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University at Buffalo
Advancing Snowmobile Technology For The Future

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ABSTRACT

As the popularity of snowmobiles has risen since their widespread introduction in the 1960s, concerns have been raised about their friendliness to the environment. To address these concerns a few aspects of the snowmobile need to be concentrated on and redesigned. The implementation of a small displacement four stroke engine is a key solution in reducing emissions and improving fuel economy. Based on these ideas, a Honda 600cc engine and an open loop Bosch fuel injection system were selected for use in the University at Buffalo's snowmobile. Due to the smaller displacement of the Honda engine, adequate horsepower and torque are not available without additional hardware to improve these characteristics. This hardware includes a turbocharger and an intercooler. In an effort to reduce noise and emissions, a three way catalyst and a reactive muffler were utilized. The resulting combination is a snowmobile that is both fun to ride and environmentally conscious.

INTRODUCTION

Due to the increasing regulations on exhaust and noise emissions on snowmobiles today, there is a need for further development of new technologies that aid in making snowmobiles cleaner and quieter. The Clean Snowmobile Challenge (CSC) is a collegiate design competition for student members of the Society of Automotive Engineers (SAE). "The intent of the competition is to develop a snowmobile that is acceptable for use in environmentally sensitive areas. The modified snowmobiles are expected to be quiet, emit significantly less unburned hydrocarbons and carbon monoxide than conventional snowmobiles, without significantly increasing oxides of nitrogen emissions. The modified snowmobiles are also expected to be cost effective. The intent of the competition is to design a touring snowmobile that will primarily be ridden on groomed snowmobile trails. The use of unreliable and expensive solutions is strongly discouraged" [1]. The maximum noise and exhaust emissions must be equal to or less than typical snowmobile to be eligible for competition points.

PROBLEMS WITH TODAY'S SNOWMOBILES

EMISSIONS PROBLEM

The predominant use of hydrogen and carbon based fuels in snowmobiles leads to hazardous emissions from the combustion process. Hydrocarbon (HC), carbon monoxide (CO), and nitrogen oxides (NO_x) are the most harmful of the exhaust molecules and must be minimized. In order to create a solution to the emissions problem, the cause of the formation of these compounds must first be understood.

Hydrocarbons are formed by five modes: incomplete combustion, flame quenching, crevice volume, absorption of fuel vapor, and high equivalence ratio. Incomplete combustion normally occurs during the expansion stroke, when the cylinder pressure drops, causing the unburned gas temperature to drop. This temperature drop causes a decrease in the burn rate. Flame quenching can be attributed to the relatively cold walls of the cylinder. The cooler walls form a boundary layer where the flame cannot penetrate, hence leaving some unburned HC molecules. Crevice volumes, such as the area between the piston and cylinder wall, can once again trap hydrocarbons in a region where the flame cannot enter. The mode of absorption of fuel vapor into the thin oil layer occurs during the compression stroke. Once the pressure gets to a certain level, the fuel vapor can be absorbed into the oil layer. Upon expansion, the fuel vapors containing unburned HC are emitted from the oil layer and exit out through the exhaust. Hydrocarbons are also known to be sensitive to the equivalence ratio,ϕ. When ϕ is greater than one, the fuel is rich and consequently HC emissions are high, conversely when ϕ is less than one, the fuel is lean and the HC emissions drop.

Carbon monoxide formation is mainly a function of the equivalence ratio. Similar to hydrocarbon formation, CO emissions will increase with a rich fuel mixture, and decrease with a lean fuel mixture [3]. The CO is formed when there is not enough oxygen present to form CO_2. Therefore, poor mixing and incomplete combustion also play a factor in increasing CO emissions [3]. Normally, CO emissions range from 0.2% to 5% in the exhaust of a spark ignition engine[4].

The formation of oxides of nitrogen are caused mostly by high temperatures and large oxygen concentrations in the combustion chamber. The high temperatures cause the nitrogen to dissociate, and NO/NO_2 forms if there is oxygen present [4]. The main engine parameters that will then influence NOx production are spark timing and equivalence ratio. Spark timing closer to top dead center causes lower peak cylinder pressure [3]. These lower pressures cause lower cylinder temperatures, which reduce NOx emissions. Advancing spark timing has the opposite effect. Equivalence ratio affects both oxygen content and temperature. Maximum temperature occurs at an equivalence ratio of 1.1, but oxygen concentration decreases as equivalence ratio rises, and is too low at $\phi=1.1$. The maximum amount of NOx emissions has been empirically found to be at approximately $\phi=0.9$ [3].

NOISE PROBLEM

Another fundamental problem with today's snowmobiles is that they are noisy. Snowmobiles produce noise both from the engine and from the other moving parts. The engine produces sound pulses from the combustion process as well as mechanical noise. Two-stroke snowmobiles are exceptionally noisy, because the variable or reed valves normally used produce sounds that are particularly sharp and unpleasant to hear. The exhaust pipes which are often used on two-stroke snowmobiles are mainly expansion chambers that help the exhaust flow and are designed for power, not noise suppression. These are usually coupled with a silencer that provides minimal noise reduction. In a four-stroke engine, the intake and exhaust pulses aren't typically as loud or sharp. However, the four stroke engine has a valve train that produces more mechanical noise than the two stroke. To make up for a lower power to weight ratio, a four stroke engine often needs the added horsepower of a turbocharger to be comparable to a two stroke engine of the same size. The turbocharger adds a high pitch "whine" when it spools up as well as small bursts of noise when the intake pressure caused by the turbocharger is released through a pop-off valve. Finally, the suspension and track that propel the snowmobile create a large amount of noise, because of many wheels and cogs that move the track and turn it combine to create high noise levels when the snowmobile is moving.

UB SOLUTION

OVERVIEW

The University at Buffalo 2005 CSC Team concept is a solution to the problem areas that are encountered by manufacturers in making snowmobiles environmentally friendly. The team chose a Polaris Pro-X chassis for the design. The design is centered on a two cylinder, four stroke scooter engine that was modified for a snowmobile application. To make up for the vehicle's power-to-weight ratio which was decreased when the two stroke engine was removed, a turbo charger was added to compliment the four stroke engine. Further modifications include a catalyst for emission reduction, an intercooler for increased performance, and a customized muffler for reduction in noise.

ENGINE

Due to the goals, of noise reduction, emissions, and fuel economy, the team chose to abandon the loud high performance stock engine and go with something different. The replacement engine chosen was that of a 2002 Honda Silver Wing scooter. The engine is a parallel twin cylinder, four-stroke, with liquid cooling. The team felt this engine meets the design goals well and that it was applicable for a snowmobile application. Being a four-stroke, the longer stroke of each piston provides higher torque. It has dual-counterbalanced shafts for reduced vibration, further yielding in substantial noise reduction coming from the engine. Also, with the engine being designed to incorporate a continually-variable transmission, the task of adapting the engine to a sequential transmission, with much added weight, was eliminated due to the existing P-85 Team Roller belt driven system on the Polaris. In this rare case the engine, for its size, is fuel injected so the throttle bodies, throttle position sensor (TPS), fuel rail, and post throttle body intake manifolds all existed for added, tuned performance. The last beneficial aspect, that will later be discussed, is the simple fact that the engine came from a scooter platform. Thus, the engine was developed by Honda with strict environmental standards in mind for operation in sensitive areas, which helped to meet one of our current goal of a cleaner snowmobile engine.

Despite the many benefits that came with choosing the scooter engine, changes must be made. Four imperative changes needed to be made for the engine to be used in the Polaris Pro-X Chassis. First, the engine mounts were incompatible with the existing mounting points on the Pro X chassis, so new ones had to be custom made. The second modification made was a compression reduction. Due to the natural aspirating design of the engine, the vacuum effect caused by the use of a turbocharger would cause pre-detonation. The solution to pre-detonation had to be to lower compression, from 10.2:1 to 8.5: 1. Additionally a tone wheel for the crank speed reference used by the fuel injection system had to be made, which was the basis for the third modification. Finally, an adapter had to be made to mate the scooter engine with the Polaris snowmobile clutch.

