P08310
Polaris Electro-Mechanically Shifting Manual Transmission

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Senior Design I

Technical Review

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Overview

The purpose of this design is to design, select, build and evaluate a dynamic shifting system for a Polaris Outlaw 525 All Terrain Vehicle. This will include a scheme to enable rapid shift mechanisms and electronics (driver, microcontroller and code).

Project Background

Polaris Industries developed the Outlaw 525 ATV as a high performance vehicle designed specifically for racing applications. The main engineering design tasks for this model were to reduce weight and increase power for maximum performance.

By pairing with KTM-Sportmotorcycle AG, Polaris designed the ATV to be fitted with a KTM 525 power plant. The finished product was released to the public in the beginning of 2007.

This design group, with direction from Polaris Industries, has begun to pursue a more advanced technological means to operate the ATV. It is seen as a disadvantage to manually shift the ATV during high speed racing applications. It is hypothesized that a more rapid shifting mechanism could be applied to speed the shifting process and provide an even faster ATV.

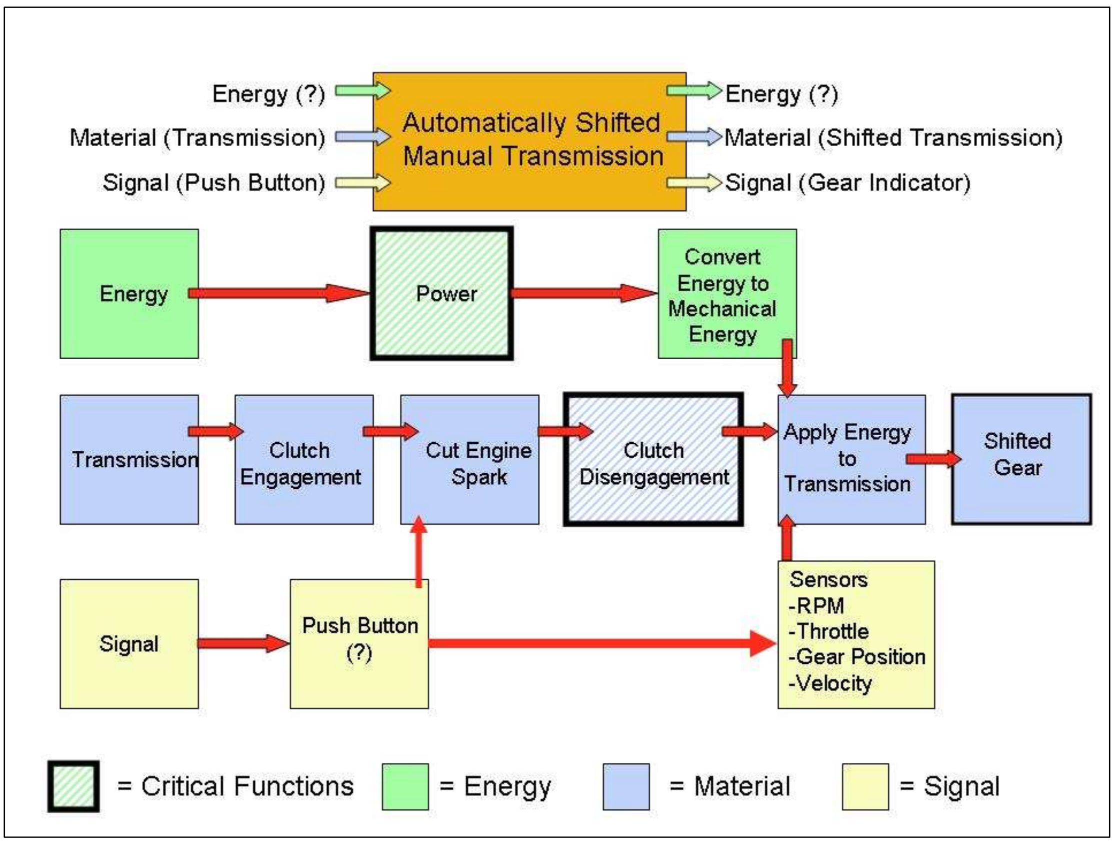


Fig 1: Functional Diagram

Mechanical Design

Excused Concepts:

The complexity involved in moving the shifting mechanism provided many challenges to the P08310 team. After a concept review and an analysis of individual abilities and knowledge regarding certain systems it was decided that many of our concepts would be difficult to accomplish in the necessary time.

A. Linear Actuation of the shift forks through the use of linear actuators and screw slides to move the individual forks was ruled to be infeasible with regards to the time requirements (<0.1 sec) for each individual shift. The complexity in the application of a system of this nature also posed a constraint.
B. The use of a stepper motor to control the shift shaft of the transmission was decided to have far too much weight and posed a size limitation given the working area.
C. The use of a customized hydraulic setup was contemplated to move each shift fork internally on the transmission. This would have required three custom hydraulic pistons to move three separate cylinders along the same physical space. The design of such a system would be beyond the scope of a 20 week project.

Pursued Concept:

A pneumatic system was chosen to be the best solution to the problem. Keeping the system external to the engine block also meant simplicity. An attachment for the existing shift arm will be made, so that the double acting cylinder will be attached in order to create a linear force that will be translated into torque. A maximum torque of 150in-lbs was given by Polaris as what is needed to overcome the resistive spring and force needed to move the shift drum which repositions the forks and thus the gears being moved. This force could be greatly reduced by removing the spring that returns the internal ratchet to the home position, ready to shift again. It was determined that the spring would be retained to further simplify the application, allowing the use of a single double acting cylinder to create both up and down shift motions. Also, by retaining the spring's functionality, the existing shift lever could be re-attached with no functionality being removed from the ATV's manual operation. A pump and tank will provide the required pressure and volume of air, while a custom cover will provide mounting for the cylinder and also create a clean environment.

Technical Analysis:

I. System Pressure Losses

The initial analysis was performed by sourcing an appropriate pneumatic double-acting cylinder that would enable the required stroke distance of +/- 0.35 inches and the necessary output force while maintaining a small size. Using the specified product, an analysis of the system was performed by starting at point A shown below and working across the system to acquire the pressure drops across each component. By determining the overall pressure losses an appropriate tank and pump could be specified for the system.

