20XX-01-XXXX

Design and Validation of a Custom Muffler

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Abstract

A custom muffler was designed for the University of Idaho Clean Snowmobile Challenge team to meet National Park Services standards for noise emissions. A flow bench and an anechoic sound box was used to conduct research on muffler design. Initial designs were simulated through SolidWorks Flow Simulation and Sidlab to compare results. The final muffler was manufactured, validated, tested, and implemented on the University of Idaho's clean snowmobile. At the Clean Snowmobile Challenge the muffler was able to reduce noise emission to 5 dBA below National Park Services noise standards and place third quietest snowmobile overall.

Introduction

The exhaust system is a critical part of engine performance and a major contributor to overall noise. This paper discusses the process of designing a muffler for a two-stroke snowmobile. However, the principals can be applied to any combustion vehicle muffler. The designed muffler was used by the University of Idaho Clean Snowmobile team (UICSC) on the Rotax 800cc E-Tec engine, as a step towards meeting National Park Service (NPS) standards for noise emissions. NPS standards say that snowmobiles can only be allowed into national parks if the sound is below 67 + 2 dBA and currently has not been met by any original equipment manufacturer two-stroke platform.

Design Criteria

The UICSC team required a muffler that would meet NPS standards, have minimal impact on engine performance, and be able to withstand extreme temperatures from a catalyst. In order to meet these goals a table of requirements was set by the muffler design team, shown in Table 1, to provide parameters for the project. The parameters were baselined off of the stock mufflers performance and characteristics.

Table 1. Design Specifications

	Goal	Acceptable
dBA Reduction	4	2
Peak Engine Power	89.5 kW (120 hp)	82.0 kW (110 hp)
Weight	9.1 kg (20 lbs)	11.3 kg (25 lbs)
Flow Rate	0.1 m^3/s (3.5 cfs)	0.09 m^3/s (3.3 cfs)
Back Pressure	4.34 kPa (0.63 psi)	5.03 kPa (0.73 psi)
Cost to Manufacture	$300	$500
MSRP	$650 (Stock)	$850
Size	40.6 cm x14.0 cm x 33.0 cm (16"x5.5"x13")	Has to fit
Robustness	982°C (1800°F) withstand winter conditions	982°C (1800°F) withstand winter conditions
CSC Competition	Meet NPS Standards, Quietest Snowmobile	Meet NPS Standards

Initial Research

Research was conducted to understand the acoustic and performance characteristics that influence the design of a muffler. The design team decided to conduct research in the areas of sound absorption material, attenuation through muffler geometry, and backpressure. As a starting point the design team looked at industry muffler design.

Industry Design

There are multiple muffler manufacturers in industry that use three techniques in muffler design. The first being an absorptive style muffler where only sound absorption material is used to convert sound energy into heat [1]. Second, a reactive style muffler uses only wave cancellation principles in the form of expansion chambers, chambers with plates, extended tubes, perforated tubes, or resonators. Lastly, a combination muffler uses both techniques. Typically, combination mufflers are the most common because they are able to effectively target a wide range of frequencies as well as produce minimal backpressure [2]. The design team decided to build a better understanding of muffler design by breaking the methods into three individual components: sound material, geometry, and backpressure.

Material Selection

When selecting a material to be used, sound material fiber size, acoustic characteristics, durability, chemical resistance, and temperature resistance need to be taken into consideration [3]. Materials that are porous make good absorbers while nonporous materials reflect sound waves back into the exhaust stream. The Knudsen number (Kn), given in Equation 1, determines the sizes of the pores and the materials ability to absorb sound.

$$Kn = \frac{l_{mfp}}{l_{char}} \quad (1)$$

l_{mfp} is the average distance an air molecule can move before colliding with another and l_{char} is the average distance between pore walls within the material. This means that the smaller the Knudsen number the better absorber it will be. An example would be ceramic foam being a better absorber than aerogel, as seen in Table 2 [3].

Table 2. Knudsen numbers for various porous materials

Material	l_{mfp}, m	l_{char}, m	Kn
Ceramic foam	7×10^{-8}	$1\text{x}10^{-3}$	$7\text{x}10^{-5}$
Metal foam	$7\text{x}10^{-8}$	$2\text{x}10^{-4}$	$4\text{x}4^{-4}$
Aerogel	$7\text{x}10^{-9}$	$5\text{x}10^{-8}$	$2\text{x}10^{-1}$

Sound material is most effective at attenuating high frequencies, but also has a minor effect on the lower frequencies. Noise is significantly attenuated when the material is one tenth the wavelength [3]. Although, the thicker the sound material the greater the attenuation.

$$c = \sqrt{\gamma * R * T} \quad (2)$$

$$\lambda = \frac{c}{f} \quad (3)$$

Figure 1 represents the thickness of sound material required to reduce the targeted frequency. First, the speed of sound, c, was calculated using equation 2, given an average exhaust temperature, T, of 538 °C

(1000 °F). Using the one tenth assumption, the sound material thickness was adjusted from one to fifteen inches and multiplied by ten to get the targeted wavelength, λ. The frequencies that would be affected were calculated by equation 3.

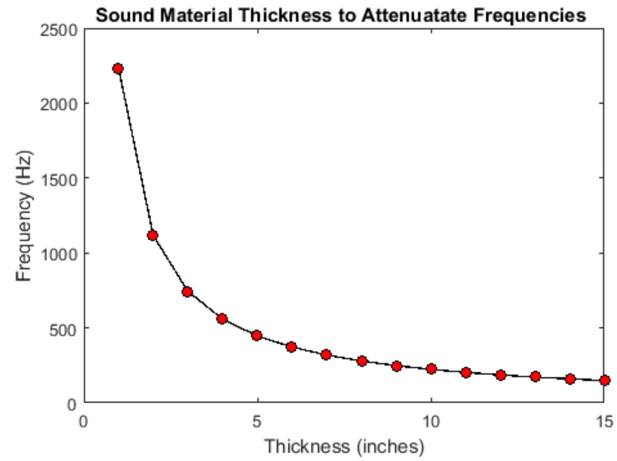

Figure 1. Sound material frequency attenuation

Besides good acoustic characteristics, the ability of that material to withstand high temperatures produced by the catalyst [up to 982°C (1800 °F)] and robustness were taken into consideration. For this purpose, a number of materials were selected for further testing and will be discussed in a later section.

Acoustics and Geometry

Acoustic impedance can be created using wave cancellation principles. Wave cancellation occurs when a wave is reflected out of phase from the initial wave causing destructive interference, represented in Figure 2 [4]. When this happens, the sound wave is partially or completely attenuated depending on the amplitude and phase of the reflected wave.

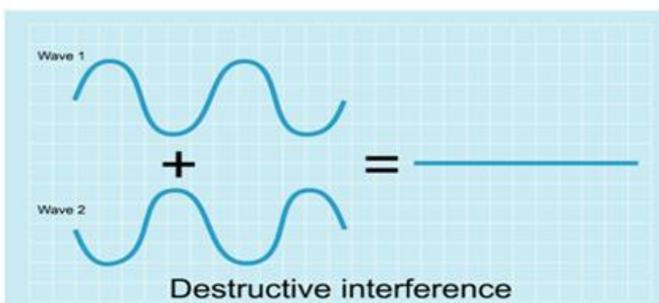

Figure 2. Sound wave destructive interference [5]

Expansion chambers can be used to attenuate sound. The volume and cross sectional area of these chambers are the primary influences of the noise attenuation. As the cross sectional area of a flow is changed, a reflection wave is sent back into the initial stream and an acoustic impedance occurs. Larger chambers attenuate lower frequencies, but have larger natural resonance. While smaller chamber attenuate higher frequencies and have smaller natural resonance [6].

Helmholtz and quarter wave resonators are a form of destructive interference and can be designed to attenuate noise at a specific frequency. Helmholtz resonator's frequency is based on the entrance length, entrance cross section area, and volume of the closed chamber, while quarter wave resonators are based solely on pipe length.

