2016 University of Waterloo Clean Snowmobile

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Abstract

The University of Waterloo Clean Snowmobile Team has developed a low-emission, low noise snowmobile by implementing a Miller cycle conversion; an integrated muffler/catalytic converter; large diameter rear idler wheels; a single-ply, 2.86 inch pitch, pre-studded track; and electronic throttle control that reduces throttling losses through closed-loop control of the turbocharger. The modified snowmobile is intended to appeal to experienced riders who demand premium power, suspension, and fit and finish.

Introduction

Snowmobiling is a popular winter recreational activity that has received substantial scrutiny for its environmental impact. Prior to the 21st century, snowmobiles were primarily powered by two-stroke engines with very little emissions control. This led to legal action being levied against the National Park Service for failing to uphold environmental regulations governing Yellowstone National Park, among others, due to the pollution caused by recreational use of snowmobiles in the parks.[1] It is from this controversy that the SAE Clean Snowmobile Challenge was conceived.

Undergraduate engineering students from across Canada and the United States compete annually to modify current production snowmobiles to reduce emissions and noise and improve fuel economy. The Challenge has also evolved to include the use of biofuels to reduce the carbon footprint of snowmobiling as well. Most recently, isobutanol blends were debuted as the competition fuel for the 2014 competition.

Preliminary Design

The University of Waterloo Clean Snowmobile team has elected to continue with its 2015 platform, consisting of a 2013 Polaris Rush chassis powered by a Weber MPE 750 turbocharged engine. This platform has proved mechanically reliable and quite robust.

Figure 1: Waterloo's snowmobile at the 2014 Clean Snowmobile Challenge

The MPE 750 turbo was used in Polaris FST snowmobiles from 2006 to 2014. This engine is capable of producing 143 hp on the 2012 factory calibration. The competition rules limit engine output to a maximum of 130 hp.[2] As such, there is little disincentive to implement solutions that may reduce the engine output.

Table 1: Stock Engine Specifications

Displacement	749 cc
Cylinder Arrangement	2 cylinder, inline
Crankpin Offset	360°
Output	143 hp at 7750 rpm
Bore	85 mm
Stroke	66 mm
Connecting Rod Length	115 mm
Compression Ratio	9:1
Intake Valve Lift	9.5 mm
Exhaust Valve Lift	9 mm
Intake Valve Opens	15 °BTDC
Intake Valve Closes	50 °ABDC
Exhaust Valve Opens	52 °BBDC
Exhaust Valve Closes	2 °ATDC
Rocker Ratio	1.43

In order to more directly gauge design alternatives against competition performance, Waterloo has generated a decision matrix that weighs six criteria against their impact on each competition event, and then against the proportion of competition points available for each event.

Table 2: Criterion weightings for evaluating design alternatives

Efficiency	23%
Emissions	22%
Weight	15%
Power	5%
Noise	15%
Reliability	15%
Novelty	5%

Each design alternative was given a numerical benefit value on a ±5 scale for each criterion. The weightings were then used to create a 'weighted benefit' to indicate the relative merit of each alternative. Several areas of focus were identified using this method:

- Miller cycle
- Electronic throttle control
- Stoichiometric fueling regime with a three-way catalyst
- Electronic continuous variable transmission
- Lightweight track
- Large-diameter rear idler wheels

Design

Miller Cycle Conversion

The spark ignition thermodynamic cycle is typically modeled using the Otto cycle, with symmetric compression and expansion ratios. Efficiency is strongly related to the geometric compression ratio of the engine. The theoretical efficiency of an Otto cycle engine is generally given in terms of the geometric compression ratio, r, and the compressibility of the working fluid, γ:

$$\eta_{th} = 1 - \left(\frac{1}{r^{\gamma-1}}\right)$$

The compression ratio is limited by the tendency of the fuel knock as well as other practical considerations. The Miller cycle is a variation of the Otto cycle in which the trapped compression ratio is deliberately made lower than the geometric compression ratio. This over-expansion is typically characterized by the ratio of the geometric compression ratio to the trapped compression ratio [4]:

$$\sigma = \frac{\varepsilon_{geom}}{\varepsilon_{tr}}$$

Martins et al have demonstrated that for a constant geometric compression ratio, efficiency decreases with increasing over-expansion and that for a constant trapped compression ratio, efficiency increases with increasing overexpansion. [4]

The geometric compression ratio is easily varied by changing the pistons. There are five commercially-available options for the stock bore diameter, listed in Table 3.

Table 3: Piston Options

Part Number (Manufacturer)	Compression Ratio	Notes
P-8494 (Weber Power)	9:1	Lateral gas ports, double pin oilers, accumulator groove
W-84118 (Weber Power)	11:1	Lateral gas ports, double pin oilers, accumulator groove
0453178 (Polaris)	9:1	FST stock piston
0453747 (Polaris)	11:1	FS stock piston
40021M08500 (Wiseco)	8.5:1	

Without having to resort to developing a new, untested piston, the only feasible option that raises the engine efficiency is to install a piston with an 11:1 compression ratio. The Weber Power component was ultimately selected.

