# UNIVERSITY OF IDAHO'S LOW-SPEED FLEX FUEL DIRECT-INJECTED TWO-STROKE REAR DRIVE SNOWMOBILE

**Final Report**

**KLK767**

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## National Institute for Advanced Transportation Technology

## University of Idaho

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16. Abstract

The University of Idaho's entry into the 2012 SAE Clean Snowmobile Challenge uses a Ski-Doo XP chassis with a low-speed 797 cc direct-injection two-stroke powered snowmobile modified for flex fuel use on blended ethanol fuel. A battery-less direct injection system was used to improve fuel economy and decrease emissions while maintaining a high power to weight ratio. A new tuned exhaust design was used to lower the engine speed while maintaining a peak power output of 79 kW (106 hp). Noise was reduced by running the engine at a lower speed, and by strategically placed sound absorbing materials within the engine compartment. A muffler was designed that reduced exhaust noise and improved engine emissions without greatly reducing power output. A rear drive system was designed to improve drive train efficiency, and the snowmobile entered the 2012 SAE CSC competition weighing 281 kg (620 lbs) wet, achieving 9.35 km/L (22 mpg) running on blended ethanol fuel, with an EPA five-mode emissions test score of 196 on E10, and a predicted J-192 sound magnitude score of 78 dBA.

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## EXECUTIVE SUMMARY

The University of Idaho's entry into the 2012 SAE Clean Snowmobile Challenge uses a 2009 Ski-Doo XP chassis with a low-speed 797 cc direct-injection two-stroke powered snowmobile modified for flex fuel use on blended ethanol fuel. A battery-less direct injection system was used to improve fuel economy and decrease emissions while maintaining a high power to weight ratio. A new tuned exhaust design was used to lower the engine speed while maintaining a peak power output of 79 kW (106 hp). Noise was reduced by running the engine at a lower speed, and by strategically placed sound absorbing materials within the engine compartment. A muffler was designed that reduced exhaust noise and improved engine emissions without greatly reducing power output. A rear drive system was designed to improve drive train efficiency, and the snowmobile entered the 2012 SAE CSC competition weighing 281 kg (620 lbs) wet, achieving 9.35 km/L (22 mpg) running on blended ethanol fuel, with an EPA five-mode emissions test score of 196 on E10, and a predicted J-192 sound magnitude score of 78 dBA. While a manufacturer's component failure caused a lower place finish at competition, the snowmobile team did achieve 9.35 km/L (22 mpg) running on E-34, passed both emissions and sound requirements, and won awards for the Best Written Paper, Best Performance, and Best Design.

## DESCRIPTION OF PROBLEM

Snowmobiling offers a great opportunity for winter recreation and exploration. Snowmobiles have traditionally been loud, with high levels of toxic exhaust emissions and poor fuel economy. Snowmobiles are often ridden in environmentally sensitive areas such as Yellowstone National Park where the adverse effects of snowmobiles can be substantial. To counter the potentially negative impact of snowmobiles, a partnership between industry, conservationists, and the snowmobiling community was created. As part of this partnership, a competition was created for college students to design a cleaner, quieter snowmobile. The Society of Automotive Engineers (SAE), the Environmental Protection Agency (EPA), the National Park Service (NPS), and the Department of Energy supported the effort and began the Clean Snowmobile Challenge (CSC) in 2000.

The 2012 CSC continued to encourage snowmobile development by mandating use of blended ethanol/gasoline fuel. The required blend range was from 10 to 39 percent ethanol by volume (E1X-E3X). Ethanol is a renewable fuel that has lower energy content per unit volume than gasoline but maintains a higher effective octane rating. Exhaust emissions from burning blended ethanol fuels differ from those of gasoline, typically with lower unburned hydrocarbons (UHC) and carbon monoxide (CO) quantities but elevated acetaldehydes and formaldehyde emissions [1]. Other challenges associated with blended ethanol fuels are creating flexible engine calibrations and managing the higher corrosion in the delicate fuel system components. This paper outlines the design strategies of the University of Idaho in engineering a solution that meets and exceeds industry standards for regulated emissions, improves efficiency, and maintains reliability.

## APPROACH AND METHODOLOGY

### UICSC Snowmobile Design

Engine Selection

For 2012, the University of Idaho Clean Snowmobile Challenge (UICSC) team chose to use a direct-injected (DI) 797 cc Rotax two-stroke engine mounted in a 2009 Ski-Doo XP Chassis. This selection was made based on the preferred power-to-weight ratio, better handling, and reduced cost and complexity of two-stroke engines. The characteristics that make two-stroke engines mechanically simple also result in lower thermal efficiency, poor low-load operation, and high exhaust emissions compared to four-stroke engines of similar output. Even with these drawbacks, it has been proven that a DI two-stroke powered snowmobile can meet and exceed the demands of the Clean Snowmobile Challenge [2]. A Ski-Doo E-TEC DI system from a stock 2009 Rotax 593 cc engine was used with a carbureted 797 cc engine block and the factory 797 cc E-TEC cylinder head. When the UICSC team began development on a 797 cc DI engine, a commercially available E-TEC system was not yet available, which necessitated using a 593 cc DI system. A custom DI cylinder head has also been developed, but is still in the testing and validation phase. Therefore, a factory 797 cc E-TEC cylinder head was purchased in late 2011 for use in the 2012 competition. To comply with the competition horsepower limit of 97 kW (130 HP) the 797 cc engine was tuned for a maximum operating speed of 6500 RPM.

One challenge of running a two-stroke engine at a lower maximum operating speed than originally designed is that the tuned exhaust system, commonly referred to as a tuned pipe, is operating outside the range of its effectiveness. A tuned pipe on a two-stroke engine is designed so that over a desired operating range two effects take place. First, a negative pressure pulse is sent back to the combustion chamber by the diverging section in the tuned pipe to help scavenge the spent exhaust gases from the cylinder as well as pull in fresh charge from the crankcase. This can be seen in Figure 1. The green area in Figure 1 shows the incoming fresh charge moving from the crankcase to the combustion chamber while the grey area shows the spent exhaust gases leaving the combustion chamber through the tuned pipe.

