0,S} + \frac{T_S}{R_S} \cdot R_{CAM,S} \cdot \tan(\phi_{CAM,S})$$

Equation 16

Hydraulically Actuated VDP CVT Model

For the purpose of the Hydraulically Actuated CVT Model (see figure #9), the power requirement for a pump to run a perfect system needs to be determined. It is assumed that a black box computer will monitor the system and adjust the pressures perfectly to obtain the correct ratio r_{CVT} and balance of equation 9. In this situation both F_p and F_s are governed by equations 17, in which A_α is the area of the pulley hydraulic piston and p_α is the line pressure for that pulley.

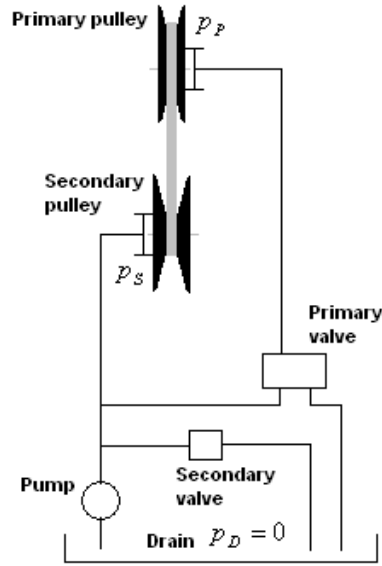


Figure 9. Hydraulically Actuated CVT

$$F_\alpha = A_\alpha \cdot p_\alpha$$

Equation 17

The average power consumption of this ideal hydraulic actuation system can be found by integrating the clamping force over the distance the CVT is actuated, the start position $s_{cvt,0}$ to the final position $s_{cvt,1}$, for both the primary pulley and secondary pulley.

$$Average \text{ _ Actuator _ Power} = \frac{\int_{s_{cvt,0}}^{s_{cvt,1}} F_\alpha(s) \partial s}{Time}$$

Equation 18

Electromechanically Actuated VDP CVT Model

The Electromechanically Actuated CVT (see figure #10) will be a modified speed sensitive CVT model with an electromechanical actuator added to the second pulley. In this model a majority of the clamping forces F_p and F_s will be provided by the weight, spring, and cam system. F_p will be equal to equation 12. The electromechanical actuator will be controlled by a black box computer that provides just enough force $F_{S,correction}$ to correct the second pulley F_s for the ideal effective gear ratio r_{CVT} . The power consumption should be greatly reduced over the hydraulic system since only one pulley will be actuated at a force $F_{S,correction}$.

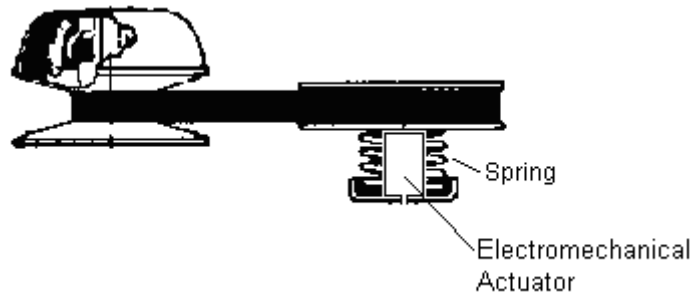


Figure 10. Electromechanically Actuated CVT

$$F_s = k_{spr,s} \cdot s_s(r_{cvt}) + F_{0,s} + F_{S,correction}$$

Equation 19

$$Average_Actuator_Power = \frac{\int_{s_{cvt,0}}^{s_{cvt,1}} F_{S,correction}(s) \partial s}{Time}$$

Equation 20

CHAPTER III

PERFORMANCE ANALYSIS

The following analysis is to show if an ideal hydraulically actuated or an ideal electromechanically actuated VDP CVT is even capable of improvements over an existing fully tuned mechanically actuated VDP CVT. This analysis is based on data collected from a Comet Industries Model 790 CVT (see appendix A) hand tuned for specific use on the University of Akron 2004 SAE Baja vehicle. This data was obtained with a specially designed inertia dynamometer (see figure #11) which was built to closely mimic the inertia of a Baja vehicle traveling on a flat surface. A brake installed on the fly wheel can be used to mimic the additional force required for the vehicle to scale a steep hill.

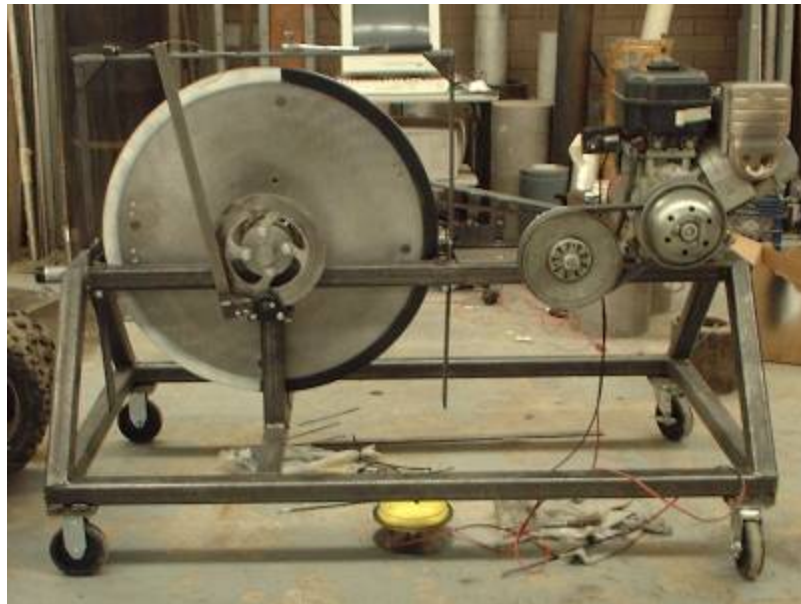


Figure 11. Inertia Dynamometer

After analyzing the data, it has been determined that the hand tuned Model 790 CVT is capable of holding the engine at its peak RPM for power output of 3100 to 3200 RPMs (see figure #13) on a flat surface between the ratios of 2.25 to 0.6 (see figure #12). This result in an efficient transmission for the Baja car at speeds over 11 mph on flat surfaces, which represents a majority of the driving conditions. However, for effective gear ratios higher than 2.25 on flat surfaces or nearly any effective gear ratio on a steep hill the CVT is not optimized. This would represent vehicle speeds less than 11 mph and any travel up a tall steep hill, which takes place a considerable amount of time.