CLUTCH ADAPTER

The Honda engine used a continuously-variable transmission clutch attached in the scooter that was driven off of the spline on the crank shaft. To transform this into a conventional snowmobile clutch an adaptor had to be fabricated to allow for the stock Polaris clutch to be used. The adaptor needed to go from the 25-tooth spline to the 3° taper that is located on the inside surface of the new clutch.

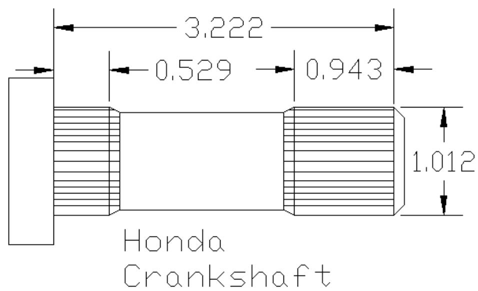


Figure 1: Stock Honda crankshaft, in inches

The clutch was bored out 0.015 inches over the major diameter of the spline of the engine because the inside of the clutch past the taper was too small to fit over the driveshaft. The blank spline was fabricated and the adapter was then turned to the 3° taper that was required.

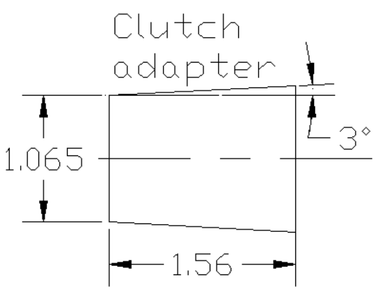


Figure 2: Adapter to fit over crankshaft, in Inches.

As seen in Figure 2 the new clutch fits over the adapter. It is secured into place by an end bolt that screws into the end of the crankshaft and presses against the oil seal sleeve on the engine (like the stock clutch did). With this alteration made it was evident that the center hole in the dynamometer would not fit over the crankshaft. Thus, a different type of adapter needed to be fabricated. The new dynamometer adapter utilizes the same taper as the clutch adapter, but it is extend 1.7 inches from the back of the taper in order to allow space for the dynamometer to fit on the crankshaft.

ENGINE MOUNTS

New engine mounts needed to be designed in order to place the Honda engine into the Polaris Pro-X bulkhead. These mounts needed to be stiff enough to hold weight of the engine and not deflect when the clutch engages, but also isolate engine vibration. The Silver Wing engine has dual counterbalanced shafts which reduce the engine vibrations, so stiff rubber mounts could be used to provide isolation of the vibrations from the chassis. The mounts also had to have adjustability for clutch alignment purposes. When choosing the positions of the mounts on the engine, the placement of the Honda engine in the scooter determined the location of the natural mounting points. When the engine was set into the bulkhead, the top front mounts rested on top of the shock tower brace and the rear of the engine conformed to the dip down from the tunnel to the bottom of the bulkhead. After seeing this, it was decided to design mounts that went from the brace to the top front mounts, and to use a cradle structure to support the back of the engine.

Front Mounts:

For the front, the shock brace was modified to accept the mounting points on the engine.

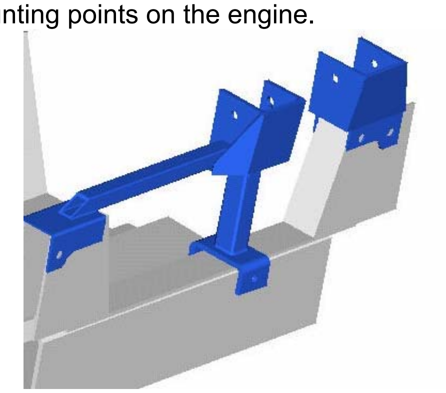


Figure 3: Solid model of front mounts.

Two pockets were designed to fit around the engine mounts. The stock rubber isolators were retained in order to isolate the engine vibrations. To eliminate the twist created when the clutch engages, the mounts needed to be wide. Since the scooter has a CVT type drive train, it too had the problem of eliminating the twist created when the clutch engages, so the mounting points are at the widest points and the rubber mounts were optimal for dampening and stiffness.

Rear Mounts:

To support the rear of the engine in the bulkhead, it was decided to use a cradle structure and two rubber isolators that attached to the sled via risers bolted into the bulkhead

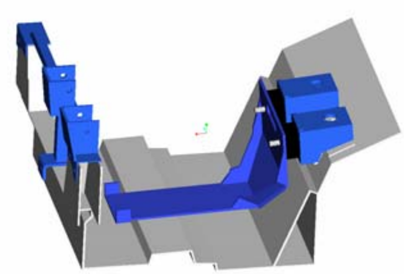


Figure 4: Engine mounts, front and rear.

The rear engine cradle and bulkhead attachments were made out of aluminum to save weight without worrying about heat because they are far enough way from the high temperatures that the front mounts are exposed to.

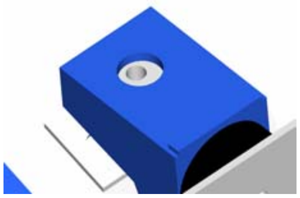


Figure 5: Adjustability of back mount for clutch alignment

Figure 5 shows the design built into the rear mount so that alignment of the primary and secondary clutches can be done. A top plate and bolt along with coarse surfaces keep the two pieces from sliding once lined up.

ENGINE MANAGEMENT SYSTEM

The engine management system selected is produced by the Robert Bosch Corporation, Small Engines Division USA. The Motronic system is based off of a version that is in production on the Polaris Frontier snowmobiles with modifications to better suit our specific design needs. This system was selected for the following reasons:

- Configuration for a parallel twin application
- Alpha-n style controller (TPS/RPM dependent Maps)
- Ambient Temperature Compensation
- Wide Range MAP Compensation (turbocharger)
- Mature Cold Start Routines
- Battery Voltage Compensation
- Programming Capabilities
- System sponsored to team for the 2005 CSC competition

Required Modifications

There were four major areas that had to be addressed for implementation of the Motronic System to the Honda FSC 600 engine. The system requires inputs for battery voltage, engine temperature, intake air temperature, throttle valve angle, manifold air pressure (MAP), and crank speed [5]. Out of these inputs there are the engine intrusive, fuel system, electronics, and calibration modifications that must be considered.

Engine Intrusive

The crank speed input to the system required the implementation of a ferrous tone wheel and a magnetic sensor (VR type) to read the tone wheel. The signal produced by this wheel is the basis for the injection of fuel and ignition timing. Since this system does not use a camshaft position sensor, it sparks and injects fuel every revolution of the engine. This is considered a wasted spark application.

The tone wheel could be either mounted externally on the output shaft of the engine or internally on the flywheel. The internal mounting was chosen for sensor reliability. An external sensor wheel would be exposed to moisture along with debris from the engine compartment and drive belt. This could cause the sensor to respond randomly or erratically after prolonged usage.

The problem presented when mounting the tone wheel internally is that it has to be mounted to the flywheel in order to be directly in phase with the crankshaft. This was accomplished by machining a tone ring with a band thickness of 2mm, turning a small step in the back of the flywheel, and pressing the ring in place. Also there was the problem of mounting the actual sensor. This was done by designing a mount that utilized the existing crank sensor mount points, which accommodates for fine radial adjustment of the gap between the sensor and the tone wheel.