Given: A pneumatic cylinder with the following specifications:

Total Supply Volume = 3.587 in^3
Total Exhaust Volume = 3.282 in^3
Maximum Shift Time = 0.0925 sec
Bore Size = 1.5 in
Rod Size = 7/16 in

Find:

a. Overall Pressure Losses
b. Required Minimum Tank Pressure

Schematic:

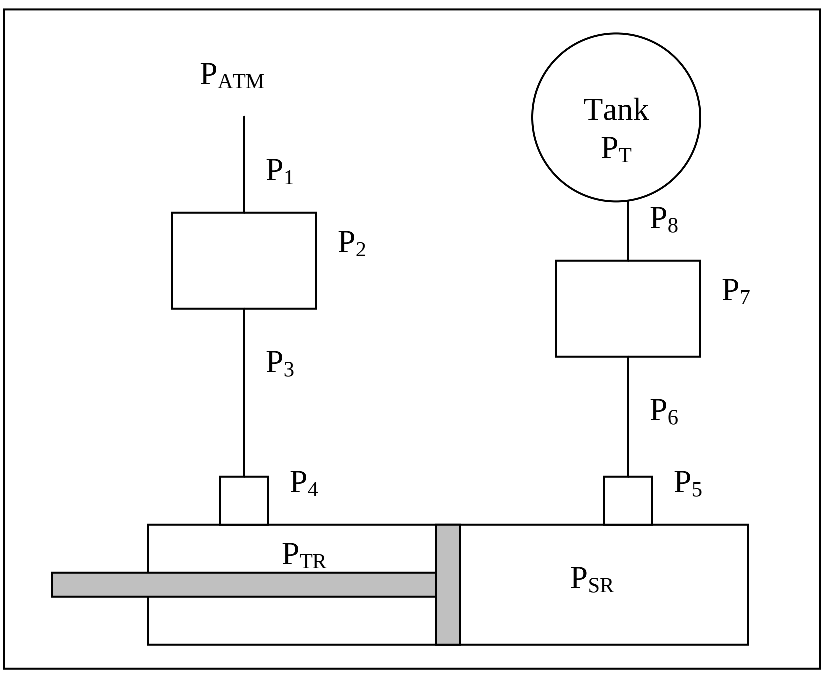


Fig 2: System Schematic

Assumptions:

- P_{ATM} = 14.7 psi
- Standard Conditions (T = 459°R, G=1)
- Total Air Line = 3 feet per side

Analysis:

Using Excel, a spreadsheet was created to calculate the total supply pressure required for this system:

Maximum Shift Time	0.1	sec						
Estimated Response Time	0.005	sec						
Solenoid Actuation Time	0.005	sec						
Maximum Cylinder Shift Time	0.0925	sec						
Atmospheric Presure	14.7	psi						
Standard Temperature	459	R						
G	1							
Valve Fitting ID	0.28	inch						
Air Line Diameter	0.25	inch						
Air Line Length	36	inch						
Air Line Friction Factor	0.02							
Cylinder Fitting ID	0.19	inch						
Supply Pressure	100	psi						
Required Force	120	lbs						
				Calculation				
Supply SCFM	9.160452	scfm						
Exhaust SCFM	8.381177	scfm						
Component	**Cv**	**ΔP (psi)**	**Pressure (psi)**		**Component**	**Cv**	**ΔP (psi)**	**Pressure (psi)**
Exhaust					**Supply**			
Valve Exhaust Port	1	4.34023			Cylinder Supply Fitting	0.6498	1.885554	
P1			19.0402302		P5			97.61718331
Exhaust Valve Fitting	1.4112	1.6826			Supply Air Line	1.223528	0.521554	
P2			20.72282976		P6			98.1387376
Exhaust Air Line	1.223528	2.056615			Supply Valve Fitting	1.4112	0.389975	
P3			22.77944474		P7			98.52871212
Cylinder Exhaust Fitting	0.6498	6.633268			Valve Supply Port	1	0.773555	
P4			29.41271286					
Cylinder Friction Pressure (psi)	1				**Total Supply Pressure (psi)**	**99.302**		
Total Resistive Pressure (psi)	30.41271							
Total Resistive Force (lb)	49.17175							
Total Resistive Force (lb)	169.1718							
Required Supply Pressure (psi)	95.73163							

Fig 3: Excel Analysis Results

Solutions:

1. From these calculations, the total minimum supply pressure is 99.3 psi which is very close to the initial pressure assumption of 100 psi.
2. The total required volume should be much greater than the volume per cycle of 3.587 in^3 to ensure the ability to perform multiple cycles immediately.

Comments:

This analysis is based on assumptions of friction factors, atmospheric conditions, and design factors. They will provide viable starting points for the system but may not be exact. This is a good reason for testing to be implemented upon purchase of the components to verify pressure losses and shift speed verification.

II. Shift Shaft Attachment Piece (P1001)

The stainless steel attachment piece, which connects the shift lever shaft to the pneumatic cylinder, will stop after 20° of rotation. At this point, 120 lbf is still applied to the component which is now fixed on the shift lever shaft. ANSYS was utilized to determine the deflection, stresses, and a design factor of safety for the component.

Solution: The entire results can be found in Appendix A

Load and Constraint Definitions

Name	Type	Magnitude	Vector
Force 1	Surface Force	120. lbf	1.574e-014 lbf -120. lbf 0. lbf
Fixed Constraint 1	Surface Fixed Constraint	0. in	0. in 0. in 0. in

TABLE 4
Constraint Reactions

Name	Force	Vector	Moment	Moment Vector
Fixed Constraint 1	120. lbf	-5.817e-007 lbf 120. lbf 3.307e-007 lbf	149.1 lbf·in	22.9 lbf·in -1.527e-007 lbf·in 147.3 lbf·in