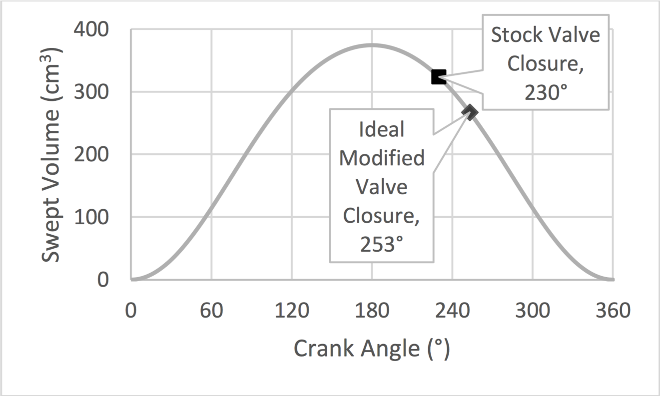


Figure 2: Intake valve closure angle

In order to reduce the trapped compression ratio, a custom camshaft was required. In order to reduce the amount of engine calibration changes required, the cam was designed to maintain the original trapped compression ratio of 9:1 and thus approximately the original peak cylinder pressure. The ideal new intake valve closure angle would correspond to the remaining cylinder sweep volume being 9/11 or 81.8% of that at the original intake valve closure angle. This results in an increase in duration of 23°. The increase in thermal efficiency is predicted to be around 3% and the loss in power is predicted to be around 14%.

The modified camshaft is, however, limited by the vendor's catalogue of available profiles. With the engine being SOHC, both the intake and exhaust profiles must be ground by the vendor. As such, both profiles end up changing. A number of profiles suggested by the vendor are listed in tables 4 and 5.

Table 4: Exhaust Cam Profiles, Difference from Stock

Profile	Lift (in)	@ .020"	@ .040"	@ .050"
E1	+0.007	+7°	+10°	+9°
E2	+0.008	-8°	-2°	-2°
E3	+0.008	+9°	+11°	+10°

Table 5: Intake Cam Profiles, Difference from Stock

Profile	Lift (in)	@ .020"	@ .040"	@ .050"
I1	+0.005		+15°	+16°
I2	+0.006		+5°	+8°
I3	+0.007		+4°	+7°

Profile E2 was selected as it changes the duration the least. Profile I1 was selected as it offers the greatest duration increase. The exhaust lobes were retarded by 4° to compensate for the reduced exhaust duration and maintain a similar amount of overlap. The intake lobes were retarded by 13° to offset the duration increase into the compression stroke while attempting to maintain overlap. The final cam specifications are presented in Table 6:

Table 6: Modified Camshaft Specifications

Intake Valve Lift	9.7 mm
Exhaust Valve Lift	9.25 mm
Intake Valve Opens	9.5 °BTDC
Intake Valve Closes	70.5 °ABDC
Exhaust Valve Opens	46 °BBDC
Exhaust Valve Closes	2 °ATDC

Throttle by Wire

Waterloo's motivation to implement electronic throttle control is derived from a desire to rethink how the powertrain reacts to user input. A snowmobile engine's power output is typically controlled using a cable directly connecting the throttle lever to the motion of the butterfly valve(s). From a functional standpoint, the throttle valve is a power modulator, and the throttle lever is a user input to demand a certain amount of power.

On a turbocharged engine, the power output is increased by increasing the density of the intake air charge. Most modern turbocharger systems are variable guide vanes. A turbocharger can also be considered a power modulator.

Throttling loss accounts for a significant portion of efficiency loss at low- and part-load conditions. For a conventionally-throttled turbocharged engine, both the piston and the compressor are working against the throttle butterfly at part-load. With electronic throttle control the throttle butterfly is physically decoupled from the throttle lever and is thus likewise decoupled from the user's power demand.

The physical design for Waterloo's ETC system includes a pair of Bosch electronic throttle bodies located where the MPE 750's ITB pair would normally be. This maximizes interchangeability between the systems, allowing them to be swapped whenever necessary. In addition, maintaining the ITB configuration of the MPE 750 preserves its responsiveness.

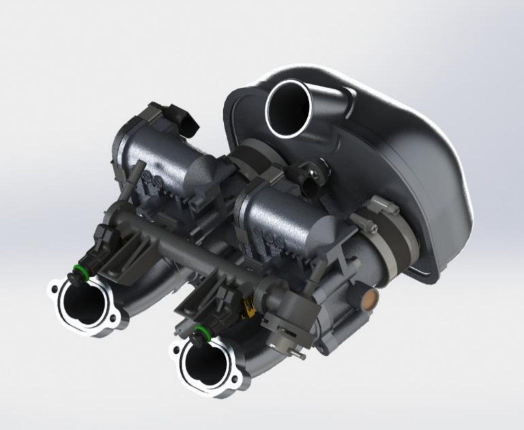


Figure 3: Electronic Throttle Body System

Waterloo's implementation of electronic throttle control modulates the last 20-30% of user power demand through the turbocharger while maintaining WOT. Figure 4 illustrates the aim output for the system at 7000 rpm.