Another attenuation method is geometric interference that uses "V" shaped, "C" shaped, or straight plates that reflect sound waves.

Backpressure and Muffler Design

The performance of a two-stroke engine is highly reliant on the backpressure of the exhaust system. Backpressure is the resistance to flow. There is a strong correlation between back pressure and noise attenuation [4]. This means typically performance suffers from a decrease in noise emissions. For maintaining the performance and minimizing calibration time, a muffler redesign should have similar pack pressure at various flow rates compared to stock.

Testing Equipment

The UICSC developed testing equipment and procedures to gain an empirical understanding of fluid flow and acoustics. The two apparatuses developed were a flow bench and anechoic sound box.

Anechoic Sound Box

To improve understanding of sound attenuation, the UICSC team manufactured an anechoic sound box testing apparatus, referred to as the UI sound box (UISB). The initial design of the UISB was based on an existing design, which was used to test the acoustic effectiveness of quarter-wave and Helmholtz resonators [7]. The anechoic sound box was designed to emit pure frequencies through a waveguide (pipe) without interference. The final UISB specifications are given in Table 3.

Table 3. UI Sound box specifications

Speakers [8]	15.88 cm (6.25 in)
Tweeters [8]	4.60 cm (1.81 in)
UI Sound Box	.23 m^3 (8.00 ft^3)
Waveguide Diameter	5.08 cm (2.00 in)
Waveguide Length	4.06 m (13.33 ft)
Studio-foam [9]	10.16 cm (4 .00in)
Audio Amplifier [10]	1100 W
Microphone Locations [11]	45.72 cm (18 .00 in), 137.16 cm (54.00 in), 289.56 cm (114.00 in), 381.00 cm (150.00 in)

The box contains amplified speakers and tweeters acting as a sound source directed into the UISB. The housing was built using 1.91 cm (0.75 in) high density fiber board internally lined with the studio-foam. Attached to the sound box is the waveguide, which extends to the environment with a removable center section. This section in the center of the pipe is removable for the purpose of testing individual acoustic components. The component length was held constant so the waveguide spanned the same distance for each test. Four microphones were placed along the pipe at the specified locations from the outlet of the UISB. The microphone's signals were analyzed using a Digilent Electronics Explorer board. Figure 3 represents the final configuration of the UISB.

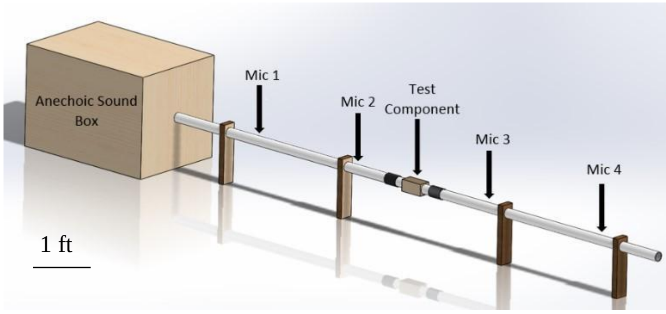

Figure 3. UICSC sound box configuration

The minimum testable frequency was calculated to be 80 Hz using equation 3 while the maximum, 4000 Hz, was found using equation 4. These equations are the wavelength and narrow pipe assumption, respectively. Wavelength is the distance between two peaks of a sound wave and constrains testing due to waveguide length.

$$\lambda = 1.71 * D \tag{4}$$

When acoustic impedance is added to the travel path of the sound wave, there is a chance for a reflected wave to occur. Within this paper, acoustic impedance represents barriers, sound absorption materials, or geometric changes. Two microphones were placed before and after the test component to account for the reflected wave [12]. Under ideal testing the distance between each microphone would be equivalent to the wavelength of the tested frequency. Measuring sound phase makes moving the microphones extraneous.

Testing Procedure

Each test was performed by sweeping frequencies from 80 Hz – 4000 Hz scaled logarithmically. The pressure and phase were recorded three different times and averaged, allowing the UICSC team to calculate the power transmission coefficient. Due to the complexity of the derivation, this equation is not shown. The power transmission coefficient is interpreted as a percentage of the sound power that is reduced due to a material or geometric change. A positive value represents a reduction in sound and a negative value an increase. These increases may come from a natural resonance of the sound chamber, waveguide, or test component [7]. For example, a transmission power coefficient of .8 indicates that 80% of the sound power is reduced while 20% passes through the component. Using the UISB, the transmission coefficient for any change of impedance can be measured meaning muffler chamber components can be tested either individually or in series.

Flow Bench

In order to test the back pressure produced by the stock muffler and individual muffler components a flow bench apparatus was used. The flow bench was developed by a previous University of Idaho Capstone senior design team [13]. As shown in Figure 4, the flow bench consists of five vacuum pumps that switch on individually and one pump with a variable switch that allows for flow rates up to 0.14 m^3/s (50 cfs).

Upon initial usage of the flow bench the calibration between flow and voltage reading was inaccurate. This led to a recalibration of the flow bench.

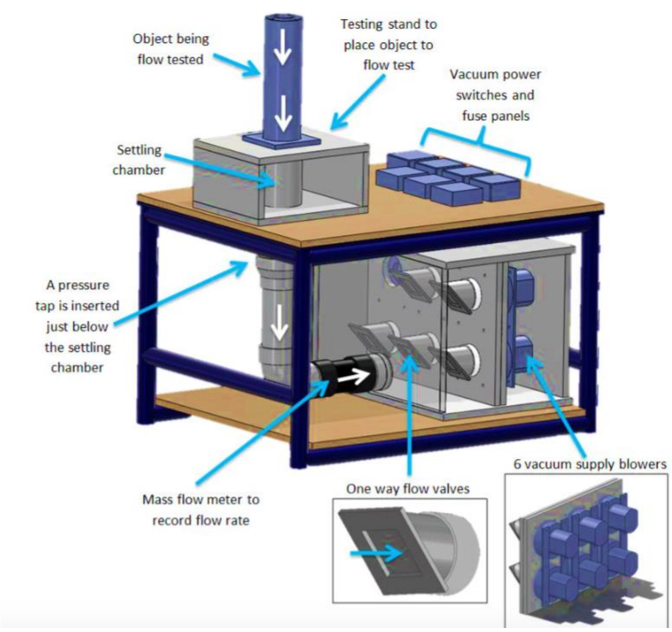

Figure 4. Flow bench test apparatus

Calibration

To calibrate the flow bench, a Superflow air flow turbine was placed on top of the bench in the component test position (see Figure 4). Using a linear calibration provided by the manufacturer, shown in Figure 5, a best fit equation was calculated, given in equation 5.

$$y = 1.0071 * x + 8.2024 \qquad (5)$$

The flow was correlated to each voltage by using equation 3 and plotted in Figure 6. This voltage was read from a Supra mass air flow (MAF) sensor on the flow bench. Twenty different flow rates were tested and recorded. The final calibration equation relating the voltage output to the flow rate is seen in equation 6.