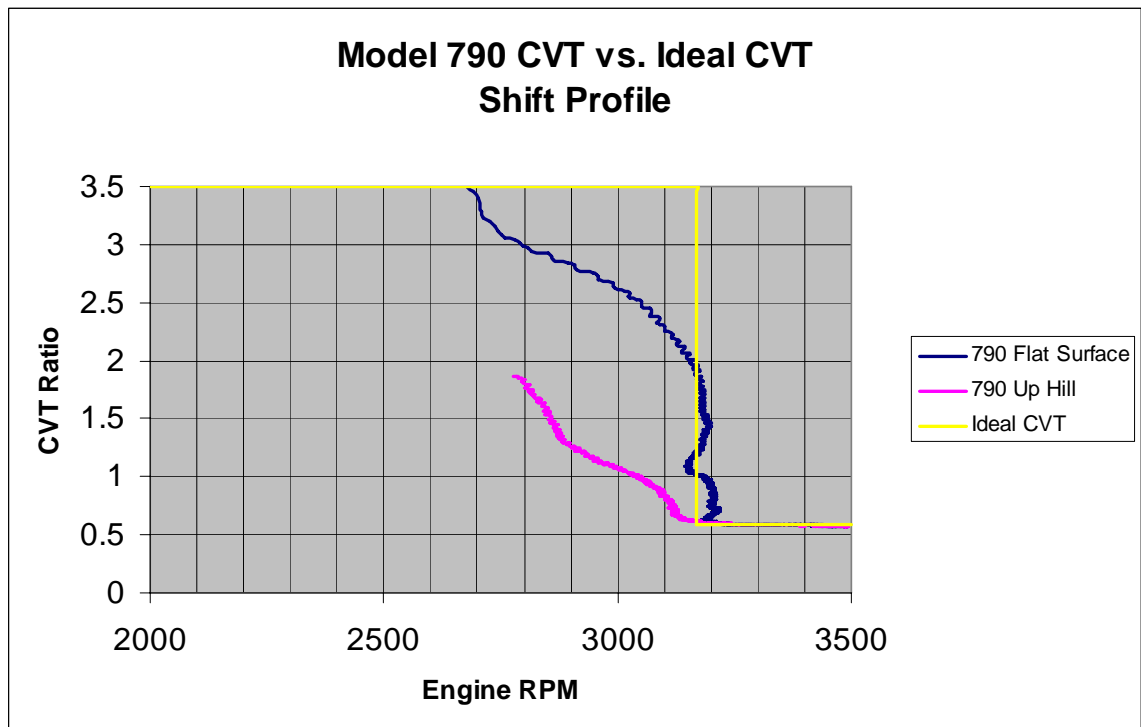


Figure 12. Model 790 CVT vs. Ideal CVT Shift Profile

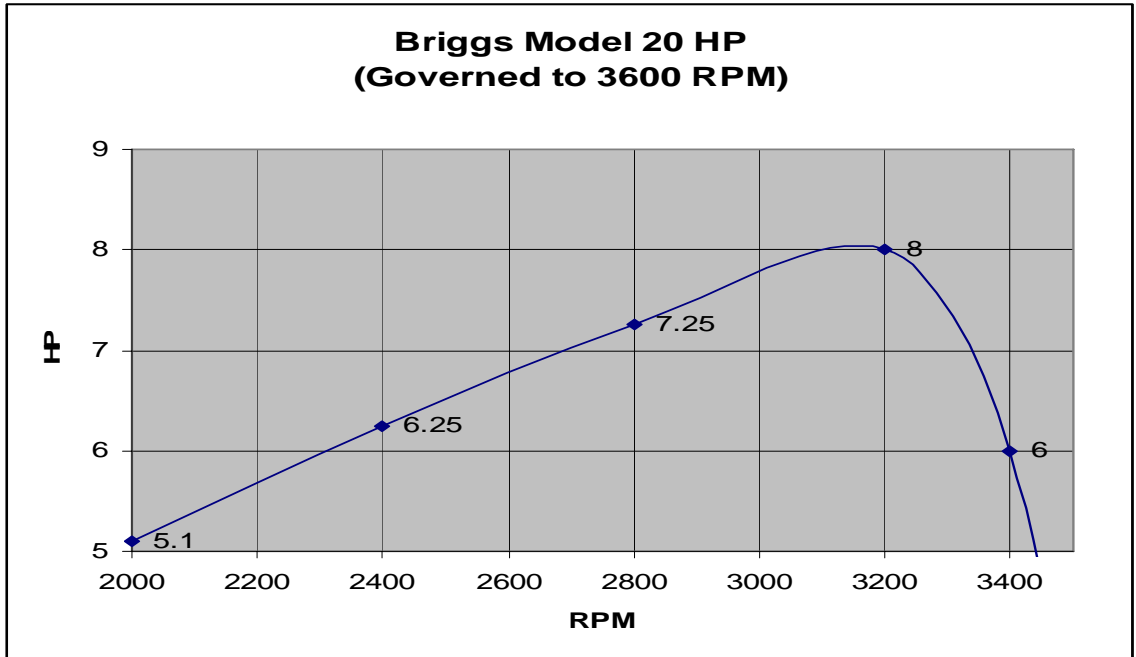


Figure 13. Engine HP Curve

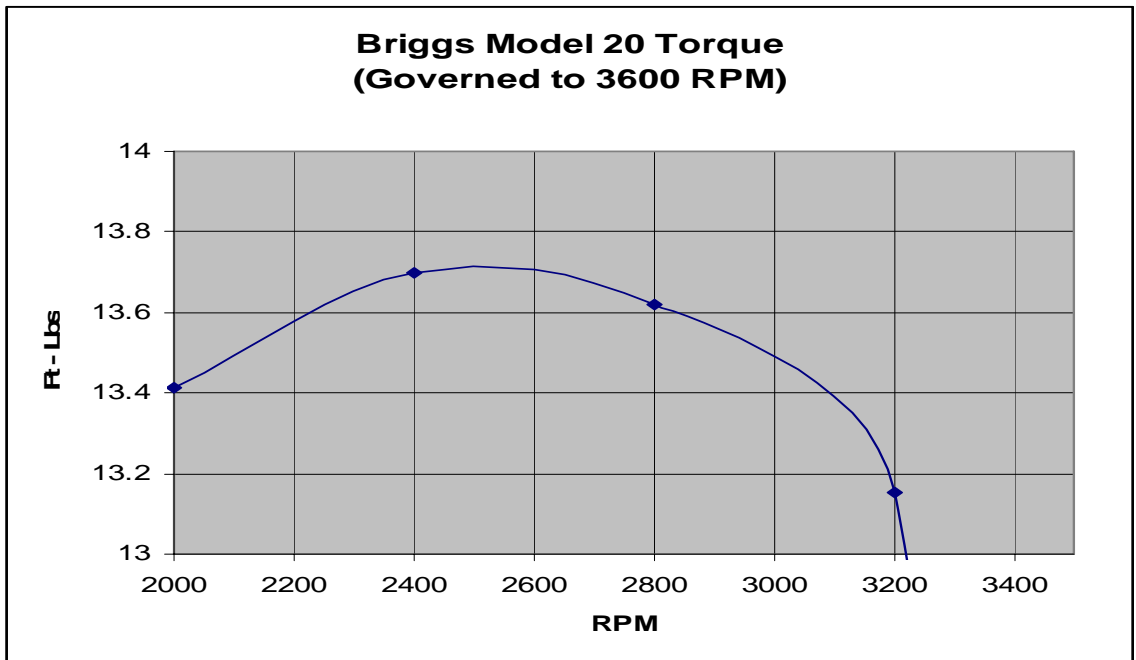


Figure 14. Engine Torque Curve

There is a potential for improvement of the 790 CVT. The question is whether or not the potential gain in transmitted power is larger than the power requirements of an actuation system. In Figure 15 there is a visual representation of how much power the 790 CVT transmits over its range of ratios vs. an ideal CVT system. For the controlled actuator to be a benefit between the effective gear ratios of 1 to 2 it would need to consume on average less than 0.06 HP to operate. At that power consumption over 1 HP could be gained for effective gear ratios from 3 to 3.5.