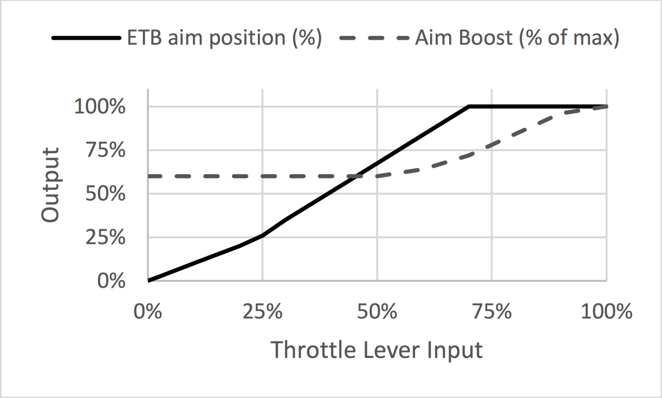


Figure 4: Throttle and Boost Target Positions at 7000 rpm

The throttle request unit selected allows the stock thumb lever to be maintained, though it requires a modified cable with a longer free length. This has four advantages over systems that integrate the position sensor into the throttle lever block:

- The throttle safety switch maintains its function
- The throttle input has a familiar feel for the user
- The wiring to the sensor is contained within the hood where it is less likely to be subject to chafing, fatigue, or damage from a crash
- Increased modularity of the ETC system – the same handlebar assembly may be installed across a range of snowmobiles with and without ETC

Electronic Continuously Variable Transmission

Mechanical Actuator

An electronically controlled continuously variable transmission can reduce drivetrain transmission losses and increase fuel economy by up to 5%. A design review showed the optimum design is a wedge to actuate the moveable sheave on the primary clutch with a linear actuator. To determine the size of the linear actuator and the load requirements the following formula was used to determine the maximum force required to actuate the clutches.

$$Actuation\ Force = \frac{\cos\theta * |T|}{2 * u * r}$$

Where θ is the angle of the clutch sheaves, T is the transmitted torque, u is the friction between the belt and the sheave and r is the radius of the belt on the pulley. Combining this with experimental data the actuation force is approximately 675 lbf for this specific application.

The final mechanical design can be seen in Figure 5. Drawbacks of this method of implementation are fairly severe. The moveable sheave is not coupled to the crankshaft, and therefore only one sheave transmits the mechanical torque to the belt. During testing this leads to slip and ultimately the greatest downfall to the system when testing at high loads.

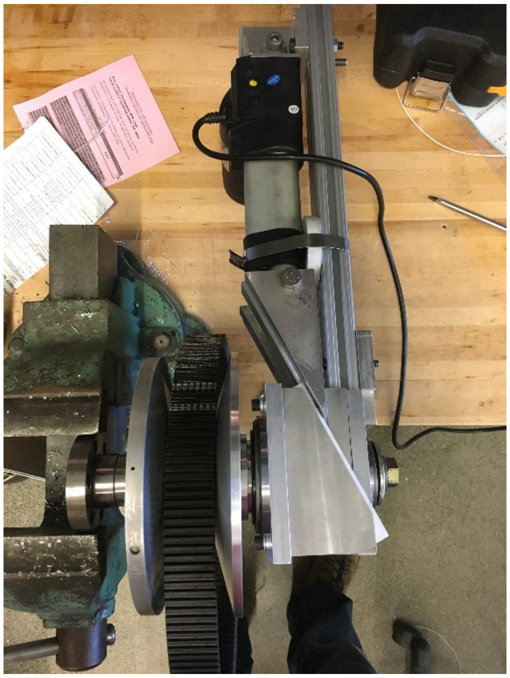


Figure 5: ECVT Mechanical System

Drawbacks of this method of implementation are fairly severe. The moveable sheave is not rationally connected to the crankshaft, and therefore only one sheave transmits the mechanical torque to the belt. During testing this leads to slip and ultimately the greatest downfall to the system when testing at high loads.

Control Scheme

To optimize the CVT for fuel efficacy at a constant power a Simulink model was created of the entire vehicle drivetrain. Parameters were selected based on physical testing to ensure model accuracy.