$$y = 4.6345 * x^{3.0784} \qquad (6)$$

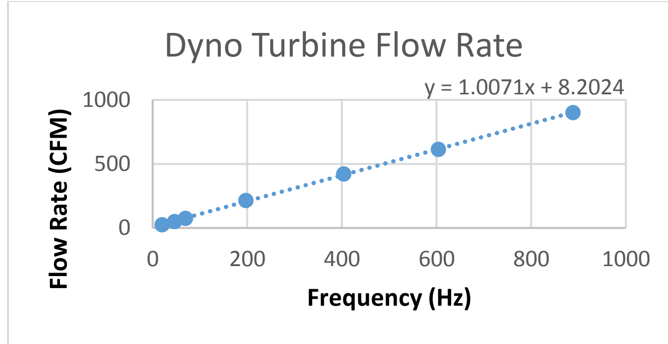

Figure 5. Wind turbine linear calibration

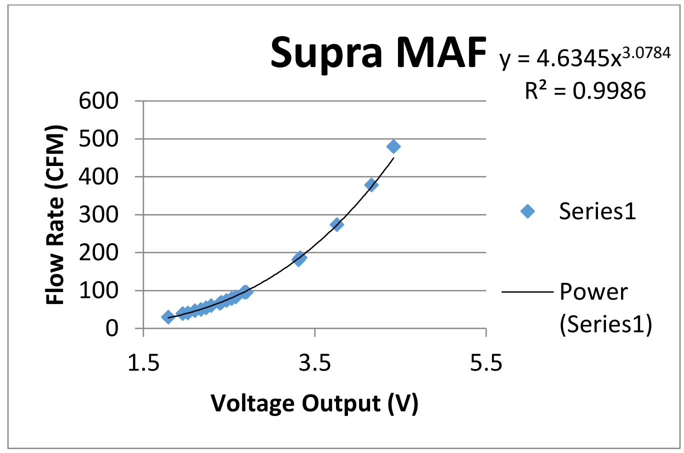

Figure 6. Calibrated flow rate of voltage output

Testing Procedure

The five typical points of operation during riding conditions were used to test the back pressure after the calibration. The RPM and displacement of the engine was used to solve for the flow rate, shown in equation (7). This assumed 100% volumetric efficiency meaning that the throttle would be fully open and no scavenging, resulting in the largest amount of flow rate the muffler would ever experience.

$$Q = RPM * N_c * Volume\ of\ cylinder \qquad (7)$$

Calibration Equipment

To validate and calibrate for the effects caused by the muffler on the engine a Borghi & Saveri eddy-current dynamometer, model FE-260-S, was used. Fuel delivery was controlled and measured by a custom fuel cart. Emissions data were collected by a Horiba MEXA-584L 5-gas analyzer. An Innovate LM-2 wideband Oxygen sensor was used to measure air fuel ratio. A water brake dynamometer was used to test the calibration in similar conditions to those at the Clean Snowmobile Challenge (CSC).

Baseline Testing

To gain a baseline for attenuation properties and back pressure the

2013 and 2017 Ski-Doo stock mufflers were tested using the UISB and flow bench. This provide empirical data on the UICSC test benches that would be compared to the prototype muffler.

Muffler Acoustic Characteristics

Using the UISB testing procedure the two mufflers were tested and the results are shown in Table 4. The table shows the power transmission coefficients for each of the tested components at frequencies of 150, 500, 1250, and 2000 Hz. These frequencies were chosen by inspecting FFTs of the snowmobile exhaust noise during cruising speeds. 150 Hz is the firing frequency of the engine at 56 kmh (35 mph), found using equation 8. It is also important to analyze the results around each of these frequencies to ensure the attenuation covers a range of frequencies.

$$f = \frac{n_c * RPM}{n_R * 60} \tag{8}$$

Table 4. Stock muffler sound test results

		Frequency [Hz]				
Configuration		150	500	1250	2000	Avg.
Stock Mufflers	2013 Ski-Doo Muffler	0.309	0.034	0.371	0.135	0.212
	2017 Ski-Doo Muffler	0.390	0.007	0.043	0.092	0.133

Muffler Back Pressure

The two mufflers were tested on the flow bench using the five test points from above. The 2011 Capstone design muffler was also tested because this muffler caused engine run ability problems due to high levels of back pressure. The results are shown in Figure 7 respectively. This showed that the prototype muffler needed to have 4.14 kpa (0.6 psi) of back pressure at full load wide open throttle to have little effect on engine calibration.

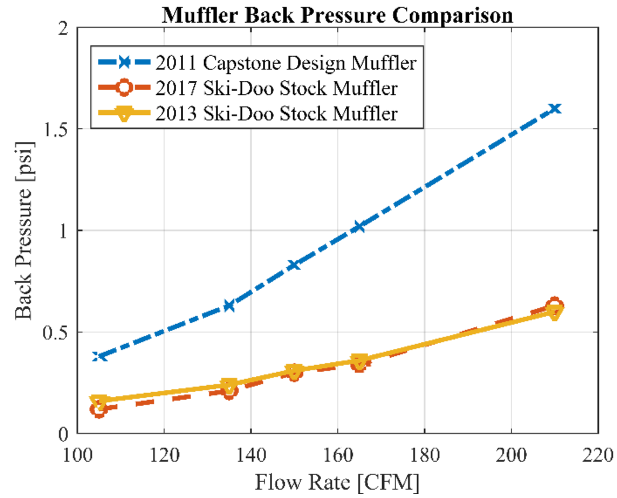

Figure 7. Muffler back pressure comparison

Modeling

Expansion chambers and other noise attenuating geometries affect the flow rate of the exhaust gases. To better understand the fluid flow of muffler components and designs, the geometries were modeled in SolidWorks and a flow simulation was conducted. This flow simulation had an inlet volumetric flow rate of 0.1 m^3/s (3.528 cfs), calculated using equation 1 above, with inlet temperatures set to 538°C (1000 °F). Using the flow simulation, several muffler components were modeled and the back pressure measured. An example of these simulations is shown in Figure 8.

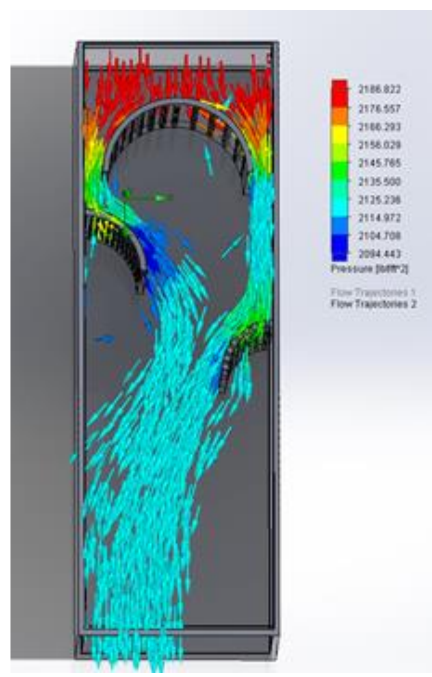

Figure 8. Flow simulation of deflection plates in a chamber

The arrows represent the flow path while the color represents the pressure. Red is the area of maximum pressure and blue is the area of lowest pressure. Several of these simulations were then compared to those tested on the flow bench to see to the correlation between simulations and the real world with the comparison results given in Table 5.

Table 5. Peak back pressure versus simulation results

Test Component	SolidWorks	Flow Bench
Simple Expansion	0.006 bar (0.09 psi)	0.02 bar (0.23 psi)
2013 Ski-Doo Stock Muffler	0.03 bar (0.39 psi)	0.04 bar (0.63 psi)

The average difference between the SolidWorks simulations and flow bench measurements was 58%. This was due to the simulations being calculated based on steady flow while the flow bench utilizes vacuum motors that create pulses. With the back pressure for different components being simulated in SolidWorks, it was not necessary to build and test every muffler possibility on the flow bench.

A model of the 2013 Ski-Doo stock muffler, was built and tested in SolidWorks, shown in Figure 9. As seen in Table 5, the back pressure from this flow test was found to be 0.39 psi. This became a target point back pressure for all muffler designs to ensure performance restrictions were met.

