# University of Waterloo Clean Snowmobile Team Design Paper

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## ABSTRACT

This report is a detailed summary of the various technologies and modifications that the University of Waterloo Clean Snowmobile Team implemented on their 2006 competition sled. The main theme for this year's competition sled is an environmentally friendly high pressure direct injected 2-stroke engine.

Team goals include the design and fabrication of a high pressure direct injection fuel system using custom and retro-fit components, a custom engine control unit, noise reduction, and drive train efficiency while maintaining aesthetics and fit and finish.

The team at UW is eager to re-join the competition after a year off, with its most competitive and innovative design to date.

## INTRODUCTION

This design paper is intended to outline the SAE Clean Snowmobile Challenge [1] entry by the University of Waterloo (UW). The goal of the UW team is to prove that two-stroke engines can be competitive in an environmentally responsible market for recreational snowmobiles. The traditional two-stroke engine is inherently louder, consumes more fuel, and produces higher emissions than a comparable four-stroke but also provides a significantly higher power-to-weight ratio and has fewer moving parts [2]. The implementation of new technology in fuel delivery and exhaust modification can minimize these drawbacks while maintaining the positive attributes of the engine. Noise reduction techniques and improvements in overall drive-train efficiency can also help to make the snowmobile more quiet and efficient.

The main consideration of the UW team in all of the vehicle improvements is the consumer. Riders want a powerful, comfortable, fuel-efficient, light-weight, predictable handling snowmobile whether they are looking for performance or touring snowmobiles. The ultimate purpose of the snowmobile is fun and this must be considered in all alterations. The UW team has been working for two years to produce an innovative design with many improvements to reach these goals.

## BASELINE SNOWMOBILE

The baseline snowmobile used by the UW team is a 2004 Polaris Pro X 550 (Figure 1). This vehicle is marketed to the aggressive trail rider and the largest segment of the snowmobile market. It is designed to be powerful, sharp-handling, and smooth on rough trails while being very durable. It is fully capable of meeting the demands of riders who desire a full day of riding in any conditions.

Figure 1: 2004 Polaris Pro X 550 baseline snowmobile.

## ENGINE

The stock engine has been removed and replaced with a 2000 Polaris 500cc US-built two-stroke. This is the same engine used by the UW team to win the 2001 CSC. Modifications have been made to this engine to allow for a high pressure direct injection fuel system to replace the stock carburetors.

FUEL SYSTEM

<u>Mechanical Fuel System</u>

The retrofitting of a high pressure fuel system to the existing snowmobile consisted of mounting injectors into the cylinder head, obtaining a high pressure gasoline pump to deliver fuel to the injectors, and replacing the existing carburetors with throttle bodies.

Two options for mounting the injectors into the cylinder head were available. The existing cylinder head could be modified, or a new head could be designed and built. Modifying the existing head would have resulted in having the injectors mounted at non-optimal locations, therefore, it was decided that a new cylinder head be built.

The new head was reverse engineered from the existing head with modifications added for injector mounting and manufacturing simplification. The new head was machined at the University of Waterloo on a three axis CNC mill and a CNC lathe. To reduce cost by reducing machining time, no milling tools smaller than one inch in diameter were used. This created a situation where a four piece cylinder head containing a base, a cover, and two removable domes was preferable over a standard two piece head. This also allows for more economical changes in combustion chamber geometry since new domes could be made and swapped without requiring a new head. Figure 2 shows a model of the custom cylinder head design.

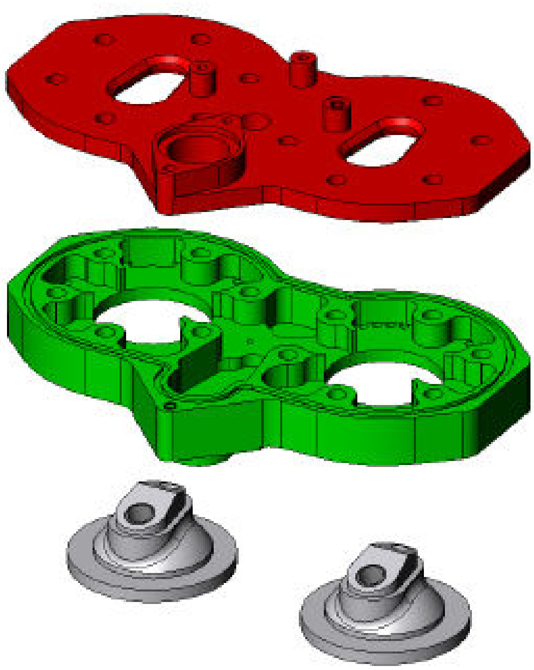

Figure 2: Four piece custom head design.

The cylinder head was made from 6061-T6 aluminum, chosen for its machinability and high temperature strength when compared to other aluminums. To ensure that the cylinder head would be strong enough under combustion, a finite element analysis was performed on the individual pieces using a combustion pressure of 725 psi [8]. It was verified that each part maintained a sufficient safety factor (1.25) under load. Figure 3 shows the dome under this load with appropriate boundary conditions.

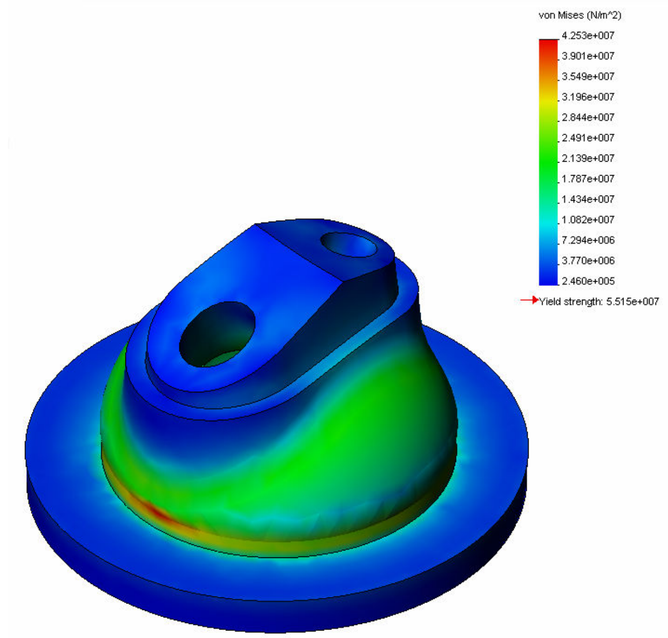


Figure 3: FEA of cylinder dome.

The cylinder head cover has three standoffs used for securing the injectors to the head. The injectors are mounted inline with springs so that they are preloaded. In this way, the total force exerted on them does not exceed the recommended maximum load of 400N. Figure 4 shows a cross section of the injector mounted into the head. The spring is not fully compressed in the model.