Fuel System

To avoid modifications to the existing snowmobile fuel tank, an in-line external fuel pump and 3.5 bar regulator were used. In addition to these components, a fuel filter and pressure gage were added to increase the reliability and lifespan of the system. This external system is ideal for this application because it takes up virtually no volume of the snowmobiles gas tank and it is very easy to maintain and make adjustments to adjust.

The Bosch Motronic system only required one minor modification to the fuel rail of the engine. Previously the fuel rail for the Honda engine had a pressure regulator integrated into the end which only allowed fuel pressure of 3 bars. As mentioned before, an inline regulator replaced this and was mounted closer to the gas tank to eliminate the need for a fuel return from the fuel rail. This allowed for a variety of fuel pressures for testing.

Electronics

When implementing the Motronic system to the engine there were eight required components (non-invasive to the engine) that include:

- Air temperature sensor (ATS)
- Engine temperature sensor, mounted in water jacket (existing mount of stock engine)
- Manifold air pressure sensor

- Ignition coil
- Engine management system (EMS) wiring harness
- Engine control unit (ECU)
- Electrical circuit utilizing a lanyard tether to disable fuel and spark
- +12v scooter/recreational vehicle battery

The throttle position sensor was used from the existing fuel management system. Only minor differences between the Bosch sensor and the original Keihin sensor were encountered. This was easily accounted for with calibration changes.

Calibrations

A properly calibrated fuel injection system is essential for enhancing both engine emissions and fuel economy. Great improvements can be accomplished by optimizing for every speed/load operating point. Starting with a completely new engine management system requires a complete remapping of the base fuel and ignition angle curves. This was initially done naturally aspirated and with the new low compression 8.5:1 forged pistons to decrease the number of variables that had to be considered when the turbo was implemented. The naturally aspirated power/torque curve can be viewed below

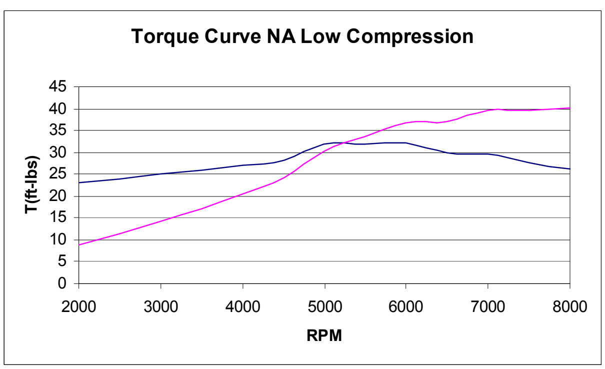


Figure 6: Naturally aspirated power/torque curves

The results of this calibration were a max torque of 32.1 ft-lbs at 6000 rpm and a max power rating of 40.1hp at 8000 rpm. These numbers turned out to be 15% lower than the stock Honda engine at 8000 rpm. This drop can be accounted for by the drop in compression and loss of tuned length exhaust headers.

Modifications were required to the software, MAP sensor, and base calibrations mentioned above to properly turbocharge the engine. Originally the fuel maps were set up for compensation of atmospheric pressures or altitude. This only gives a pressure range between 0 and 14.2psi. However the pressure range that we require is between 0 and 23psi with a boost pressure of up to 8.5psi with the addition of the turbo charger. Therefore the software had to be changed to allow this higher pressure and fueling multiplier greater than 1. Turbocharging requires more fuel due to the intake charge being pressurized and denser. In addition to the software changes, the Honda MAP sensor was inadequate, so it had to be replaced with a higher pressure range sensor. Due to the limited variety of ranges available, a 43.5psi or 3 bar sensor was used. The voltage this system is directly related to pressure, therefore fuel is compensated for increase or decrease of pressure. Below in Figure 7, the relationship between voltage and pressure during calibration of the engine is demonstrated. The last modification that had to be completed was changing the base fuel and spark maps. The base fuel maps had to be multiplied by a small percentage to account for a delay in the MAP compensation. Additionally the base spark curves had needed an overall decrease in spark advance to avoid pre-detonation within the cylinders

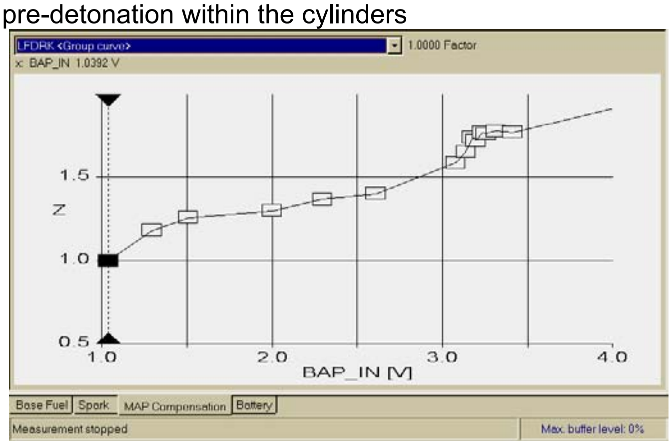


Figure 7: 2D Manifold Air Pressure (MAP) compensation curve. The Z is the time of injection in msec and the BAP_IN is the voltage input from the MAP sensor.

TURBOCHARGER

Naturally aspirated, a 600cc four-stroke engine, like the one from the Silver Wing, does not produce enough power for a snowmobile. To make the snowmobile competitive and fun, the power output has to be increased. A turbocharger can significantly increase an engine's output. It uses the momentum of the exhaust gases to compress the intake air. With the air pressurized, the engine draws in more oxygen per stroke. This additional oxygen allows more fuel to be injected. Hence more power is generated upon combustion.

A Garrett GT12 turbocharger was selected as the best turbo for this engine. It is a wastegated, water cooled turbo rated for engines producing 50 to 130 horsepower, with a displacement of 0.4 to 1.2 liters. One key feature that the GT12 turbo has is an adjustable boost control. This easy to use boost control allows for performance tuning to the desired horsepower.

The controller is set so that the turbo provides a maximum pressure boost of 8.5 psi at 6500 rpm. This was a compromise between turbo efficiency and fuel consumption. Figure 8 shows the compressor map for the GT12 turbo. The lug line for the snowmobile setup is in red square line, while the dotted line shows the line to follow for maximum efficiency. In order to follow the line better, the snowmobile would have to run with considerably higher boost pressures. Higher pressures are avoided for this competition because the power produced with 8.5 psi is sufficient (figure 9). Increasing the boost, and thus the power, would cause the engine to consume too much fuel and create more emissions. The important thing to be seen from the map is that the turbo avoids the choke region demarcated by the right-most line. The choke line represents the flow limit. When a turbocharger is run deep into choke, turbo speeds will increase dramatically while compressor efficiency will plunge, causing very high outlet temps. This will compromise turbo, engine and exhaust durability.