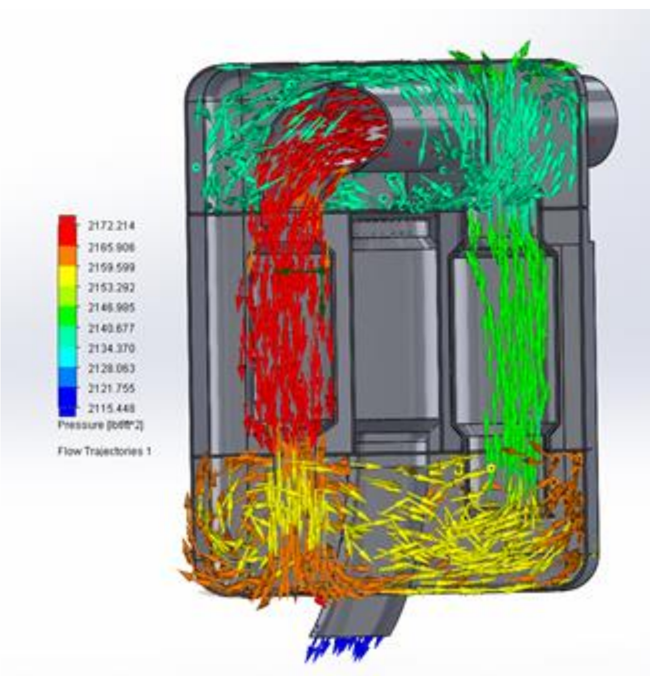

Figure 9. 2013 stock muffler fluid flow

Testing Results

The testing strategy used within this paper included first testing components individually and then taking the best components and testing them in combined configurations. The components and configurations that were built and tested are listed below.

- Different sized, same cross sectional area expansion chamber
- Same volume, different cross sectional area expansion chamber
- Varying entrance and exit locations in expansion chamber
- Same hole size, different hole density perforated sheet
- Different hole size, same hole density perforated sheet
- Beveled pipe (bellmouth) on inlets and outlets
- Six different high temperature sound absorption material
- Dense versus loose packed sound material
- Geometric interference in the form of “V” and “C” shaped plates
- Helmholtz/quarter-wave resonators
- Expansion chambers in series orientation
- Expansion chambers in parallel orientation

Acoustic Impedance Devices

All the items above were tested using the UISB to see the best attenuation devices. The results of the selected components, which represent a small portion of those tested, are shown in Table 6. The underlined red components were used in the final muffler design.

Table 6. Transmission coefficients from individual component tests

		Frequency [Hz]				
Configuration		150	500	1250	2000	Avg.
Expansion Chamber Size [cm x cm x cm (in x in x in)]	**7.6x7.6x20.3 (3x3x8)**	0.008	0.828	0.491	-0.194	0.283
	10.2x10.2x10.2 (4x4x4)	0.296	-0.179	0.518	0.01	0.161
	10.2x10.2x30.5 (4x4x12)	0.287	0.522	-0.037	-0.372	0.100
	15.2x15.2x20.3 (6x6x8)	0.211	-0.31	0.055	0.381	0.084
Sound Material with Weight [g (oz)]	**Fiberglass Blanket 25 (0.89)**	0.062	-0.014	0.135	-0.565	-0.096
	Fiberglass Blanket 50 (1.76)	0.627	0.765	0.073	-0.177	0.322
	Ceramic Blanket 25 (0.89)	0.382	0.234	0.114	-0.343	0.097
	Ceramic Blanket 50 (1.76)	0.367	0.236	0.121	-0.165	0.140
Perforated Hole Density	1/4 in 22%	0.256	0.299	0.227	-0.157	0.156
	1/4 in 40%	0.247	0.119	0.014	-0.13	0.063
	1/4 in 58%	0.288	0.358	0.682	-0.207	0.280
Geometric Interference	Round Same Size	0.316	0.238	0.332	0.026	0.228
	V Same Size	0.48	0.113	0.339	-0.000	0.233
	Round Descending Size	0.319	0.161	0.191	-0.081	0.148
Three Expansion Chambers Placed in Series [1=7.6x7.6x20.3 (3x3x8), 2=15.2x15.2x20.3 (6x6x8)]	**1,1,2**	0.265	0.157	0.041	-0.188	0.069
	1,2,1	0.243	-0.087	0.163	-0.469	-0.038
	2,1,1	0.133	0.206	-0.1	0.084	0.081
Bell Mouth Extension Tubes	Intake	0.424	-0.557	-0.063	-0.473	-0.167
	Outlet	0.216	0.267	0.013	0.165	0.165
	Both Intake and Outlet	0.618	-0.244	-0.298	0.165	0.060

Back Pressure

Table 7 shows the results from testing the back pressure of the five different expansion chamber components. It was determined that it was a combination of the cross-sectional area and volume that contributed to the backpressure of the expansion chamber. As the length of a chamber with the same cross-section increases so does the back pressure.

Table 7. Back pressure for expansion boxes

Chamber Size	Back Pressure (psi)
3x3x8	0.14
4x4x4	0.19
4x4x8	0.22
4x4x12	0.3
6x6x8	0.23

Initial Muffler Design

Mufflers implementing the concepts learned from the UISB tests and flow simulations were designed within SolidWorks. The idea was to include within the design the muffler components that maximized sound loss while limiting the back pressure to 16% greater than that of the 2013 stock muffler. These designs were also restricted to the size available within the snowmobile chassis. Other areas of consideration within muffler design beyond performance included manufacturability and cost of materials.

Design Procedure

The following strategies contributed to the largest reduction in sound: expansion chambers with larger cross-sectional areas, ordering expansion volumes from small to large in series, “V”-shaped geometric interference plates of the same size, extension tubes on the outlet to chambers, a long internal flow length, and the use of ceramic fiber sound material. Using this information, the UICSC team designed six mufflers that were simulated through Solidworks. An example of one design is seen in Figure 10.

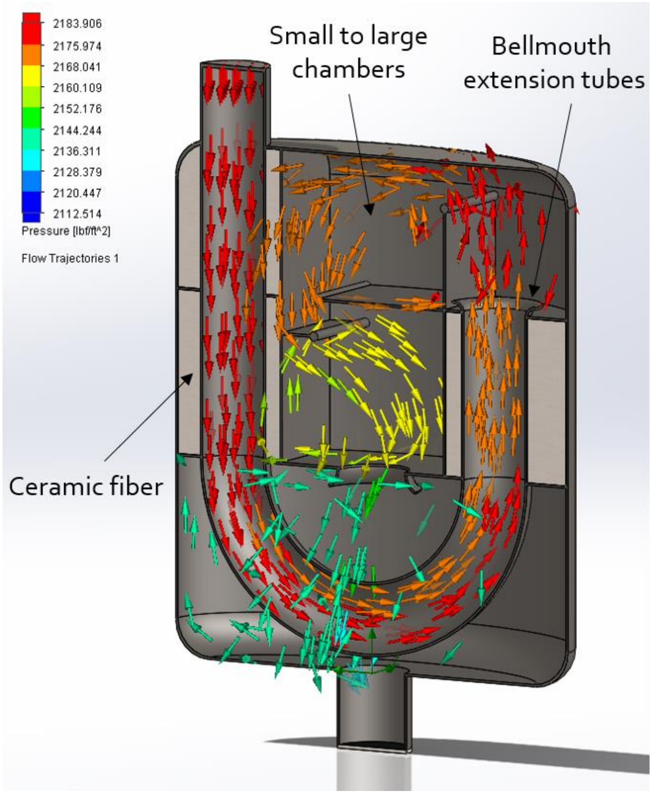

Figure 10. Red Dawn (AM)^2 flow simulation

The top three models were selected through a design matrix comparing flow, potential acoustic ability, cost, and manufacturability. These selected models were acoustically simulated within Sidlab to determine the transmission loss of each model.

$$TL = 20 * log \frac{P(1)}{P(r)} \tag{9}$$

Equation 9 represents transmission loss where P(1) and P(r) are the pressure amplitudes measured at distances of 1 m (3 ft) and r [6]. The distance r used in the Sidlab simulation covers the distanced travel through the tuned pipe and muffler equaling 2.4 m (8 ft). All testing within Sidlab was completed assumed 56.3 kmh (35 mph) at 30% throttle. This gives an accurate representation of typical snowmobile cruising speeds. The results for the Sidlab simulations are shown in Figure 11 and Appendix A. A positive transmission loss signifies the amount of decibel reduction achieved through the use of the muffler.