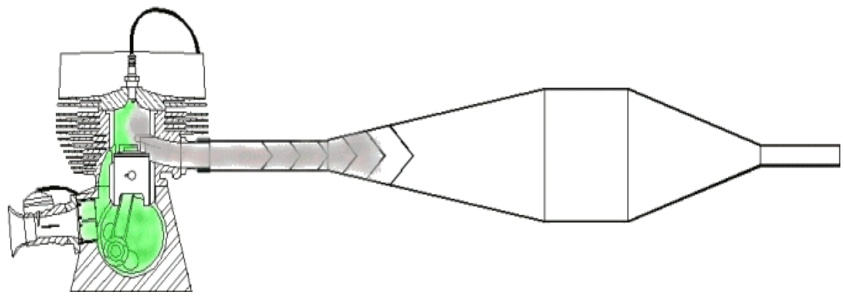

*Figure 1: Negative pressure pulse from tuned pipe*

Once the pressure waves reach the converging section of the tuned pipe, the second effect takes place. The compression of the exhaust gases through the converging section sends a positive pressure pulse back through the exhaust, pushing any residual fresh charge that may have entered the exhaust system back into the combustion chamber. This can be seen in Figure 2.

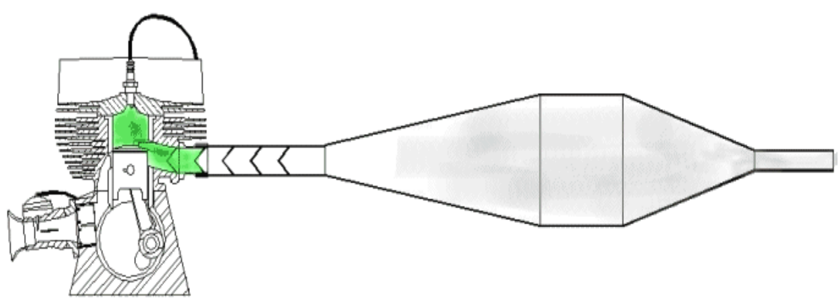

*Figure 2: Positive pressure pulse from tuned pipe*

These pressure waves travel at the local speed of sound in the exhaust system, which has a strong dependence on exhaust gas temperature (EGT). When the engine speed is lowered below the original tuned pipe design specs, the pressure pulses arrive at the combustion chamber earlier than desired, resulting in poor scavenging and low charge purity. To negate these effects, the UICSC team designed a tuned pipe to operate in a range closer to the operating range chosen. Adding length before the diverging section of pipe and in between the converging and diverging sections causes the tuned pulses to take longer to travel through the tuned pipe and enter the combustion chamber at the correct time. In order to determine the amount of length needed, a computer model was made in Matlab that used gas dynamics fundamental geometry as well as engine port and tuned pipe geometry to reach an optimum design. The results of the Matlab model were verified using equations for effective tuned length from the Jennings and Bell texts [3,4]. Figure 3 shows the correlation of the Effective Tuned Length (ETL), measured from the piston face to the end of the converging cone, for different operating speeds as a function of EGT.

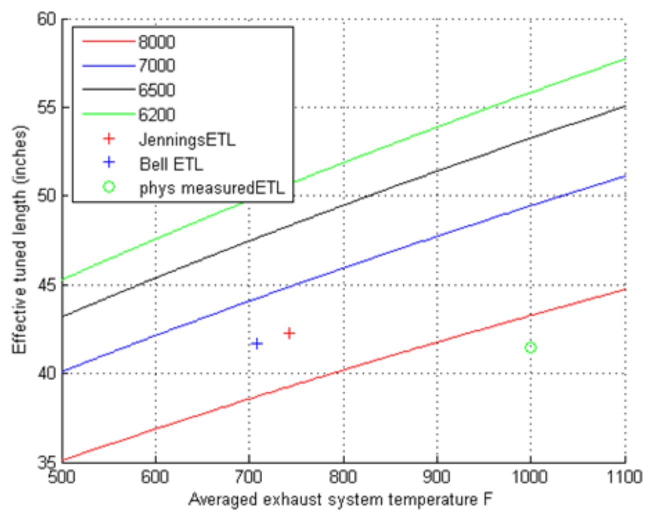


*Figure 3: Effective Tuned Length vs. Exhaust Gas Temperature*

The sloped lines correspond to different operating speeds while the physically measured ETL and EGT were used to back out an operating speed as shown by the green dot. This is higher than the expected design speed of the tuned pipe due to the assumption that the EGT measured directly after the combustion chamber was approximately equal to the average EGT in the tuned pipe. However, Jennings and Bell both suggest a mid-370°C (700°F) EGT as a first round approximation. Using this and the operating speed of 7500 RPM resulted in the red and blue points of Jennings and Bell respectively. The UICSC team determined that adding a 7.62 cm (3 in) section before the diverging cone and a 15.24 cm (6 in) section between the converging and diverging cones, for a total of 22.86 (9 in) added to the ETL, would produce a tuned pipe well matched for the 797 cc engine operating with a maximum engine speed of 6500 RPM. Figure 4 shows the completed tuned pipe.

*Figure 4: Completed tuned pipe used in the 2012 CSC competition*

When the modified tuned pipe was installed, the peak horse power from the engine at 6500 RPM increased approximately 6% from 75 kW (100 HP) to 80 kW (107 HP) as can be seen in Figure 5. Figure 5 also shows that over the 5500 RPM to 6500 RPM range the power delivery is smoother for the modified pipe vs. stock pipe. This translates into a more user-friendly ride that does not sacrifice performance.

