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## **SWITCH-MODE CONTINUOUSLY VARIABLE TRANSMISSION WITH FLYWHEEL ENERGY STORAGE**

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### **ABSTRACT**

A hybrid drive train significantly improves energy efficiency of ground vehicles. While numerous auxiliary hybrid power sources have been researched, few are capable of the energy and power density of a flywheel coupled with a continuously variable mechanical transmission. The primary challenge of a flywheel hybrid system is a transmission capable of coupling a high speed flywheel to the drive train of the vehicle. A novel solution to this challenge is a switch-mode continuously variable transmission that utilizes a rapidly switching clutch to transmit power. This system, the mechanical analog of a DC-DC boost converter circuit, utilizes a flywheel, a high frequency clutch, an anti-reversing ratchet, and a spring to vary the output torque. The switch-mode continuously variable transmission is demonstrated through an idealized finite difference model, created from the dynamic system of equations. The model is used to demonstrate the system behavior in a passenger car subjected to road loads in various conditions. The output of the model demonstrates pulses in the output torque as a result of the rapidly switching clutch. This output ripple is smoothed to an acceptable level by the torsion spring. From this preliminary analysis the on-off continuously variable transmission offers an efficient, energy dense, and power dense hybrid vehicle drive train alternative.

### **INTRODUCTION**

The increase in global energy consumption combined with a decreasing supply of petroleum elevates the importance of improving the energy efficiency of all products. A major component of global energy consumption is transportation, which consumes 4.8 billion barrels of crude oil per year. Of the transportation industry, passenger cars consume 2 billion barrels of oil per year with a value of over \$200 billion [1].

This paper focuses on a hybrid vehicle drive train consisting of an on-off continuously variable transmission with flywheel energy storage, which enables a significant improvement in the efficiency of ground transportation vehicles.

A hybrid vehicle improves fuel efficiency by leveraging the ability to store energy in a form other than liquid fuel. The drive train of a hybrid vehicle contains two sources of power, typically an internal combustion (IC) engine that cannot recover energy and a second power source that is capable of recovering energy. Hybrid vehicles improve economy in three primary ways. 1) When a vehicle is stopped, the internal combustion engine is switched off to eliminate idling losses. 2) During braking events, such as stopping for a traffic signal or descending a grade, the kinetic energy otherwise dissipated in the friction brakes is stored in the energy storage device. This stored energy is then used to re-accelerate the vehicle. 3) The engine can be aggressively managed to only operate at or near the most efficient operating condition, which typically occurs at high power levels for internal combustion engines. Because this power level is often higher than the required power at the wheels, a portion of this power is used to propel the vehicle and the additional power is stored in the energy storage device. Once the energy storage device reaches a high level of charge, the engine is shut off and the vehicle is propelled by using energy from the energy storage device.

A hybrid vehicle with an auxiliary power source capable of storing large amounts of energy and absorbing or delivering the energy at high levels of power enables further improvements in the economy of a vehicle. The maximum average power requirements of a passenger vehicle occur during high-speed highway driving or when climbing a grade.

For a 4 door passenger car, these maximum average power requirements rarely exceed 25 kW (34 HP). In a conventional drive train, most passenger cars have engines that produce near 150 kW (200 HP) of power, which is fully utilized only during short-term rapid acceleration. This maximum acceleration is often driven by the consumer demand and not by the acceleration required for daily driving. A hybrid drive train with high power and energy capabilities allows the engine to be sized to meet the maximum average power requirements, while the auxiliary power source is used to achieve the maximum transient power requirements. A high power auxiliary power source allows decreasing the IC engine size, which decreases vehicle weight, reduces emissions, and reduces fuel consumption, while still enabling the rapid acceleration desired by consumers.

A wide variety of auxiliary hybrid power sources and energy storage devices have been previously explored including: batteries, hydraulic accumulators, flywheels, elastomeric springs, capacitors, and others. Three of these options will be discussed further: electric motor/generators combined with batteries for energy storage, hydraulic pump/motors combined with hydraulic accumulators, and a continuously variable mechanical transmission with a flywheel for energy storage.