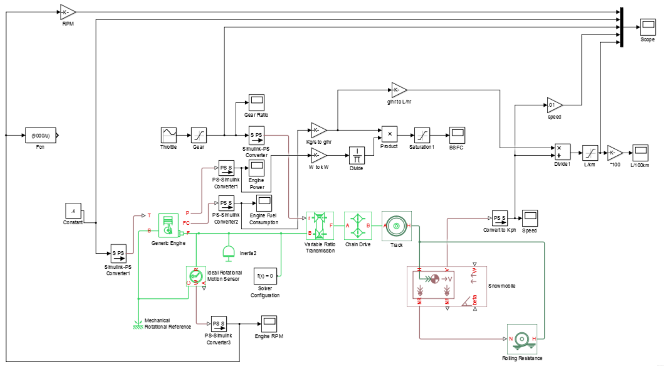
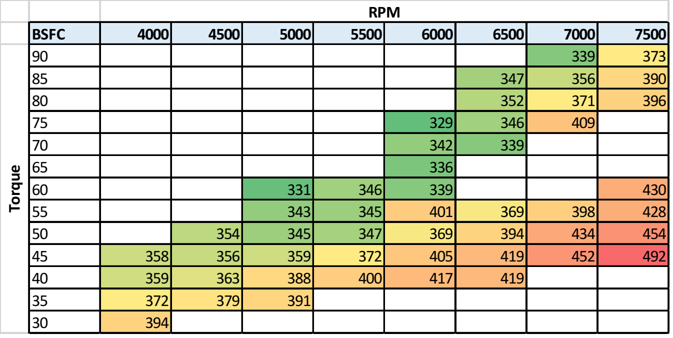


Figure 6: Simulink Model of Snowmobile

To accurately predict fuel consumption and power output dyno testing was completed. Recoding injector duty cycle and engine power and torque, BSFC in lb/hp*h was calculated at a set number of RPM and torque values. These values are shown in Figure 7.



		RPM							
	BSFC	4000	4500	5000	5500	6000	6500	7000	7500
Torque	90							339	373
	85						347	356	390
	80						352	371	396
	75					329	346	409	
	70					342	339		
	65					336			
	60			331	346	339			430
	55			343	345	401	369	398	428
	50		354	345	347	369	394	434	454
	45	358	356	359	372	405	419	452	492
	40	359	363	388	400	417	419		
	35	372	379	391					
	30	394							

Figure 7: BSFC vs Torque and RPM

Using these values in our Simulink model an optimization was performed to minimize fuel consumption at a constant power at a specific gear ratio. The optimization that was performed at a constant throttle while the gear ratio was varied. By choosing a path where both speed and fuel efficiency were both at a compromised maximum was the determined gear ratio. At time (2) in Figure 8 is where the optimized value occurs, at the specific gear ratio showed by the red curve.

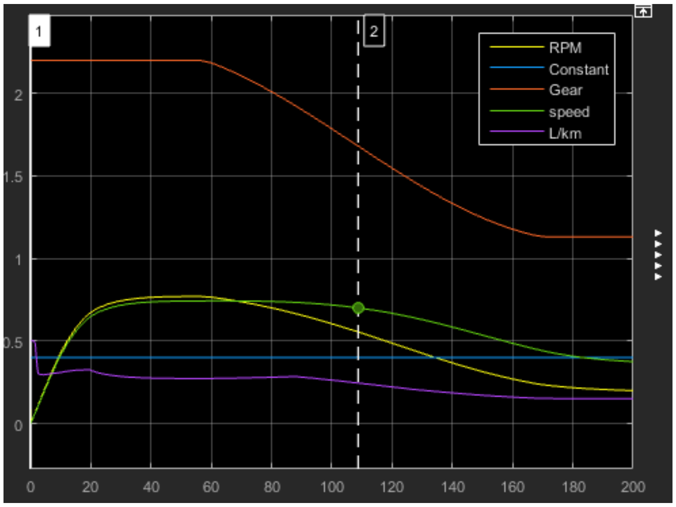
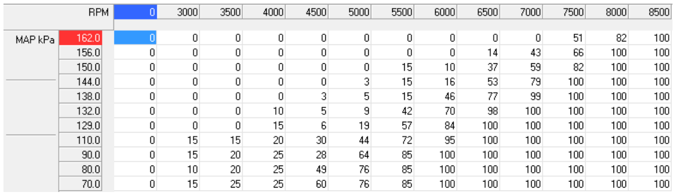


Figure 8: BSFC vs Torque and RPM

To implement the manifold pressure has to be correlated to engine torque. This is done by testing and averaging values of manifold pressure across a range of load and rpm values by dyno testing. By combining these results together and implementing in a lookup table in the M400 ECU the snowmobile is able to have theoretically optimized the gear ratio for an electronically actuated continuously variable transmission. The final lookup table is shown in Figure 9.