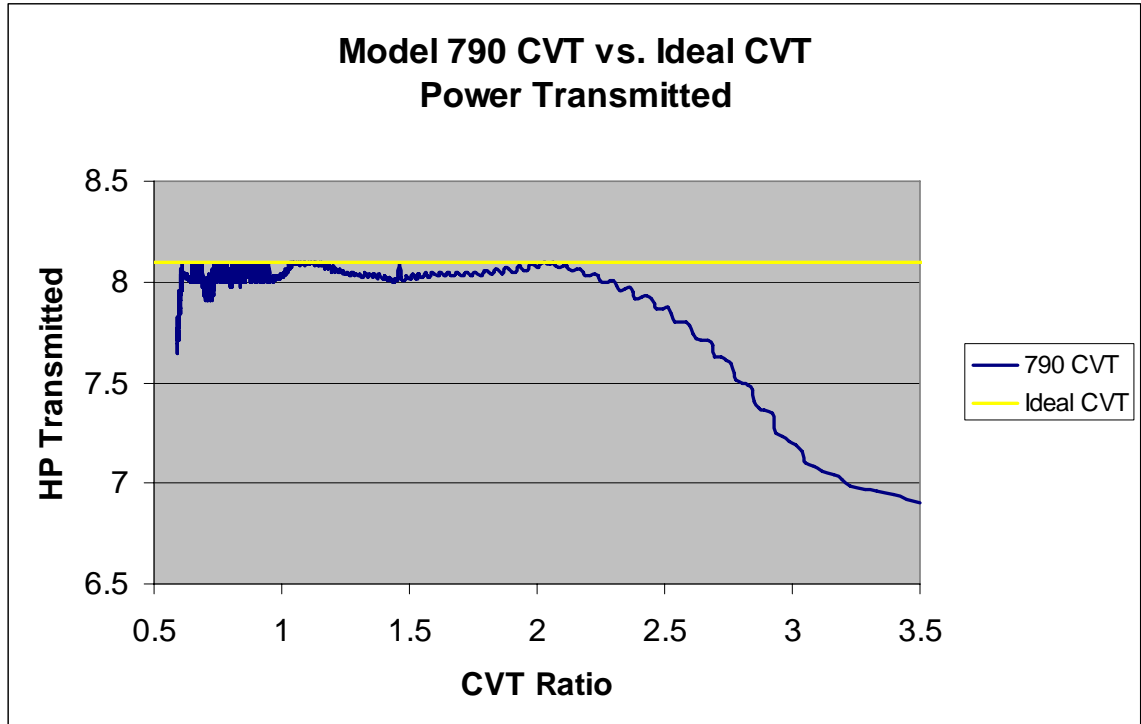


Figure 15. Model 790 CVT vs. Ideal CVT Power Transmitted

Consider the case of a fully hydraulic actuated VDP CVT. In order for this to be viable its required average operational power needs to be less than 0.06 hp to out perform the 790 CVT between the effective gear ratios of 1 to 2. Starting by evaluating how much power it would require a fully hydraulic VDP CVT to copy the force generated by the 790 CVT secondary pulley as it shifts ratios. The secondary pulley clamping force $F_{S,790}$ is governed by equation 15 but actual values for the spring force are used in equation 22. The required energy to fully shift the CVT can be found by integrating the clamping force $F_{S,790}$ over the range of motion for the pulley

plates in equation 23. The average required power to actuate is found in equation 24 by dividing the energy to shift the CVT by the time it takes for it to shift. A hydraulic system would require on average 0.0019 HP to actuate the secondary pulley.

$$F_{S,790} = 42 \frac{lb}{in} \cdot s_{s,790}(r_{cvt}) + 30lb$$

Equation 22

$$\int_{0in}^{1.19in} F_{S,790} = 0.009556 \text{ } _{Btu}$$

Equation 23

$$P_{S,790} = \frac{0.009556 \text{ } _{Btu}}{7 \text{ } _{sec}} = 0.0019 \text{ } _{hp}$$

Equation 24

To determine the average power requirement to hydraulically actuate the primary pulley, it is convenient to consider the relationship of clamping force F_p and F_s are related by thrust ratio $\kappa(r_{cvt}, \tau_{S,estimate})$ as shown in equation 9. The thrust ratio constantly changes as the CVT shifts but for a worst case scenario assume $\kappa(r_{cvt}, \tau_{S,estimate})$ equals the highest valve from figure #6, which is 1.7 (see equation 25). Because both the primary and secondary pulleys actuate almost the same distance in the same amount of time, the ratio of power required for the primary pulley in relationship to the secondary pulley shown in equation 26 is the same as the relationship of clamping force from equation 27. Add the average power for the primary pulley (equation 26) and secondary pulley (equation 24) in equation 27 to find a fully hydraulically actuated CVT replacement for the 790 CVT would only require 0.0052 hp (<<0.06 hp MAX allowable) with a worst case thrust ratio of 1.7. This means the ideal fully hydraulic actuation system could be used to replace the 790 CVT in this scenario.

$$F_p = 1.7 \cdot F_s$$

Equation 25

$$P_p \approx 1.7 \cdot P_s = 0.0032 \text{ } _{hp}$$

Equation 26

$$P_p + P_s = 0.0052 \text{ _hp} \ll 0.06 \text{ _hp _MAX}$$

Equation 27

Since an ideal full hydraulically actuated CVT has low enough power requirements to replace the 790 CVT, an electromechanically actuated CVT, as described on page 16, is assured to have even lower power requirements since the mechanical portion (weights, springs and cams) will do most of the work. Considering an ideal electromechanically actuated CVT in which the actuator is only added to the secondary pulley, based on equation 24 the required power would only be 0.0019 hp. If a spring was added to the secondary unit to provide part of the clamping forces, the power requirements of an electromechanically actuated CVT could be reduced further.

CHAPTER IV

EXPERIMENTAL DATA

An experiment was conducted to evaluate and confirm the potential gains that can be obtained with an electromechanically actuated CVT vs. a standard mechanically actuated CVT. In order to confirm the analysis from chapter III, the electromechanically actuated version of a CVT would need to demonstrate a net gain in power transmission over its mechanically actuated counterpart. The setup of this experiment can be seen in Figure 16.

The motor is a 10 hp Briggs & Stanton 4 cycle gasoline motor. It was chosen for its similarity to the engine used in the SAE Baja completion, which is an ideal application for this experimental transmission. The primary difference between this motor and the motor used in the competition is a factory installed 10 amp 12 volt alternator. It was intended for this alternator to be the power source for the actuator but a 12 volt batter was used instead for convenience.

A Comet Industry 790 CVT was used as the control unit and the base for the modified unit since it is the preferred CVT used by the University of Akron Baja racing team. It can handle up to a 16 hp 4 cycle engine and shift to any ration between 3.38:1 and 0.54:1, which is the widest range of ratios commercially available for this application. It is very versatile since it can be tuned for a particular application with various weights, cams, and springs.

The linear actuator was purchased from Fircelli Automations. They offer a variety of actuators in length, force, and speed. Through experimentation it was determined that the actuator would be required to produce up to 80 lbs of force at a speed up to 0.4 inches per second. At first glance the model FA-100-S-12-4" with 100 lbs of force at 0.625 inches per second seemed like the ideal choice but it was feared that the duty cycle of the electric motor wouldn't allow for continuous operation at 80 lbs of force. A more robust model FA-PO-150-12-4" was chosen instead because its max force capability of 150 lbs at 0.5 inches per second made it

more likely to survive a continuous application of 80 lbs of force while marinating the necessary speed. The model also featured a potentiometer for detail measurement of its position, which was not used in the experiment.

Control of the actuator was performed by a combination of a laptop computer, high speed data acquisition unit DI-148U, and programmable RC car motor controller unit Tazer 15T. The data acquisition unit monitored the speed of the 10 hp engine and relayed the information to the laptop software. When the laptop software determined the selected threshold for engine speed had been surpassed it signaled the data acquisition unit to turn on one of its digital output channels. Once the RC car motor controller detected the open channel of the data acquisition unit it applied full power to the linear actuator. In a production unit the computer and data acquisition unit would be replaced with a single dedicated programmable chip but this was unnecessary for the experiment.

A control run was first made with the unmodified model 790 CVT per the test procedure in Appendix F. The goal is for the CVT to transmit the power from the 10 HP engine at full throttle to the flywheel. CVT shift profile, power transmitted, and acceleration of the flywheel were record for future comparison.