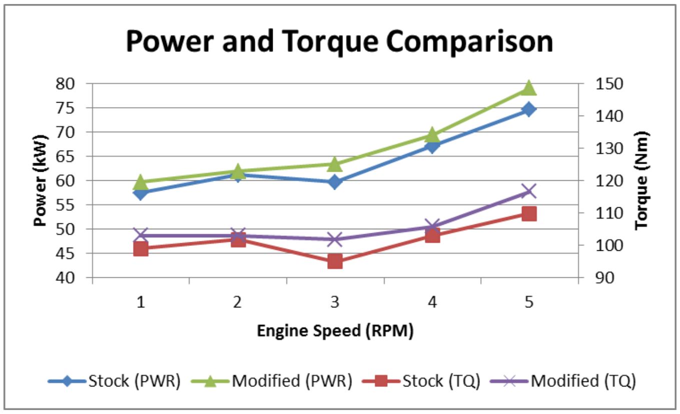


*Figure 5: Power and torque comparison between stock and modified tuned pipes*

Off-tune points still exist below the effective operating range of the tuned pipe. Operating the engine at off-tune points results in short circuiting of the unburned fresh fuel and air charge.

In previous years, the 2D RAVE exhaust valve used a two-position guillotine blade to help regulate the flow of exhaust by lowering the exhaust port height at off-tune points in the operating range. The UICSC team chose to use a 3D RAVE valve that has a three-position guillotine and additional plugs that block the exhaust transfer ports at low loads to increase volumetric efficiency. The 3D RAVE valve was first available from Ski-Doo in 2008 for the 797 cc engines. Figure 6 shows a comparison between the 2D RAVE and 3D RAVE exhaust valves.

*Figure 6: The 2D RAVE (left) and the 3D RAVE (right)*

The extra midrange position of the 3D RAVE helps to increase the engine's efficiency over a greater RPM range. Shown in Figure 7 is a comparison of brake specific fuel consumption (BSFC) between the 2D RAVE and 3D RAVE valves using E10. BSFC is a measure of how much fuel is being used per unit power and time. It is a measure of engine efficiency, with a lower BSFC resulting in more efficient power delivery. Previous comparison tests of the 3D RAVE valves showed an improvement at every mode point, with an average BSFC improvement of 16.6% on a DI 593 cc Rotax engine [5]. Similar improvements are expected on the 797 cc engine used for the 2012 competition entry. 3D RAVE valves first became available on the 797 cc engine in 2007.

Inductive Ignition System

For 2012, the UICSC team chose again to use an inductive ignition system. An inductive ignition discharges energy continuously into the fuel-air mixture as opposed to the multiple strike strategy of a capacitive discharge system. This design was chosen due to the added activation energy required for the combustion of ethanol and the added flexibility in engine calibration.

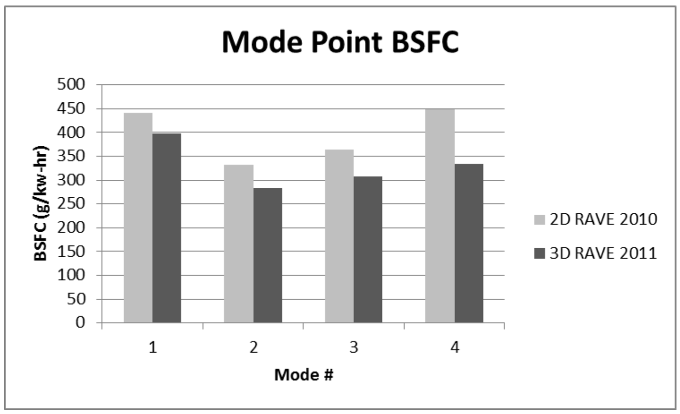


*Figure 7: BSFC RAVE valve comparison*

Oil Control and Engine Lubrication

Traditional two-stroke snowmobile engines use a total-loss oiling system. Either the oil is premixed with the fuel or the oil is pumped into the inlet-air stream where it mixes with the incoming fuel. As the fresh air/fuel/oil mixture travels through the crankcase, an oil film is deposited on the surfaces. Any oil that does not attach to a wall is scavenged into the combustion chamber. This system does not require oil filters, oil changes, or a sealed crankcase.

The 2012 UI DI engine uses an electronic total-loss oil injection system from a Ski-Doo E-TEC snowmobile. This system eliminates premixing of oil and fuel and only delivers oil to specific locations. Less oil is required in a DI engine because the oil is not diluted by fuel in the crankcase. With the precision control possible using the electronic pump, oil consumption was reduced by approximately 50% over traditional carbureted two-stroke engines.

Fuel Delivery System and Flex Fuel

Due to a SAE CSC 2012 rule requiring all spark ignition engines to be fueled with blended ethanol fuel, a major design goal for the 2012 SAE CSC competition was to tune and modify the UICSC DI snowmobile to run on a blended ethanol fuel (E1X–E3X) [6]. The properties of ethanol are much different than gasoline and require different calibration strategies to take advantage of the benefits of the higher ethanol blend fuels, i.e. the lower measured exhaust emissions and greater knock resistance. However, there are some drawbacks to using ethanol, such as increased corrosion, increased fuel flow requirements, and difficult cold starting.

In order to take advantage of the benefits of ethanol, a flex fuel system with ethanol content feedback must be provided to the Engine Management Module (EMM). For 2012, the UICSC team used a continental flex fuel sensor and a custom circuit to send information about the ethanol content of the fuel to the EMM. The EMM was then programmed to use the signal from the circuit and make adjustments to injection angle, injection quantity, and spark timing in order to take full advantage of the ethanol content of the fuel.

Blended ethanol fuel has a higher heat of vaporization than gasoline and therefore requires more energy to vaporize and mix before ignition [7]. Under temperate ambient conditions this is not normally an issue. However, when blended ethanol fuels are used in low temperature environments, such as in a snowmobile application, cold start becomes difficult due to poor atomization of the fuel. Use of the E-TEC injectors in a stratified calibration strategy has proven to offer reliable cold starting even while using blended ethanol fuel at temperatures down to –10°C (14°F).