Fig 4: Loading & Constraints

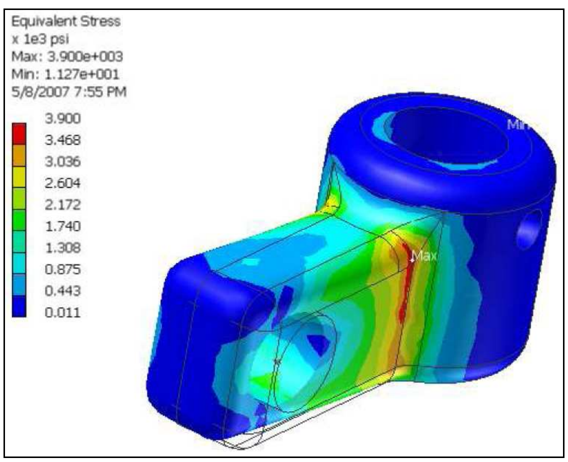


Fig 4: Equivalent Stresses

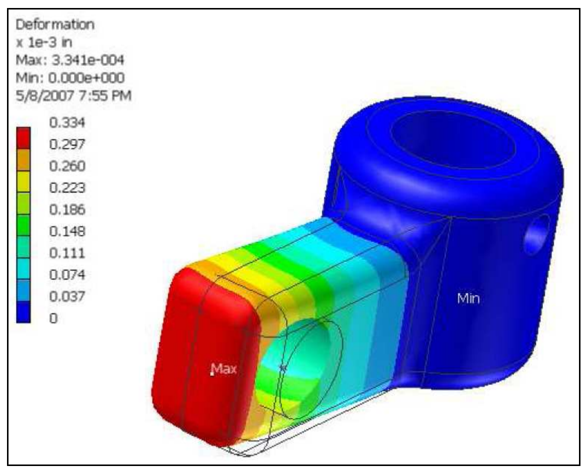


Fig 5: Deformation

Structural Results:

Name	Minimum	Maximum
Equivalent Stress	11.27 psi	3900 psi
Maximum Principal Stress	-386.6 psi	4249 psi
Minimum Principal Stress	-4078 psi	449.3 psi
Deformation	0. in	3.341e-004 in
Safety Factor	9.297	N/A

Fig 6: Structural Results

Comments:

Based on the ANSYS results the component will have a minimal deflection and will not yield. The minimal factor of safety of 9.3 shows it is over designed to ensure it will not yield. The deformation of 3.341e-4 inches shows it has practically no deformation. From this it can be said that the component will sustain the necessary loading based on static loading.

III. Cylinder Delay Time

Once a shift occurs there is a pressure differential within the cylinder. For example, when the cylinder is in the position shown in figure 7 the pressure in the exhaust side begins to bleed out its exhaust line while on the supply side the pressure will begin to increase. Due to the volume of the supply side being close to zero its pressure will increase rapidly. The time delay occurs mainly due to the slow exhaust time on the exhaust side. Once the pressures become equal, and continue to change, the piston will now be able to move in the push direction.

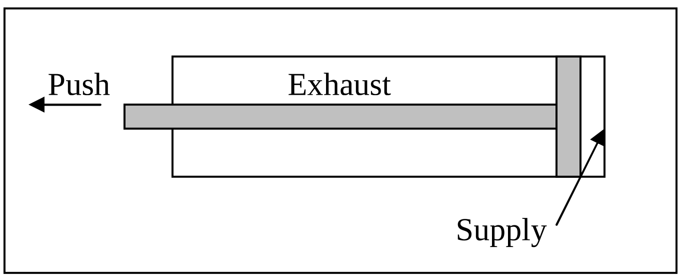


Fig 7: Delay Time Scenario

The results for the delay time can be easily found experimentally once our components are selected.

IV. Clevis Pin Analysis

The pin that holds the attachment part (P1001) onto the existing shift lever shaft will be a point of interest regarding yield stresses. The pin was analyzed as a beam under pure bending. The portion of the pin within the shift shaft is considered fixed while the portion within the attachment piece is under bending. The point of interest is the location where the pin leaves the shift shaft and enters the attachment.

Upon analysis it was found to have a stress at this point due to bending of 3785.2 psi which is much less than the material’s yield stress of 53700 psi. Based on this it is determined that the pin will have the sufficient strength to withstand the necessary loading.

Mechanical Risk Assessment:

Current areas of risk include weight and cost limitations along with other areas of continued analysis. These risks are primarily a result of not meeting a specification. The specifications can be found in Appendix B. Some of the notable risks include:

I. Weight Limitations

The specified weight given by Polaris Industries for this design is a maximum additional weight of 7 lbs. At this point the components specified have the following weights:

Component	Qty	Weight (lbs)
Cylinder	1	0.8
Solenoid Valves	2	0.74
Pump	1	2.5
Tank	1	0.833
Attachment	1	0.25
Existing Shift Arm	1	-0.326

This gives us a working total of 5.5 lbs. In addition to these components we will need to add brackets, air lines, fittings, and protective covers. This will be a challenge to meet and may be resolved using alternative materials or parts.

II. Cost Specifications

The specified cost requirements for this design was given by Polaris Industries as $250 as a final cost to manufacture. Appendix C contains the entire bill of materials for each known component. It should be noted that these costs are per part and will decrease per a 5000 count purchase.

The current working total at this point is approximately $295 which is already more expensive than our specified value. This may be resolved by sourcing alternative components or by receiving alternative quotes based on a 5000 piece order.

III. Analysis Risks

- The delay time of the cylinder is a mechanical risk due to the lack of prior analysis of the component. It will be addressed through contact with the supplier and by further analysis of the situation.
- It was discussed whether or not to purchase a tank or to design our own. A tank was specified due to the fear of laws enforcing regulations on the design of pressure vessels.

Mechanical Failure Modes:

The possible modes of which the pneumatic system may fail were discussed upon our team and were determined to be:

1. Shift attachment failure (P1001)
2. Individual component failure
3. Lack of necessary air
4. Pump operational failure including downtime
5. Contaminants in the system causing failure
6. Air line failure

These failures also have been present during the design phase in order to best prepare the system to combat these occurrences.

1. Shift attachment failure
   It was found that the attachment piece will not fail during static loading. The life of the component is unknown.

2. Component Failure
   Each component was examined and cycles per life were attempted to be found. The life of each component is still being researched between each supplying company.

3. Lack of Air
   The tank specified was based on the idea of having enough air to shift the ATV five times without the need to pump additional air into the tank. Given this and the specifications for the pump found in Appendix D the tank will supply enough volume of air and the pump will be able to rapidly replenish the system to prevent this occurrence.

4. Pump Operating Failure
   The pump specified has a duty cycle of 9% at 100 psi. This indicates that 9% of the time this pump will be required to operate to maintain a ½ gallon tank.

5. Contaminants
   Contaminants are an issue because if dirt or other foreign materials get into the system components will begin to fail more rapidly and may cause catastrophic failure. His will be contained utilizing covers over the components and a filter to filter the inlet air.