Electric hybrid vehicles have been the first hybrid technology to be mass produced for the commercial passenger car market. A strength of electric hybrids is the high energy density of electric batteries, at 220 kJ/kg and 540 kJ/kg for nickel-metal hydride and lithium ion batteries respectively [2], allowing for a large quantity of energy to be stored in relatively compact and lightweight batteries. While electric hybrids have been successful, demonstrated by the increasing number of vehicles models with hybrid power trains, there are significant drawbacks to an electric hybrid. The most substantial shortcoming of the electric systems is the limited power density of both electric motor/generators and batteries at approximately 30-100 W/kg [3]. The low power density results in the electric motor/generators being underpowered and heavier than desired. Due to this limited power density, a large portion of the braking energy during an aggressive braking operation is dissipated through friction brakes instead of being stored in the drive train. Similarly, batteries cannot be charged and discharged quickly without shortening their usable life, limiting their power density and energy recovery rates.

Utilizing hydraulics for the auxiliary hybrid power source provides improvements in power density over electric systems at the expense of energy density. Hydraulic pump/motors and accumulators have a higher power density of approximately 500-1000 W/kg [3]. This power density allows the hydraulic pump/motors to be sized to capture nearly all of the braking power, even during aggressive braking events. The energy density of hydraulic systems is the primary limitation at

approximately 5 kJ/kg for modern composite accumulators. The developing technology of an open accumulator enables theoretical improvements in energy density of 20 times [4] over current accumulators.

Another auxiliary hybrid power source, the topic of this paper, is a flywheel coupled with a continuously variable mechanical transmission. This system combines the high energy density of the electric system with extremely high power density. The energy density of flywheel systems is approximately 325 kJ/kg [5] which is similar to the most advanced batteries. The power density of flywheel systems is extremely high because it is only limited by the torque capabilities of the mechanical components.

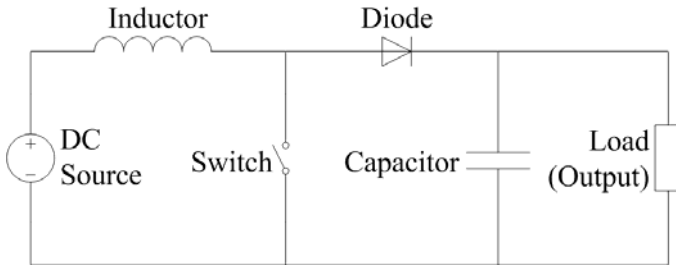
Using flywheel energy storage in a ground vehicle poses a few challenges. One issue with using a flywheel in ground vehicles is the gyroscopic force created when changing the pitch, yaw, or roll of the vehicle. This gyroscopic force can be canceled out by using two flywheels rotating in opposite directions or by mounting the flywheel in a gimbal. A second challenge, also applicable to any form of energy storage, is the hazard of releasing the energy in an undesirable way, as possible in a vehicle collision. This hazard must be mitigated by providing the appropriate safety shielding around the flywheel and designing the system to dissipate energy in a safe manner in the event of an accident.

One of the prime challenges that has hindered flywheel energy storage in vehicles is a transmission capable of coupling the high speed flywheel, typically rotating at 20,000 to 50,000 RPM, with the wheels. The transmission for this application must be efficient, continuously variable, and have a very large reduction ratio. Previous work in this field has proposed continuously variable transmissions using hydraulics, belt drives, and toroidal drives, with marginal success [6]. A novel solution to this issue is a switch-mode continuously variable transmission that uses rapid switching of a clutch to transmit power. A switch-mode transmission using two clutches and a spring connected to ground was previously described and tested by Gilbert et al. This system transmitted low levels of power through a limited range of 1:3.2 and 1.3:1 [7]. This design did not enable energy storage and relied on oscillating the spring connected to ground for operation.

As an enabling solution to a flywheel hybrid vehicle, a novel switched-mode continuously variable transmission is presented in the following section. This transmission uses a single clutch, a flywheel, and a spring to provide a digitally controlled continuously variable transmission with significant energy storage capability.