	RPM	0	3000	3500	4000	4500	5000	5500	6000	6500	7000	7500	8000	8500
MAP kPa	162.0	0	0	0	0	0	0	0	0	0	0	51	82	100
	156.0	0	0	0	0	0	0	0	0	14	43	66	100	100
	150.0	0	0	0	0	0	0	15	10	37	59	82	100	100
	144.0	0	0	0	0	0	3	15	16	53	79	100	100	100
	138.0	0	0	0	0	3	5	15	46	77	99	100	100	100
	132.0	0	0	0	10	5	9	42	70	98	100	100	100	100
	129.0	0	0	0	15	6	19	57	84	100	100	100	100	100
	110.0	0	15	15	20	30	44	72	95	100	100	100	100	100
	90.0	0	15	20	25	28	64	85	100	100	100	100	100	100
	80.0	0	10	20	25	49	76	85	100	100	100	100	100	100
	70.0	0	15	25	25	60	76	85	100	100	100	100	100	100

Figure 9: Output position of linear actuator vs. map

This optimization is for a specific snowmobile with a specific set of clutches, snowmobile weight and other important parameters. The system was tested, and was able to move forward and showed increased fuel economy at slow constant speeds. However due to the untested reliability the decision not to run it at the 2016 clean snowmobile challenge was decided due to reliability concerns.

Figure 10: ECVT installed on 2016 snowmobile

Exhaust System

Catalytic Converter

In 2014, Waterloo's snowmobile was equipped with a 3-way catalytic converter that is commonly used on road vehicles, connected in series with a custom muffler. While this provided effective exhaust gas treatment, the catalyst was long and occupied most of the physical space inside the body of the snowmobile, leaving the muffler to be mounted below the running board. This caused several problems. First, the muffler required a large heat shield around it to prevent accidental contact with hot surfaces. Also, the muffler would constantly make contact with the snow, which caused thermal fatigue from heating and cooling. Finally, the muffler became damaged when the snowmobile rolled over during competition, creating extra work for team members who had to fix it.

For 2016, Waterloo has made modifications to the 2015 exhaust system in order to make better use of space and improve emissions and noise. A shorter catalytic converter allows the converter to be earlier in the exhaust system freeing more space for noise reduction.

The catalytic converter that is being used is a three way converter that uses a reduction catalyst and an oxidation catalyst to convert harmful exhaust gases into less harmful forms. The reduction catalyst uses precious metals to reduce nitrogen oxide emissions. A precious metal coating reacts with NO_x molecules to produce nitrogen gas and water. The oxidation catalyst reduces carbon monoxide and unburned hydrocarbon emissions by burning them over platinum and palladium using oxygen in the exhaust gas.[5]

Muffler Design

Several design features have been used in order to maintain a lightweight exhaust system. The final muffler dimensions are approximately 11” tall, 5” deep and 10” wide. By keeping the muffler small in size, less than 2 square feet of steel sheet was used, resulting in lower cost and weight than many stock mufflers. During material selection, aluminum had been considered, due to its low density and ease of manufacturing. However, the last muffler that the team built was made of aluminum and it had broken during dyno testing at competition. Failure was likely due to thermo-mechanical fatigue, proving that aluminum would not be a reliable long-term material choice. Stainless steel has much better resistance to fatigue, and so it is the material that the 2016 muffler is made with. The muffler wall thickness is 0.050” and is 23% lighter than 0.065” stainless steel sheet, which is typically used for mufflers.

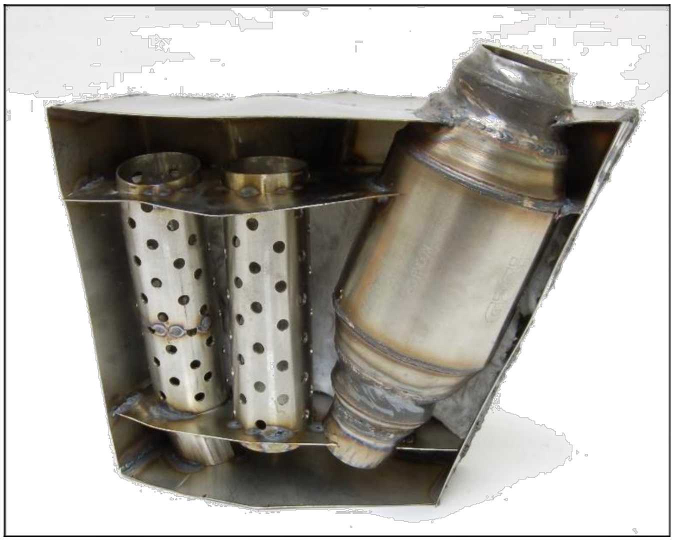


Figure 11: View of internal of baffle tubes, converter and chambers

The internal structure of the muffler is designed to attenuate a range of frequencies while maintaining minimal backpressure. There are three separate chambers within the muffler which allow sound waves to diffuse against each other, as well as two pipes with a pattern of 0.375” diameter holes that diffract sound waves. The general arrangement of baffle plates and tubes in the muffler is very common in automotive systems and has been shown to provide superior sound attenuation. The muffler contains Smelko Fiberfrax blanket, a ceramic fiber insulation that has excellent sound absorbing and heat reflecting properties, and is also lightweight and fireproof [6]. This absorbs much of the high to mid-range frequency sounds in the exhaust flow and also insulates the catalytic converter.