6. Air Line Failure
   The specified air line is made of stainless steel which will prevent most failures we would normally see by using plastic or rubber air lines.

Electrical Design

Solenoid driver:

The solenoid driver as shown in figure 8 uses a MOSFET as a switch which will supply or cutoff the solenoid from the power source. The solenoid is modeled by the series resistance and inductor and the microcontroller is modeled by voltage source V3. The output from the microcontroller which operates at 3.3 V will be amplified by an op-amp so that the voltage applied to the gate of the MOSFET will be as high as possible to drive the MOSFET fully on. The output waveform for the driver circuit is shown in figure 9.

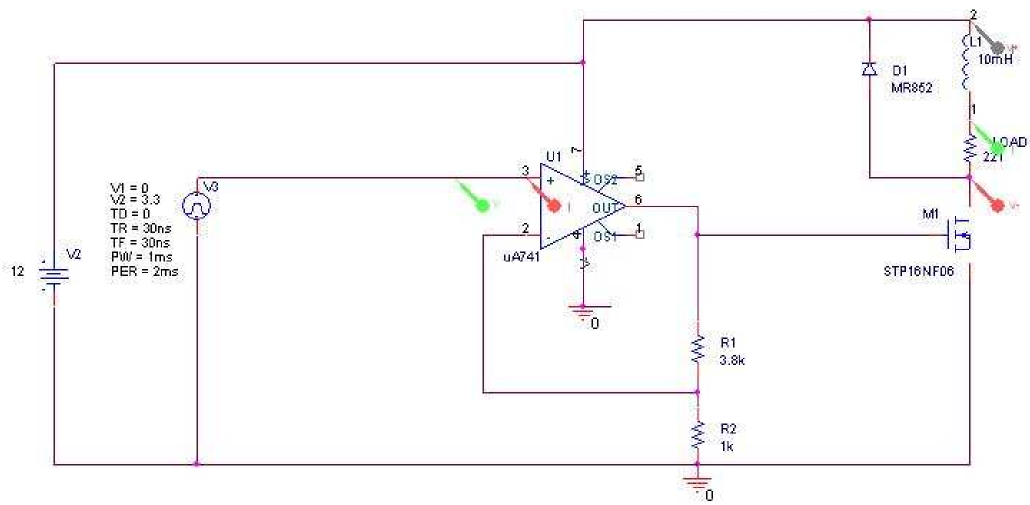


Figure 8: Solenoid driver circuit.

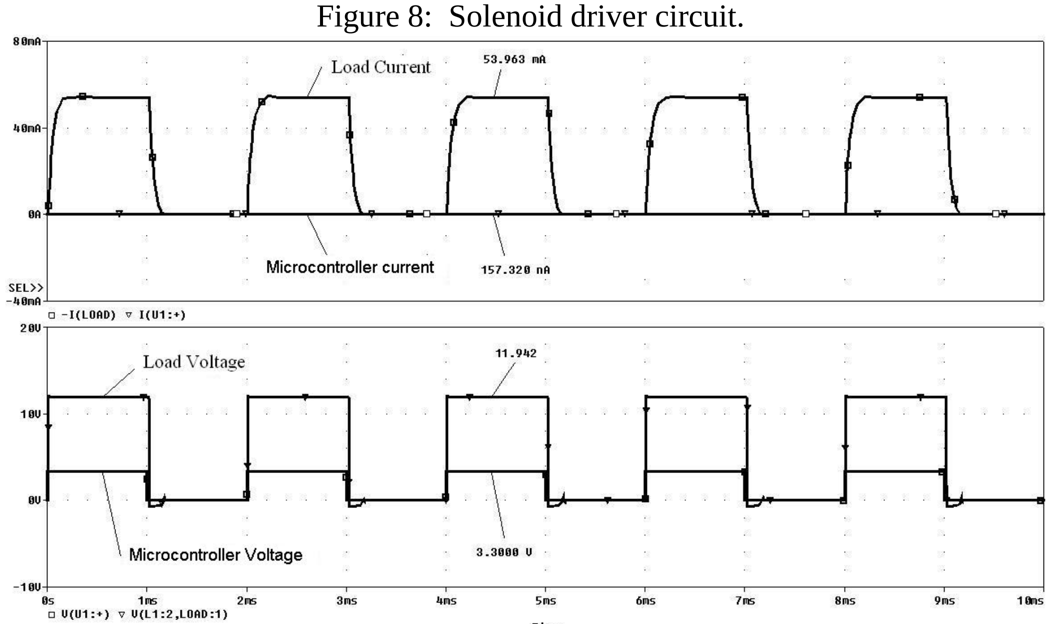


Figure 9: Solenoid diver input and output waveforms.

LED driver:

Because LED requires only a small forward voltage, the LED will be driven by buck converter supply voltage. A simple MOSFET switch will be used to turn the LED's on or off and the MOSFET will be controlled by the microcontroller. The LED driver circuit is shown in figure 10.

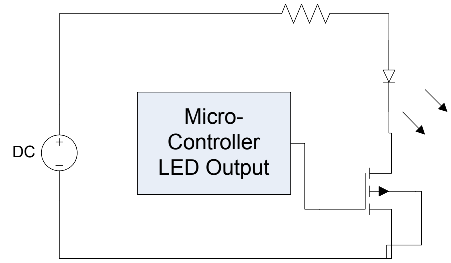


Figure 10: LED output driver circuit.

Inductive RPM sensor input:

Figure 11 demonstrates the circuit designed to recognize voltage spikes coming from the inductive pickup and amplify these signals for the microcontroller to recognize. The voltage spikes from the inductive pickup peaks at about 300 mV and is too low for the microcontroller to recognize. Minimum input voltage to the microcontroller's digital input is 1.5 volts. The input source to the operational amplifier represents the output of the inductive pickup and the output of the amplifier will be fed to the microcontroller.