Calibration Strategy

Engine calibration for blended ethanol fuel was completed using a Borghi & Saveri eddy current dynamometer, Lambda sensor, knock sensor, exhaust gas temperature sensors, precision fuel flow measurements, in-cylinder pressure traces, and a Horiba emissions analyzer. Because of excess air in the exhaust stream due to the nature of a DI two-stroke engine, the Lambda sensor alone was not adequate for tuning. Once the lean/rich limits were found, the Lambda sensor provided a guide to creating a smooth engine map for operation on

E1X–E3X fuel. The in-cylinder pressure trace and knock sensor was used to detect detonation while tuning and to monitor heat release rates. Emission tuning was completed using a Horiba five-gas analyzer. The strategy for engine calibration focused on BSFC and improving run quality throughout the map, followed by emission reduction at each of the mode points, without sacrificing run quality.

The emissions reduction was accomplished using optimization principles. The EPA's emissions score function was simplified to Equation 1 defined as the objective function.

$$F(x) = \frac{\left(\frac{6UHC + NO_x}{1.5} + \frac{CO}{4}\right)}{TQ} \quad (1)$$

The objective function F(x) is found using unburned hydrocarbons (UHC), oxides of nitrogen ($NO_x$), and carbon monoxide (CO) taken directly from the emissions analyzer measured in parts per million (ppm). The torque (TQ) is measured from the dynamometer. The factor of 6 on UHC comes from the use of hexane instead of methane as the calibration fuel for the emissions analyzer. Using a pattern search algorithm changing injection angle and quantity, and reducing the grid size by 50% each iteration, the objective function was minimized such that the EPA emissions score will be maximized.

## Engine Emissions

In order to compare the effects of hardware and calibration changes made by the UICSC team, a completely stock Ski-Doo E-TEC 797 cc engine running on E10 was used as a baseline for an EPA 5-mode emissions test. At each of the 5 modes, data were collected regarding the exhaust emissions, torque, Lambda, throttle position, and BSFC. These values are referred to as the "baseline" for the engine. After the baseline was completed, the UICSC muffler, and a catalyst without reactive coating, was installed and a 5-mode emissions test was run. Modifications to the baseline calibration were made to compensate for the increased flow restriction and reduced scavenging efficiency caused by the muffler. Finally, time was spent tuning each mode point using the search algorithm in order to further reduce emissions. These values are referred to as "Configuration 1." Figure 8 shows a comparison of the EPA five-mode test of the two separate configurations against UICSC's 2011 competition entry as well as the UICSC's 2007 competition entry. The 2011 engine configuration consisted of the UICSC cylinder head and 593 cc cylinders with a 3D RAVE exhaust valve running on E22 [5]. The 2007 entry consisted of the UICSC cylinder head, older cylinders with a 2D RAVE exhaust valve, an active catalytic converter from Aristo, and ran on E10 [8].The baseline and Configuration 1 use the newer 3D RAVE system running on E10.

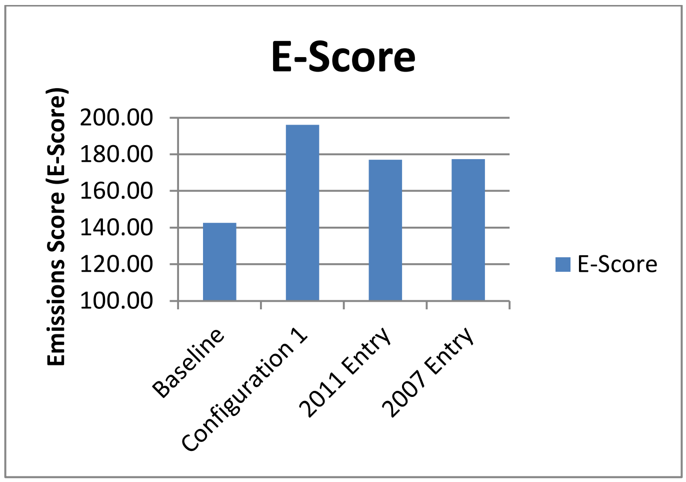


*Figure 8: Comparison of muffler and calibration effects on emissions using the EPA five-mode test*

The catalyst was a cylindrical design 8.9 cm in diameter and 11.4 cm long with 46.5 cells/cm$^2$ (3.5 in x 4.5 in, 300 cells/in$^2$) and did not have any reactive coating on the substrate. The catalyst was provided by Aristo Catalyst Technologies. The inactive catalyst acted as a thermal reactor where a second combustion event was taking place in the exhaust. This phenomenon was attributed to the large decrease in emissions and the substantial increase in the E-Score over previous years. These results led the 2012 UICSC team to not use an active catalyst and rely solely on the thermal reactor properties of the inactive catalyst. The weighted emissions, as well as an E-Score comparison, are shown in Figure 9. The 2012 competition entry produces significantly less engine emissions than previously, due to the inactive catalyst. The 2012 calibration strategy strictly focused on reducing UHC+CO as long as UHC+$NO_x$ remained below the NPS of 15 g/kW-hr. This strategy resulted in a score of 196 in the EPA emissions test, a combined UHC+$NO_x$ of 10.2 g/kW-hr, and produced less than 25 g/kW-hr of CO.