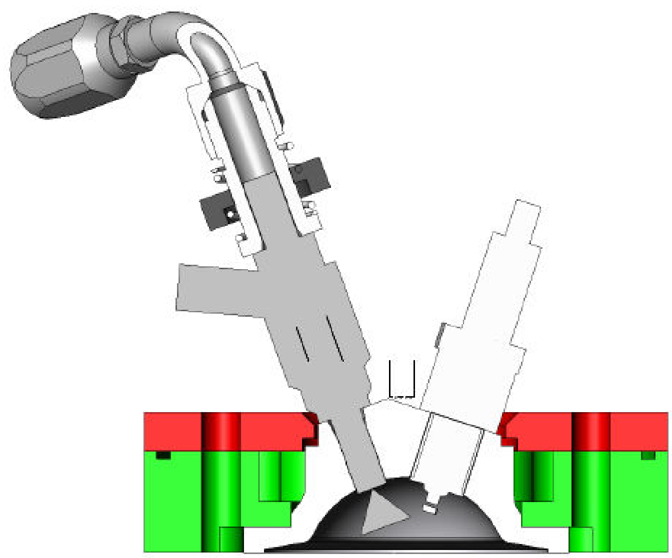

Figure 4: Cross section of injector in head.

Also visible in Figure 4 is the spray angle of the injector. This spray cone has been aligned such that the spray does not come into contact with the spark plug.

Fuel is delivered to the injectors by a series of fuel pumps that progressively increase the fuel line pressure. These pumps have been obtained from a model year 2000 Yamaha HPDI marine outboard engine (Z2000TXRY). The full system is pictured in Figure 5.

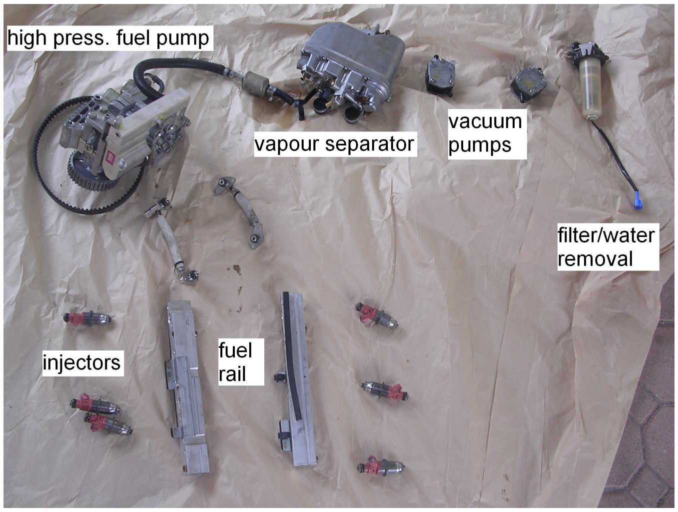

Figure 5: Yamaha high pressure fuel system.

With the exception of the filter, injectors, and fuel rail, this entire system was retrofitted to the snowmobile. The following outlines how this fuel system delivers fuel to the injectors.

1. Fuel is gravity fed from the gas tank to the two vacuum pumps. They are activated by an impulse vacuum line from the crank case. They pump fuel into the vapour separator
2. The vapour separator contains an electric fuel pump that pressurizes fuel to 350 kPa (~50 psi). A pressure regulator, interlocked with the throttle bodies, ensures that this pressure is constant. The vapour separator also has inlets for fuel returning as overflow from the high pressure pump. Lastly, oil is metered into the vapour separator via a solenoid oil pump to ensure the high pressure pump is well lubricated.
3. The high pressure pump is a mechanical piston pump driven by a cam and roller. It pressurizes fuel to 5 MPa (~725 psi). An integrated pressure regulator maintains this pressure; overflow fuel is sent back to the vapour separator. There are two high pressure outlets from the pump. They are linked enroute to the injectors to ensure pressure equalization.

The high pressure fuel pump has a maximum flow rate of 1.86 L/min at 3100 rpm. This is more than sufficient for this application [9].

Linking the high pressure pump to the engine was performed using belts. Belts were chosen because of their simplicity compared to gears which require complex housings and oil baths. A series of belts was required because of the large difference between the engine's maximum rpm and the pump's maximum rpm. The engine will run at 8500 rpm. The mechanical fuel pump has a maximum rpm value of 3100. This equates to a speed reduction of 2.74. The only feasible place to create pulley geometry on the engine was by tapping into the starter recoil. The final solution was to incorporate an HTD8 42 tooth pulley into the recoil ratchet. Figure 6 shows the transformation.

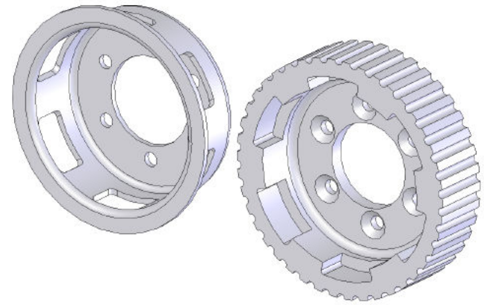
Figure 6: Old ratchet / new ratchet with pulley geometry.

A 42 tooth pulley was chosen because of space constraints, however, using a 42 tooth small pulley translates into a 115 tooth large pulley after the 2.74 speed reduction. A 115 tooth pulley is 11.5 inches in diameter making it prohibitive considering the belt speed is 155 ft/s (at 8500 rpm); therefore, a jack shaft was designed to tackle the reduction in two stages. The first stage is 42 teeth to 64 teeth, the second reduction is 22 teeth to 40 teeth (the stock pulley on the high pressure mechanical pump). This gives a final reduction of 2.77 and a maximum pump speed of 3068 rpm.

All belts were sized based on the load rating of the original drive system implemented by Yamaha, calculated in consultation with the Gates Corporation to be a maximum of 10 hp. Accordingly, the first belt system uses a 20mm HTD 8 timing belt with a power rating of 30 hp. The second belt uses a 25mm STD 8 timing belt with a power rating of 15 hp. The jack shaft itself and its mounting were also sized accordingly.

Fuel Mapping

Direct injection delivers fuel directly into the combustion chamber allowing greater control over how the fuel will be arranged and atomized and over combustion efficiency compared to port injection or carburetion [2]. This serves to significantly reduce emissions, increase fuel mileage and maintain or increase power.

The constraints that the system must follow for the fuel mapping to be considered successful is based on past data from CSC competitions in the emissions event:

- Decrease emissions of $CO_2$ and $NO_x$ by a minimum of 50% from stock engine.
- Decrease fuel consumption by a minimum of 25% from stock engine.
- Maintain a minimum of peak power of stock engine.