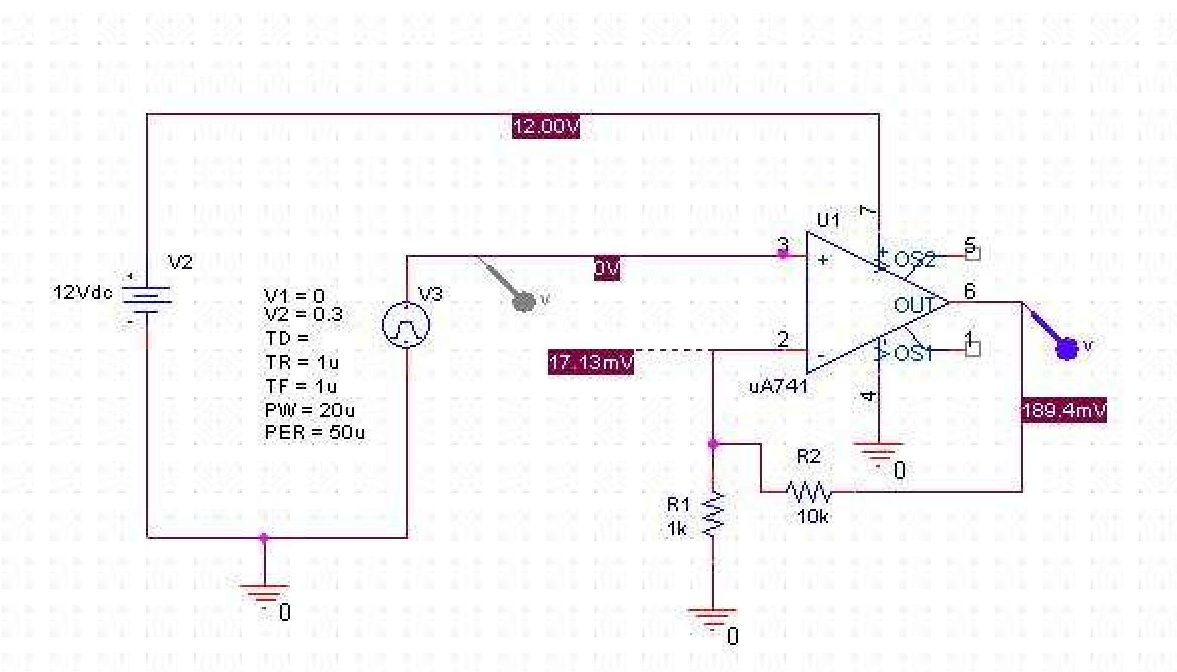


Figure 11: Circuit to amplify input from inductive pickup

Figure 12 shows the simulation of this scenario. Ideally, the 300 mV input is amplified to about 3 volts with the design of Figure 11. These amplified pulses will be the input to the microcontroller and will be used to find the RPM of the ATV.

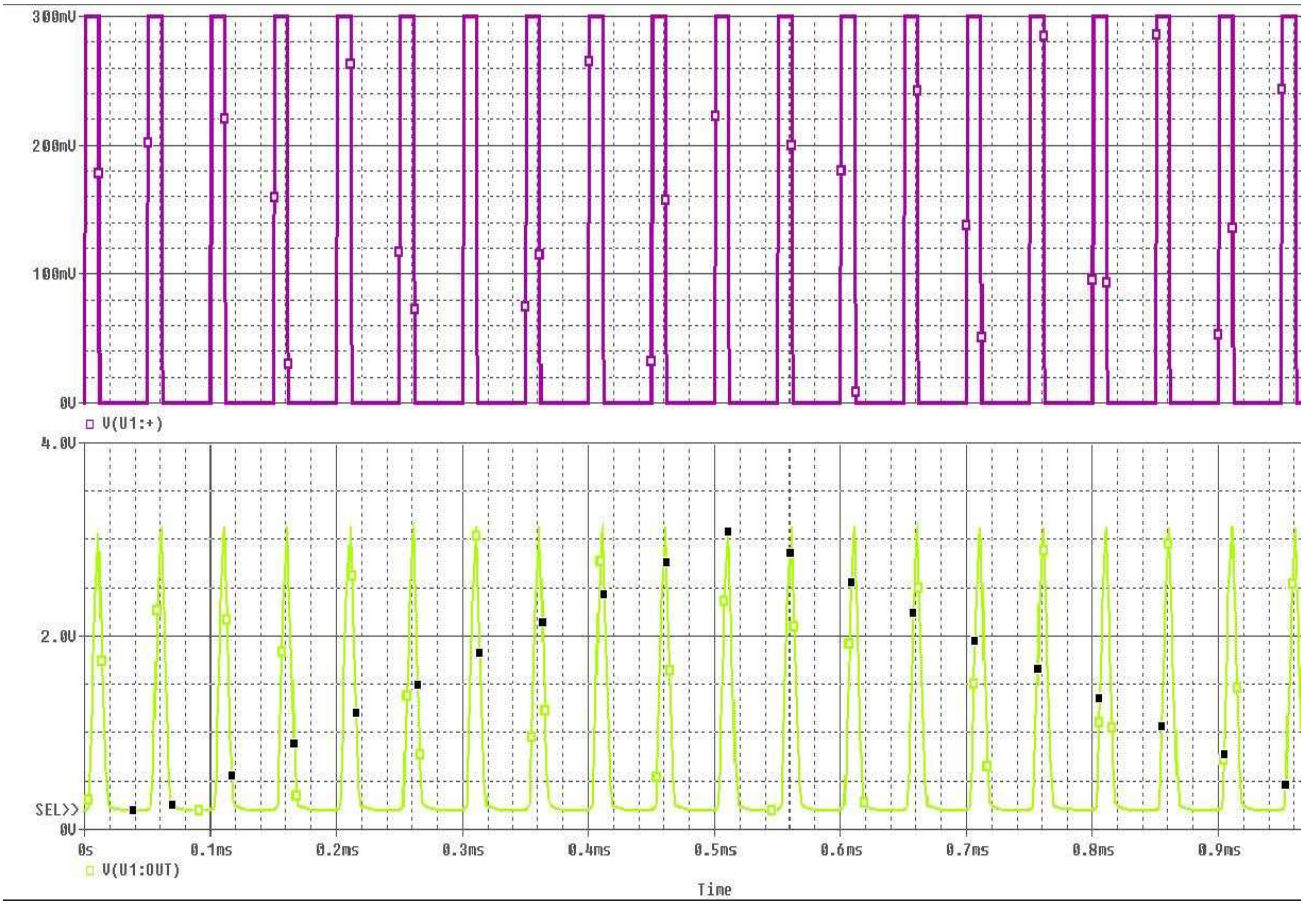


Figure 12: Simulation results of amplified signal from the inductive pickup

The calculations for determining the RPM are shown below, Δt is the time between the pulses from the inductive pickup. The time between pulses is determined by the microcontroller by using an interrupt on change input pin. When a pulse occurs the microcontroller will move to an interrupt subroutine that saves the value of a timer and then reset the timers count value. When the next pulse occurs the timer value is again saved and reset. Using the timer value the amount of time that elapsed between pulses can be determined by using the following equations.