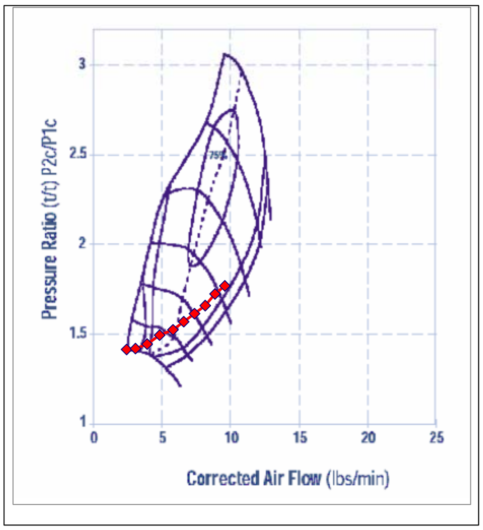


Figure 8: Showing turbo compressor map with snowmobile setup lug line superimposed [10]

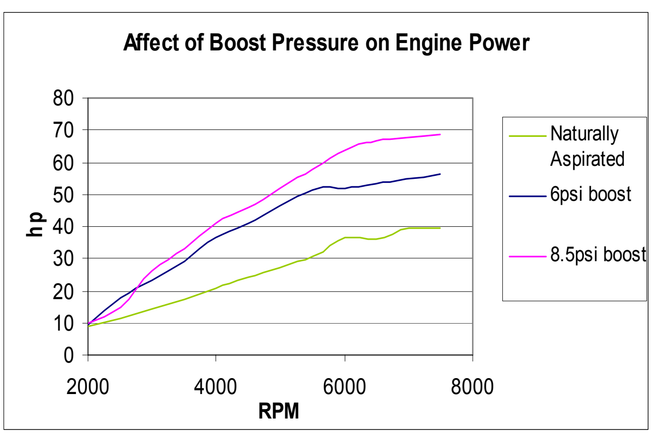


Figure 9: Showing power curves for the engine at different intake pressures

For the compressor map lug line the following equations were used [10]:

1) Pressure Ratio – Ratio of absolute outlet pressure divided by absolute inlet pressure.

$$PR = \frac{Boost + \Delta P_{Intercooler} + Atmos.}{Atmos - \Delta P_{AirFilter}}$$

2) Corrected Airflow – Represents the corrected mass flow rate of air, taking into account air density (ambient temperature and pressure).

$$CorrectedFlow = \frac{ActualFlow\sqrt{(AirTemp + 460)/545}}{P_{barometric}/13.95}$$

3) Actual Airflow – The ideal gas law was used to approximate intake airflow

$$\dot{M} = \frac{P}{RT} \cdot RPM \cdot V_{cylinder}$$

The results of the turbocharged calibrations were a max torque of 57.98, ft-lbs at 6438 rpm and a max power rating of 69.9 hp at 6257 rpm. This correlates to a 36.8% increase in power over the stock configuration and a 46% increase over the naturally aspirated configuration. A comparison can be viewed in figure 10 below:

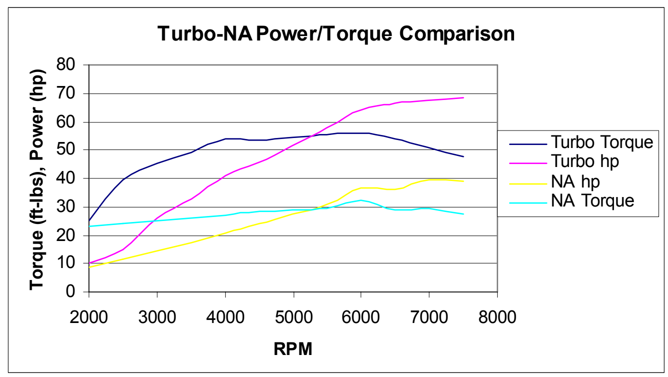


Figure 10: Comparison between naturally aspirated and turbocharged configurations

To prevent the turbo from overheating, coolant has to be pumped through the bearing housing. The bearings also require oil to flow through them to ensure lubrication. Both fluids enter the midsection of the turbo on one side and exit on the opposite. Since this turbo is for a specific automotive application, the middle block is designed to mate with the engine's respective oil and coolant ducting. In the snowmobile, the turbo is mounted separate from the engine, so an inlet and an outlet block had to be made with fittings for the oil and coolant hose attachments. The coolant line simply taps into the engine's coolant circuit. This integration of the coolant loop avoids the need for another pump and heat exchanger. Similarly, the oil used in the turbo is linked to the engine's oil system by an oil scavenger pump. The oil coming out of the drain on the turbo is at 0psi (gauge) and therefore needs to have a pump or gravity feed back to the crankcase of the engine. The specification for gravity feed is between one to two inches of drop for 12 inches of run with no sharp 90° bends. Due to tight space limits and desired geometry under the hood, the proper oil drain could not be achieved. The turbo oil system required the use of a pump. The pump that was chose to put into the snowmobile was a “J” style 8000 Series from Weldon Pump. The reasoning for this style pump over other pumps was the high torque for oil during cold operation, low current draw during operation, self priming, resistance to high temperatures, and a flow rate great enough to keep the turbo from being congested with oil [11]. The placement of the turbo was chosen so that minimal restriction occurred both at the exhaust and intake sections. Therefore the turbo was mounted in front of the sled directly after the exhaust headers with a smooth transition to the intake.

INTAKE DESIGN

Due to the addition of a turbocharger to the stock naturally aspirated Honda silver wing engine a new intake had to be designed. Considerations for the intake design included: maximization of the limited space available for the air box, obtaining the coldest densest air possible from the surroundings, minimization of the pressure losses through the charge tube and intercooler, and creating an esthetically pleasing intake system.

The first problem that we needed to address was the design and fabrication of an air box within the belly pan of the snowmobile. The placement of the turbo dictated where the air box needed to be placed due to the orientation of the turbo inlet. After measurements were taken the air box was fabricated.

The next problem addressed was how to get the coldest air possible to the inlet of the turbo. Originally our group intended to obtain the air by running a tube across the belly pan under the turbo, headers and catalyst to a hole in the hood that was used as the air intake on the stock snowmobile. We soon realized that this wasn’t a good design since the headers reached temperatures in excess of 1600 degrees Fahrenheit. We then modified our design to incorporate a ram air style air intake. We ran a tube from the air box to the front of the snowmobile and cut a hole in the hood to gain access to the ambient air. This design allowed the cold ambient air to have a direct path to the air filter, without being heated by the headers or catalyst. A K&N air filter was chosen to filter the air due to its high flow ability and superior filtering ability.

In order to minimize the pressure loss in the intake of the system we tried to minimize any sharp turn in the intake. The charge tube that went from the turbo to the intercooler was mandrel bent so that a smooth flowing transition could be obtained. If the mandrel bend tube wasn’t used we would have had to cut the tube at an angle and weld it together to make the transition from the turbo to the intercooler. This would have created sharp turns for the air flow and would have also increased the roughness of the tube at the weld points. These would have made the pressure loss in the intake greater.

The pressure loss in the intake system was calculated through the use of flow through tube equations and by modeling the intercooler at a tube bank. The overall pressure loss was calculated to be around 1 psi.

By calculating the pressure loss in the intake charge tube and the intercooler we were able to estimate a loss of approximately 1 psi.

Calculations for pressure loss:

<u>Intake charge tube pressure loss;</u>

$$\Delta P = f * \frac{\rho\, Um^{2}}{2D} * (X_2 - X_1)$$

Um= max air velocity, max volumetric airflow=236 CFM, with the diameter of pipe equal to 1.5 inches Um= 97.6 m/s

D= diameter of pipe=1.5 inches=. 0381 m

(X2-X1)= length of pipe= 25 inches= .635 m

Density=. 913 (kg/m^3) at max intake temp of 383 K

f is found by solving for the Reynolds number in the pipe

$f = (.790 * \ln \mathrm{Re} - 1.64)^{-2}$, for turbulent flow

$$\mathrm{Re} = \frac{4\dot{m}}{\pi * D * \mu},$$

$\mu = 222.67 * 10^{-7}\, Ns/m^2, \dot{m} = .102 kg/s$

Re=153082 so we have turbulent flow in the pipe

f =. 019

After placing the above numbers into the equation for the change in pressure we calculated a pressure drop of 1377.03 pa, which equals <u>.2 psi.</u>

.2 psi is a reasonable estimate for the pressure loss in the intake tube. The mandrel bent tubing allowed for the smooth airflow through the intake charge tube.