The fuel delivery is controlled by injector timing. The basic principle of tuning a fuel map is determining injector timing at various engine speed and throttle opening points. The injectors obtained from Siemens Automotive deliver a standard flow rate at various fuel rail pressures:

- 0 rpm – 50bar pressure: 6mg/ms flow rate
- 9000 rpm – 120bar pressure: 60mg/ms flow rate [4]

Using this information, the injector timing for any point can be found by the following calculation:

- Volume of one cylinder required to be filled by air:
  250cc
- Air fuel ratio for internal combustion engines:
  14.7:1 (optimal - stoichiometric)
  11:1 (rich - maximum throttle, low rpm)
  16:1 (lean – no throttle, high rpm) [4]
- Density of air (assumed):
  1.29 $kg/m^3$ [5]
- Mass of air present in combustion chamber:
  = volume*density
- Mass of fuel required to be determined:
  A/F ratio = mass air/ mass fuel [6]
- Injector timing:
  =mass fuel / flow rate

Figure 7 shows a standard fuel map for internal combustion engines. The graph shows engine speed on the x-axis, throttle opening on the y-axis, and injection timing on the z-axis. With a 2000 rpm idle point, the engine will run rich up to this point. An engine operates on the principal that an optimum equilibrium A/F ratio (14.7:1) exists. As seen in Figure 7, the mixture must be richened or leaned to speed up or slow down the engine from any point on this line. The engine management application created by the team has the capability to mesh the fuel map in between specified points. Therefore, several key points must first be determined, from which a rough map can be meshed and further tuning can be done.

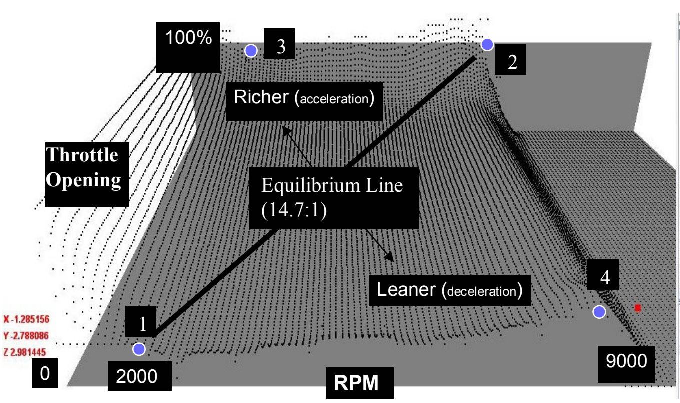


Figure 7 – Standard Fuel Map Template [4]

Following the calculation procedure outlined above, preliminary injection timing for points 1, 2, 3, and 4 are calculated in Table 1.

Table 1 – Standard Point Injection Timing.

| | | RPM / % Throttle (point) | | | |
|---|---|---|---|---|---|
| | | 2000/0 (1) | 2000/100 (3) | 9000/0 (4) | 9000/100 (2) |
| A/F ratio | | 14.7 | 11 | 16 | 14.7 |
| cylinder volume | cc | 250 | 250 | 250 | 250 |
| air density | kg/m3 | 1.29 | 1.29 | 1.29 | 1.29 |
| mass of air | mg | 322.5 | 322.5 | 322.5 | 322.5 |
| mass of fuel | mg | 21.94 | 29.32 | 20.16 | 21.94 |
| injector rate | mg/ms | 6 | 6 | 60 | 60 |
| injector time | ms | **3.66** | **4.89** | **0.34** | **0.37** |

From this point, the following steps can be taken to tune the fuel mapping of the engine:

1. Input calculated injection times into fuel map for points 1, 2, 3, and 4 as a preliminary starting point.
2. Run engine at 2000 rpm and no throttle (point 1) with calculated injection time of 3.66 ms
3. Once at operating temperature, monitor engine for basic operation characteristics:
   a. NVH (noise-vibration harshness) – richen and lean mixture at small increments (.1 ms) to achieve smoothest possible operation. [4]
   b. Exhaust gas temperature – richen mixture if exhaust gas abnormally hot, lean if exhaust gas is abnormally cool
   c. Exhaust oxygen sensor – richen mixture if excess oxygen exists
   d. The best possible balance of these 3 characterizes is desired.
4. Once engine is running well at load point 1, proceed to load point 2 with 9000rpm and 100% throttle. This should be done by slowly increasing throttle and monitoring NVH. This step will require some modulation on and off the throttle as engine will run into speed limiter as it tries to speed up past 9000rpm. Same monitoring factors as mentioned in step 2 apply.
5. Load points 3 and 4 can then proceeded.

Once this basic tune level is achieved, further precision of the tuning can be attempted. The basic tune level ensures that the engine can run reliably without major cooling and vibration issues and can be completed without the use of a dynamometer. Further work can be done on a dynamometer where monitoring of power output and emissions is possible. More points inside the map mesh can be found using the same calculation procedure and then tuned. Ten to twenty additional points throughout the map will be sufficient to properly mesh the fuel map and create smooth operation and performance.

At this stage, the UW team has not been able to tune the engine on a dynamometer; this additional tuning will be completed for CSC 2007.

## ELECTRONIC CONTROL UNIT

The Electronic Control Unit (ECU) for the snowmobile is built from a fresh design. Existing stand-alone control units are available yet do not give the team enough freedom to precisely time and control the injection periods. A high pressure direct injection system in a small engine application needs a few milliseconds of injection. Creating our own ECU allowed us to gain 16,384 points of precision on our fueling cycle.

The ECU is powered by an Atmel Mega 128 chipset. Atmel manufactures microprocessors for industrial applications, as well as for hobbyists. A development kit was donated to the team, and work on creating a Robust and reliable ECU began. After finalizing the requirements for inputs and outputs, the circuit board with the pin headers was supplied by BDMIRCO. The system runs @ 16 MHz with native code in assembly and C based languages.

Control System Integrating

The ECU uses two main inputs to determine the proper injection cycle. A load sensor detects the throttle angle, or air vacuum strength. A revolution speed is also sent as the second input to help scale the magnitude of change in fueling needed.

Extra sensors have been added to monitor the current condition of the motor. Each cylinder has its exhaust temperature probe and a fuel pressure sensor. Also included are oxygen and coolant temperature sensors. The ECU checks each of these periodically. If any begin to pass certain prescribed thresholds, warning lights will be shown. If any of the values rise over the soft limit, the ECU either goes into a limited operation mode, or will completely shut-off, based on the severity.

The ECU also connects to 2 separate triggers, in which the ECU encodes the current position of the pistons in the combustion cycle. The ECU begins its injection cycle when the exhaust ports close and finishes upon the prescribed pulse width. After such time the cylinder will reach the top of its stroke, with the atomized fuel pressurized. The spark plug, controlled by the capacitor discharge ignition (CDI), then ignites the mixture. There is very little fuel waste in this method of controlling the fueling.

Control Loop

The ECU has 2 main operating modes, open and closed loop. Open loop mode takes into account throttle position, air vacuum, and rotational speed. Each of these points is related to a fixed look-up table, which holds the proper fueling amount. The “fuel map” is stored in an array sequence in the EEPROM, and can be programmed and changed on the fly, via the ECU’s comm. port. This method is used in cold start and testing operations, or any other time it’s deemed that closed loop control will be unstable. Closed loop control monitors the air fuel ratio of the exhaust, and adjusts the fueling injection pulses based on demand. When an increase in throttle is detected, the ECU will attempt to speed the motor up by introducing more power, by adding more fuel into the mixture. The loop will attempt to adjust the pulse, until a healthy fueling “Rich” condition exists to create the extra power needed to accelerate. Upon detecting a stable throttle load condition, the closed loop circuit shortens the pulses back until the air fuel mixture has reached its optimum efficient value.