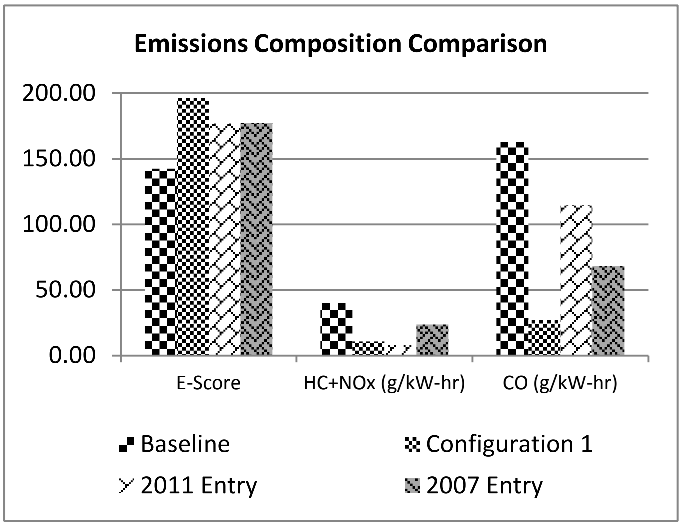


*Figure 9: A five-mode weighted emission and E-score comparison*

Before an inactive catalyst was deemed acceptable, the UICSC team researched catalyst break-in cycles in order to get more accurate emission results. A fresh catalyst has a tendency to remove far more emissions than one that has been run on an engine for an extended period of time. Therefore, the UICSC team decided to perform an experiment to test different aging techniques that would mimic engine aging practices. This would result in a catalyst aged to the equivalent of 161 km (100 miles) and allow a more accurate representation of emissions over the life of the catalyst. Two methods were used as a comparison to an engine-aged catalyst. The first method was a thermally aged catalyst in which the catalyst was placed in an oven for an extended period of time to age the substrate. The first catalyst was thermally aged for 24 hours at 760°C (1400°F) [9]. The second method used was suggested by Aristo and was done at their facility and consisted of hydrothermal aging. This process includes baking the catalyst in an oven while simultaneously running water vapor through the substrate. This test was again run at 760°C (1400°F) for 24 hours with an air/water mixture blowing through the substrate. To see how accurate the UICSC's thermal and Aristo's hydrothermal aging processes were, they were compared to a third catalyst that ran on the dynamometer for an equivalent of 161 km (100 miles).

A Horiba MEXA-584L five gas analyzer was used to collect exhaust emissions. This analyzer has an accuracy of ten percent. For each catalyst, three one minute long samples were collected at the same operating point of the engine. The experiment was run at 5500 RPM and 30% throttle on a 593 cc DI two-stroke engine. This point was chosen because it mimics a typical cruise point when trail riding. After collecting all of the data, a one-way ANOVA analysis was performed with the Tukey's method in Minitab to compare the three different aging processes.

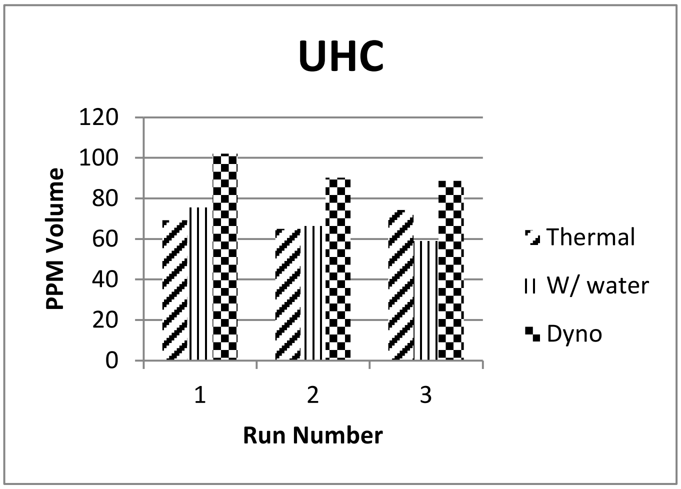


*Figure 10: Hydrocarbon emissions after catalyst aging*

The average UHC emissions are shown in Figure 10. From the Minitab analysis of the UHC emissions, the thermal and Aristo's hydrothermal aging processes were statistically the same with 95 percent confidence intervals, but they were not the same as the dynamometer aging process. The analysis of the $NO_x$ and CO emissions showed that all three of the aging processes were the same with 95 percent confidence intervals. This shows that for CO and $NO_x$ emissions, a thermally or hydrothermally aged catalyst performs statistically the same as an engine aged catalyst, but the thermal and hydrothermal aging process does not perform the same for UHC emissions. If a catalyst aging process were to be used, a thermally aged catalyst offers the greatest benefits for the UICSC team. This is because the catalyst can be aged in house without taking valuable time on the dynamometer.

## Drivetrain Design

Seeking to improve upon the fuel economy of its snowmobile, the UICSC team decided to research areas to reduce drive train losses for 2012. One area of interest was the idea of a rear drive system. On a traditional snowmobile, the track is driven from the jackshaft to a driveshaft with drivers located towards the front of the snowmobile tunnel. Figure 11 shows a traditional snowmobile drive system removed from the chassis and the location of the driver is highlighted by an arrow.

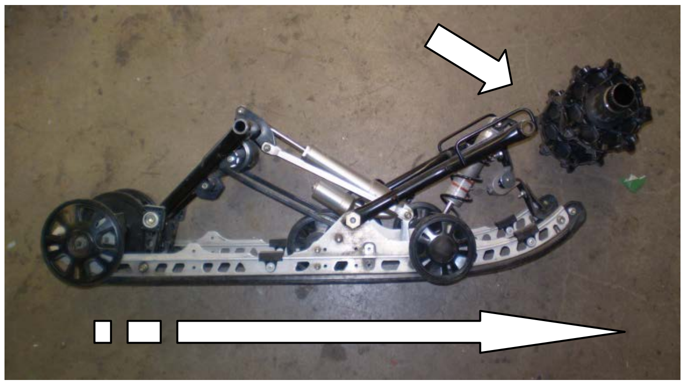

*Figure 11: Traditional snowmobile drive system with direction of travel shown*

This configuration causes the top portion of the track to be under tension while the driving surface becomes slack when the snowmobile is moving forward, resulting in wasted motion in the drive system. Another inefficiency of the traditional drive system is the weight transfer during hard acceleration and braking. Under acceleration, a snowmobile with a traditional drive system has a tendency to compress the rear suspension and extend the front suspension, reducing the contact force between the skis and the snow. This leads to poor handling characteristics under heavy acceleration and potentially a loss of control. Under braking, the rear suspension tends to extend and the front suspension compresses, leading to a weight transfer that is biased towards the skis reducing the brakes' effectiveness.

The UICSC team wanted to make a rear drive system that would be a bolt-in replacement to the stock chassis, and use as many of the stock components like the shocks and track as possible. Several designs were considered when designing the rear drive system, ranging from a gear, belt, or chain driven design. After comparing the three designs, the team decided to go with the chain drive design due to low complexity, cost, manufacturing time, and assembly time.