<u>Pressure loss through intercooler:</u>

In order to calculate the pressure loss through the intercooler the intercooler was modeled as a bank of aligned tubes. From this model we have;

$$\Delta P = Nl * \chi * \frac{V_{\max}{}^{2} * \rho}{2} * f$$

After taking measurements from the intercooler core we found;

Nl= number of tube banks = 7

Effective cross sectional area if the intercooler $= .14\,ft^2$

$$V_{\max} = \frac{St}{St - D} V = 13.98 m/s$$

<u>Estimations</u>:

D= diameter of tubes=. 25 in.=. 0064 m

St= vertical distance between tubes = .65 in.=. 0165 m

V= velocity of air= 8.56 m/s

χ, f found through Re in the intercooler

$$\upsilon = 24.55 \times 10^{-6}$$

$$\text{Re} = \frac{V_{\max} D}{\upsilon} = 36444.8$$

It is then found that that $\begin{matrix} f = .8 \\ \chi = .8 \end{matrix}$ through the use of available graphs that relate the Reynolds number and dimensions of the tube banks to the values of f, χ

After placing the above numbers into the pressure difference equation we found the pressure loss to be .1 psi. This calculation is rather conservative, as it doesn't account for the fins in the intercooler or the fact that there are plates that carry the coolant, not tubes. Taking these into account we estimate a 5% error in this model of the intercooler. Therefore our actual estimation of pressure loss in the intercooler is .5 psi.

<u>Overall pressure loss in the intake system;</u>

$$\Delta P_{tube} = .2 psi$$
$$\Delta P_{intercooler} = .5 psi$$
$$\Delta P_{total} = .2 + .5 = .7 psi$$

INTERCOOLER

The type of intercooler that would be optimal for the new design is a water to air intercooler for the following reasons:

- Small and compact
- Location or mounting position
- Allows sealed hood design to reduce noise
- Allows for a more constant intake temperature

The biggest disadvantage to this type of intercooler is the added weight from an additional coolant system. However it still allows for a sealed hood mentioned later.

The intercooler was designed to decrease pressure losses by having an orientation that eliminates the 180 degree bend in the charge tube from the turbo. The air enters the intercooler at an angle of 90 degrees to the flow path through the intercooler and is smoothly expanded into the fins. The exit of the intercooler is the opposite of the entrance with the directed flow being 90 degrees again from the exit of the core of the intercooler. This completes the 180 degree bend necessary for air delivery to the engine. Theoretically, by using the intercooler as a type of expansion chamber, it slows the flow down, and associated pressure losses are less when changing flow directions with smaller flow rates. The geometry of the intercooler that fits within our spatial limits can be viewed in the figure below:

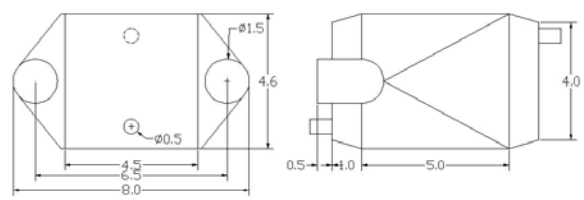


Figure 11: Intercooler Geometry

The projected goal for engine intake temperatures was a constant temperature of 25 degrees Celsius. The intercooler has potential for cooling at a heat transfer rate of 74.6 kW. The temperature at the exit of the turbocharger under full loads ranges between 45 and 110 degrees Celsius. This correlates to a necessary heat transfer rate of 55.7 kW to keep the intake temperature at 25 degrees Celsius. Therefore, the intercooler will not be the limiting factor when trying to cool the intake temperatures.

The new design problem that surfaces is sizing the snow to air heat exchanger that cools the coolant from the intercooler. First the convection heat transfer coefficient is calculated. This is done starting with equation (1) [2] seen below. For this specific situation we have a laminar flow within the heat exchanger. This is determined from the small Reynolds number computed using equation (2) [2]. The use of a hydraulic diameter is required in these calculations due to the rectangular shape of the coolant flow cavity within the exchanger. Since the exchanger is not infinite in length Le, it is assumed that there are entry region affects. Since the flow is laminar and there are entry affects, the Nusselt number is computed from equation (3) [2]. From here the average convection heat transfer coefficient, hi, is calculated using equation (4) [2]. From here, equation (1) is solved for the surface area [2]. Finally, the length of the heat exchanger Le can be solved for, given a desired heat transfer rate q. Equations are shown below:

1. $q = hi\ As\ LMTD$ [2]

$$As = \frac{q}{\mathrm{hi}(LMTD)}$$

$$Le = \frac{As}{\mathrm{N}(2h + w + s)}$$

hi = internal convection coefficient
As = surface area
Le = Overall length of heat exchanger
N = Number of fins
h = height of fin
w = thickness of fin
s = fin spacing width

2. $$Red = \frac{\rho\ V\ Dh}{\mu}$$ [2]

3. $$Nud = 1.86\left(\frac{Red\ Pr}{\frac{L}{\mathrm{D}}}\right)^{\left(\frac{1}{3}\right)}\left(\frac{\mu}{\mu\ s}\right)^{.14}$$ [2]

4. $$Nud = \frac{hi\ Dh}{k}$$ [2]

k = thermal conductivity

The results from the calculations above reveal that the intercooler needs to be 34.5 inches in length with 0.375 inch fins to achieve a heat transfer rate of 55.7 kW. The results from the optimization can be view below in figures 12 and 13. Assumptions for these calculations are:

- External surface of heat exchanger is covered in snow (deep snow conditions)
- Ambient temperature is 0°C
- Surface temperature of heat exchanger is 0 °C with the assumption that the outer heat transfer coefficient is very large.
- Steady-state heat conduction
- Heat transfer only through bottom of heat exchanger (the top and sides are considered to be insulated
- Negligible radiation exchange between heat exchanger and surroundings
- Negligible fouling factor on heat exchanger surfaces
- Constant properties
- Constant internal volume flow characteristics at 5 gal/min
- Uniform convection coefficient over outer surface (with or without fins)
- Fixed optimized fin length of 3/8”

The actual length of the heat exchanger that was used is 40 inches, with the same fin dimensions as used in the calculation. Therefore adequate cooling can be achieved using this snow to water heat exchanger. In testing, a range of temperatures between 25 and 33 degrees Celsius were recorded, which is an acceptable range for calibration purposes.