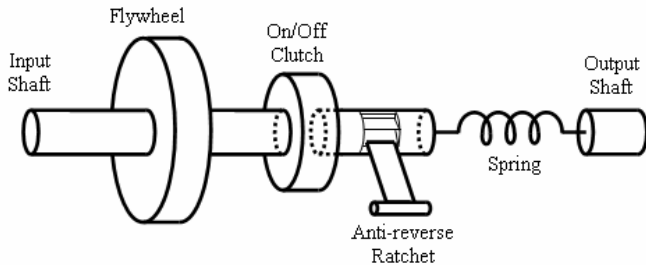
## APPROACH

The switch-mode pulse width modulated continuously variable transmission is a mechanical equivalent of a DC boost circuit. The boost converter circuit, shown in Fig. 1, uses two energy storage devices, an inductor and a capacitor, a high speed on-off switch, and a diode to produce an output voltage that is larger than the input voltage. The switch, typically a transistor, is rapidly switched between the on and off states. By varying the duty cycle, defined as the proportion of the time the switch is in the on-state versus the time of a switching period, the average output voltage is controlled.



**Fig. 1:** A boost power converter increases the voltage from the input source to the output by controlling the duty ratio of the switch. The high-speed switch, typically a transistor, switches the circuit between two distinct modes. When the switch is closed the inductor charges and the load is driven by the capacitor. When the switch opens, current continues to flow through the inductor, through the diode, and charges the capacitor.

The switch-mode on-off CVT is formulated by replacing the electrical components of the boost converter circuit with mechanical components. The inductor and capacitor or the energy storage devices are replaced with a flywheel and torsion spring. The transistor is replaced with a high speed clutch and the diode is replaced with an anti-reversing ratchet. The mechanical equivalent can be seen in Fig. 2.



**Fig. 2:** Mechanical Equivalent to DC Boost Converter

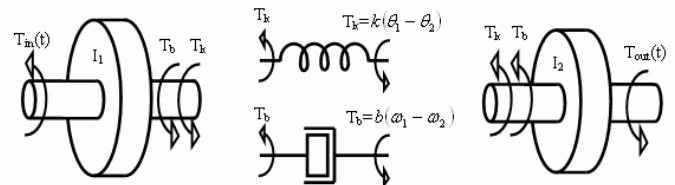
The operation of the switch-mode CVT contains two distinct states. During the off-state, the clutch is disengaged and any torque applied to the input shaft results in increasing the angular velocity of the flywheel. During this mode, the

ratchet prevents backward rotation of the input side of the rotational spring, while the output side of the spring applies a torque to the output shaft that is proportional to the angular deflection of the spring. If the system is left in the off-state, the angular deflection of the spring will reach a zero value and thus no longer apply torque to the output shaft. In this situation, the ratchet will allow the input side of the spring to freely rotate in the direction of the output shaft, preventing the application of a negative torque on the output shaft by the spring. When the clutch is engaged, the angular velocity of the input side of the spring becomes the same as the flywheel, which is assumed to be higher than the output shaft angular velocity, resulting in an increase in the angular deflection of the spring. While in the on-state, the angular deflection of the spring increases, which increases the torque applied to the output shaft. By rapidly switching between these two states the output torque is controlled by modulating the duty cycle.

One of the primary roles of the two energy storage devices is to act as filters. The inertia of the flywheel minimizes the amplitude of variation in the angular velocity of the input shaft as the clutch is engaged and disengaged. Similarly, the spring acts to smooth the torque ripple resulting from the pulsing engagement of the clutch. The amplitude of the torque ripple of the output shaft is a function of the clutch switching frequency and the rate of the spring, as discussed below.

## METHODS

To model the system, a finite difference model was created from the dynamic system of equations for the on and off states. To aid in developing the equations of motion, free-body-diagrams are created of the individual components, as shown in Fig. 3. For the free-body-diagrams the inertia of the flywheel and the clutch are combined and represented by  $I_1$ . A further simplification is made by combining the any inertia associated with the output shaft into a second flywheel with inertia  $I_3$ . For this preliminary model, the output of the clutch is assumed to have negligible inertia and the rotational spring is modeled as a perfect spring and a rotational damper.



**Fig. 3:** Simplified free-body-diagrams of the components of the pulse width modulated continuously variable transmission.