The 790 CVT was then modified for the addition of the electromechanical linear actuator for the purpose of precisely controlling the CVT's ratio. Data acquisition used to monitor the experiment was also used to trigger the operation of the linear actuator when the 10 HP engine RPM exceeded 3170 RPM, which has been determined to be the optimum engine speed for peak power generation. The same test per the procedure in Appendix F was conducted on the modified CVT as the unmodified.

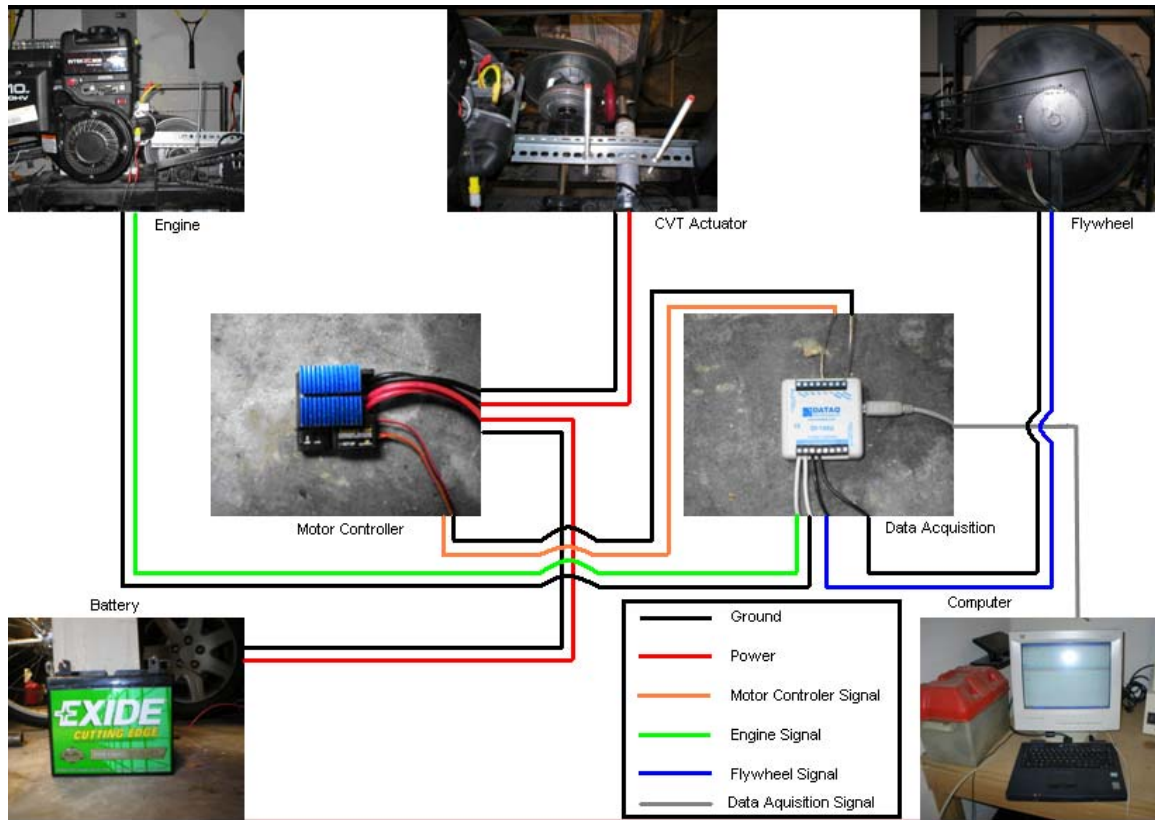


Figure 16. Computer Controlled Actuated CVT Experiment Diagram

Shift profiles of the 790 CVT and the electromechanical CVT are shown in Figure 17. The results indicate the electromechanical CVT shift profile was closer to an ideal CVT between the ratios of 3.5 to 2 than the 790 CVT. As a result of the electromechanical CVT shifting closer to the ideal CVT shift point, the engine is allowed to operate at a more efficient speed for power production. Figure 18 shows the improvement in power transmission obtained. Initially the electromechanical CVT transmitted in excess of 1.2 HP more power over the 790 CVT between the effective gear ratios of 3.5 to 3.25, a 17% improvement. Due to the selected linear actuator initially having insufficient speed to allow the electromechanical CVT to maintain the peak engine performance, the advantage decreases to only 0.5 HP average improvement for the electromechanical CVT from the effective gear ratios of 3.25 to 2.75. During the remainder of the CVT effective gear ratios the electromechanical CVT maintains an average of 0.045 HP improvement over the 790 CVT. The power to operate the linear actuator was found to be 0.027 HP, see Figure 19, which is less than the 0.06 hp max threshold previously determined. Thus the electromechanical CVT maintains a net power gain over the 790 CVT for the entire shift profile, allowing the flywheel to accelerate faster, see Figure 20.