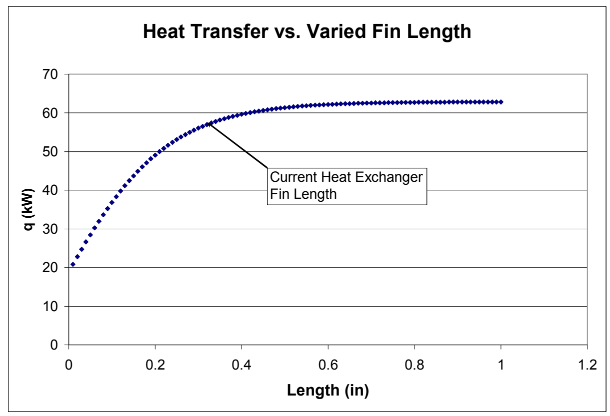


Figure 12: Fin Length Optimization

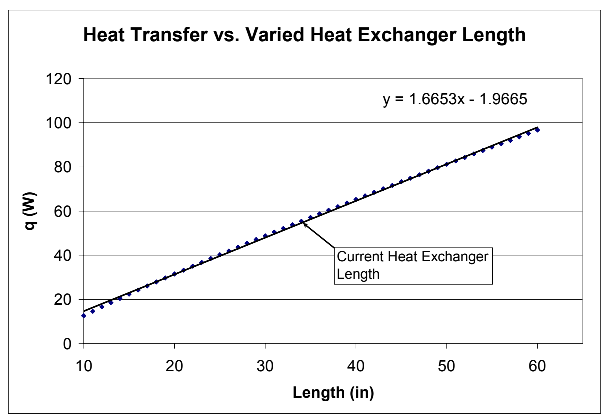


Figure 13: Exchanger Length Optimization

RESONANCE TUNING

Resonance tuning is a very important aspect of this design. The intake runner length, measured from the entrance of the intake system to the back of the intake valves, can be optimized for maximum engine efficiency and performance. In this case, because the design employs an intercooler, the length is measured from the exit of the intercooler to the back of the intake valves. This optimum length is based on the amount of time it takes for an expansion wave coming from the intake valve to travel the length of the intake system, reflect at the entrance of the intake plenum, and return to the valve. If the intake length is optimized, this intake-ramming wave will return to the intake valve just as it is closing which forces the intake charge into the cylinder. If this is not tuned properly it can work the opposite way and hinder your flow which will significantly change the operating characteristics of the engine. In order to calculate this tuned length the Helmholtz resonance model is used. This model is an electrical circuit analogy developed by H.W. Engelmann for manifold design applications. Basically it calculates the frequency at which this intake ramming effect mentioned above will happen. This frequency tends to be twice that of the natural frequency of the piston. Since it is a parallel twin application we can assume that it is a single cylinder engine and due to its 360 degree crank configuration there is no need to consider out of phase frequency effects.

- The first step is to calculate the effective volume or the volume of the piston mid stroke:

$$V_{eff} = \frac{V_D}{2}\frac{(CR+1)}{(CR-1)}$$

V_D = Displacement Volume

CR = Compression Ratio

- Next we define the system as a system defined by inductances and capacitances:

$$Inductance\,(I_i) = \left(\frac{L}{A}\right)_i$$

L = Length of the Runner

A = Cross-sectional Area

- As in circuits these inductances can be calculated separately for different cross-sectional areas and lengths and summed up:

$$(I_e) = \left(\frac{L}{A}\right)_i + \left(\frac{L}{A}\right)_{i+1} + ... + \left(\frac{L}{A}\right)_n$$

- So now the separate inductances can be calculated for the port, intake runner, throttle body, and plenum. With these values it is possible to calculate the inductance ratio:

$$a = \frac{\left(\frac{L}{A}\right)_2}{\left(\frac{L}{A}\right)_1} = \frac{I_{T.body} + I_{plenum}}{I_{port} + I_{runner}}$$

- Next the capacitance ratio is determined which is the ratio of the secondary volume to the primary or effective volume:

$$b = \frac{V_{eff}}{(n-1)\,x\,V_{runner}}$$

(n-1) = Volume of intake runners that are ineffective

- Next the Induction system resonances must be calculated:

$$f_1 = \frac{1}{2\pi}\sqrt{\frac{A-B}{2\,x\,a\,b\,(IND)_1\,x\,V_{eff}}}$$

$$f_2 = \frac{1}{2\pi}\sqrt{\frac{A+B}{2\,x\,a\,b\,(IND)_1\,x\,V_{eff}}}$$

$$A = (a\ b + a + 1)$$

$$B = \sqrt{(a\ b + a + 1)^2 - 4\ a\ b}$$

$(IND)_1 = I_{port} + I_{runner}$

- Determine the primary resonance:

$$f_p = \frac{1}{2\pi}\sqrt{\frac{1}{I_{port} + I_{runner}}\,x\,V_{eff}}$$

- Determine the frequency ratios:

$$X_1 = \frac{f_1}{f_p} \qquad X_2 = \frac{f_2}{f_p}$$

- Determine the tuning peaks:

$$N_1 = X_1\,x\,N_p \qquad N_p = K_1\,x\,C\,x\sqrt{\left(\frac{A_1}{L_1 V_d}\right)\left(\frac{CR-1}{CR+1}\right)}$$
$$N_2 = X_2\,x\,N_p$$

A_1 = Average area of runner and port

K_1 = 642

$L_1 = L_{port} + L_{runner}$

C = Speed of Sound (340 m/s)

**All formulas are base off of Heywood's Versions [3]

Completing the calculation for the original naturally aspirated engine the second two tuning peaks N_1 and N_2 are not needed due to the fact that there isn't a plenum. Therefore the primary tuning peak (N_p) is the only relevant peak that needs to be calculated. This tuning peak correlates to the piston speed at the engines peak torque which came out to be 8160 RPM. The actual peak of the engine naturally aspirated is 8250 RPM which is only off by 1.09 percent.

The calculation for the turbocharged application had to take into consideration the added intake runner length to the intercooler and the intake plenum or intercooler in this case. This is where the other two tuning peaks (N_1 and N_2) come in to the picture. If the frequencies are large enough to be within the operating range of the engine then they could cause multiple peaks within the engines torque curve. In this case the frequencies were very low meaning that the effects of the intake plenum were negligible. However the increase in intake plenum length by the primary tuning peak equation drops the peak torque to 6089 RPM. When dynoing the engine very similar results were achieved, with a peak torque value at 6100 RPM.

EXHAUST SYSTEM

<u>Catalyst</u>

The three main criteria for the selection of the catalyst were the surface area in the monolith, the backpressure produced, and it had to be a three-way catalyst. In these conditions, the three-way catalyst has the ability to

convert hydrocarbons, carbon monoxide, and nitrogen oxides into water, carbon dioxide and nitrogen gas.
In conjunction with Lubrizol, the most efficient three-way catalyst was selected based upon the engine intake airflow (SCFM). Flowrate can be approximated by:

$$\text{Flowrate(SCFM)} \cong \frac{EngineDisplacement \times RPM \times V_E}{14.16 \times EngineCycle}.$$

A value of 2 was substituted for V_E because the engine is a four stroke, turbocharged, and intercooled. The RPM used is that of the maximum RPM (6500) for the engine. Once the SCFM was determined, the correct size catalyst could be selected from Lubrizol.

Muffler

To control the noise emitted from the engine exhaust, the muffler from the Honda Silver Wing was used. It functions as a reactive muffler designed to dissipate sound waves. It was selected because it is designed for use with the Silver Wing 600 engine. It is constructed to act as a broadband attenuator. This type of muffler is effective because the exhaust noise is composed of many different frequencies. A broadband attenuator reduces the amplitude of the frequencies with the highest peaks across the entire range of noise produced. Its construction involves three reactive chambers connected by pipes of various sizes. There is also a glass pack around the internal pipe just before the outlet. This is a section of the pipe where it is perforated and is surrounded by an outer shell containing sound absorbing fiberglass fill. The muffler's body is three layers: an outer shell, a perforated inner shell, and sound absorbing fill in-between. Together, they inhibit sound waves from exiting through the muffler shell by creating a barrier and making them travel through the chambers. Finally, the slim oval design fits well into the exhaust section of the chassis.

However, the muffler could not be used as is. The muffler needed to be modified to reduce the backpressure from the turbocharged engine. The chamber dimensions, the lengths of the passages between the chambers, and the inlet and outlet pipe lengths were kept the same, but the pipe diameters were increased to reduce pressure. The passages were not all the same diameter, so the sizing modification focused on the most restrictive one. The passage between the second and third chamber was originally 1.125 inches in diameter. This is sized for the 50 horsepower stock engine. For a 100 horsepower engine, a pipe of 2 inches is recommended [7]. Interpolating for 70 horsepower which is what the modified engine is producing, it was determined the smallest pipe in the exhaust should be 1.5 inches. The diameters of the other passages were increased proportionally to maintain the relative sizes. The glass pack was also remade for the larger outlet pipe.