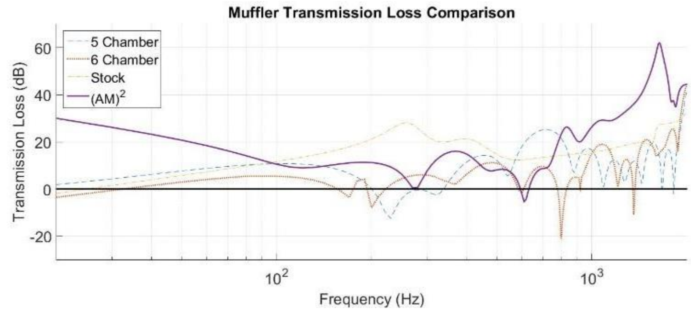

Figure 11. Muffler Sidlab results

Final Design Selection

The acoustic results from Sidlab had a major impact on the design chosen. The UICSC muffler design is called Red Dawn (AM)^2. The Red Dawn (AM)^2 muffler performed better than stock at higher frequencies, which are weighted higher in the A-weighted scale, as well as performing better at the lower frequencies [6]. Though the backpressure was higher than stock, it was deemed to be within an acceptable range. Additionally, this muffler was selected as the easiest to manufacture as well as a cost effective solution. Red Dawn (AM)^2 is modeled in Figure 12.

Through the design process round edges were placed to ensure the smoothest flow characteristics and reduce the chance of creating high frequency noise [14]. The muffler chosen was built with seven chambers, three absorptive style chambers using densely packed ceramic fiber [15] and four expansion chambers. It was seen that a higher number of chambers had a positive effect on sound reduction. The chambers were ordered from small to large and implemented one deflection plate within the central chamber. A 180° Mandrel bend was used to guide the flow as well as increase the overall travel length within the muffler. Additionally, two beveled extension tubes were added on the outlet of two chambers to aid with the flow and acoustic properties.

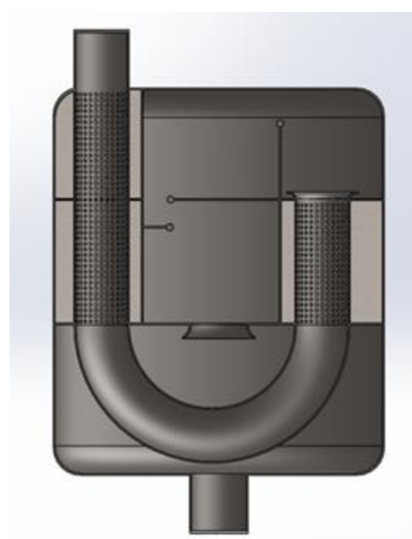

Figure 12. Red Dawn (AM)^2 muffler configuration

Muffler Proof of Concept

Once the final design was selected, preparations were made for manufacturing. A list and amount of required material were compiled along with an estimation on the time it would take to complete the build. Detailed drawings were used to develop a manufacturing plan, as represented in Figure 13. The Red Dawn $(AM)^2$ muffler was then manufactured using the University of Idaho's Mechanical Engineering shop space and tools.

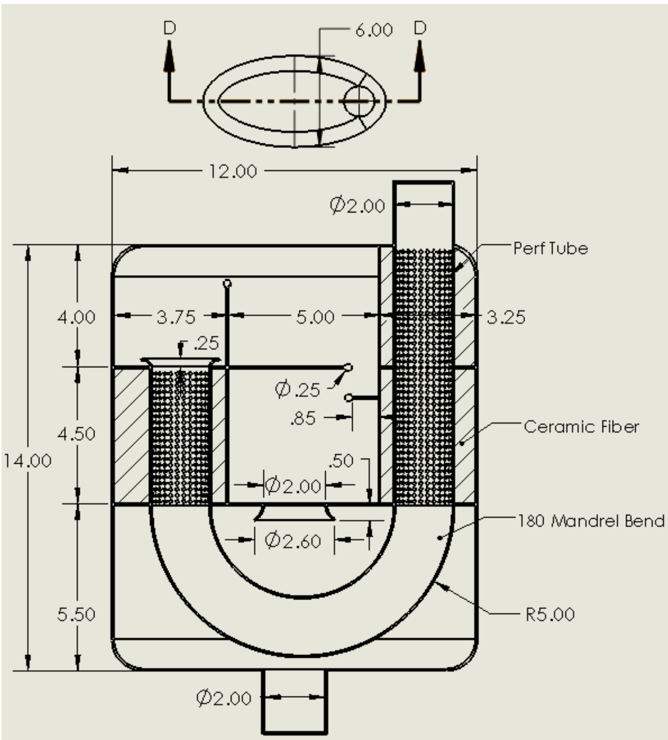

Figure 13. Dimensions of $(AM)^2$ muffler

Catalyst Implementation

Previously the catalyst was implemented in the exit of the muffler for ease of packaging. This was achieved by replacing a chamber of the stock muffler with a catalyst, without quantifying changes in acoustic performance, weight, and added back pressure. In this configuration, the catalyst had reduced emission conversion efficiency during competition. The redesign of the muffler provided the opportunity to move the catalyst to the entrance of the muffler, as shown in Figure 14, increasing the exhaust gas temperature entering the catalyst. These increased temperatures allow the catalyst to convert emissions over all engine operation.

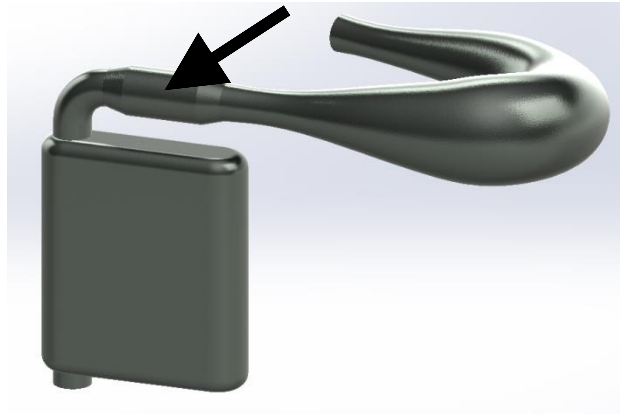

Figure 14. Relocation of catalyst

Bill of Materials

Table 8 shows the materials needed to build Red Dawn $(AM)^2$. The muffler shell, several internal plates, the beveled extension tubes, and the heat shield were purchased from Ski-Doo. Other parts such as the perf sheets and 180° mandrel bend were purchased through McMaster Carr.

Table 8. Bill of materials for $(AM)^2$ muffler

Red Dawn $(AM)^2$ Final Muffler	
Description	**Amount Needed**
Ceramic Fiber	**0.2 kg (0.4 lbs)**
Fiberglass	**0.2 kg (0.4 lbs)**
18 Gauge Steel	**0.46 m^2 (4.9 ft^2)**
2" OD Tubing with 1/6" Wall Thickness	**76.2 cm (30 in)**
2" OD Perf Tube 1/8" Holes 40% 16 Gauge	**33.0 cm (13 in)**
Aluminum 15 Gauge	**0.35 m^2 (3.8 ft^2)**
¼" Steel Rod	**40.6 cm (16 in)**
2" 180° Mandrel Bend	**1 piece**
Weld Time	**4.5 hours**

Manufacturing

The Manufacturing of Red Dawn $(AM)^2$ started with the housing for the catalyst; the cones and shell were manufactured at and provided by Starting Line Products. A flange was attached to allow for easy removal of the catalyst for testing or if damaged. See Figure 15.

Figure 15. Housing for catalyst

The internals of the muffler were then welded together starting with the top plate assembly. The top plate, made from stock parts, contained a hole opening for the perforated tubes. The hole was initially too large, so a moon shape plate was cut from sheet metal and welded onto the opening to get the desired diameter of two inches. A three inch section of perforation tube was then attached to this opening. Lastly, a chamber wall was welded onto the top plate to allow separation between expansion chambers.