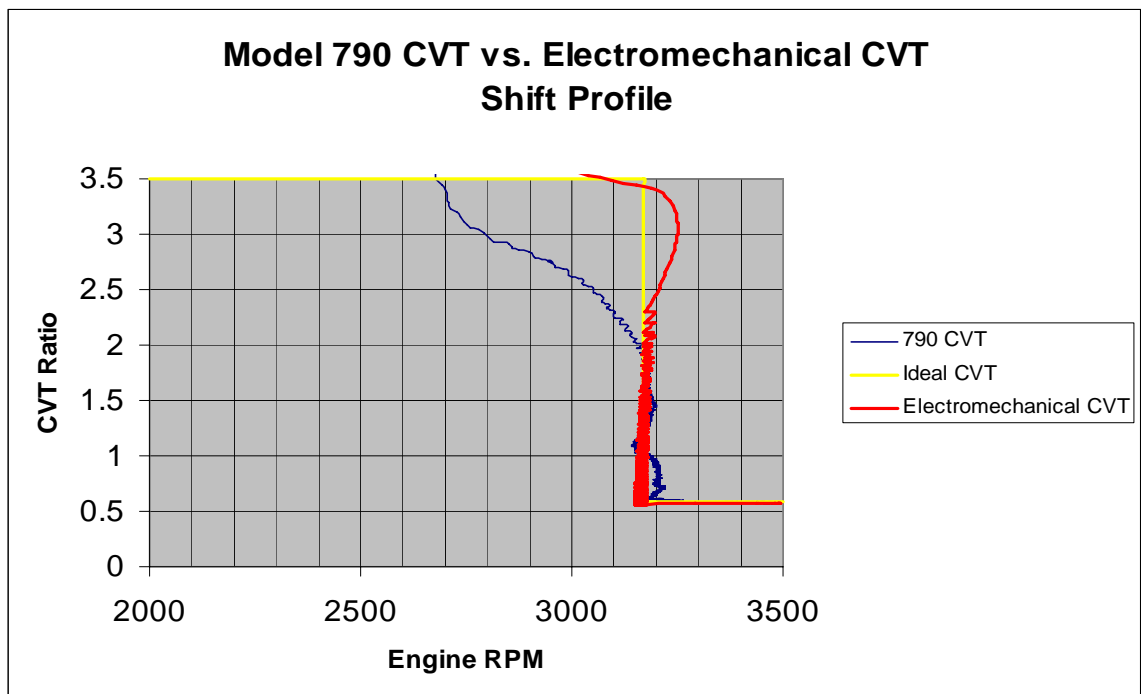


Figure 17. Model 790 CVT vs. Electromechanical CVT Shift Profile

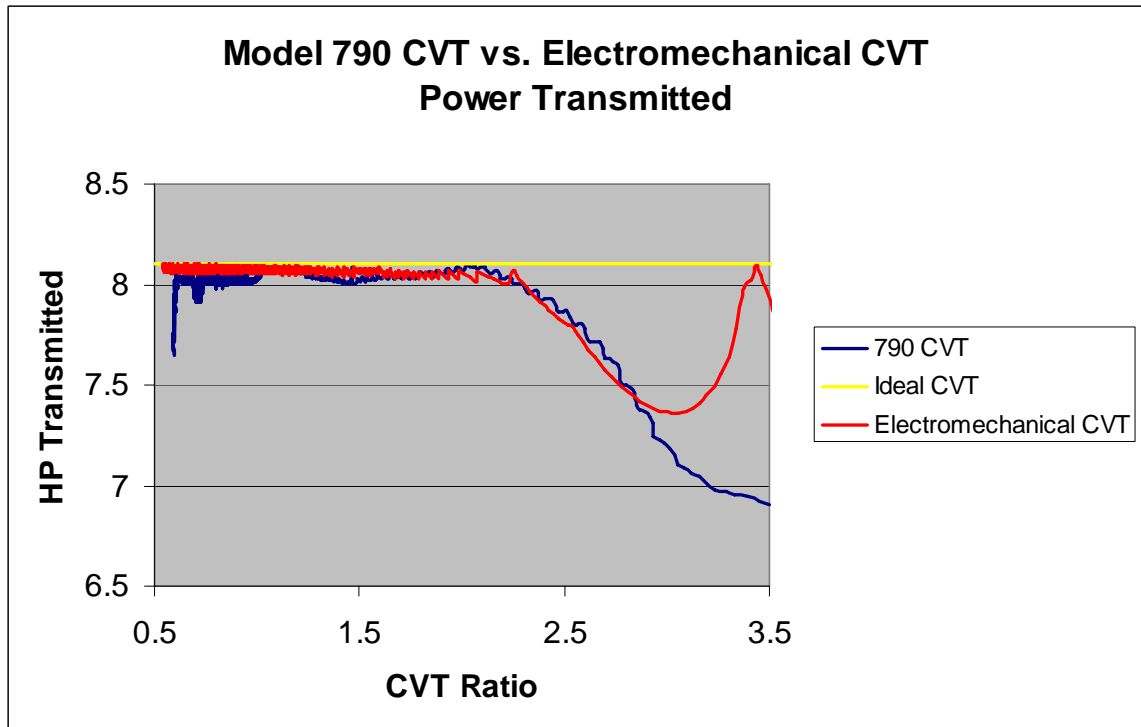


Figure 18. Model 790 CVT vs. Electromechanical CVT Flywheel Acceleration



Figure 19. Linear Actuator Power Consumption

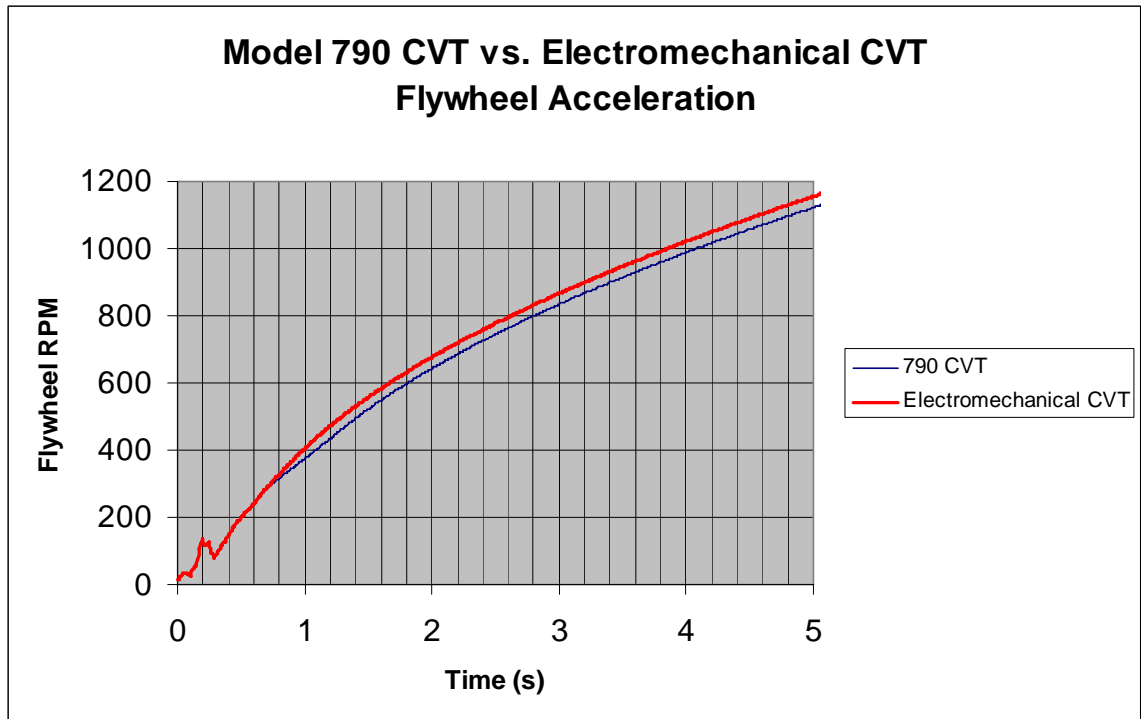


Figure 20. Model 790 CVT vs. Electromechanical CVT Flywheel Acceleration

Future design improvement could be made to the experimental setup such that a working prototype could be built for a testing ground such as Baja SAE. A slave hydraulic piston could be directly integrated into the secondary CVT pulley such that the linear actuator could operate the CVT via a master hydraulic piston located in a protective area of the vehicle. The laptop, data acquisition, and motor controller could be integrated into one programmable chip. Power to operate the system could be provided by an alternator attached to the engine. In case of system failure, the hydraulic system could include an electromechanical valve which when open would reduce the pressure to atmospheric and allow the CVT to operate similar to a fully mechanically operated CVT.

Basic control logic could effectively control the CVT. Up-shifting could occur when the RPMs of the engine exceeded a desired set point. The CVT should down-shift when the engine is below a desired set point. For system stability a small gap may be desired between the up-shift and down-shift set points. Limit switches built into the linear actuator would ensure that over travel does not occur in either CVT shifting directions.

CHAPTER V

CONCLUSION

Based on the analysis and experimental data in this thesis, it is possible to make a gain in performance and efficiency for CVTs via computer controlled actuation of the transmission by either implementing a hydraulic system or an electromechanical system. In an experiment it was demonstrated that an electromechanically actuated CVT operating on 0.027 HP transmitted as much as an additional 1.2 HP or 17% more power than an equivalent mechanically actuated CVT. This additional power could be used to accelerate a small vehicle faster than what would be achieved by a non-computer controlled actuated CVT.

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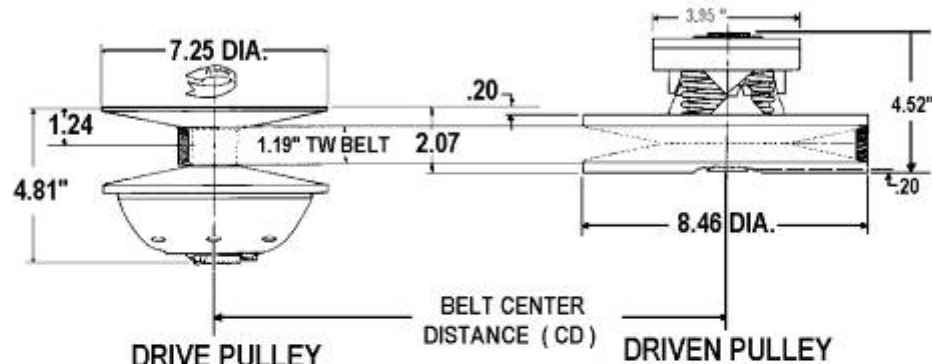
APPENDICES

APPENDIX A
COMET MODEL 790 CVT



MODEL 790 AUTOMATIC TORQUE CONVERTER

LAYOUT DIMENSIONS:



SPECIFICATIONS

MAX ENGINE RATING: MAX RPM:
2-Cycle - 30 H.P. 10,000
4-Cycle - 16 H.P. 5,500

BORE SIZES SPEC. NO.
DRIVE - 1" 302424C
DRIVEN - 3/4" 302603C

PULLEY RATIOS:
LOW - 3.38:1 (SHOWN ABOVE)
HIGH - .54:1
OVERALL - 6.26:1

BELT SELECTION:

PART NUMBER	BELT C.D.
300780C (704054)	9.41"
300634C (704055)	10.41"
300637C (704060)	11.50"
300638C (704061)	11.84"

DISCOUNTED PRICES APPLY TO
MIN-BAJA PROJECT ONLY.