When the clutch is engaged, the system is in the on-state. During this state, the system can be represented by the following dynamic equations:

$$I_1\alpha_1 - b(\omega_2 - \omega_3) - k(\theta_2 - \theta_3) = T_{in}(t) \quad (1)$$

$$I_3\alpha_3 + b(\omega_2 - \omega_3) + k(\theta_2 - \theta_3) = -T_{out}(t) \quad (2)$$

$$\omega_1 = \omega_2 \quad (3)$$

where  $\theta_1$ ,  $\theta_2$ , and  $\theta_3$  are the angular position of the input shaft, input side of the spring, and the output shaft respectively,  $b$  is the damping rate of the spring,  $k$  is the spring rate, and  $T_{in}$  and  $T_{out}$  are the input and output torques respectively. The angular velocity and angular acceleration are expressed by  $\omega$  and  $\alpha$  respectively. By collecting terms, Equations 1 and 2 can be expressed as:

$$I_1\alpha_1 - b\omega_2 - k\theta_2 + b\omega_3 + k\theta_3 = T_{in}(t) \quad (4)$$

$$b\dot{\theta}_2 + k\theta_2 + I_3\ddot{\theta}_3 - b\dot{\theta}_3 - k\theta_3 = -T_{out}(t) \quad (5)$$

When the clutch is disengaged, the system is in the off-state. During this state, the spring is disengaged from the flywheel and the left side of the spring is held stationary by the ratchet. In this state, the dynamic system equations simplify to:

$$I_1\alpha_1 = T_{in}(t) \quad (6)$$

$$b\omega_2 + k\theta_2 + I_3\alpha_3 - b\omega_3 - k\theta_3 = -T_{out}(t) \quad (7)$$

$$\omega_2 = 0 \quad (8)$$

In the case that the stored energy in the spring is zero, which occurs when  $\theta_2 - \theta_3 \leq 0$ , the ratchet allows the input side of the spring,  $\theta_2$ , to freely rotate. For this case, Equations 7 and 8 change to:

$$I_3\alpha_3 = -T_{out} \quad (9)$$

$$\theta_2 = \theta_3 \quad (10)$$

To develop the model around a real work situation, a standard passenger sedan was considered. A typical sedan weighs approximately 1500 kg, has a frontal area of approximately 2.16 m<sup>2</sup> and a coefficient of drag of 0.26. [8]. The output flywheel energy storage was calculated based on the kinetic energy of the sedan at 31.3 m/s (70mph) as per equation 11.

$$E_{kinetic,linear} = \frac{mv^2}{2} \quad (11)$$

where  $m$  is the mass of the vehicle and  $v$  is the velocity.

The moment of inertia of the output flywheel,  $I_3$ , is calculated by setting the kinetic energy of the vehicle, as defined in equation 11, equal to the kinetic energy of an equivalent flywheel, equation 12, with an angular velocity based on a 0.75m diameter tire.

$$E_{kinetic,rotational} = \frac{I_3\omega_3^2}{2} \quad (12)$$

The inertia of the input storage flywheel was sized to store enough energy to accelerate the vehicle to 31.3 m/s (70mph) assuming a flywheel angular velocity of 2100 radians per second (20,000 rpm) which is well with the capabilities of modern carbon fiber energy storage flywheels. [9]

The torsion spring constant was calculated based on the power output of 75 kW (100) at 4.5m/s (10 mph). At this power level it was desired to have no more than two spring wraps between the input and the output. The currently modeled torsion spring constant can be obtained through the use of a conventional torsion spring, a shaft in torsional loading or a mechanical assembly of commercially available compression springs. The torsion damping coefficient used is based on empirical data of cam follower systems, where 6% of critical damping is typical [10].

For this simplified analysis, the clutch is modeled as having no slip during engagement and disengagement and negligible inertia. The clutch is switched between the on and off states at a frequency of 20 Hz. This frequency was chosen to minimize the output torque ripple while still being near the capabilities of current clutch technology. The transmitted torque is varied by changing the duty ratio. Due to the large difference in angular velocity between the energy storage flywheel and the output shaft, the duty ratio is typically less than 2% for this system.