As an additional sound reducing feature, a reactive muffler “flap” valve has been added to the end of the exhaust pipe that block part of the exhaust flow and reduces noise output. The valve is a spring supported flap that stays closed when there is little or no exhaust flow and progressively opens with increased engine speed. This component is most effective for decreasing noise at idle and low speed operation.

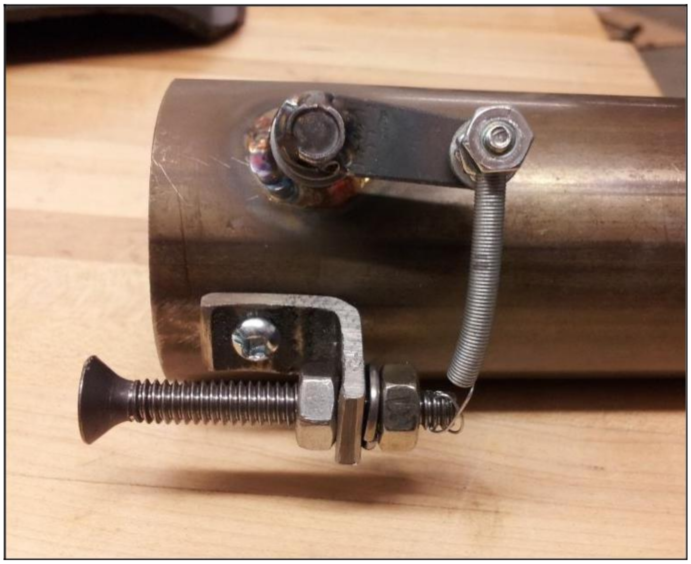


Figure 12: The muffler valve is built with an adjustable spring tension that controls the opening motion of the valve

The spring tension is adjustable, such that it can be set to open at different engine speeds, depending on rider preference, and can be left open to decrease backpressure.

The main design requirements of the muffler include size, weight and cost. In order to achieve favourable sound attenuation and backpressure properties, the muffler has been designed to mimic a similar structure to common automotive mufflers, without specifically using quantifiable parameters. Because of this design method, these two attributes are not explicitly listed in the table of design requirements for 2016, as shown in Table 7:

Table 7: Muffler design requirements and achievements

Requirement	Achievement	Pass/Fail
Fit inside body panels of the snowmobile	Final muffler fits inside body panels	Pass
Weigh less than 15 lbs with a goal of 13 including catalytic converter	Total weight: approx. 11 lbs	Pass
Cost team less than $200	Total cost: about $100	Pass

The muffler meets all design requirements for cost, weight and size. Future objectives will include quantitative sound attenuation, backpressure, and heat radiation. These such objectives will require more extensive testing and analysis of the 2016 design, which will be carried out at the competition and on campus at UW during the months following. This analysis will provide the team with important information for improving the exhaust system for future competitions.

Engine Control and Tuning

The Polaris FST engine is usually controlled using a Bosch M7.4.4 engine control unit. Unfortunately, modifying the calibration and control strategy on this ECU is beyond the means available to the UW Clean Snowmobile Team. Instead, a Motec M400 engine control unit has been selected to allow live calibration of the injection and ignition systems. The M400 also allows high-resolution logging of engine control sensor values and control outputs. A knock module has been added to replicate the original ECU's ability to retard ignition timing when knock is detected.

A Motec PDM15 power distribution module has been implemented to provide supervisory control of the snowmobile's electrical system. This has allowed rigorous testing of the supervisory logic via fault insertion and monitoring of various channel states. The PDM also allows the team to more easily diagnose electrical faults.

The ECU, PDM, throttle-by-wire controller, and dash datalogger are connected using a CAN bus, allowing distributed control of the snowmobile's systems.

Tuning from Logged Data

Due to limited availability of dynamometer time, the fuel maps have been tuned using data logged during field testing. This iterative process was conducted by recording the engine speed, manifold pressure, air-fuel equivalence ratio (λ), and engine temperature. The snowmobile was driven under varying load conditions with closed-loop fueling disabled to encounter as many load sites (speed and pressure) as possible.

Figure 13: Data collection procedure

The collected data was filtered to include only samples where the engine was fully warmed. The average air-fuel equivalence ratio was estimated for each load site configured in the fuel map and compared against the closed-loop aim table to calculate an adjustment to the base map. This process was repeated until convergence. Within three iterations of this process (around one hour), all regions of the fuel map accessible with normal driving were tuned to the target values.