Safety Integration

In order to meet safety regulations, and to protect the rider from any abnormalities, a safety system must be in place. The snowmobile is affixed with a kill switch, as well as a tether system. These devices are intended to stop operation in any event, either by depressing the kill switch, or in the even that the rider has been dislodged from the sled.

These inputs are tied directly into the ECU, which affix to an input pin which requires a logic level of 1 (5 volts). Upon every injection cycle, the check for these conditions is made. If any of the safety devices are defeated, the ECU will go into a shutdown mode, and the injection pulse waves will cease. The engine will not receive any fuel, and inherently come to a stop. This event also cuts power to the CDI, which will stop the spark plugs from igniting further.

FUEL AND LUBRICATION REQUIREMENTS

The snowmobile is designed to run on premium gasoline as available at nearly all trail-side stops. The 10% ethanol blend provided at the competition is compatible with the engine.

The two stroke oil used is the Legend ZX-2R provided by Legend Performance. It is a commercially available, mineral-based hybrid oil that claims to be completely combustible, smokeless, and biodegradable.

COOLING

The 500cc two-stoke engine is liquid cooled. Due to the reduction in airflow in the engine compartment caused by the noise insulation (described later), additional cooling is required compared to a traditional 500cc snowmobile. Since the chassis was designed to house a fan cooled engine, the entire cooling system has been designed from the ground-up.

Almost all of a snowmobile’s systems are housed in the engine compartment, therefore it is desirable for the cooling system to be as compact and lightweight as possible.

The cooling system must be able to maintain a desired operating temperature of approximately 93°C [7], in a wide range of weather conditions including varying snow conditions and temperatures (-40°C to +10°C).

There are two main types of heat exchangers to choose from for the cooling system. These are the plate type and tube and shell type. Both are pictured below in Figures 8 and 9. Both of these types have been utilized by the UW team.

Figure 8: Tube & shell type heat exchanger

Figure 9: Plate type heat exchanger.

Tube and shell heat exchangers, better known as radiators, use the surrounding air to cool the engine coolant via convection. They are made up of a series of fins that are brazed to flattened tubes that circulate the hot fluid [7]. As the mass flow rate of air increases, the rate of heat transfer also increases. This means that as the snowmobile accelerates the amount of heat that is removed increases.

Plate heat exchangers for snowmobiles mainly rely on slush or snow being thrown onto their surface by the track to transfer heat. Unlike radiators, plate exchangers rely mostly on the principles of conduction to transfer the heat. Plate exchangers, or slush boxes, do not perform as well when on ice or marginal snow due to the small amount of snow churned up. The installation of studs can help this by breaking chips off of hard packed surfaces and ice to be thrown by track lugs to the heat exchangers. Radiators work regardless of snow conditions but are more susceptible to damage due to their fragile design. The use of both types will provide the advantages of each to allow for adequate cooling in any condition.

For both types of heat exchangers there are optimal placements. Since radiators rely on air flow for heat transfer, the optimal position is perpendicular to the flow. As the angle of the radiator strays from the perpendicular position, the effective area of the radiator is reduced as shown in Figure 10 where $h_1$ is the effective height.

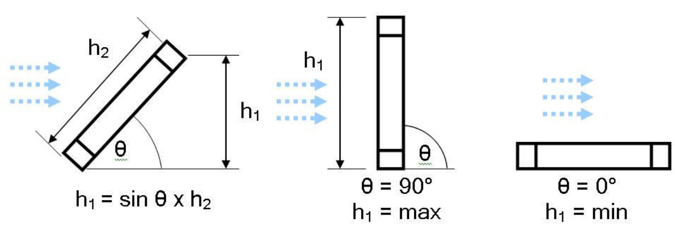


Figure 10 - Radiator Effective Area

For the radiator to be in the direct path of the airflow, it must be mounted somewhere in or on the hood where there are no obstructions (Figure 11). With the addition of scoops or cowls it is also possible to redirect the airflow which would allow for the radiator to be mounted in various orientations.

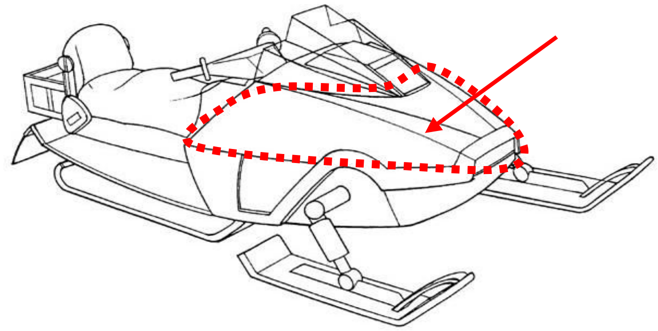

Figure 11: Radiator placement on snowmobile.

When considering the placement of the plate heat exchanger, the orientation is not as important as long as the entire surface is being covered by snow or slush. The best place to mount a plate heat exchanger is on the under side of the tunnel above the track (Figure 12). As the track propels the sled forward, it throws a significant amount of snow up into the tunnel.

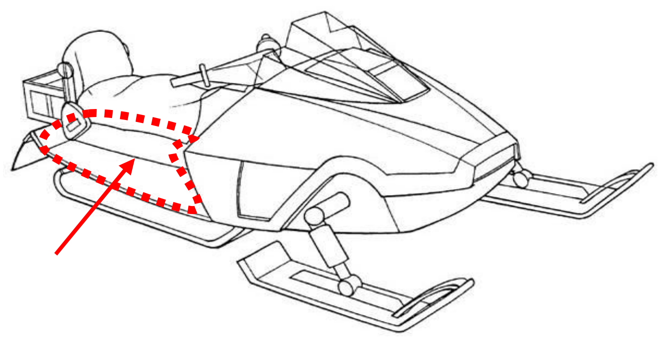

Figure 12: Slush-box placement on snowmobile.

<u>Radiator and Tunnel Design</u>

On the 2003 UW CSC snowmobile, an 8” x 6” radiator was used to sufficiently cool a 600cc engine. It can safely be assumed that the same size radiator will provide adequate cooling to the 500cc engine. Measurements were taken from the chassis assembly with the engine and drive system installed to determine the available space above the clutches for a tunnel to house a radiator. Figures 13 a and b show the hood and radiator tunnel modeled in Unigraphics.

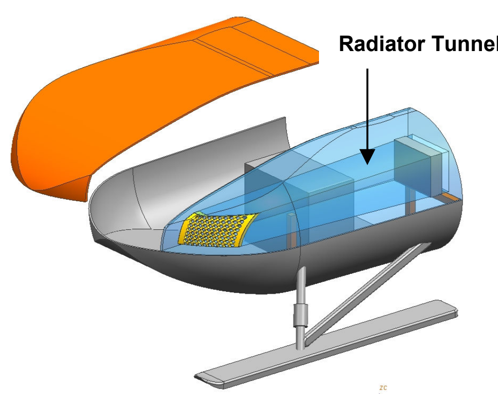


Figure 13a: Radiator tunnel – side view

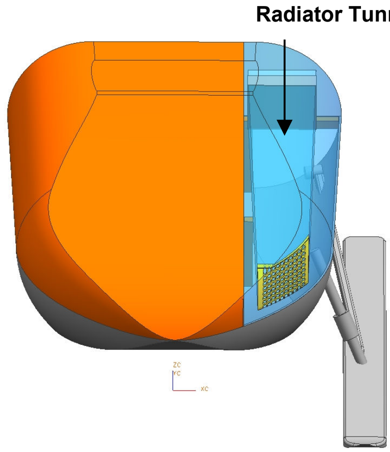


Figure 13b: Radiator tunnel – front view

The tunnel is bolted to the snowmobile chassis and the hood attaches as normal with space for the tunnel.