$$\frac{\Delta t \text{ ms}}{1\text{Rev}} \times \frac{1\text{ s}}{1000\text{ ms}} \times \frac{1\text{ m}}{60\text{ s}} = \frac{\Delta t \text{ minutes}}{60000\text{ Revolution}} \quad \therefore \quad RPM = \frac{60000}{\Delta t}$$

Taking the worst case scenario of a maximum of 10000 RPM, the voltage spikes would come in at a rate of one every 6 ms. Because of the nature of the RPM sensor the sampling rate will change with the change in the RPM's.

Throttle position input:

The throttle position can be determined from the analog signal that is produced from a sensor on the intake manifold. This signal will be reduced by a voltage divider to produce a voltage swing of 0 to 3.3 V so that the microcontroller can do an analog to digital conversion (ADC) and determine a normalized throttle position value. So that there is no current loss on throttle position signal to the ATV motor controller, a unity gain op-amp circuit will be used to buffer the sensor signal and the voltage divider. The circuit used to condition the throttle position input signal is shown in figure 13. The sampling rate will be the same as the RMP sampling rate to achieve a correlation between RMP values and throttle position.

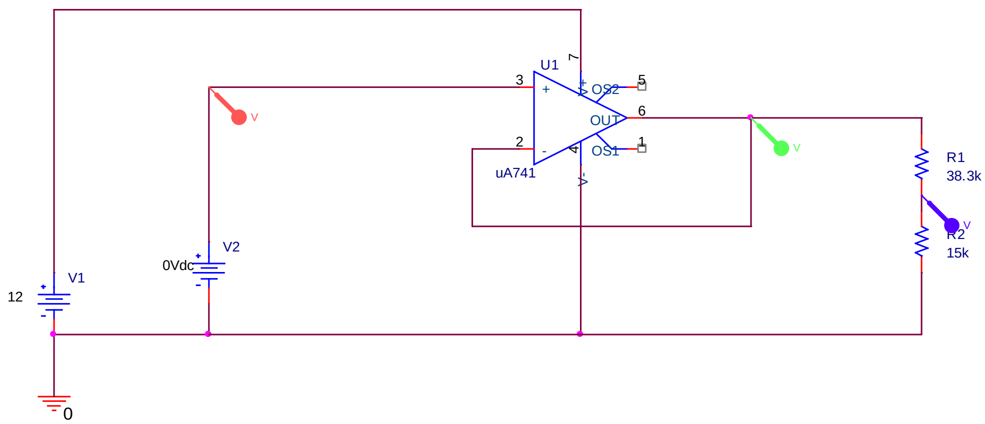


Figure 13: Signal condition for throttle position input.

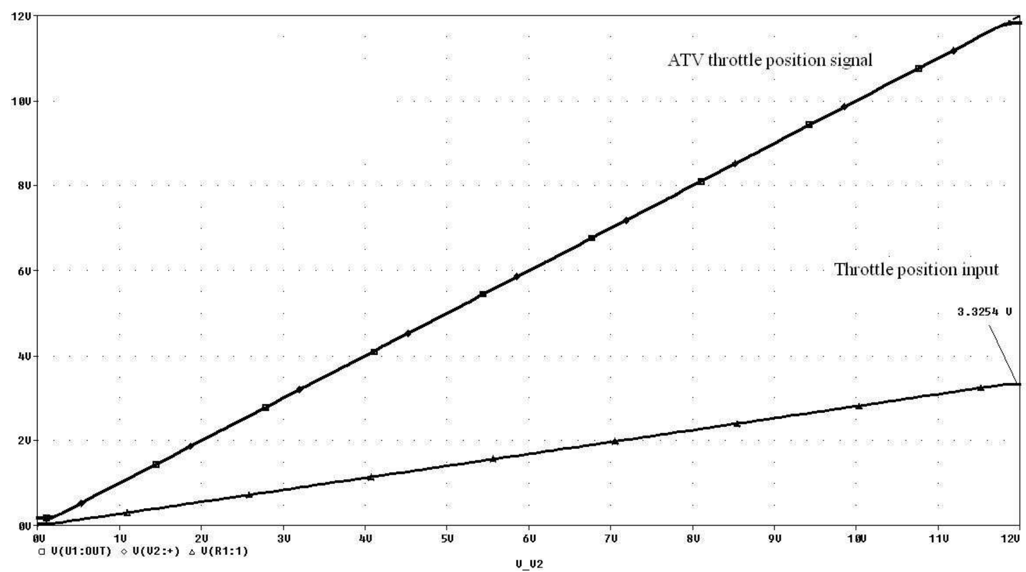


Figure 14: Throttle position input waveform.

Push button input:

The pushbuttons will be input into the microcontroller after a hardware de-bouncing circuit which consists of a low pass filter. The input will produce an interrupt on logic level change so that the input pin does not need to be polled constantly. There will be two push buttons: one for shifting up and one for shifting down. The pushbutton schematic is shown in figure 15.

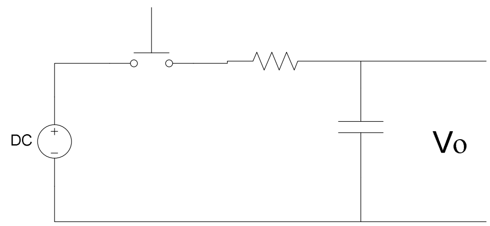


Figure 15: Pushbutton de-bouncing circuit.

Block Diagram:

Figure 16 shows a block diagram of the electronic circuit that would control the transmission shifting. The respective circuits are as they have been shown above. The final circuit will be placed together on a PCB board and contain additional parts such as resistors, inductors, and capacitors. Actual components values may vary due to hardware tolerances that are present in the simulations.