The bottom assembly included the attachment of eight inch perforated tube, the mandrel bend, and chamber walls to the bottom plate. The three needed holes within the bottom plate were already cut into the stock part to the desired dimensions. The perforated tube was welded onto the far right hole followed by the 180° bend which was attached to the far right and far left holes on the bottom of the plate. The top plate assembly was welded to the bottom assembly giving shape to the central chamber. Two chamber walls making up the first and second chambers were then welded onto the bottom plate and perf tube. Lastly, a deflection plate with ¼ inch rounded edges was welded onto the far right chamber wall, finishing the internals of the muffler, shown in Figure 16.

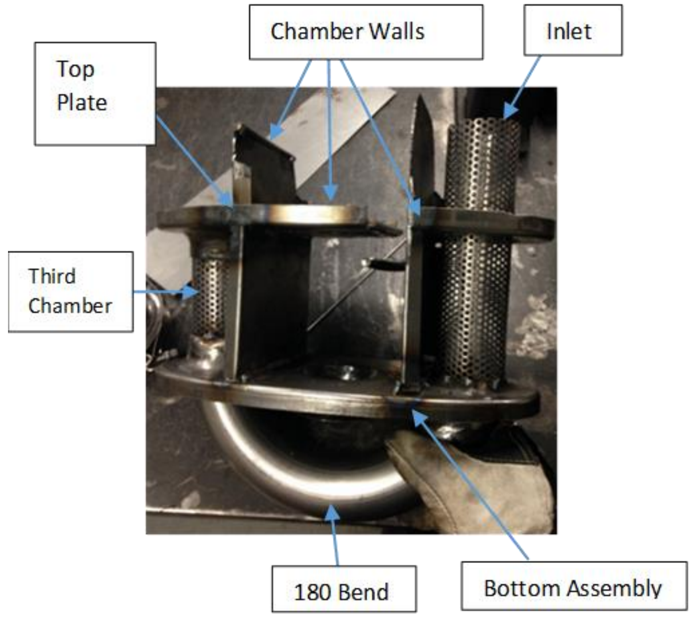

Figure 16. Muffler internals

A shell manufactured for a stock 2017 Ski-Doo muffler was modified and welded around the internals. A cone was then welded onto the inlet of the muffler to help improve flow from the catalyst. The top and bottom shell parts were welded first followed by the placement of the ceramic fiber sound material. The sound material was packed in the chambers containing perforated tubes. It was packed as densely as possible and held in place with masking tape. The middle shell was then placed and welded onto the muffler to complete manufacturing, shown in Figure 17.

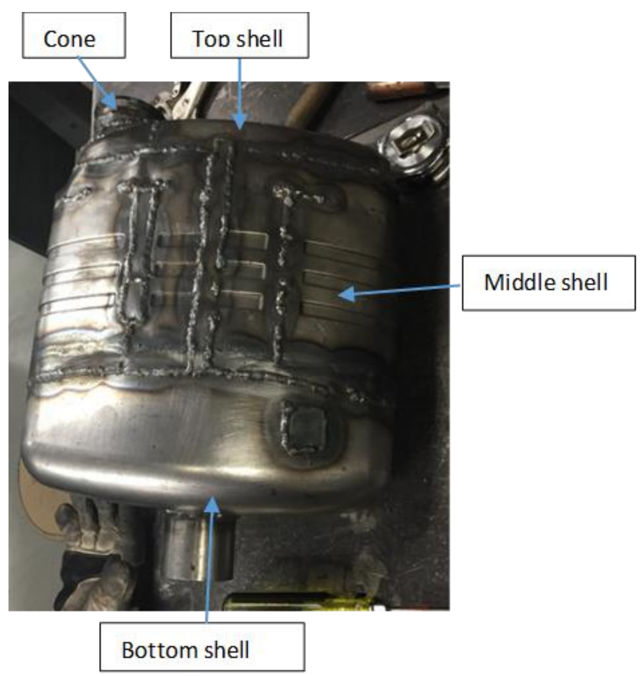

Figure 17. Complete muffler without heat shield

The catalyst housing was then mounted onto the muffler inlet with a Marman clamp to insure a complete seal and placed in chassis to ensure final placement. Figure 18 shows Red Dawn $(AM)^2$ in chassis.

Figure 18. Muffler in chassis leak test

Lastly, ¼ inch fiberglass matting and an aluminum heat shield, from Ski-Doo, were measured and placed around the muffler to give it additional protection, robustness, and aesthetics. This shield was very important for the protection of snowmobile chassis components including the side panels, oil well, and wiring harness. The final product is shown in Figure 19.

Figure 19. Complete muffler with heat shield

Validation

Validation of Red Dawn $(AM)^2$ was needed before it could be used by the UICSC team for the CSC competition. This included validation of acoustic ability, back pressure, and performance on chassis.

Sound Bench Testing Results

The UISB was used to validate Red Dawn $(AM)^2$. At lower frequencies there was improved performance over stock, and at higher frequencies it either performed as well or better at most points, shown in Figure 20.

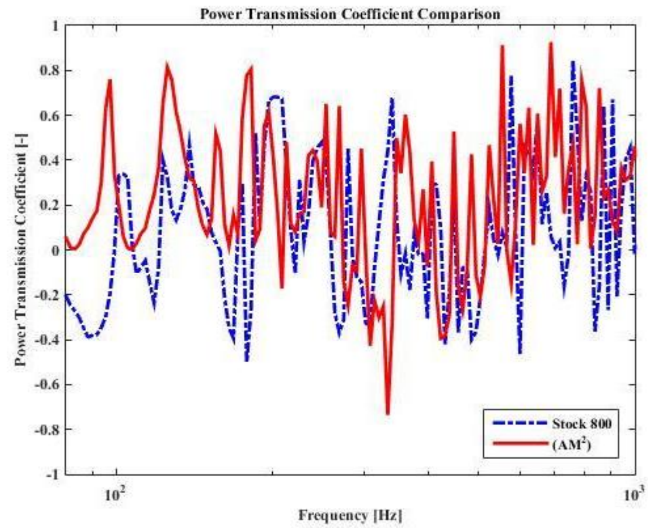

Figure 20. UISB muffler comparison

Back Pressure

The measured back pressure of the 2017 muffler with catalyst is 52% higher than stock at the max flow rate of the engine. This resulted in very little power loss from the engine after recalibration. Table 9 represents the various configurations tested.

Table 9. Back pressure comparison to stock

Configuration	Back Pressure	Change from Stock
2013 Ski-Doo Stock	.04 bar (.63 psi)	0%
2017 Ski-Doo Stock	.05 bar (.73 psi)	16%
Red Dawn $(AM)^2$	.05 bar (.76 psi)	21%
$(AM)^2$ With Catalyst	.07 bar (.96 psi)	52%

Dyno Calibration

A water brake dynamometer, was used to test the robustness and durability of the final muffler to insure survivability at riding conditions. The dynamometer was set to run at the five different operating mode points of the snowmobile to test the muffler at each condition, especially when riding at full power or wide open throttle (WOT). The dyno operated at WOT for several 2 - 3 minute intervals. The muffler would be inspected before, during, and after the tests to note down areas of concern where improvements needed to be made. The testing of the muffler can be seen in Figure 21.

Figure 21. Muffler dyno testing

On Snow Testing

The muffler was tested using the J1161 Society of Automotive Engineering snowmobile sound test with a result of a 1 dBA loss compared to the 2013 Ski-Doo muffler. The J1161 test is a constant 56.3 kmh (35 mph) drive by test where noise levels of a snowmobile are recorded from 15.24 m (50 ft) away. It is using this test that NPS standards must be met.

While testing the $(AM)^2$ muffler during riding conditions, the excess of heat added by the catalyst created worries that chassis components had the potential of being damaged. Due to this, an additional heat shield was added. Heat tape was also added to the side panel as a protection barrier.