	LIST	NET*
DRIVES :	\$253.00	\$ 152.00
DRIVENS:	250.00	150.00
BELTS :	68.00	41.00

*SHIPPING / TAXES: "FOB QDS" OR "FRT. COLLECT" / ADD 8.25 PERCENT TO
PURCHASES IN CALIFORNIA ONLY.

NOTE: NEVER OPERATE A TORQUE CONVERTER WITHOUT A SUITABLE SAFETY SHIELD.

NOTE: DO NOT RUN AN ENGINE EQUIPPED WITH A DRIVE PULLEY IF THE BELT IS NOT
ENGAGED WITH THE DRIVEN PULLEY.

DISTRIBUTED BY:

QDS P.O. BOX 6910, ALHAMBRA, CALIFORNIA 91802 TEL. (626) 293-5770 / FAX. (626) 281-3392

APPENDIX B

LINEAR ACTUATOR FA-PO-150-12-2"

PRODUCT INFORMATION



Firgelli Automations, 3888 Sound Way, Bellingham, WA 98227 USA

Tel, 604-542-8945

Fax 1-866-226-1649

Email sales@firgelliauto.com

www.FirgelliAuto.com

- Low noise design
- Enhanced corrosion resistance
- Aluminum outer and inner tube
- Zinc alloy housing
- Powder metal gears
- Lubrication for longer life
- Small compact Design
- Low Price

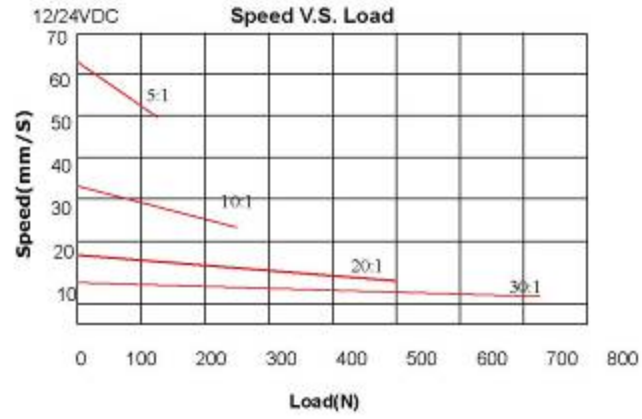


Specifications

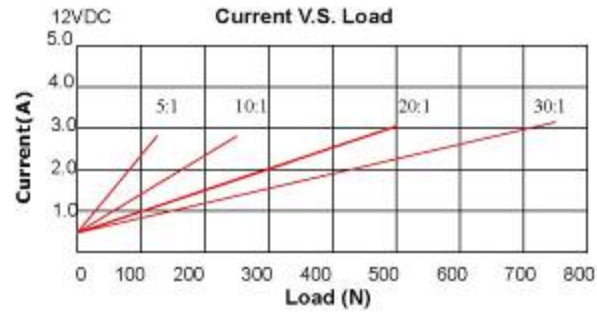
Model	FA-PO-20-12-xx" (High Speed)	FA-PO-100-12-xx" (Standard Force)
Input Voltage	12 VDC	
Load Capacity	20 lbs	100lbs
Static Load	2 x max load capacity	
Stroke Length	2" to 12"	
Speed at No Load	2"/sec (50mm/s)	1"/sec (12mm/sec)
Feedback	10K ohm 3 wires Potentiometer	
Clevis Ends	6.3mm diameter	
Screw	ACME screw	
Gear Ratio	5:1	20:1
Duty Cycle	20%	
Operation Temperature Range	-26°C-65°C (-15°F-150°F)	
Limit Switch	Built-in (Factory Preset) Not movable	
IP Grade	IP54 (dust and splash proof)	

PRODUCT INFORMATION

LOAD/SPEED

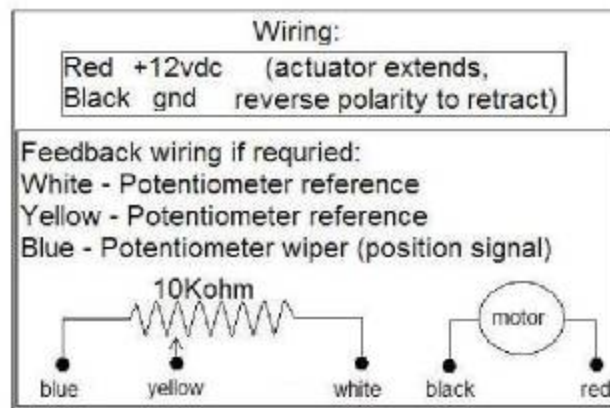
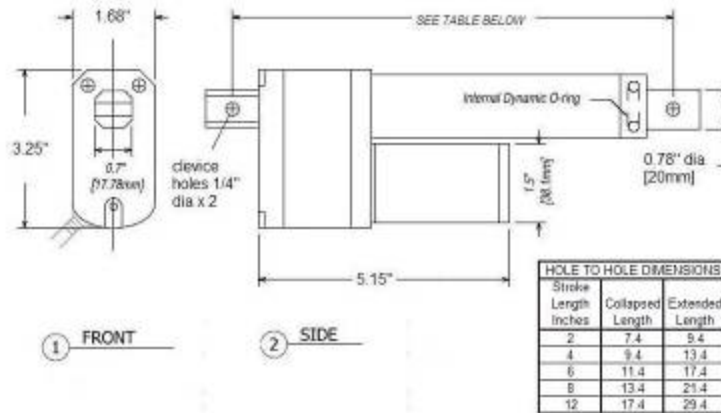


LOAD/CURRENT



PRODUCT INFORMATION

Dimensions:



- This model actuator uses Firgelli Automations MB1 Brackets on both ends of the actuator.
- Visit http://www.firgelli.com/product_info.php?cPath=70&products_id=54 for more details

Please visit our website for latest prices and to purchase
www.FirgelliAuto.com

APPENDIX C

TAZER 15T FORWARD/REVERSE ECS



Instruction Manual

Thank you for choosing the Dynamite brand. We know that you, the customer, see the reason that we are in business. Your satisfaction is our number one priority. With this in mind, we have produced this product to be of the highest quality performance and reliability, and at a "value-packed" price. We hope it provides you with years of enjoyment in your radio/R/C project.

Features

- High power PPT control with precision forward and reverse.
- High frequency design did not smear speed transmission.
- Thermal Overload Protection. Prevents damage due to overcurrent conditions that automatically resets.
- Pre-wired with Targa battery plug, bullet style motor connectors, and universal receiver plug that fit JR, Futaba 2 and Hi-Tec radios.
- Designed to operate with stock motors (15 turn or higher).
- Push button programming makes set-up a breeze.