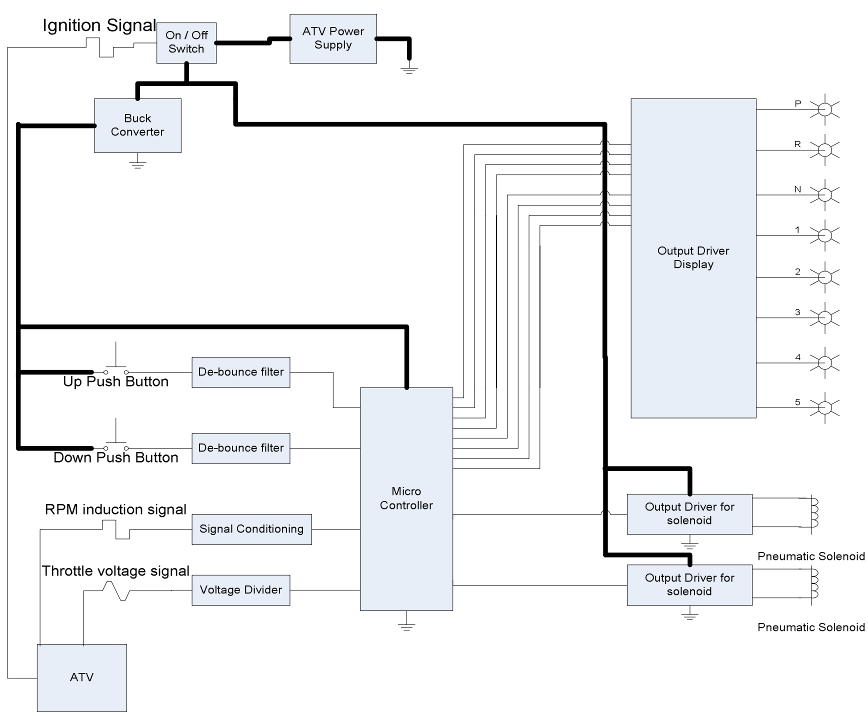


Figure 16: Block diagram of the electronic circuit.

Microcontroller:

The microcontroller that is going to be used is the MSP430FG4618. This chip was chosen because it has 8 kilobytes of RAM. This allows for two 16-bit RAM tables of 1024 memory location, and allows for an equally large amount of general usage memory location. The microcontroller has a hardware multiplier which saves programming time because multiplication does not have to be done in software. The microcontroller has a twelve channel 12-bit analog to digital controller (ADC) for the throttle position input. It has the ability to generate interrupts on pin logic level changing for inputs from the 3 pushbuttons and from the inductive RPM sensor. There are 10 16-bit timers (the most available in a MSP430) which reduces programming limitations. The microcontroller will also provide 48,000 instruction cycles for a program to determine if it is safe to switch gears when the user asks to do so.

- 8 KB of RAM
- Hardware Multiplier
- 12-bit ADC
- Interrupt on logic level change
- Ten 16-bit timers
- 80 Input/Output.
- 16 MIPS

The microcontroller memory is in two main sections. Flash program memory and RAM, which are separate from one another in the microcontroller architecture. RAM can be broken into three sections: general purpose RAM and two RAM tables used to store past values of engine RPMs and throttle position. A memory map showing this is shown in figure 17 and a state diagram is shown in figure 18.

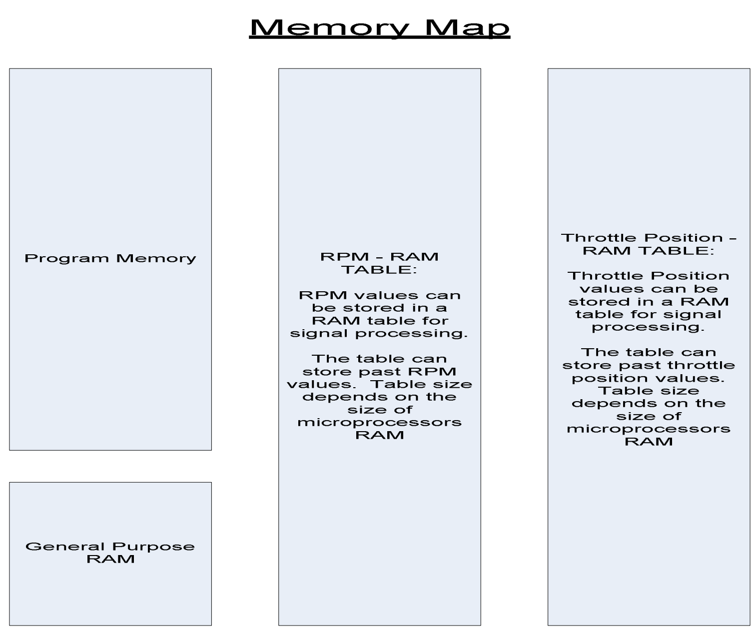
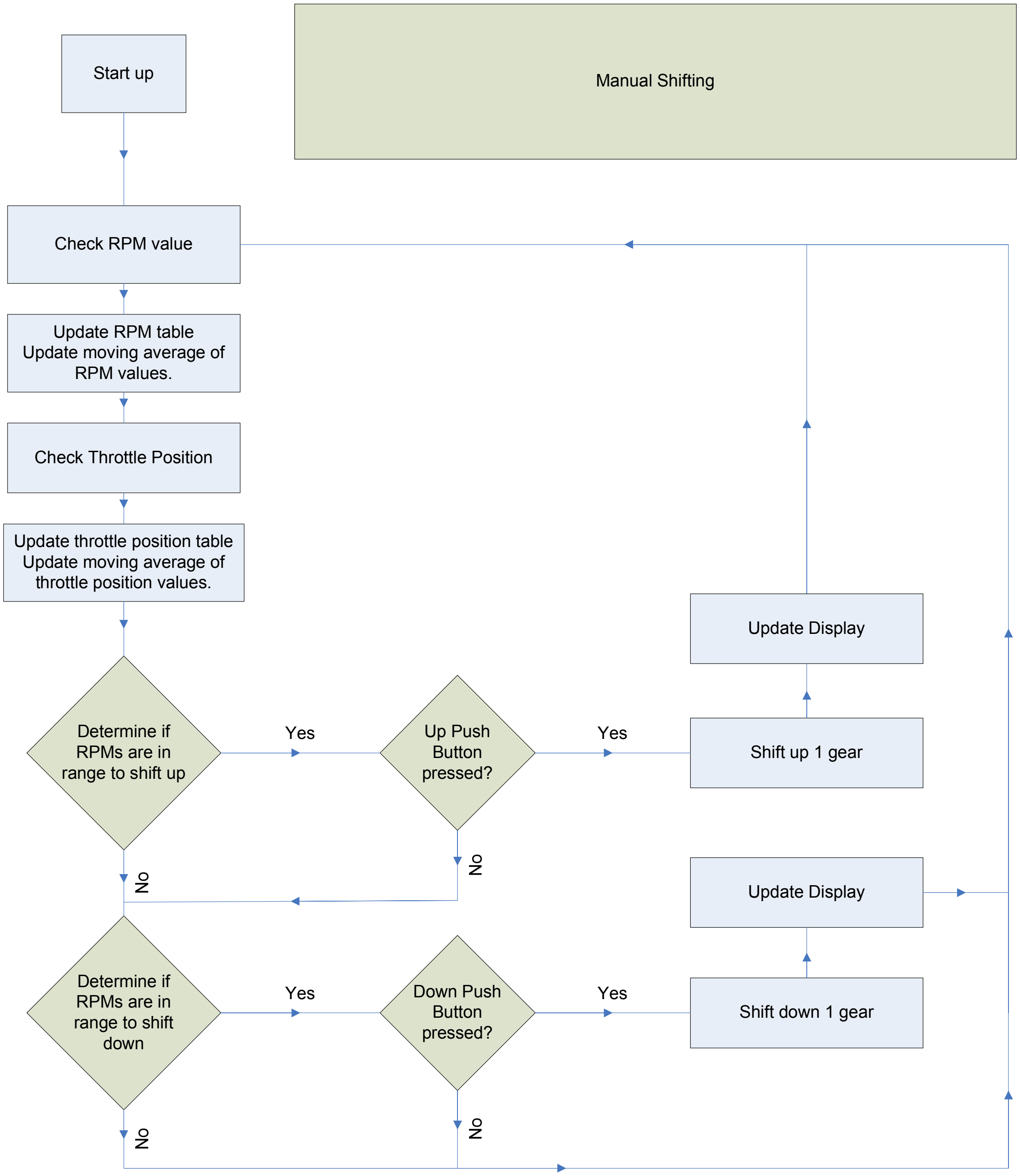