Power

With the increase in back pressure only 74.6 kW (100 hp) was able to be produced. However, this is not solely due to the muffler. The increased back pressure from the catalyst played a major role in this power reduction. This reduction was seen as acceptable.

Weight

A key selling point of the two-stroke platform is its power-to-weight ratio. Reducing weight results in better fuel economy, improved dynamic performance, and decreased rider fatigue. The muffler weighed 8.7 kg (19.2 lbs) or 0.91 kg (2 lbs) less than stock allowing for less total vehicle weight.

Cost

The muffler design had a cost of $325, which is well within the acceptable manufacturing cost. The manufacturing suggested retail price (MSRP) of the muffler was $488, which is below the stock MSRP of $650.

Concentric Muffler

Along with manufacturing the $(AM)^2$, another muffler called the concentric muffler was designed, seen in Figure 22. This was built by the University of Idaho facilities to be used as another validation point for the $(AM)^2$ muffler versus stock. Beyond that, out of our muffler designs, the concentric muffler had the lowest back pressure. Compared to stock it had 55% lower back pressure. The final muffler can be seen in Figure 23.

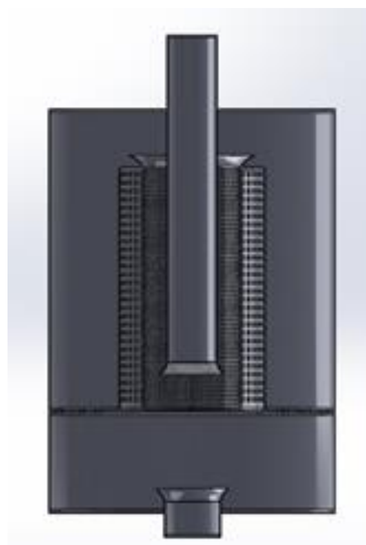

Figure 22. Concentric muffler model

Figure 23. Concentric muffler

After it was built, the concentric muffler was tested on the UISB and compared to the stock muffler and the $(AM)^2$. Seen in green on Figure 24 and Appendix B, the concentric was louder than both the stock (yellow) and $(AM)^2$ (red) mufflers at low frequencies. It performed similarly to the $(AM)^2$ muffler at higher frequencies.

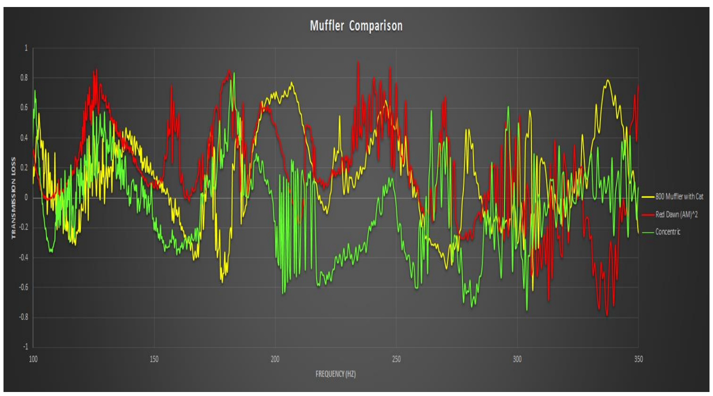

Figure 24. Muffler comparison UISB test

Summary/Conclusions

Red Dawn $(AM)^2$ reduced noise emissions by 5 dBA compared to stock. It maintained the performance of the stock system as no significant power loss could be directly attributed to the muffler. The muffler was able to reduce the weight and cost compared to stock. Table 10 shows the results of the $(AM)^2$ muffler compared to the design goals set at the beginning of the project.

Furthermore, the technique of using the UISB to compare sound reduction in various components was implemented successfully. This will provide a reliable testing apparatus and method that may be needed for future acoustics work.

Table 10. Competition results

	Goal	Acceptable	Competition Results
dBA Reduction	4	2	5
Peak Engine Power	89.5 kW (120 hp)	82.0 kW (110 hp)	74.6 kW (100 hp)
Weight	9.1 kg (20 lbs)	11.3 kg (25 lbs)	8.7 kg (19.2 lbs)
Flow Rate	0.1 m^3/s (3.5 cfs)	0.09 m^3/s (3.3 cfs)	0.1 m^3/s (3.5 cfs)
Back Pressure	4.34 kPa (0.63 psi)	5.03 kPa (0.73 psi)	5.24 kPa (0.76 psi)
Cost to Manufacture	$300	$500	$325
MSRP	$650 (Stock)	$850	$487.50
Size	40.6 cm x14.0 cm x 33.0 cm (16"x5.5"x13")	Has to fit	Fits
Robustness	982°C (1800°F) withstand winter conditions	982°C (1800°F) withstand winter conditions	1093°C (2000°F) withstand winter conditions
CSC Competition	Meet NPS Standards, Quietest Snowmobile	Meet NPS Standards	Met NPS Standards, 3rd Quietest

Future Work

Analysis

After extensive use on the dynamometer and at the UICSC competition, an analysis of the muffler was needed to view the durability. The outer heat shield was removed and viewing ports were cut into the sides of the muffler. Figure 25 shows the location of the view ports. Both chambers previously contained sound material, but upon inspection, sound material was found to be missing from the first port. This is thought to be due to the following factors:

1. The ceramic fiber in the first two chambers was located directly after the catalyst so saw higher temperatures than the ceramic fiber in the third chamber.
2. Unlike fiberglass which comes in long strands, the ceramic fiber used comes in small strands that are densely packed together. The holes in the perf tube or gaps between the chambers due to the heat cycling may have allowed for some material to escape.
3. The gaps between the first two chambers and the center chambers that formed due to heat cycling allowed flow through the sound material. This could have resulted in the sound material degrading at a much faster rate.

Figure 25. Muffler destructive analysis

Along with missing sound material, cracks and holes were found in locations where the stresses and heat were high. A hole was located at the end of the piping where the flow met the upper wall. This hole was due to vibrations, impacts, and brittleness. Cracks were located around the outlet of the muffler because the metal was repeatedly plastically deformed due to external forces and the weight of the emission pipe.

Recommendations

After the analysis, the design of this muffler has the potential to be improved. Higher grade steel and better welding would have better resistance to heat and vibrations. A higher temperature ceramic fiber could be used to better withstand catalyst temperatures. A mounting technique would allow for better support for the muffler reducing weight bearing stresses. For the design itself, adjusting the outlet so the pipe doesn't block the flow would decrease the back pressure of the muffler. Larger volume and size of the muffler would allow for better attenuation especially at lower frequencies, but chambers designed for peaks of exhaust noise emissions is highly recommended.

The UISB works very well as a comparison test, but could be improved. Slight changes were seen in testing data with changes in environmental conditions (snow versus no snow as an example) and temperature changes. There is a chance for noise to be reduced in the system, potentially by improving the circuit or the math model.

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Acknowledgments

The University of Idaho CSC Team would like to thank our many supporters: 509, AeroLEDS, ANSYS, Auralex, Avid Products, BASF, BCA, Between the Lines Designs, Biketronics, Bombardier Recreational Products, Boyeson Engineering, C&A Pro, Cameron Aviation, Camso, Cascade Clutch, E-Lab, Elk Butte Recreation, ETAS, Fokus Graphics, Heraeus, HMK, Idaho Motocycle Specialties, IceAge, Iron Cupcake, ISSA, Klim, Knowles, KRC, Latah County Snow Drifters, Makita Power Tools, Millennium Technologies, Modular Exhaust, Polymer Technologies, Silent Running, Slednecks, Starting Line Products, Snowest, Spokane Winter Knights, Thunder Products, Unifrax, Valley Powersports, Wagstaff, Western Power Sports, Washington State Snowmobile Association, the National Institute for Advanced Transportation Technology, the Mechanical, Electrical and Computer engineering departments, Finns, Justin, Nick, Drew, Alex, Sam, Chris, Crystal, David and Sonia Eyre, Russ Porter, Bill Magnie, Dr. Dan Cordon, Dr. Karen DenBraven, Dr. Mike Santora, Dr. Steve Beyerlein, Dr. Michael Anderson and the many others that, with their strong support, made this project possible.