Coolant Routing

Coolant will flow out of the top of the engine and into the radiator tunnel under the hood. From here it will flow through a long, narrow slush box mounted underneath the left running board close to the suspension. This is connected to the rear slush box, located under the back of the tunnel where the most snow will be churned up by the track lugs. Coolant then flows back to the engine's coolant pump through a duplicate running board slush box on the right side. The coolant reservoir is connected between the last heat exchanger and the coolant pump, and at the top to the outlet from the top of the engine. Flow out of the top of the engine is directed either to the coolant bottle to be recirculated to the coolant pump, or to the heat exchangers by a mechanical thermostat depending on fluid temperature.

INTAKE

The air intake manifold is composed of a bottom resonator chamber, filtering tubes and top coupling chamber. Figure 14 shows the intake components.

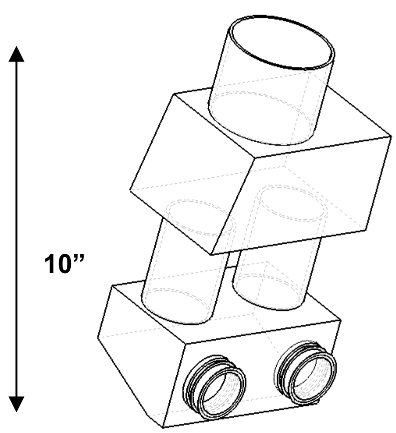


Figure 14: Intake.

Aside from delivering air to the engine, the purpose of the air intake manifold is to reduce noise emissions through the low pass filter formed by the resonating chamber and filtering tubes. The primary cause of noise in the air intake system comes from the reed valves that control the admittance of the air-oil mixture in the crankcase. Specifically, a high frequency noise is emitted from the reed petal vibrations when the reed valve is in the open position. From elasticity theory calculations it can be deduced that the natural frequency of the reed petals is ~833Hz, necessitating a low-pass acoustic filter that will attenuate the noise of the reed petal vibrations.

The simplest low past filter is of the chamber/tube type and typically has two attenuation bands: between $f_i$ - $2f_i$ and between $4f_i$ - $7f_i$. This applications calls for an $f_i$ of 180Hz so that the second attenuation band is between 720-1260Hz. This will reduce the noise coming from the vibration frequencies of the reed valve petals. The low pass filter is modeled by the following equation where $a_0$ is the speed of sound in air at room temperature. For the filter to have the second attenuation band of 720-1260Hz, the volume of the chamber must be $V_b$ = 1600cm³. The diameter and length of the tube is subject to the space available under the hood. A diameter of $d_p$ = 3" was chosen, with a tube length of $L_p$ = 5". $a_0$ assumed to be 343m/s.

$$\pi V_b\left(L_p + \frac{d_p}{4}\right) = \left(\frac{a_0 d_p}{4 f_i}\right)^2$$

Two low pass filters were installed, one for each throttle body valve, coupled in the top chamber that draws airflow from the air filter. Each filter is formed by a PVC tube connecting the top and bottom chambers, with a metal insert separating the two in the bottom chamber.

EXHAUST SYSTEM

Silencer Design

For the 2006 competition, an entirely new silencer was designed from the ground up, designed with the intention of reducing noise and maximizing the use of engine compartment space.

The silencer was designed with multiple chambers, each of different sizes. Since each specific volume and design of chamber muffles a certain range of frequencies, this design makes the most logical sense for an application with varying exhaust frequencies. In each chamber, sound waves resonate back and forth, canceling each other out. The chambers were designed as large as possible, with one wall per chamber lined with sound absorption material. The outlet pipe is located at the middle of each chamber because it has the highest probability of noise cancellation at that location. Sound and pressure travels through the chambers through perforated tubes. The perforated tubes are enclosed within chambers with loosely packed sound absorption material. To maximize the silencing efficiency, it is absolutely critical that the inlet and outlet tubes not face each other to avoid sound traveling from the inlet straight to the outlet.

The silencer design must not interfere with existing engine components and correlate with the tuned pipe. To determine the maximum amount of space available in the hood compartment, a 1:1 scale model (Figure 15) was fabricated to represent the new silencer design. Made from wooden skews and glue, the model provided confirmation that the design will fit into snowmobile. It sat comfortably in the left side of the sled with no interference to other components. From this model, the layouts of chambers for the silencer were derived.

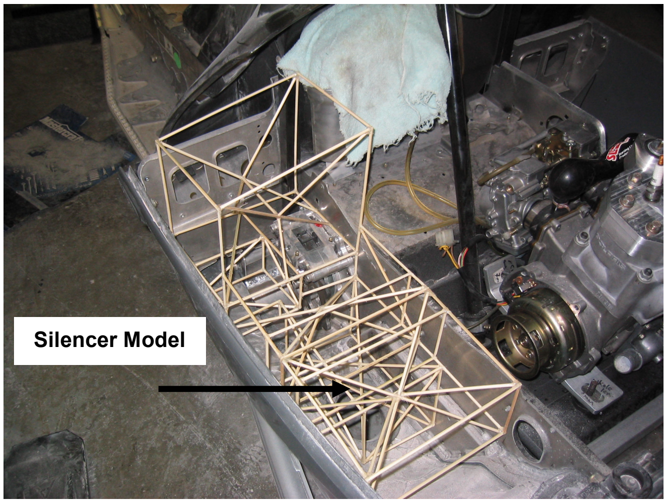


Figure 15 – 1:1 model of silencer.

The model as seen above evolved into the finalized design shown in Figure 16.

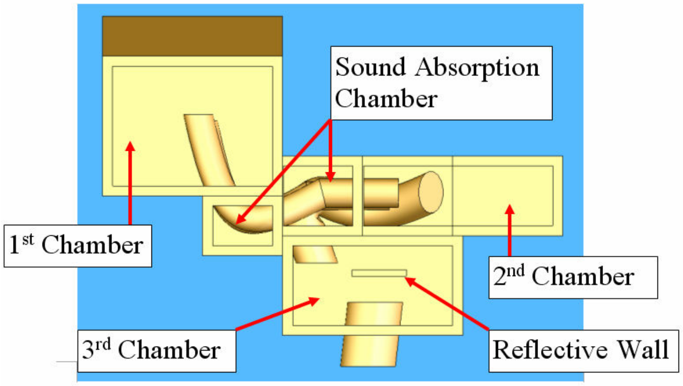


Figure 16: Final silencer design.