The output torque applied to the output shaft to propel the vehicle is modeled based on the tractive force for a passenger vehicle. The tractive force is comprised of four components, the aerodynamic drag, rolling resistance, road grade, and the inertial forces of acceleration. The inertial forces are accounted for in the equivalent inertia of the output flywheel. For this simplified analysis, the road will be assumed to be level, eliminating the road grade term. The aerodynamic drag is calculated from:

$$D_A = \frac{1}{2} \rho v^2 C_D A \quad (13)$$

where  $D_A$  is the aerodynamic drag,  $\rho$  is the mass density of the air,  $A$  is the frontal area, and  $C_D$  is the drag coefficient of the vehicle [11].

The rolling resistance of the vehicle is described by:

$$R_X = f_r W \quad (14)$$

where  $R_X$  is the rolling resistance,  $f_r$  is the rolling resistance coefficient, and  $W$  is the weight of the vehicle [11]. There are multiple mechanisms that contribute to the rolling resistance of the tires, requiring a precise rolling resistance model to include dynamics of the vehicle. As an approximate model of the rolling resistance coefficient of the tires on concrete, the following equation is used:

$$f_r = f_o + 3.24 f_s \left( \frac{v}{44.7} \right)^{2.5} \quad (15)$$

where  $f_o$  is the basic rolling resistance coefficient and  $f_s$  is the speed effect rolling resistance coefficient [11]. Note that the velocity is in units of meters per second in equation 15.

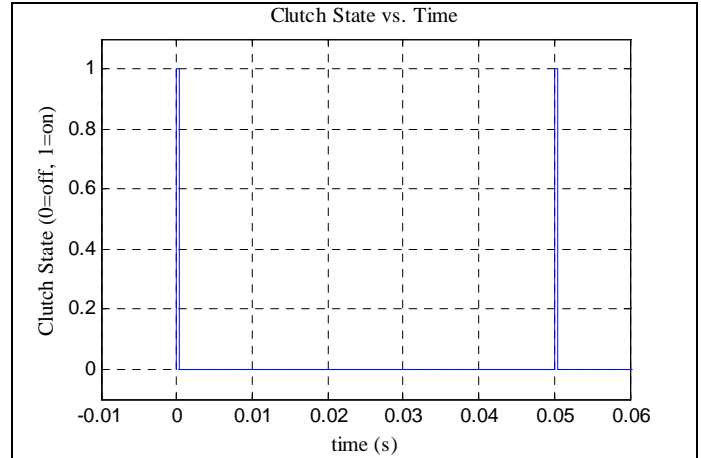
The model parameters used in the finite different model are summarized below in Table 1.

**Table 1: Input parameters for the simulation.**

Table of Model Parameters			
Variable	Description	Initial Value	Units
$\theta_1$	Angular Position of Shaft 1	0	rad.
$\theta_2$	Angular Position of Shaft 2	0	rad.
$\theta_3$	Angular Position of Shaft 3	0	rad.
$\omega_1$	Angular Velocity of Shaft 1	2100	rad./sec.
$\omega_2$	Angular Velocity of Shaft 2	0	rad./sec.
$\omega_3$	Angular Velocity of Shaft 3	0	rad./sec.
$I_1$	Moment of Inertia of Storage Flywheel	0.328	kg*m <sup>2</sup>
$I_3$	Moment of Inertia of Output Flywheel	211	kg*m <sup>2</sup>
$f$	Clutch Switching Frequency	20	Hz
duty	Duty Ratio of Clutch	0.9	% On
dt	Finite Difference Time Step	0.01	ms
$k$	Torsional Spring Rate	497	(N*m)/rad.
$b$	Torsional Damping Coefficient	0.5	(kg*m <sup>2</sup> )/sec.
$m$	Vehicle Mass	1500	kg
$A$	Vehicle Frontal Area	2.16	m <sup>2</sup>
$C_d$	Vehicle Coefficient of Drag	0.26	(unitless)
$d_{ia}$	Diameter of Vehical Tire	0.75	m