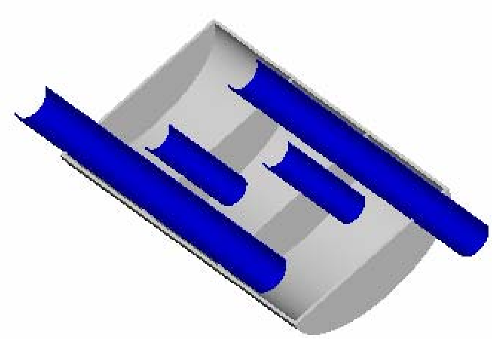


Figure 14: Cross Section of Muffler

Although the modifications will probably increase the sound transmitted through the muffler, it is assumed that since it is contained within the snowmobile hood and sends exhaust towards the ground, the engine noise will not be greater than it is on the scooter.

ELECTRICAL SYSTEM

The layout of the wiring system is that of three separate wiring harnesses integrated together. This was due to the existence of three discrete circuits; one from the stock Polaris snowmobile, the Honda Silver Wing 600, and the Bosch Fuel Injection system. The stock Honda Silver Wing battery, starter, and engine indicator sensors comprise the main power supplying circuit. The Bosch Fuel Injection runs off a separate independent circuit, with the battery as the source using the ignition key as the primary switch. The wires left from the factory snowmobile are the hand warmers, tether switch, and tail light. To display the exhaust temperature, boost pressure, and oil temperature, EGT, Inc. digital monitoring displays used. In addition an oil pressure gauge was also installed to ensure sufficient pressure was maintained for the engine and turbocharger.

The twelve volt battery from the Honda Silver Wing 600 provided enough potential difference for the electrical system on the snowmobile, to be used as the power supply. The original Polaris instrument panel was removed and the dash unit from the Honda Silverwing Scooter was used which consisted of a tachometer, speedometer, and warning indicators. A Hall Effect sensor runs the speedometer. It reads or counts the spinning teeth on the gearing from the Honda transfer case. In-order to recalibrate the speedometer to read the snowmobile main shaft, a ten tooth sensor gear was installed in place of the stock snowmobile cable driven speedometer. However, switching to the Bosch Fuel Injection system, the output signal produced operates at half the needed input frequency for the Honda dash. This is the result of the Bosch Fuel Injection controls ignition to the coil by low side drive. The Honda tachometer still reads the input from the coil but displays only half of the RPM's the engine is truly operating at. The solution to this was with the use of a simple frequency multiplier. The frequency multiplier multiplies the frequency by a factor of two so that correct RPM can be read.

The fuel injection system was a unit from Bosch Technologies. This Bosch electronic control module, coupled with a ECM (Engine Control Module) allows the user to adjust fuel curves, ignition, and monitor engine diagnostics during the dyno testing sessions. Majority of the factory wiring from Polaris was removed with the exception of the handwarmers, kill switch, and taillight. Power supplied for these components runs into a circuit with fused relays that was not necessary for the Honda system. The snowmobile's cooling system consists of two electric water pumps. The stock water pump, caused a mechanical interference due to the snowmobile's increased clutch size, opposed to the Honda clutch. The coolant for the intercooler runs on a separate system, therefore, a second water pump is needed. The two water pumps are connected to an illuminated three-way switch located just below the dash, but easily accessible. During cold engine starts the engine itself can be brought up to operating temperatures sooner without initial coolant flow. For the novice rider the switch can be left on so that once ignition is turned on, the water pumps turn on and stays on. In the case for coolant temperatures that could be detrimental, the Bosch ECM shuts the motor down to avoid engine damage.

For closer monitoring of actual exhaust and engine temperatures, EGT digital displays were used, rather than the simple OEM Polaris warning light. The EGT digital display can also function as an analog sensor. This additional feature allows the team to monitor the boost pressure at the intake manifold by connecting a 5 volt regulator to the boost sensor and sending a signal to the EGT. To obtain temperature readings, liquid thermocouples were installed in the oil and water lines. For a complete wiring schematic see “Appendix – A”.

HOOD

A new hood was necessary because the stock Polaris Pro-X hood proved to be inadequate in a few areas. The change from a two-stroke engine to a four-stroke engine meant a significant change in necessary clearances, so this issue had to be addressed. Also, the stock hood did not provide adequate noise reduction; this can be attributed to the thin plastic and multitude of hood vents.

The new hood was designed to provide adequate space for the new engine and also to keep weight and noise to a minimum. To create this new hood a plug had to be made to model the mold form. The stock hood was placed on the sled, and the necessary clearances for the engine and turbocharger were measured. With these clearances, foam and body filler were used to build the hood up to the necessary shape. Once completed, the plug was used to form a mold. From the mold the new hood was made. After sound testing with an electronic stethoscope, it was determined that a large amount of sound was being emitted from the space between the windshield and the hood. To eliminate this problem the windshield was molded into the hood. Instead of using the inadequate plastic of the stock hood, Kevlar enhanced with PDP was chosen because of its superior strength and noise reduction capabilities. PDP is a constrained layer damping foil for laminating into fiberglass panels to improve sound transmission loss and absorb structural acoustic energy [8]. It interacts with the structure of the composite material to dampen sound waves. To further reduce noise, acoustic melamine was added to the underside of the hood. The particular melamine we used is called Hushcloth Melamine and was chosen for its light weight, excellent sound absorption, and high heat resistance. The high heat resistance is particularly important because the lack of hood vents leads to extremely high under-hood temperatures; and even if the melamine's maximum temperature is reached, it will char rather than burn. Other types of sound absorbing foams are much more flammable and would create a fire hazard. While other materials have higher sound absorption coefficients (figure 15), the Hushcloth melamine is the best foam for fire resistance and absorbs sound better than other fire resistant materials, like fiberglass. It also works the best in the 1 kHz to 4 kHz range, which is the most comman weighted frequency in the dBA scale with our engine set up.

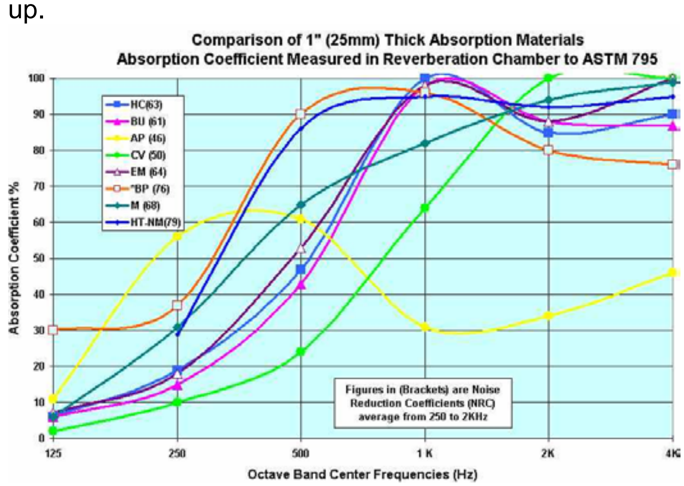


Figure 15: Sound transmission through various commercially available sound absorbing materials. Melamine foam (M) (dark green) is used in the snowmobile hood.

SUSPENSION

The stock suspension in the Polaris Pro-X chassis is an edge suspension with Ryder-FX shocks. The stock suspension is designed for mogul race applications therefore it is set up very stiff and does not absorb bumps as well as a trail sled could. To correct this problem with the stiff ride we had the shocks re-valved to allow for a more even dampening rate allowing the sled to ride smooth over small and large bumps. Race applications use a very strong spring in the rear of the sled so that the suspension does not bottom out over a very large jump. The strong springs makes for a rough ride over medium to small bumps because there is not enough force for the spring to bend and absorb the bump, we corrected this by replacing the stock spring with a Polaris 800 XC-SP spring which is weaker and proven to absorb bumps very well in the edge suspension. The other benefit to having an edge suspension under the rear end of our sled is that the transfer created by the suspension is much better than previous suspension and better than some others on the market now. Transfer is the ability of the suspension to take power from the motor and put it to the ground. The edge suspension does this by keeping the same amount of track on the ground, thus keeping friction as high as possible between the ground and the track.