Each chamber is sized in order to resonate and cancel different sound frequencies. There are two sound absorption chambers housing the perforated tubes which direct the sound waves through the three resonator chambers. The reflective wall just before the exhaust outlet prevents sound waves from entering the third chamber and flowing straight out. This will force the waves to resonate and cancel before exiting the system. The thick outer walls represent sound absorption material used to cover the outside casing of the overall silencer. The silencer was constructed out of steel to withstand high temperatures and increase durability.

Catalytic Converter

For simplicity and due to time constraints a proven design that was used on the UW team's 2001 CSC winning sled was applied to the 2006 competition sled. The design consists of a 3.0" diameter tube style catalytic converter that is 3.5" in length, welded onto the end of the tuned pipe. At this location, the performance of the tuned pipe is not affected and exhaust temperature is sufficient to support catalyst operation. The catalytic converter is attached to the silencer by a flanged connection.

From past competitions, it is estimated that the temperature of the catalytic converter could reach as high as 750°C. It is for this reason that significant amounts of heat shielding are installed around the tuned pipe and catalytic converter. The heat shielding consists of a metal shield with insulation wrap.

## DRIVETRAIN

DIRECT DRIVE

The chaincase/jackshaft combination have been replaced with a direct drive unit from Radical Machines Inc. (RMI) (Figures 17a,b). This unit features a stub for the secondary clutch that is connected by a gearbox to the drive shaft. Weight savings from the unit is roughly ten pounds of rotating mass and increases available under-hood space by eliminating the jackshaft and chaincase and moving the brake to the end of the driveshaft. Relocating the brake also adds safety since the system no longer relies on a chain connection for stopping. The unit uses ten tooth drive sprockets to reduce track bending radius thus increasing efficiency.

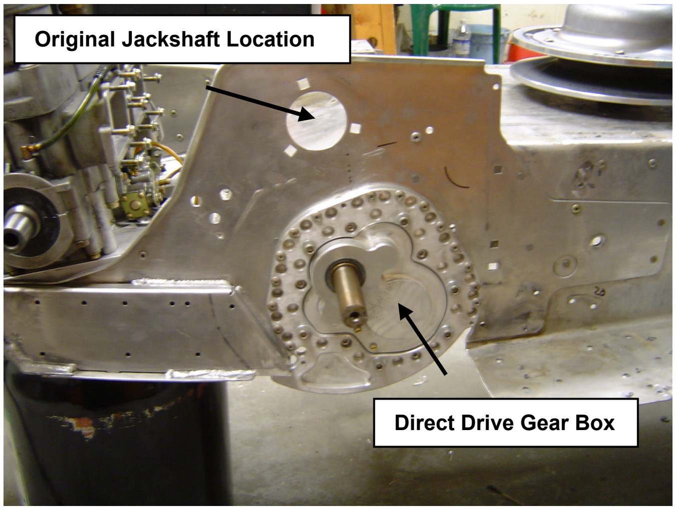


Figure 17a: RMI direct drive – clutch side.

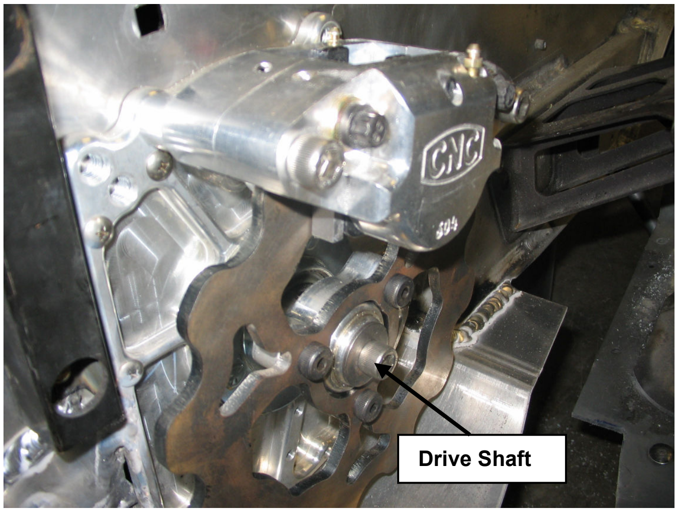

Figure 17b: RMI direct drive – brake side.

The installation of this unit required extensive modification to the foot rests and trailing arm mounts, as discussed later.

The direct drive was installed by leaving space around the drive shaft for the ten tooth drive sprockets plus a track with studs and an inch of clearance so that no damage would occur if the track were ever to ratchet. Also, the secondary clutch stub had to be spaced 12.5 inches from the crankshaft in order to use a Polaris Edge RMK belt available at any local dealer.

## SKIS AND SUSPENSION

### SKIS

The stock Polaris skis were replaced with a set of USI SPX plastic skis (Figure 18). These are lighter and offer more control by featuring a deeper keel and hiding the front of the carbide runner to reduce darting. They also feature a stepped bottom to reduce drag at higher speeds.


Figure 18: USI SPX skis [1].

### FRONT SUSPENSION

The Edge IFS front suspension is unchanged except for the trailing arm mounts which are within an inch of the stock position. These were modified to incorporate the direct drive system and are described in the chassis modification section.

## TRACK, TRACK SUSPENSION, AND TRACTION

### TRACK

The stock Polaris track has been replaced with a 121 inch long Camoplast 9833H high-performance trail track with 1.25 inch lugs. This track is designed to perform well in all snow conditions. The lug height allows for maximum traction while still allowing the sliders to be lubricated when driving on packed roads and provides minimal drag at higher speeds. It is fully clipped and features a lightweight clip design.

### Traction

For added traction and safety, 96 Woody's carbide-tip studs have been installed on the center belt of the track. These studs significantly improve traction especially on icy surfaces allowing for more control and safety. The pattern used incorporates double backer plates in a staggered V-pattern as shown in Figure 19.

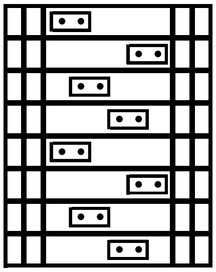
Figure 19: UW team stud pattern.

This design prevents the studs from being torn from the track if a spinning track catches a rock or other solid object. The pattern utilized is staggered to create the maximum number of "scratch lines" possible to maximize traction. The studs have been installed according to CSC Rule #4.5.3 with tunnel protectors to protect the tunnel and heat exchangers.

### TRACK SUSPENSION

The rear suspension (Figure 20) remains mostly stock featuring Ryde FX remote reservoir shocks and 13.5 inches of travel to provide superior bump absorption and handling. The only modification was removing the idler wheels along the slide rails to reduce noise and friction and installing an eight inch billet wheel kit on the rear axle to reduce the track bending radius and improve efficiency.

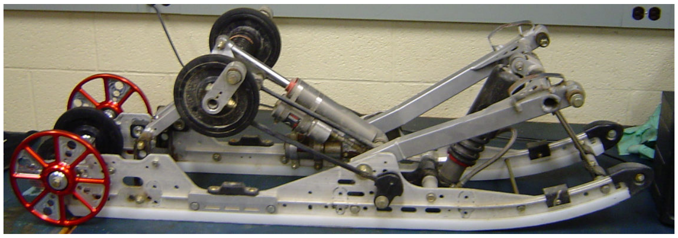
Figure 20: Rear suspension.