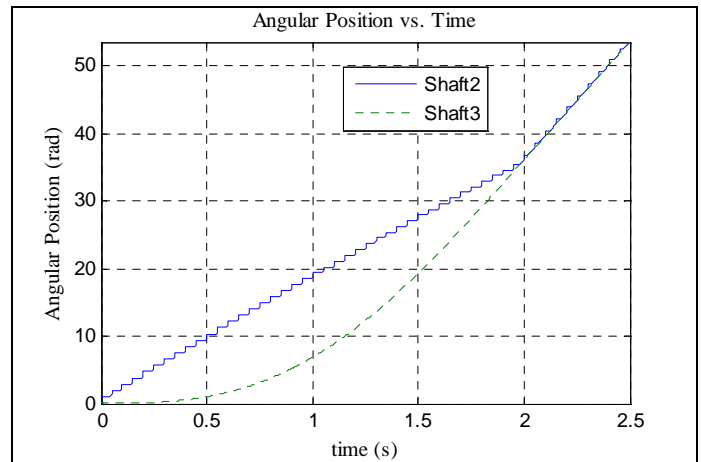
## RESULTS

The finite difference model was used to simulate a passenger automobile performing a maximum power accelerating from a stop to 13.4 m/s (30 mph). Using the equivalent output flywheel, this final velocity occurs at 35.8 rad/s. In the simulation, the flywheel initially stores enough energy to accelerate the car to 31 m/s, neglecting drag, and no additional power is added to the system by the engine. This simulation was run using a fixed duty ratio of 0.9% and a fixed time step of 0.01 ms. The switching behavior of the clutch is presented in Fig. 4.

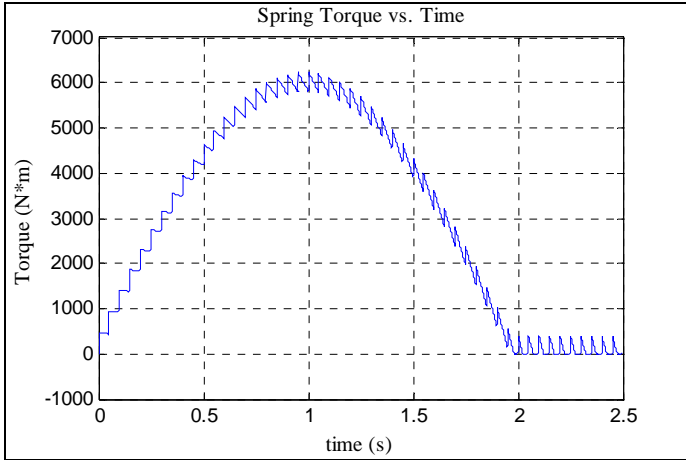


**Fig. 4:** Clutch state as a function of time. Note the short duty cycle of the system.

The relationship between the angular position of the two sides of the rotational spring, shown in Fig. 5, is important to the system behavior. The fixed frequency pulses of the clutch result in advancing the angular position of shaft 2 by momentarily engaging it with the storage flywheel. The angular position of shaft 3, the output shaft, lags behind that of shaft 2 as the spring winds up due to the effective inertia of the flywheel attached to shaft 3. The difference between the angular position of the two shafts determines the spring torque applied to the output shaft, as seen in Fig. 5. Note that there are additional torque pulses transmitted by the damping in the spring, yet this influence is fairly minimal.