Tuning using field data has had a number of substantial advantages to the team. First, field testing and calibration happen concurrently, allowing more immediate feedback of the calibration's effect on drivability. Second, chassis or drivetrain issues are identified and corrected earlier in the season. Lastly, it allows more team members to participate in the calibration process.

Flex Fuel System

Isobutanol blended gasoline was introduced as the competition fuel for 2014. Given that isobutanol's stoichiometric air/fuel ratio is lower than gasoline's, the base fueling for the engine needs to be adjusted accordingly (Figure 8). Furthermore, because the blend ratio is not fixed, the fueling must also be adjusted dynamically.

Historically, two flex-fuel strategies have been employed by teams: closed loop operation with a wideband oxygen sensor or fuel composition sensing (with or without closed loop feedback). For Waterloo's 2016 snowmobile, as in the past two competitions, the latter is used.

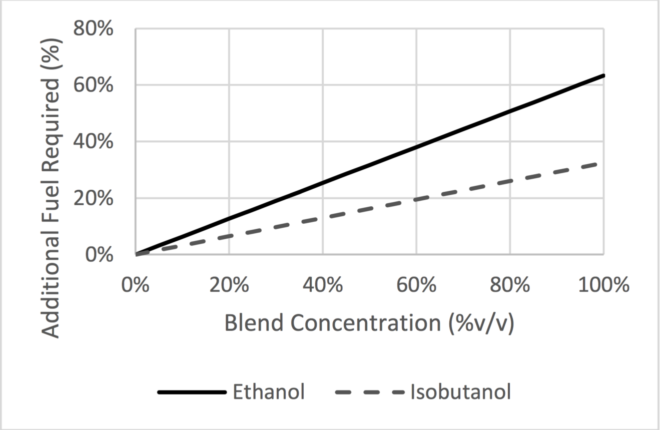


Figure 14: Additional fuel required vs blend concentration for ethanol and isobutanol

Sensing the fuel composition directly has the advantage of allowing a tighter limit on the amount of long-term compensation the closed loop system is allowed to accumulate, preventing damage in the event of a wideband sensor failure. Moreover, keeping the systems separate allows the aim lambda value to be varied based on the fuel used. This allows, for example, leaner mixtures to be used when the fuel is predicted to have a higher knock index.

Recent developments in fuel sensor technology have made composition sensing substantially more economical. Where the Siemens sensor the team has used in the past cost between CAD$300-600, General Motors part number 1357729 costs approximately CAD$80. This new sensor outputs the same signal as the previous one and thus, with the exception of requiring a different connector, is a drop-in replacement and is well-documented.

Both versions of the sensor are designed for ethanol blends and thus their applicability to butanol blends is untested. The sensor outputs a frequency-based signal based on the capacitance across an annular section of fuel which varies linearly with the composition of the fuel. It thus stands to reason that, if the dielectric constant of isobutanol is between that of gasoline and ethanol, the sensor can still be used. Moreover, if the difference in stoichiometric air fuel ratio is equivalent to the difference in dielectric constant (Table 8), the same fuel compensation table could be used and the snowmobile could run both ethanol and isobutanol blends.

Table 8: Fuel Properties

Fuel	**Stoichiometric AFR**	**Dielectric Constant**
Gasoline	14.7	2
Ethanol	9	24.55
Isobutanol	11.1	16.68

Based on the relative stoichiometry and dielectric properties of the three fuels and on the sensor output, it is evident that the sensor can be used for isobutanol. It is also evident that the system could only be adequately calibrated for one of ethanol or isobutanol as the difference in compensation becomes pronounced at blends as low as 20% ethanol or 30% isobutanol (Figure 9).

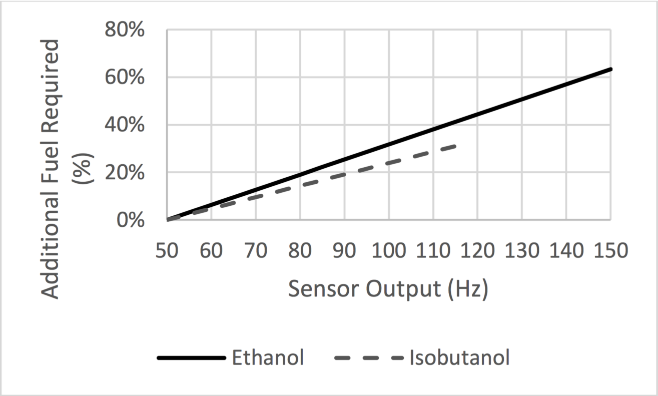


Figure 15: Additional fuel required vs sensor output for ethanol and isobutanol blends

In order to have full control of the fueling strategy, a Motec M400 engine control unit is used. Flex fuel compensations are achieved through a 3D transform table which also accounts for the change in density with fuel temperature.