# FRAME AND BODY

SNOW FLAP

The snow flap has been replaced by a larger one from USI that is slightly wider and long enough to touch the snow with a rider on it as per CSC Rule #4.6.1. It is retained by two non-elastic straps of rubber belting to maintain contact with the snow surface during operation.

HOOD

Hood modifications include removing the gauge and headlight portion and fixing them permanently to the body so that the hood may be removed without disconnecting any wires. The main objective of this was to add simplicity for the emissions event but it also allows for easier access to the engine compartment for repairs.

CHASSIS MODIFICATION

The installation of the direct-drive system described previously required modifications to the foot rests and trailing arm mounts, all of which are structural members of the chassis.

The drive unit mounts to the chassis using circular plates bolted to the tunnel. These plates interfere with the trailing arm mounts and thus, new mounts were designed. These mounts incorporate guards for the secondary clutch, which now has a low point below the low point of the chassis, and the brake which is now located on the end of the driveshaft and is also lower than the chassis. These guards were designed to protect the brake and secondary clutch against minor impacts from uneven terrain.

<u>Direct Drive Mounting Plate Strength</u>

The first thing to determine was whether or not extra strength was needed for the chassis with the new mounting plates. The loading for buckling was determined assuming the snowmobile was travelling at 100 km/h (roughly 60 mph) and went over a jump. Upon landing, the suspension system bottomed out; the spring was forced to its full compression. After bottoming out, the section of the chassis with the new mounting plate and guard strikes the uneven trail surface. The velocity of the snowmobile changes by 20 km per hour with an impulse time of 100 ms, created by the impact. If the snowmobile with a rider were to weigh 350 kg (approximately 770 pounds), the force on the chassis and mount plate at that moment would be:

$$F = ma \qquad \text{Where } a = \frac{\Delta v}{t},$$

$\Delta v$ is 5.56m/s (20 km/h)

t = 0.1s

m = 350kg

The force acting on the chassis will be 19.44kN. The load on the chassis will cause that section to fail due to buckling long before it will fail in compression. Failure in buckling is governed by the critical force equation:

$$P_{CR} = \frac{\pi EI}{(KL)^2} \quad [10]$$

Where E = modulus of elasticity of 6061-T6 aluminum = 68.9GPa
I = 2nd moment of inertia of a member = $1/12ab^3$ = $8.43 \times 10^{-9}$ m$^4$
K = end condition constant = 0.5 in this case
L = length of the member under compression = 0.254m

The critical force for the smaller of the two mounting plates is 113.1 kN. Given that the critical force is over five times that of the actual loading on the mounting plate, the plate is not likely to fail in buckling during normal operation of the snowmobile.

There is also a possibility that the bolts holding the brake mounting plate to the chassis will shear under the load as well. Given that there are eleven bolts on the smaller of the two mounting plates, each being an M4 bolt made of steel, this is not likely. The yield point of low-carbon steel like AISI 1020 steel is 350MPa [11]. The stress on each bolt, assuming the same loading conditions as above will be 140MPa. They will not fail on either plate, as the larger plate has thirteen bolts mounting it to the chassis.

<u>Trailing Arm Mount Strength</u>

The trailing arm mount must be able to withstand the same conditions as above. Given that the suspension system will be absorbing the majority of the sled's vertical momentum, the loading conditions are slightly different. If the sled was to hit a hump, it would also have its forward velocity reduced. Assuming that the forward velocity was reduced by 40 km per hour with the same impulse time, the force acting on the bracket through the trailing arm would be double that of the chassis, or 38.9 kN.

Given that the bracket geometry is virtually identical to that of the old bracket, it is no more likely that it will fail than the old one would have. The concern is the new mounting. If the new bracket is welded on, will the welds withstand the load conditions? The welds around the mounting shaft to connect it to the main bracket body will be made of ER4043 aluminum filler. This filler metal has an ultimate tensile strength of 145 MPa and yield strength of 70 MPa; calculations are shown in the Appendix. The force from the trailing arm will most likely be in the direction along the bracket and not from the inside out.

The bracket will be attached to the existing chassis using welds as opposed to rivets in the case of the old bracket.

The welds on the new bracket are approximately 53 cm in length. With the same weld size of 4 mm, the shear stress will be 27MPa.

Fatigue Considerations

During normal operation of the snowmobile, the loading will be considerably less than the values calculated above. For an average ride of four hours, there may be only three or four times that a major load will be experienced by the system, and it will almost always be on the bracket, as it takes a great deal of force to bottom out the suspension. Assume that casual bumps are hit about four times a minute; that means that there are about 250 small impacts for every large impact. If each of the small impacts is one tenth the intensity of the major impacts, the fatigue stress range equivalent will be:

$$S_{re} = \left( \sum_{i=1}^{n} \alpha_i \cdot S_{ri}^{m} \right)^{\frac{1}{m}} \text{ [12]}$$

Where α = ratio of cycles i to total number of cycles

$S_{ri}$ = stress range for cycle i

m = fatigue curve slope constant = 3.45 for fillet welds

This equation results in a stress range equivalent of 29.1 MPa. This is higher than the allowable value of 13 MPa for aluminum fillet welds, so the welds will fail before the baseline limit of five million cycles. The baseline limit translates to five thousand trips, or twenty thousand hours of operation. To find out the exact number of cycles the parts will last, the following equation is to be used.

$$S_{re} = C_f \cdot N^{\frac{-1}{m}} \text{ [12]}$$

Where $C_f$ = fatigue curve intercept = 1100 MPa for fillet welds

N = number of cycles

m = fatigue curve slope constant = 3.45 for fillet welds

This results in a life of 277,000 cycles. This translates to 277 trips, or 1108 hours of operation.

Given that most trail riders only use their snowmobiles on weekends when there is snow, there are only twelve to fifteen weekends per year for the average snowmobiler in the Great Lakes region. If each one traveled for sixteen hours each weekend, or four "trips", that means that the bracket would fail after about four and a half years. This is about equal to the average life of a snowmobile of four to five years.

Final Assembly

The new chassis components, fully assembled, will appear as below in Figure 21X.

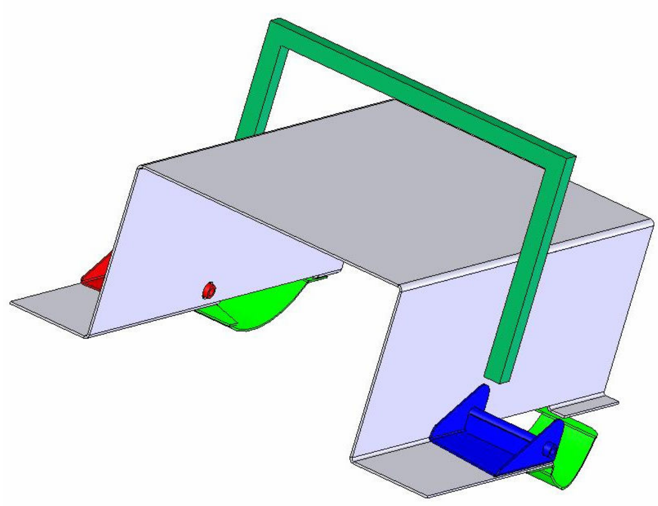

Figure 21: Chassis Assembly

The symmetrical trailing arm mounts are shown in blue and red connecting to the brake and clutch guards shown in green. All sheet metal parts are made of 10 gauge 5052 aluminum alloy. Components will be welded together with the ER4043 aluminum filler metal using the gas tungsten arc method.