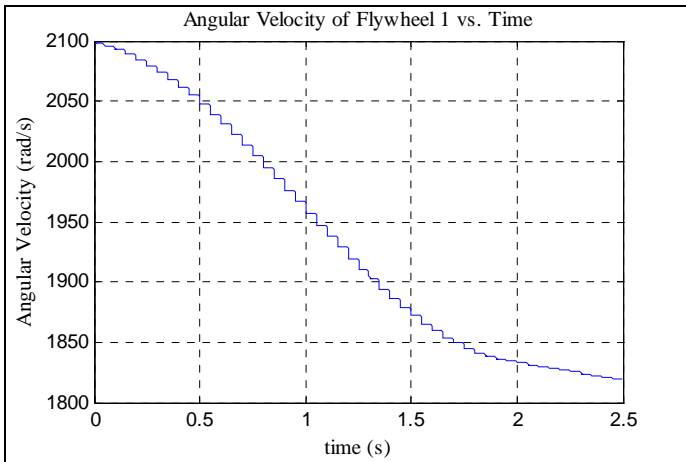


**Fig. 5:** Difference of input and output of spring as a function of time.

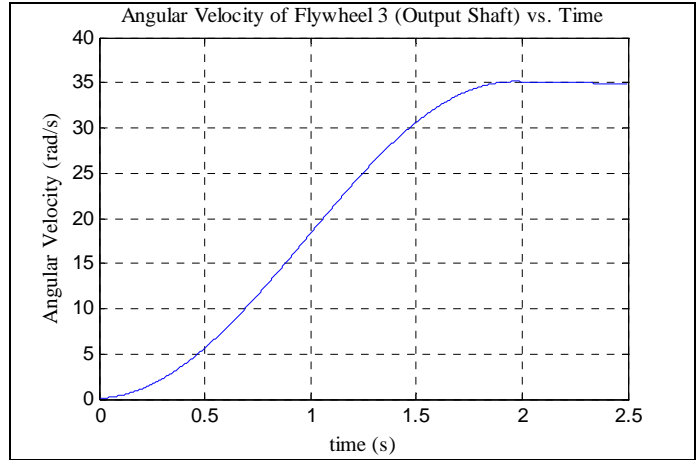


**Fig. 6:** Spring torque as a function of time.

The applied spring torque results in decreasing the angular velocity of the main flywheel, Fig. 7, and increasing the angular velocity of the output shaft, Fig. 8. Despite the torque pulsations due to the operation of the clutch, the output shaft velocity is relatively smooth due the inertia of the vehicle. Note that the continued decrease in the angular velocity of the main flywheel after a time of 2 seconds is due to energy dissipated through road loads and spring damping.

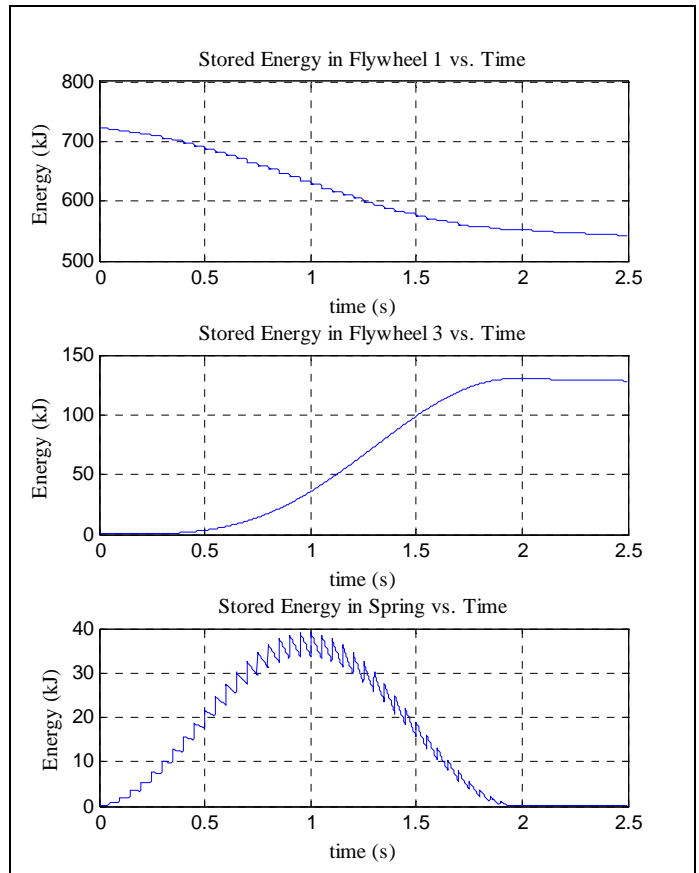


**Fig. 7:** Angular velocity of storage flywheel as a function time.



**Fig. 8:** Angular velocity of the output flywheel as a function of time.

Because there are multiple energy storage devices in the system, it is interesting to observe the flow of energy through the drive train. Fig. 9 presents the energy stored in the main flywheel, the energy stored in the output flywheel that is equivalent to the kinetic energy of the vehicle, and the energy stored in the spring.