Specifications

Operation	Proportional forward, proportional reverse with braking delay
Input Voltage	4-cell (4.8 volts) to 7-cell (8.4 volts) DC
Peak Current	110 amperes
Continuous Current	110 amperes
Full-On Resistance	0.006 ohms
Frequency	2 MHz
BEET output	4.8 VDC, 1 amp max
Overload Protection	Thermal
Dimensions	1.5" x 1.5" x 1.8" (40 x 40 x 27mm)
Weight	1.80 oz (50g)

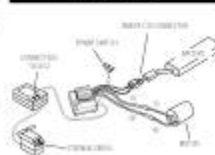
Mounting the Speed Control

Mount the Tazer 1ST ESC in the location specified by your vehicle's instructions. Use the double-sided foam tape (included) to secure the speed control in position.

NOTE: Be sure all wiring connections can be reached prior to mounting.

Use double-sided foam tape to secure the side of the switch to a convenient location on the chassis or shock tower.

Wiring Diagram



Wiring the Receiver

NOTE: The Tazer 1ST uses the motor battery to supply power to the receiver through the servo wires. There is no need for a separate receiver battery.

- See your radio's instruction manual for proper connection. Typically, channel 2 is used to control the throttle.
- There are 3 wires included in the universal receiver connector. It is directly compatible with JR, Hitec, Aerobics 2, Hitec RC and Futaba 2 systems.
- Older Aerobics or RC style systems must use a modified wiring order.
- The positive (red) and negative (brown) wires must be reversed to operate with most radio systems. To reverse the wires, turn the plug on a servo's potentiometer to flip up the pin on the bottom wire out of the plug. Repeat on the red wire and replace in the opposite position.
- Install the plug, referencing the wire colors on the steering servo for proper polarity.

Motor Capacitors

If the motor you are using (15 turn or more) does not have a capacitor already attached, you must install them. The two other capacitors help prevent motor noise which becomes radio glitches. Some motors have internal capacitors that are not suitable without removing the motor. Check the motor's instructions to confirm if capacitors are present in your motor. If your motor does not have capacitors, they can be purchased at your local hobby shop. Follow the directions below for proper connection.

- Using a small, flat, stiff pin or wire on the motor can be used to connect the positive and negative tabs. (See figure 2.)
- Take one of the capacitors and solder one wire to the positive motor tab with the other wire positioned near the soldered area on the motor housing. (See figure 3.)
- Use another capacitor and solder one wire to the negative motor tab with the other wire positioned near the soldered area on the motor housing. (See figure 3.)

4. Solder both wires to the motor housing. (See figure 4.) A high voltage soldering iron may be necessary.

5. Solder the third capacitor from the positive to the negative motor tab. (See figure 5.)

6. Use side cutters to remove excess capacitor wires.

Some forward only ESCs require a diode (diode attached to the motor) for motor suppression. Be certain there is a diode attached to the motor when using the Tazer 1ST or any other reversing electronic speed control.

Connecting the Motor Wires

If your motor has pre-attached radio lead connections, press the servo motor wire into the plug that is wired to the positive motor terminal.

The black wire is connected to the servo motor in the negative motor tab (see figure 6).

If your motor does not have radio lead connections, you may purchase them from your local hobby shop. "Motor Connector Wire (2, 14, 30)".

Connecting the Battery

The Tazer 1ST comes pre-wired with a Targa style connector, compatible with most battery packs. (See battery packs from 4-cell (4.8 volt) to 7-cell (8.4 volt) as a battery pack.)

- Re-charge the switch is in the "off" position.
- Connect a fully charged battery pack to the speed control's battery connector.

Adjusting the Transmitter

NOTE: Refer to the radio instructions for specific information on transmitter set-up.

- Set the "throttle reverse" switch to the FORWARD position. (This may need to be reversed on some brands of radios.)
- Set the "throttle trim" in the CENTER position.
- Set the "throttle exponential" to 100%.
- Set the "REV" to 100%.
- If your transmitter has an adjustable trigger (or stick) position, move it to the THROTTLE position.

Speed Control Programming

NOTE: While in the programming mode, no power is applied to the motor.

- Turn on the transmitter's power switch. (Be sure the transmitter batteries are fully charged.)
- Turn the ESC switch on.
- Press and release the set-up button with the included programming tool. The red and green LEDs will light.
- Move the throttle to the full throttle position and press the programming button. The green LED will remain lit and the red LED will go out. If the ESC does not seek throttle movement in 2 seconds by the transmitter, it will exit the programming mode and you will have to begin again.
- Now move the throttle to full reverse and press the programming button. The green LED will glow and the red LED will go out, indicating programming is complete.
- Return the throttle stick to neutral and press the programming button. The green LED will glow and the red LED will go out, indicating programming is complete.

During normal operation, the green LED indicates forward and the red LED indicates full forward and full reverse.

Troubleshooting Guide

Symptom	Solution
Beeping servo, speaker, but the motor does not run	Programming is not complete. Re-program the ESC by following the programming instructions. Speed control connected to receiver incorrectly. Refer to manufacturer's instructions. Motor defective. Test motor independently, repair or replace as needed. Low batteries. Charge as needed. Overload Protection engaged. Check motor and connections.
Beeping and motor do not function	Receiver wired incorrectly. Check polarity and connection of control plugs. Radio interference. Set radio to function channel. Batteries discharged. Recharge or replace.
Full speed not attainable	Transmitter adjusted incorrectly. See radio instructions for proper adjustment. ESC programmed incorrectly. Re-program.
Motor slows but will not stop	Transmitter may be set improperly. See radio R/C manual. ESC program does not match manufacturer's programming ESC.
Reduced radio range / interference	Radio capacitor broken, leaking. Repair or replace. Radio motor. Move receiver further away from ESC, radio and wiring. Transmitter defective. Re-program batteries. Interference from modified or in use radio frequency. Reconnect or change radio to channel (see manufacturer's instructions).

Warranty Information

Dynamite warrants this product to be free from defects in materials and workmanship from the original date of purchase. The Tazer 1ST will be repaired or replaced (at no charge for the first 365 days upon inspection by the Dynamite Service Center. This warranty is limited to the original purchaser of this electronic speed control and is not transferable. This warranty will not cover ESCs that have been modified, misused or tampered with or used for other than its intended purpose. Equipment repaired or replaced after the first 365 days will require a payment of \$5.00 to cover shipping and handling for warranty repair or replacement.

The Dynamite Tazer 1ST does not come disassembled, assembled or installed damage under any circumstances. However, we will void your warranty on parts that are not listed in:

- Reverse polarity damage caused by connecting the battery pack backward, plugging battery into the motor connector wires, etc.
- Abuse or removal of the battery plug or connectors.
- Flipped or double wires due to shock or poor construction.
- Use of less than 4-cell (4.8 volt) or more than 7-cell (8.4 volt) battery packs.
- Damage due to tampering with any of the electronic components.
- Damage caused by water, moisture or foreign objects (e.g., oil, dirt, etc.).

If you have any questions please call us toll free at 1-877-834-0222.

If you Tazer 1ST is in need of inspection or repair, please also fill out packet to:

Dynamite Service Center
48N Dynamite Drive
4148 Fiddlers Road
Champaign, IL 61812-0222

Please include a completed Service and Repair Card. Checked from our website at www.dynamitehobby.com, or include the following information with your Tazer 1ST:

- Full Name
- Mailing address
- Daytime phone number
- Original Purchase Receipt
- Date, location and description of products included in the package
- Description of product's construction
- Payment information including Credit Card number, cardholder name as it appears on the card, and expiration date

Non-Warranty Repairs

Check if your repairs cost more than 50% of the retail price you paid, you will be provided with an estimate showing the cost of repairs. Money orders will be accepted for the amount of the repairs, repairs are associated shipping and handling.