The clutch and brake guards are very similar in design, with the clutch guard being slightly larger. These two guards will be cut out and then bent to the required shape prior to welding onto the main chassis and mounting plates. The two guards are to ensure that no foreign material can be thrown into the brake or clutch, as well as protecting them from impact.

## NOISE REDUCTION

### NOISE INSULATION

A significant amount of noise resulting from vibration of engine components is emitted from the engine compartment. A sound absorbing fabric provided by 3M Canada will be applied to the hood and belly pan to contain this noise. This lightweight fabric, specifically the 3M AU 4020-2 High Performance Acoustic Insulation, will add less than four pounds to the snowmobile's weight. It is designed for a wide range of applications including automotive and industrial. This material was selected due to its high absorbtion coefficient for the problematic frequencies in the range of 1000-3000 Hz, found in previous tests. It will be applied using a high temperature adhesive with added heat shielding applied to the areas close to the exhaust system.

### RUBBERIZED UNDERCOATING

A thin layer of rubberized asphalt undercoating has been applied to the underside of the tunnel and the chassis under the engine to reduce the resonance produced in these semi-enclosed areas. This added less than one pound of weight. It is expected that the coarse surface of the coating will reduce the reflections of sound waves. This coating was also applied to the hood before the application of the insulation so that both a firm and soft layer of noise control material would be present.

OTHER ITEMS

Other noise reduction strategies described in other sections include the multi-chamber exhaust silencer, quiet intake design, and rear suspension modifications.

## CONCLUSION

The University of Waterloo Clean Snowmobile Team is eager to return to the SAE Clean Snowmobile Challenge with its most competitive and innovative design to date.

A high pressure direct injection fuel system on a two stroke engine inside of an efficient chassis coupled with noise reduction techniques and exhaust after-treatment will provide a very environmentally snowmobile that will prove that the sport of snowmobiling is capable of moving towards environmentally friendly designs.

## ACKNOWLEDGMENTS

The University of Waterloo Clean Snowmobile Team would like to thank our sponsors, if not for them our team would not survive.

Title Sponsors
- Ontario Federation of Snowmobile Clubs
- PCB Piezotronics

Partners
- Legend Performance
- Radical Machines Inc.
- Siemens
- Waterloo Engineering Endowment Fund

Friends
- Brisk Spark Plugs
- Camoplast
- Dana – a division of Long Manufacturing
- Dayco
- Easymove
- Jaws Performance
- Mountain Machines Performance
- Ontario Snowmobiler
- SledPro Distributing
- Ultimate Sports Inc.
- Woody's

We would also like to thank the Society of Automotive Engineers for creating the opportunity to modify snowmobiles and address the environmental concerns they create as part of our education.

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## APPENDIX

Failure in Buckling

$E := 68.9\cdot 10^{9}\,Pa$

$b := \frac{7\cdot 2.54}{100}\cdot m$

$b_1 := b - 4\cdot 0.007m$

$K := 0.5$

$b = 0.178\,m$

$b_1 = 0.15\,m$

$L := \frac{10\cdot 2.54}{100}\cdot m$

$h_1 := 0.01m$

2[nd] area moment of inertia

$L = 0.254\,m$

$h_2 := 0.008m$

$t := 0.0032m$ $\quad l := \frac{5\cdot 2.54}{100}m$

Critical load allowed

$$I := \left(\frac{1}{12}\cdot b\cdot h_1^{\,3}\right) - 3\left[\left(\frac{1}{12}\right)\frac{b_1}{3}\cdot h_2^{\,3}\right]$$

$I = 8.425\times 10^{-9}\,m^4$

$$P_{cr} := \frac{\pi\cdot E\cdot I}{(K\cdot L)^2}$$

$P_{cr} = 1.131\times 10^{5}\,N$

$$I_{old} := \frac{1}{12}b\cdot l^3$$

Failure in Bending

Maximum Load Assumption

Assume That a snowmobile with one rider weighs 300kg (660lbs), going 100km/h hits a bump on a 45 degree angle, which slows the forward speed by 20km/h in one tenth of a second

$m_{sled} := 350kg$

$\tau_{max}$

$a := \frac{20}{3.6}\cdot 10\frac{m}{s^2}$

$F := m_{sled}\cdot a$

$sd := 350\cdot 2.2$

$F = 1.944\times 10^{4}\,N$

$sd = 770$

The strut to the mounting bracket is on a 15degree angle

$l_w := 0.004m$ $\quad L := \left(\frac{0.019}{2}\right)^2 \cdot \pi m + 0.220m + 0.020m$ $\quad F_y := 130 \times 10^6 Pa$

$F_b := F \cdot 2$

$A_m := \frac{l_w}{\sqrt{2}} \cdot L$ $\quad L = 0.24\,m$

$A_m = 6.796 \times 10^{-4}\ m^2$

$we := .146 + .126 + .120 + .121$

$we = 0.513$

$\frac{F_b}{A_m} = 5.722 \times 10^7\ Pa$

$A_{m1} := \frac{l_w}{\sqrt{2}} \cdot we\,m$

$\frac{F_b}{A_{m1}} = 2.68 \times 10^7\ Pa$

Area of bolt

$d_{bolt} := 0.004m$

$A_{bolt} := \pi\left(\frac{d_{bolt}}{2}\right)^2$

Stress per bolt on all 11 bolts on brake mounting plate

$A_{bolt} = 1.257 \times 10^{-5}\ m^2$

$\sigma := \frac{F}{A_{bolt} \cdot 11}$

$\sigma = 1.407 \times 10^8\ Pa$

$S_{r1} := \sigma \cdot \frac{1}{Pa}$ $\quad S_{r1} = 1.407 \times 10^8$

$S_{r2} := \frac{\sigma}{10} \cdot \frac{1}{Pa}$ $\quad S_{r2} = 1.407 \times 10^7$

Fatigue considerations

$\alpha_1 := \frac{4}{1000}$ probability of major bump

$\alpha_2 := 1 - \alpha_1$

$m_f := 3.45$

Fatigue life/stress eqn

$S_{re} := \left(\alpha_1 \cdot S_{r1}^{m_f} + \alpha_2 \cdot S_{r2}^{m_f}\right)^{\frac{1}{m_f}}$ $C_f := 1100 \cdot 10^6$

$S_{re} = 2.909 \times 10^7$

$N := -m_f \ln\left(\frac{S_{re}}{C_f}\right)$

$N = 12.532$

$N_e := e^N$

$N_e = 2.771 \times 10^5$