A fatigue analysis was performed on the front driveshaft of the rear drive to determine if it would be able to withstand the cycling stresses that are applied to it during use. The front driveshaft is designed with a fully reversed fatigue stress in bending due to the shaft rotating with a force applied by the track and chain tension. To determine the fatigue stress that was applied, a static stress analysis was performed where the stresses due to bending and torsion were taken into account to determine a principal stress for Mohr's circle (Eq. 2).

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{[\frac{\sigma_x - \sigma_y}{2}]^2 + {\tau_{xy}}^2} \qquad (2)$$

After calculating the principal stress, the fatigue analysis was performed. First the modified fatigue limit, $S_{fmod}$ (eq. 3), was calculated for a smooth shaft, taking into account surface, loading, and size effects.

$$S_{fmod} = k_{surface} * k_{size} * k_{load} * 0.5 * S_u \qquad (3)$$

Once the modified fatigue limit for a smooth shaft was known, the fatigue limit for a shaft with a keyway was calculated. To calculate this, the fatigue notch factor, $K_f$ (eq. 4), was first calculated to incorporate the stress concentration, $K_t$ , due to the keyway:

$$K_f = 1 + \frac{K_t - 1}{1 + \frac{\sqrt{\rho}}{\sqrt{r}}} \qquad (4)$$

where $\rho$ is the characteristic length and $r$ is the radius at the notch root. Then the fatigue notch factor was used in eq. 5 to determine the fatigue limit of the shaft with a keyway.

$$S_{f,keyway} = \frac{S_{fmod}}{K_f} \qquad (5)$$

The last step before calculating the fatigue life was to calculate the slope of the stress-life model. This was calculated using eq. 6.

$$B = \frac{\log[\frac{S_{f,keyway}}{S_u}]}{\log[10^6]} \qquad (6)$$

The fatigue life was then calculated using eq. 7.

$$N_f = [\frac{\sigma_1}{S_u}]^{[\frac{1}{B}]} \quad (7)$$

The plot in Figure 12 illustrates the fatigue stress vs. life difference between a smooth shaft without a keyway and a shaft with a keyway.

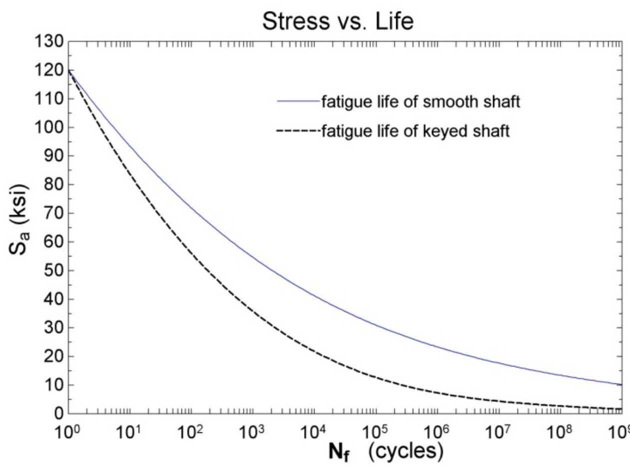


*Figure 12: Maximum cyclic stress vs. number of cycles until failure*

The principal stress that was calculated for a 3.8 cm (1.5 in) 4340 steel shaft was 14150 kPa (2052 psi), which resulted in a fatigue life of $4.292*10^8$ cycles. If the snowmobile's engine were running at 8,000 RPM, after going through the secondary clutch and the chaincase, the front driveshaft would be spinning at 3,000 RPM. At 3,000 RPM, the front driveshaft would be able to last 2384.4 hours at full throttle. Assuming at full throttle the snowmobile would have a speed of 129 km/hr (80 mph), that shaft would last 307,000 km (191,000 miles). The final design is shown in Figure 13.


*Figure 13: UICSC rear drive system*

Weight

Due to a change in the competition rules, the overall weight of the snowmobile is no longer directly awarded points. However, the UICSC team has always strived to keep their machine light for several reasons. First, a light snowmobile will achieve better fuel economy. Lower weight also improves dynamic performance and reduces rider fatigue. Lightweight is also an important aspect of marketability. As snowmobile manufactures continue to reduce the weight of their machines in response to consumer needs, the UICSC team must as well. The snowmobile entered the 2012 SAE CSC competition weighing 281 kg (620 lbs) wet. Although this is slightly heavier than a production snowmobile, the added weight was deemed acceptable to meet competition challenges. The majority of the additional weight came from the rear drive system and sound absorbing material, both of which were used to meet other competition goals.

Fuel Economy

Previously, fuel economy testing was done using a track dynamometer that could accurately control engine load and speed as well as eliminate track slippage. This resulted in inflated fuel economy numbers that did not give an accurate description of on-snow performance. For the 2012 competition, fuel economy was measured during on-snow testing to eliminate this variation. The 2012 UICSC snowmobile attained an average fuel economy rating of 9.35 km/L (22 mpg) over a 69 km (43 miles) endurance run.

Noise Reduction

For the UICSC snowmobile to be competitive in the noise event, the entire range of human hearing had to be addressed. There are four main sources of noise in snowmobiles: 1) mechanical noise emitted from the engine and drive system, 2) track and suspension noise, 3) air intake noise, and 4) engine exhaust noise.

One method for reducing sound emission in the past has been to add sound material wherever possible. This was effective in suppressing noise but added unnecessary weight to the chassis. In 2008, a test apparatus was constructed to evaluate sound deadening material effectiveness [10]. It allowed sound deadening material to be selected based on general frequencies to be attenuated. To improve on this and determine the most effective use of the sound material, coherence and impedance testing have also been implemented. Coherence testing takes an overall sound sample of the snowmobile and compares it to a local sound sample taken from locations of interest on the chassis. The test determines the percentage of sound at a frequency that contributes to the overall sound pressure level (SPL) of the snowmobile. After testing a variety of materials, a coherence test was used to determine where a material with certain properties should be placed, making more efficient use of space and saving weight. Coherence testing not only helps with sound deadening material but it also aides in chassis modifications. Knowing where the bulk of sound energy was emitted from and the difficulty of damping the sound helped us determine priority areas, making more effective use of time and sound-damping materials. Equation 8 is the general equation for coherence.

$$\gamma^2(f) = \frac{\left|G_{xy}(f)\right|^2}{G_{xx}(f)G_{yy}(f)}\left(0 \leq {\gamma_{xy}}^2 \leq 1\right) \qquad (8)$$

Mechanical Noise

There are several sources of mechanical noise. These include the clutches, chain drive, and the engine. Mechanical noise emits from the engine compartment through vibrations in the belly pan, panels, and hood as well as from vents in the hood and body panels.