Drivetrain and Chassis

Idler Wheels

Typical track idler wheels are required to rotate at considerably high speeds whenever the track is spinning. Frictional losses in bearings, assuming a constant coefficient of kinetic friction at trail speeds, are proportional to the angular speed of the idler wheel which is, in turn, inversely proportional to the diameter of the wheel. Moreover, any viscous friction in the bearing will increase with speed. The rear idler wheels are constantly subject to force applied by track tension, the weight of the snowmobile, and by transient force from acceleration. As such, the bearings of these rear wheels present the highest potential for loss-reduction. Furthermore, the track is constantly deformed as it passes over the rear wheels. Energy is converted to heat as a result of the bending strain imposed on the track. Assuming the losses are roughly proportional to the curvature (inverse of radius) of the track passing over the rear wheels, an increase in rear wheel diameter will reduce such losses. The stock rear wheel diameter is 180mm (7.125 inches). A modified diameter of 8 inches would be easily accommodated by an offset axle without requiring a longer track or any modification of the skid frame. Using the theoretical loss relations assumed above, the reduction of steady-state losses at the rear idler wheel (neglecting inertial effects) is as follows:

$$\%\, reduction \propto \left(1 - \frac{D_{old}}{D_{new}}\right)$$

Using the above formula, the percent reduction in frictional losses and bending losses for substituting 8 inch rear wheels is approximately 10.9%. The aluminum extrusion used for the Rush's slide rails is tall enough to accommodate a taller rear wheel section (the front arm mounting section is much taller). The only foreseeable added cost would be for larger rear wheels.

Figure 16: Offset axle and 8 inch rear idlers

A new offset axle was made, taking into account a number of lessons learned from the team's first attempt:

- A small amount of clearance has been added surrounding the skid rail to allow track tension to be adjusted without disassembling the unit
- Plated fasteners are used to reduce corrosion
- Flats have been added to the shaft so that the inner subassembly may be assembled more easily
- Pockets have been added to the outer blocks to reduce deflection of the tension adjusters

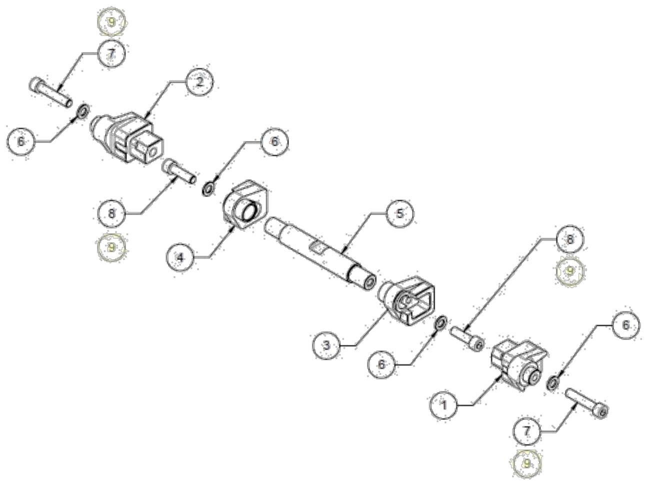


Figure 17: Offset Axle Assembly Diagram

Track and Driveshaft

The base snowmobile uses an older 2.52” pitch track, with a total length of 121” and a lug height of 1.25”. In 2014, Waterloo used a pre-studded track with the factory pitch and length and with a lug height of 1.22”. In 2015, Waterloo elected to instead use a track with a 2.86” pitch and 120” length, substituting the drive shaft for that of a 2010 Polaris Rush to accommodate the broader pitch. This lightens the track by 3 lb compared to 2014.

The motivation for using a pre-studded track is to reduce track slippage in packed snow, ice, and bare conditions. This offers improved control of the snowmobile and reduced losses to slipping the track.

Table 9: Comparison of track options

Track	**Length/ Pitch (in)**	**Lug**	**Weight**
2013 Rush 600 Pro-R (Hacksaw)	121/2.52	1.25	35
Waterloo 2014 (ICE Attak XT)	121/2.52	1.22	40
Waterloo 2015 & 2016 (ICE Attak XT)	120/2.86	1.22	37

Cost

Due to the Rush Pro-R being discontinued as of the 2015 model year, the 2016 600 Pro-S was selected as the base snowmobile with an MSRP of USD $11099. Based on the Clean Snowmobile Challenge MSRP guidelines [2], Waterloo’s modifications add USD $3671 to the retail price of a premium performance snowmobile for a total of USD $14770.

Conclusions

In summary, the University of Waterloo Clean Snowmobile Team has built on its 2015 platform to produce a four-stroke performance snowmobile with excellent potential for low emissions, good fuel economy and reliability.

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Definitions/Abbreviations

BTDC	before top dead center
ATDC	after top dead center
BBDC	before bottom dead center
ABDC	after bottom dead center
SOHC	single overhead cam
ETC	electronic throttle control
ETB	electronic throttle body
ITB	individual throttle body
WOT	wide open throttle
ECU	engine control unit