Figure 17: Microcontroller memory map.



Gear	Frist	Second	Thrid	Forth	Fifth
Min. RPM value for Up shifting	3000	3000	3000	3000	NA
Max. RPM value for down shifting	NA	7000	7300	7600	7600

Figure 18 – State diagram for the control program with RPM limits.

Making use of the RPM table in memory a moving average can be done on the stored values. The moving average can be used to do some filtering on the RPM values. Comparing the average value to the most recent value a measure of the change in RPM can be determined.

The main goal is to design the electrical controls to operate the transmission based on the user input. However the design will be scalable so that it may be upgraded to be used for a fully automatic system if a fully automated system is proved to be feasible. If the main goal is completed with extra time remaining in the quarter, the microcontroller can be used for data acquisition. The data acquired data can be used to determine the feasibility of a fully automated system.

Risks:

One of the risks that were determined at the time of the concept review was that a microcontroller would not be able to handle the signal processing and physical control of the transmission. Because of the simplicity of the physical controlling aspect and because of the slow input signals the microcontroller will be sufficient. For fully automating the transmission the mathematical functions required to determine the proper shifting time may require more computation time. For this reason a DSP with the same capabilities as the microcontroller but with a faster clock frequency that is possible to achieve with a microcontroller has been chosen as a back up plan. The model number of the DSP is TMS320F2802-60 and it is made by Texas Instruments.

A risk more serious then not having enough processor power to automate the shifting of the transmission is that a function for shifting cannot be determined due to the extensive possibilities of different scenarios that would require a shift. The only inputs available to the microcontroller are RPM, throttle position, and push buttons. These inputs may not contain all the information needed to determine every shifting scenario. An example of such scenario is going up a hill. In this case, the driver would want to shift down to get more torque, while the microcontroller may decide to up shift due to throttle position and RPM. Also, it depends on the rider on how he/she wants to ride downhill, get off throttle and let the engine get out of gear or break and slow down. Such shifting dynamics get too complicated so the initial goal of shifting on push buttons will be implemented.

The major risk encountered in trying to achieve our stretch goal, shifting with no user input, is that an algorithm couldn't be determined which would take into effect infinite riding scenarios. At the present time there is no information available to determine if shifting without user input is a possibility or not. A few of riding scenarios are listed below:

1. Racing
    a. Start of race

        i. Neutral to 1^{st} through 5^{th} gear, full throttle
        ii. Shifts fast, Late shifting at high rpm
        iii. Keep rpms high for more torque
    b. Full speed into a turn
        i. Braking hard
        ii. Down shifting possibly 5^{th} to 3^{rd} or 2^{nd}
        iii. Off throttle almost all the way
    c. Exit turn
        i. Full throttle
        ii. Again shifting at high rpm
    d. Jumps
        i. In air, no load on engine
        ii. Use brake and torque for rotation of bike
        iii. Leave in one gear for this control?
    e. Landing
        i. Initial slowdown of rpm
        ii. Jump on throttle back to high rpm
    f. Up hill
        i. Full throttle
        ii. Down shift for higher rpm?
        iii. High load on engine
        iv. Slowing down
    g. Down hill
        i. Full throttle
        ii. Faster time between shifts because of increase in acceleration

2. Normal/Backyard
    a. No need for shifting at high rpm
    b. Lower torque
    c. Smoother throttle control
        i. No "jumping" on throttle
        ii. Constant speed
    d. Braking
        i. Uses engine as brake
        ii. Very light braking
        iii. Usually coming to a stop if braking is used
        iv. Hard braking mostly because of an obstacle in the way
    e. Mostly trail and long straight riding
        i. Long continuous speeds
        ii. Small hills

3. Odd situations
    a. Very large hills
        i. Coasting down using small amount of throttle
            1. could mean rider wants to coast down without using gas
            2. take engine out of gear

ii. Going down with no throttle engagement
    1. rider wants engine to “brake” for him
    2. does not want to “ride” the brakes
iii. Possibly no acceleration but high throttle for 1 or 2 minutes up hill