Absorption and redirection were the two methods used to reduce emission of noise through body vibration. The test apparatus was used for sound deadening material properties, combined with on-snow J-192 testing, and it was found that a material consisting of various density foams and rubber with a reflective heat barrier was the most effective.

To contain and redirect noise, all hood and side panel vents that were not necessary for engine compartment cooling were sealed. Those needed were fitted with directional vents to reduce direct noise emission and maintain airflow through the engine compartment. To allow for ample airflow with substantial sound insulation, new, larger, stock panels were fitted as well as hood scoops to help force cooling air through the remaining vents. In addition to the added sound insulation room, these panels allowed for the creation of exhaust systems that would not have fit within the stock side panels.

Noise coherence testing was used to select sound-damping materials for the body panels. The UICSC team designed a testing box to determine the best sound deadening material. A piece of plastic with similar properties to that of the stock Ski-Doo XP body panels was used as a baseline. White noise was directed through the material using a 15.24 cm (6 in) speaker and a model spectrum analyzer. An accelerometer was placed on the outside of the plastic panel to determine how much of the noise generated by the speaker was causing the panel to vibrate and add to the overall noise level. The unmodified panel results are shown in Figure 14. The panel vibration accounted for 3.6% of the overall sound sample of white noise at frequencies from 0-3.25 kHz.

A piece of 3-ply sound deadening material from Polymer Technologies was applied to the panel and the experiment was run again. The results are shown in Figure 15. The results show that the damped panel accounted for 1.9% of the overall sound sample, which shows a reduction of 47% in panel vibration. The frequencies used in this study account for the fundamental and harmonic frequencies of an engine operating at 8,000 RPM. Similar trends in noise transmission were also observed at lower frequencies.

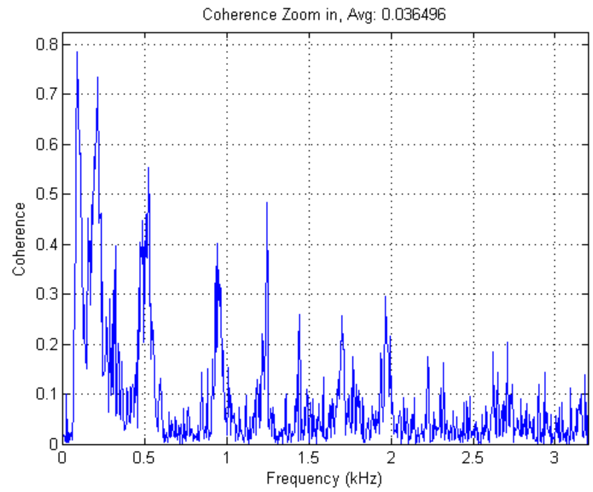


*Figure 14: Coherence of un-damped panel subjected to white noise at low frequencies*

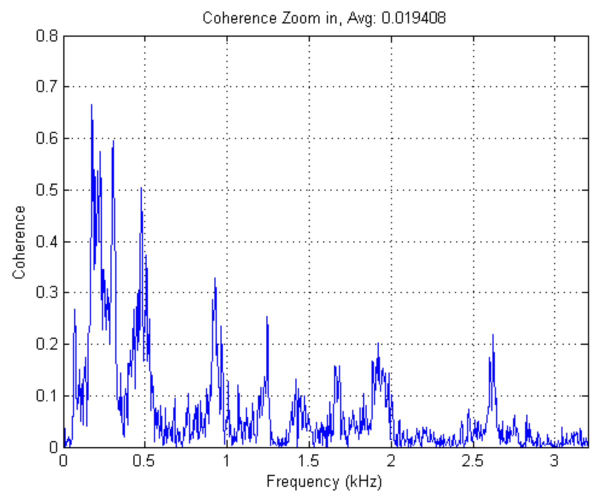


*Figure 15: Coherence of damped panel subjected to white noise at low frequencies*

Exhaust Noise

In previous years, reducing the sound of the exhaust system came through testing of different combinations of tuned pipes, mufflers, and Helmholtz resonators [5]. For 2012, the UICSC design team decided to take a two-step approach. This first step was lowering the maximum operating speed of the engine. Lowering the operating speed from 8,000 RPM to 6,500 RPM greatly reduced the overall sound level and frequency of the sound from the snowmobile.

This resulted in a lower sound magnitude (dB) on the A weighted scale used during competition.

For the second step, the UICSC team looked into a product that has been used in other branches of the power sports industry. Hushpower, a division of Flowmaster, Inc., has designed several mufflers for ATVs, motorcycles, and road vehicles. These mufflers use convergent and divergent perforated cones to direct sound while allowing exhaust gas to flow through as shown in Figure 16.

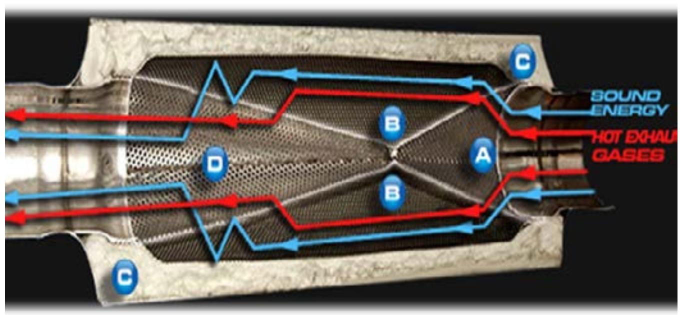


*Figure 16: Cutaway view of Hushpower muffler*

Changing the muffler on an engine can change the backpressure the engine experiences and affect the performance of the engine. A flow bench was constructed as shown in Figure 17. A flow bench is used for testing the aerodynamic performance of engine components. Its main use is for testing intake and exhaust ports on internal combustion engine heads. The device also tests the air passage qualities of air filters, manifolds, carburetors, and mufflers. The reason for testing a component would be to increase airflow for improved volumetric efficiency and reduced pumping loss. In the case of mufflers, the wrong backpressure can cause backfiring, loss of power, and in extreme cases cause the engine to stop completely. The stock muffler’s backpressure was tested along with the newly constructed muffler. Figure 18 shows the flow testing results.