Definitions/Abbreviations

CSC Clean Snowmobile Challenge

DI direct-injected

FFT fast Fourier transform

MAF Mass Air Flow

MSRP Manufacturer’s Suggested Retail Price

NPS National Park Service

PAF Polydamp Acoustical Foam

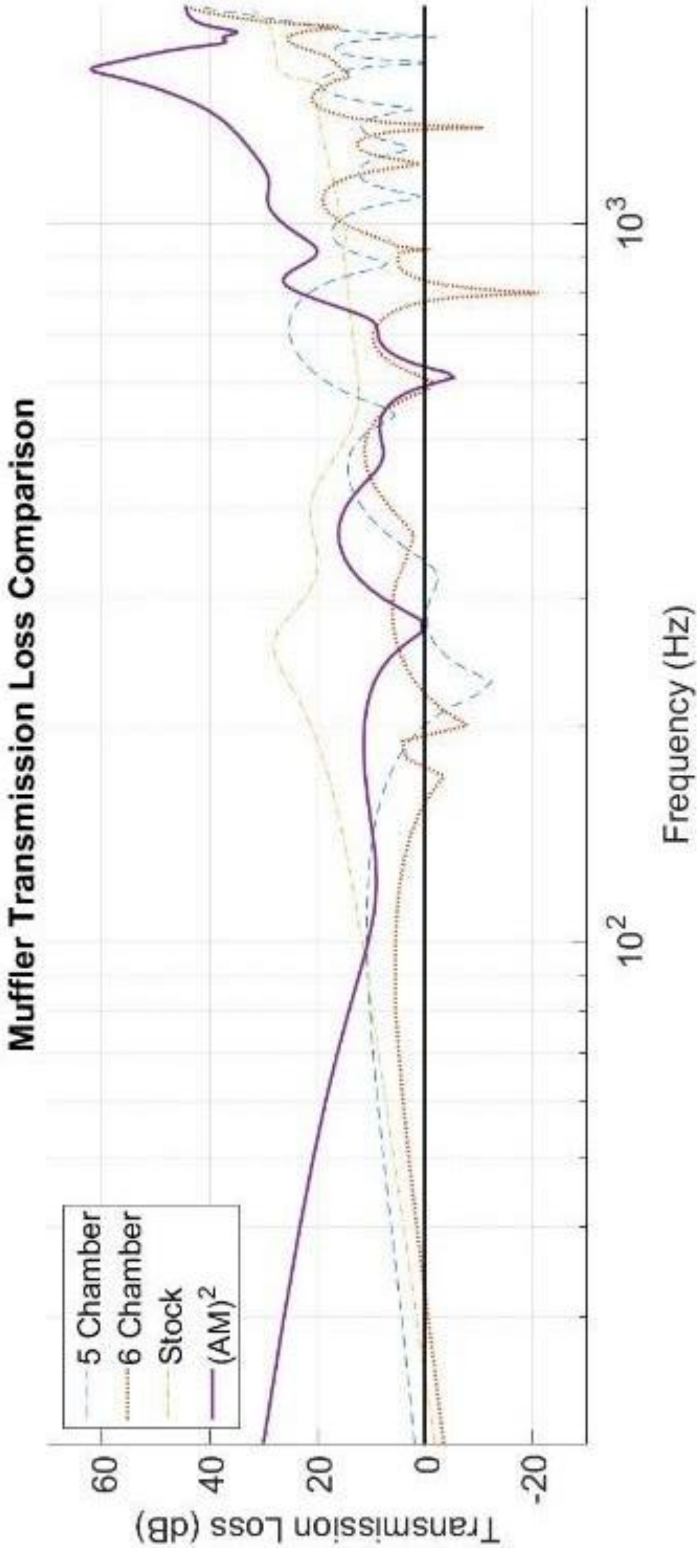
SAE Society of Automotive Engineers

UICSC University of Idaho Clean Snowmobile Challenge

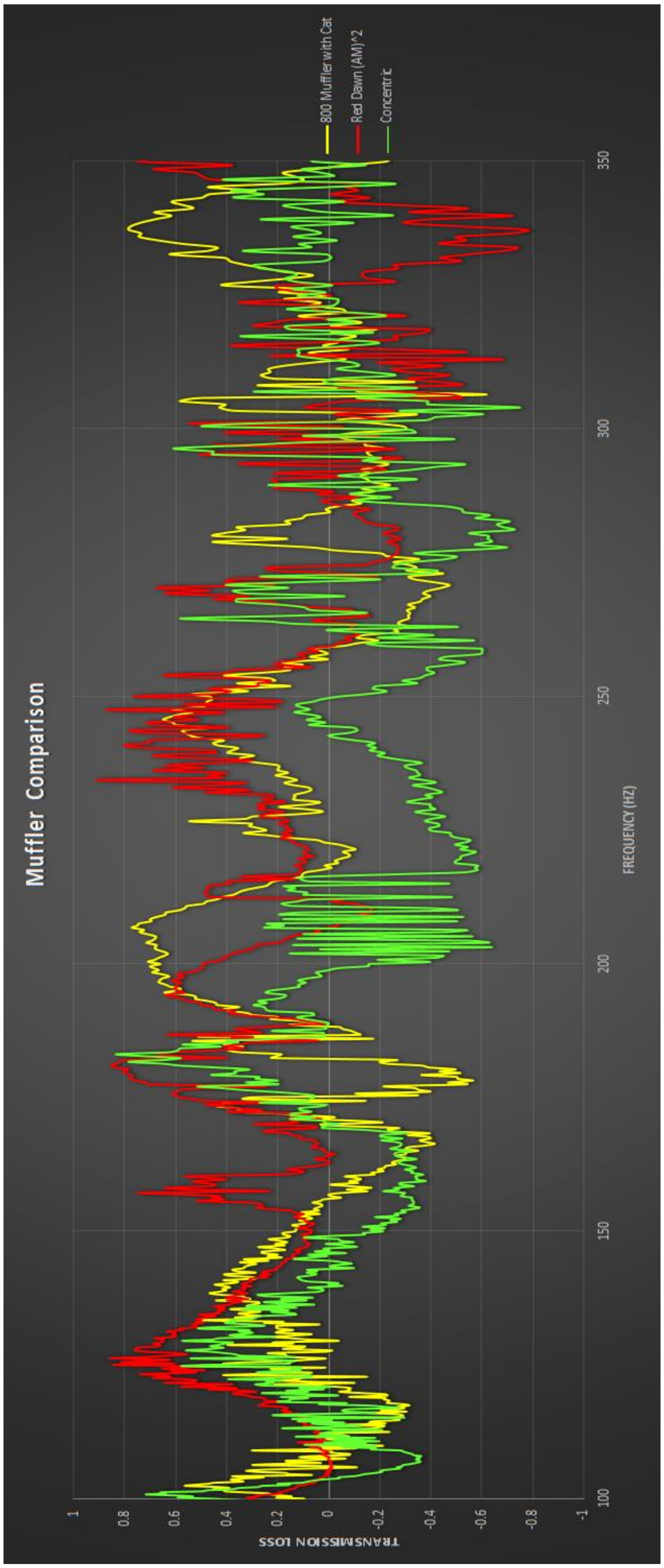
UISB University of Idaho sound box

WOT Wide Open Throttle

Appendix A


Muffler Transmission Loss Comparison
5 Chamber
6 Chamber
Stock
(AM)2
Transmission Loss (dB)
60
40
20
0
-20
Frequency (Hz)
10^2
10^3

Appendix B



Muffler Comparison
800 Muffler with Cat
Red Dawn (AM)^2
Concentric
TRANSMISSION LOSS
FREQUENCY (HZ)