Web Product Information by Dynamite Hobby Inc.
www.dynamitehobby.com
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APPENDIX D

DI-148U DATA ACQUISITION SYSTEM

DI-148U Starter Kit



Low Cost, Compact Data Acquisition Kit

Convenient USB Interface

8 ± 10 V Analog Single-Ended Inputs

Six Bidirectional TTL Ports for General Purpose Control

10 Bit Resolution

Up to 14,400 Hz Sample Rate



The DI-148U breaks new ground in price and performance, offering advanced features usually reserved for more expensive instruments. They include, but are not limited to, a channel scan list, high sample rate throughput, and an advanced computer interface. These features combine to produce a robust instrument that can be applied to nearly any data acquisition situation where pre-amplified signals need to be acquired to a PC.

This extremely inexpensive instrument offers eight single-ended analog input channels (each with a fixed ± 10 volt full scale range), analog-to-digital conversion resolution of 10 bits (allowing a minimum voltage sensitivity of ± 19.5 mV), six bidirectional TTL ports that may be used for general purpose control, and a maximum sample throughput rate of up to 14,400 Hz. No other instrument offers so much for so little.

Features

Easy to Connect and Use

Connect the DI-148U to any local laptop or desktop PC. Power is derived from the PC through the USB interface so no external power is required. Two, built-in, 8 position screw terminal connectors allow easy and secure access to all signal I/O connections without the need for extra options.

Wide Sample Throughput Range

Throughput ranges from sub-Hertz to over 14,400 Hertz allow the DI-148U to connect to a wide range of both static and dynamic signals.

Compact

Small size—66D \times 66W \times 28H mm (2.6D \times 2.6W \times 1.1H inches)—allows the DI-148 to fit comfortably in crowded instrumentation cabinets, desktops, and other tight locations.

Self Powered Advantage

All DI-148 instruments derive their power directly from the host PC eliminating the need for an external power adaptor and connections—perfect for use in automotive and other portable environments where power is unavailable.

Built-In, Bidirectional Port

A built-in bidirectional port allows programmable discrete inputs and outputs for control.

Free Data Acquisition Software

Our WINDAQ/Lite data acquisition software offers real time display and disk streaming for the Windows environment. Their real time display can operate in a smooth scroll or triggered sweep mode of operation, and can be scaled into any unit of measure. Event markers with comments allow you to annotate your data acquisition session with descriptive information as you're recording to disk.

Raise your productivity to new heights with WINDAQ's unique multitasking feature. Record waveform data to disk in the background while running any combination of programs in the foreground—even WINDAQ Playback software to review and analyze the waveform data as it's being stored! WINDAQ/Lite recording and playback software is provided free with every DI-148U purchase. WINDAQ/Lite recording software is limited to 240 Hz sample rate when recording to disk. The extra cost WINDAQ/Pro High Speed option allows you to record at rates up to the speed of the instrument.

DATAQ Instruments, Inc. • 241 Springside Drive • Akron, Ohio 44333 • Tel: 330-668-1444 • Email: support@dataq.com • www.dataq.com

Specifications

Analog Inputs

Number of Channels:	8
Channel Configuration:	Single-Ended
Measurement range:	$\pm 10\text{V}$
Accuracy:	$\pm 0.25\%$ of FSR
Resolution:	$\pm 19.5\text{mV}$
Input Impedance:	$200\text{K}\Omega$
Input bias current:	$50\text{ }\mu\text{A}$ for $\pm 10\text{V}$ input, single channel
Max. normal mode voltage:	20 V pk to pk
Channel-to-channel crosstalk rejection:	-60db
Gain temperature coefficient:	$100\text{ppm}/^\circ\text{C}$
Offset temperature coefficient:	$5\text{ }\mu\text{V}/^\circ\text{C}$
Digital filtering:	Over-sampling average

A/D Characteristics

Type:	Successive approximation
Resolution:	10-bit
Monotonicity:	$\pm 2\text{LSB}$
Conversion Time:	$70\text{ }\mu\text{s}$

Scanning Characteristics

Max. throughput sample rate:	14,400
Min. throughput sample rate:	0.0137534 Hz
Timing accuracy:	100 ppm of sample rate
Max. scan list size:	10 entries
Sample buffer size:	256

Digital I/O

Channels:	4 bidirectional ports
Output voltage levels:	Min. "1" 3V @ 2.5mA sourcing Max. "0" 0.4V @ 2.5mA sinking
Output current:	Max. source -2.5mA Max. sink 2.5mA
Input voltage levels:	Min. required "1" 2V Max. allowed "0" 0.8V

Calibration

Calibration cycle:	One year
Calibration method:	Digital calibration with scale and offset constant

General

Input connectors:	Two eight position terminal blocks
Operating Environment:	0°C to 70°C
Enclosure:	Molded ABS plastic
Dimensions:	$2.6\text{L} \times 2.0\text{W} \times 1.1\text{D}$ inches $66\text{W} \times 50\text{W} \times 28\text{D}$ mm.
Weight:	3 oz. (85 gr.)
Power Requirements:	80mA max. @ 5 VDC . No external power required. Power derived from communications cable.

DI-148 Analog Inputs (Typical)

Input Impedance > $200\text{K}\Omega$

241 Springville Drive
Akron, Ohio 44333
Phone: 330-448-1444
Fax: 330-444-5434
www.dataq.com

Ordering Guide

Description	Order Number
DI-148U Starter Kit 8-channel data acquisition module with USB interface.	DI-148U

Data Acquisition Product Links

(click on text to jump to page)

[Data Acquisition](#) | [Data Logger](#) | [Chart Recorder](#) | [Thermocouple](#) | [Oscilloscope](#)

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APPENDIX E

BRIGGS & STRATTON MODEL 20 ENGINE

RACING ENGINES: MODEL 20

For over 20 years, Briggs & Stratton has worked with the Society of Automotive Engineers (S.A.E.) in providing power for their Mini Baja and Supermileage programs. Starting for the 2000 season the S.A.E. Mini Baja Program began using the Model 20 INTEK as their engine of choice, replace the Model 19 after 20 plus years as the workhorse...

BRIGGS & STRATTON
MODEL 20



ENGINE SPECIFICS

[Illustrated Parts List](#) 

[Outline Drawings/Dimensions](#) 

[Price guide](#) 

General Specs

Technical/Torque Specs

- Model/Type: I/C - 202400
- Displacement: 18.6 cu. in. (305 cc)
- Cylinder: Slant, 30 degree
- Valve Design: Overhead Valve Design
- Bearings: DU Magneto DU PTO (ball bearing optional)
- Bore Type: Dura-Bore™ Cast Iron Sleeve
- Bore: 3.120 (79.25 mm)
- Stroke: 2.438 in. (61.67 mm)
- Compression Ratio: 8.0 to 1
- Oil capacity: 26 fl. Oz. (.7 liters)

APPENDIX F

EXPERIMENT TEST PROCEDURE

List of equipment:

1. Custom inertia dynamotor “fly wheel”
2. 10 hp Briggs & Stratton engine model 20
3. DI-148U data acquisition unit
4. TAZER 15T forward/reverse ECS
5. Linear actuator FA-PO-150-12-2”
6. Comet model 790 CVT
7. Laptop computer with software for DI-148U data acquisition unit
8. 12 volt battery

Test setup:

See Figure 16. “Computer Controlled Actuated CVT Experiment Diagram” on page 16

Procedure:

1. Initiate data collection of data acquisition software.
2. Apply brake to fly wheel such that its angular velocity is zero.
3. Simultaneously release fly wheel brake and apply full throttle to the 10 hp engine.
4. Allow the system to run until the fly wheel has achieves max angular velocity indicated by a lack of acceleration.
5. Simultaneously release the engine throttle and apply the fly wheel brake until the fly wheel angular velocity is zero.
6. End data collection of the data acquisition software.