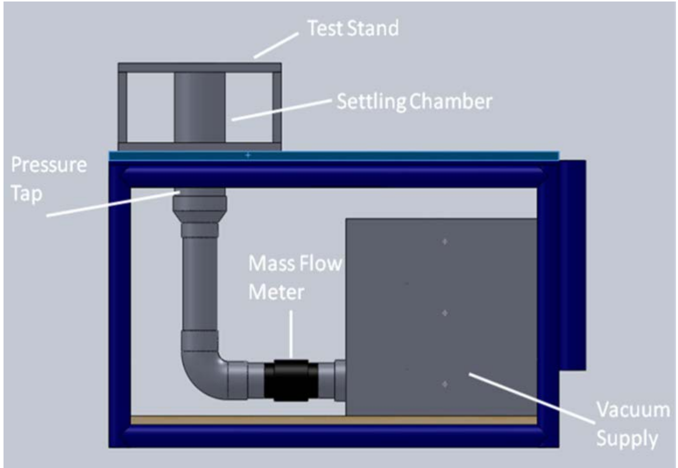


*Figure 17: Solid model of flow bench design*

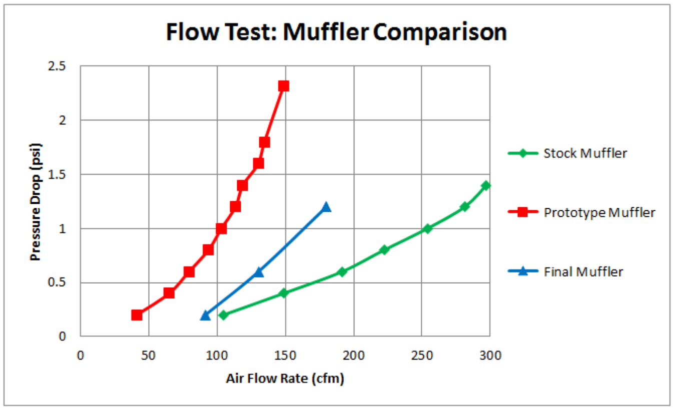


*Figure 18: Flow testing results of stock muffler and UICSC muffler*

The UICSC muffler created an increase in backpressure that negatively affected engine performance. With the UICSC muffler, the engine produced 40 kW (54 hp), while the 2012 UICSC engine and stock muffler produces 80 kW (107 hp). Therefore, for 2012 the UICSC ran a modified stock muffler that incorporated an inactive catalyst.

## FINDINGS, CONCLUSIONS, AND RECOMMENDATIONS

### Final Approach

No one method adequately reduced noise, so combinations of several methods were implemented in the final sound reduction approach for 2012. Selective sound deadening material, low engine speeds, and a modified muffler were used. Implementation of all of these methods yielded an average score of 79 dBA using the SAE J-192 procedure. Previous clean snowmobile challenges have shown that UICSC's sound pressure meter consistently reads 2 dBA higher than the Head Acoustic products used at competition. Therefore, a score of 77 dBA was expected at competition.

### MSRP

The base price for a stock 2012 Ski-Doo MX-Z 800 E-TEC is $12,099. With all modifications included, the manufacturer's suggested retail price (MSRP) of the 2012 UICSC DI, totaled $14,632. Chassis components that add to the MSRP were justified by sound absorption, increased performance, and sponsor product awareness. The price of rear drive components totaled $560, including material, sprockets, and chains. The exhaust modifications totaled $150, which includes an inactive catalyst and heat shielding.

### Summary and Conclusions

The University of Idaho has developed a cost-effective flex fuel DI two-stroke snowmobile engine capable of running on E1X–E3X blended ethanol fuel. The DI two-stroke snowmobile maintains the mechanical simplicity and low weight avid riders enjoy, without sacrificing the clean and quiet characteristics necessary to meet current and upcoming standards. The UICSC design produces 80 kW (107 hp), is lightweight at 281 kg (620 lbs) wet, and achieves a fuel economy of 9.35 km/L (22 mpg). Overall, sound production measured using the SAE standard J-192 was reduced from 82 dBA to 78 dBA. Since a measurement of 77 dBA was anticipated, the performance was close to that predicted. With future regulations coming for manufacturers, consumers will expect clean and quiet snowmobiles. However, increased fuel economy, a better power-to-weight ratio, and a general enjoyable riding experience are important characteristics that consumers demand. The 2012 UICSC low-speed flex fuel DI two-stroke snowmobile is an economical response to that demand.

At competition, the UI snowmobile achieved an emissions testing E-score of 125. Fuel economy measured during the emissions event was second best of the competition.

A manufacturer’s component failure caused a lower place finish at competition. The snowmobile team achieved 9.35 km/L (22 mpg) fuel economy running on E-34, passed both emissions and sound requirements, and won awards for the Best Written Paper, Best Performance, and Best Design.

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## DEFINITIONS/ABBREVIATIONS

**SAE** Society of Automotive Engineers

**EPA** Environmental Protection Agency

**NPS** National Park Services

**CSC** Clean Snowmobile Challenge

**UHC** Unburned Hydrocarbons

**CO** Carbon Monoxide

**$NO_X$** Oxides of Nitrogen

**UICSC** University of Idaho Clean Snowmobile Challenge

**DI** Direct-Injected

**ETL** Effective Tuned Length

**EGT** Exhaust Gas Temperature

**BSFC** Brake Specific Fuel Consumption

**EMM** Engine Management Module

**MSRP** Manufacturer's Suggested Retail Price