**Fig. 9:** Energy stored in the input and output flywheels and spring as a function of time.

## DISCUSSION

Given a step input of a given duty ratio, the on-off CVT demonstrates a promising response. As shown in Fig. 5, the angular position of the output side of the spring lags the input until approximately 2 seconds. At this time, the angular position of the two sides of the two sides meet and the ratchet is allowed to freewheel for a period of time between pulses. Following the matching the angular position, angular velocity of the output shaft, Fig. 8, reaches a constant value and the additional energy supplied by the spring simply reacts the road load.

The simulation presented in this paper demonstrates how a high power auxiliary hybrid power source can meet transient power requirements. Throughout the simulation, no additional power was added from the primary power source. Instead, energy was extracted directly from the flywheel for the high power transient requirement. This enables the engine in a hybrid vehicle to be downsized to meet only the maximum average power requirement, saving weight, reducing fuel consumption, and improving emissions.

For the switch-mode CVT to operate in a hybrid vehicle, a method of storing regenerative power in the flywheel is required. During regenerative events, such as a braking for a traffic signal or descending a grade, the output torque transfer is reversed. For simplicity, the CVT architecture described above, utilizes a ratchet on the input side of this spring. The reversing torque during regeneration would not be reacted by the CVT as the ratchet would allow the input side of the spring to freewheel. To allow regeneration, a brake needs to be added to the system to replace the ratchet. During regeneration, this brake is engaged and the spring begins to deflect, applying braking torque to the output shaft. The energy stored in the spring is transferred to the flywheel by disengaging the brake and engaging the clutch simultaneously. The input side of the spring will rotate at the same angular velocity as the flywheel and apply torque to the flywheel. The brake can then be re-applied and the clutch released to apply more braking torque to the output shaft.

The above simulation requires very fast clutch operation. At a switching frequency of 20 Hz and a duty ratio of 0.9%, the clutch is only on for 0.45 milliseconds at a time. This requirement is well beyond conventional clutch designs and would require a carefully designed clutch. Another option is to include a speed reduction gear box in the system. The gear box could be placed between the flywheel and the clutch creating: a decrease in the difference in shaft velocities between the input and output of the clutch, a decrease in the energy loss in the clutch due to slippage, an increase in the potential duty cycle of the clutch, and a decrease in the lag of the system with an increased effective spring rate.

Ongoing research is exploring the sources of energy loss in the system including clutch slippage, spring damping, bearing losses, and others. The clutch engagement and disengagement profiles are currently being characterized as a function of clutch type, inertia bias and relative shaft velocities. These parameters will be integrated into the model to more accurately determine the clutch torque and energy losses. Current research is also examining the damping and physical design of the torsion spring. This work will allow for a more detailed understanding of the spring energy lost due to internal spring damping. This work will allow the bearing loads to be determined and thus bearing losses can be calculated. Finally, detailed flywheel sizing calculations will allow vacuum losses for the flywheel to be estimated.

One unique feature of this system is the output torque pulsations. This torque ripple is quite pronounced, as seen in Figure 6, yet does not cause large ripples in the output velocity. The prime reason that the output velocity does exhibit larger ripples is due to the large effective inertia of the vehicle. The magnitude of the torque ripple can be modified primarily through changes in switching frequency and spring rate. This is an area of future work to determine a level of torque ripple that is acceptable to the operator. Unlike a conventional automobile piston engine there are no high speed reciprocating masses and all components of the system can be designed such that they are dynamically balanced.

The presented results are the open loop response of the system to a fixed duty ratio. In future implementations, adding a control loop to the system will enable velocity or torque tracking based on the desired response. This “drive by wire” system will be able to closely follow the operator’s input through control of the clutch duty ratio. A second feedback control loop can be used to monitor the energy storage level of the entire system to determine when the engine should be operated.

## CONCLUSION

In this paper a novel switch-mode continuously variable transmission is presented. This drive train enables an energy and power dense flywheel hybrid vehicle by creating an efficient coupling between the flywheel and the wheels. By using the energy stored in the flywheel for high power transient requirements, the engine can be greatly downsized, decreasing weight, increasing the energy conversion efficiency of the engine, and decrease the emissions. Numerous areas exist for further research and development of this promising technology.

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