TRACK

A major contributor to the overall noise of the sled is the drive train and track. The stock track of the Polaris Pro-X chassis was 14 inches wide with a 1.5 inch profile Camoplast track. In order to reduce the noise from the source extensive testing was done to analyze the influencing factors. Through analysis a correlation was found that as profile height increased by 0.25 inches the sound level raised approximately 1dBa. The next area that was analyzed was the stiffness of the track lugs. Because of limited ability to obtain various stiffness compound tracks only one could be tested, and it was a compromise between being soft but still durable. The stiffness of the rubber is measured on a durometer scale. Track lugs conventionally range between 90 and 110 durometer stiffness. Camoplast, a popular track manufacturer, was willing to work with the UB CSC 2005 team on making a prototype track for testing and research. The base prototype track lugs measures 80 durometer and the tip of the lugs measure 60 durometer. Additionally, the track belt is 0.0625 inches thicker. The ideal behind these adjustments is that by having a softer lug there will be less ground contact noise and a thicker belt will help suppress the problem frequencies by blanketing or absorbing them with the increased mass. To further decrease the noise levels created by the track, we placed Visco-Elastic Material inside the tunnel to absorb noise created by the vibration in the suspension. We also encased water resistant melamine foam in millennium plastic and riveted it to the bottom of the running boards to isolate the noise inside the suspension area. The results of testing with these new parameters reveals a 1.3 dB decrease in overall noise levels emitted.

RESULTS

EMISSIONS REDUCTION

Emissions data was obtained under constant loads and RPMs. The tests were done at three RPMs (4095, 4745, and 5395 RPM) and three engine loads (149, 267, and 404 kPa) which were estimated to be common measurements of the sled during normal operation. In order to calculate the specific emissions of the engine, the RPM, fuel mass flow rate $(\dot{m}_f)$, and torque (T) needed to be measured. From these measurements the power (P), then the brake mean effective pressure (BMEP) and the specific emissions could be calculated as follows:

$$P = 2\pi x RPM x T$$

Where V_{sw} is the engine displacement

$$BMEP = \frac{P}{V_{sw} RPM}$$

$$Specific\ Emissions of\ \beta\left(\frac{g}{kW*hr}\right) = \frac{[\beta(ppm)*10^{-6}]\left[Molecular Weight\left(\frac{g}{mol}\right)\right]\left[\frac{\dot{m}_f\left(\frac{14.6}{\phi}+1\right)}{28.8-0.2\phi}\right]\left[3600\frac{s}{hr}\right]}{Brake Power}$$

Note: β is the emission molecule of interest.

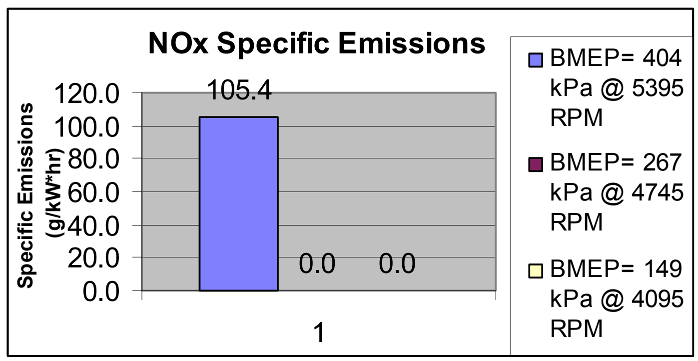
$$MW\ of\ HC = 16\frac{g}{mol}$$

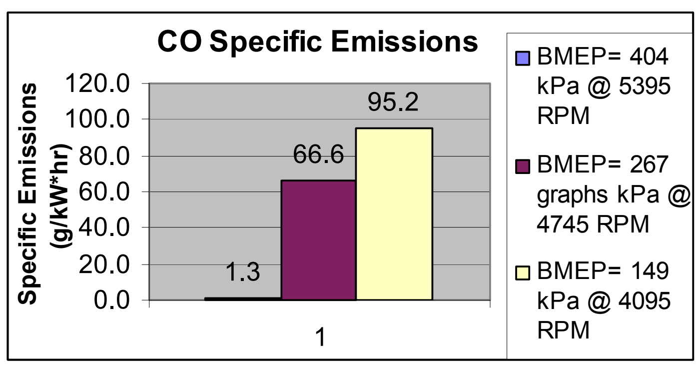
$$MW\ of\ CO = 28\frac{g}{mol}$$

$$MW\ of\ NOx = 30\frac{g}{mol}$$

Note: The MW of NO_x is approximated at 30 g/mol because the vast majority of NO_x particles emitted from a SI engine are NO.

EMISSIONS TESTING RESULTS





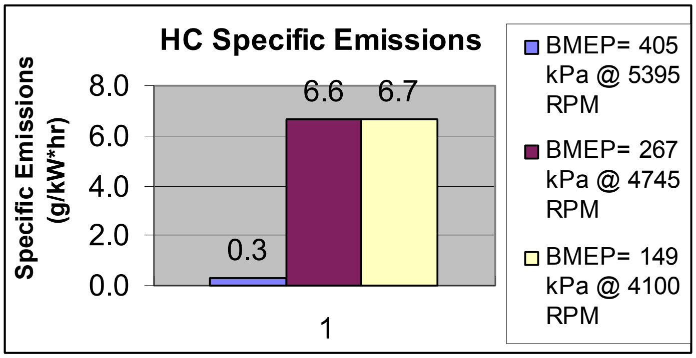


Figure 16: Emissions testing Results

NOISE REDUCTION

To measure the effectiveness of the sound reduction strategies in the snowmobile design, simple sound tests were performed according to SAE J192 standard for snowmobile noise testing. This is not specifically how the test is performed for the competition, but is accepted as a valid way to measure snowmobile noise emissions, and can be performed with a single sound pressure meter. Measurements were made at the midpoint of a 500 ft., full load acceleration run. For comparing results, noise emissions were measured for the stock Polaris Pro-X snowmobile, the snowmobile with the quiet track installed, the snowmobile at last years competition, and with 2005 modifications had been added.

Date		12/29/04	12/30/04	Mar-04	2/25/05
Conditions		Trace	Trace	hardpack	hardpack
Upgrades		Stock sled	new track	CSC 2004	After 2005 Mods.
		noise (dB)	noise (dB)	noise (dB)	noise (dB)
Run 1	left	80.5	79.4	78.3	77.1
	right	83.3	81.2	79.5	78.3
Run 2	left	80.2	79.2	77.9	76.8
	right	82.9	81.9	79.1	78.4
Run 3	left	80.4	79	78.2	76.9
	right	83.1	81.5	79.4	78.5
Avg.	left	80.4	79.2	78.1	76.9
	right	83.1	81.5	79.3	78.4
	total	81.7	80.4	78.7	77.7

Figure 17: University at Buffalo's Sound testing results

CONCLUSION

The concept of reducing the noise and emissions of a snowmobile with out greatly reducing the performance is a complex task. For reducing the emissions the strategy chosen was a four-stroke engine with a three-way catalyst. With these changes implemented the emissions from the setup are significantly less than that from a conventional two-stroke snowmobile. With the Honda engine and the catalyst implemented into the snowmobile the emissions should have lower emissions than that of most stock snowmobiles. The strategy for noise control was a modified Honda Silver Wing muffler and various types of sound dampening/barrier materials. Through testing and analysis it is clear that the steps taken to reduce noise were effective. By decreasing the backpressure from the muffler, and increasing the engine intake charge with a turbocharger and an intercooler the performance is still comparable to that of a commercial trail riding snowmobile. The UB 2005 CSC Team's design shows the potential for snowmobiles to be cleaner, quieter, and environmentally friendly, while still being entertaining.

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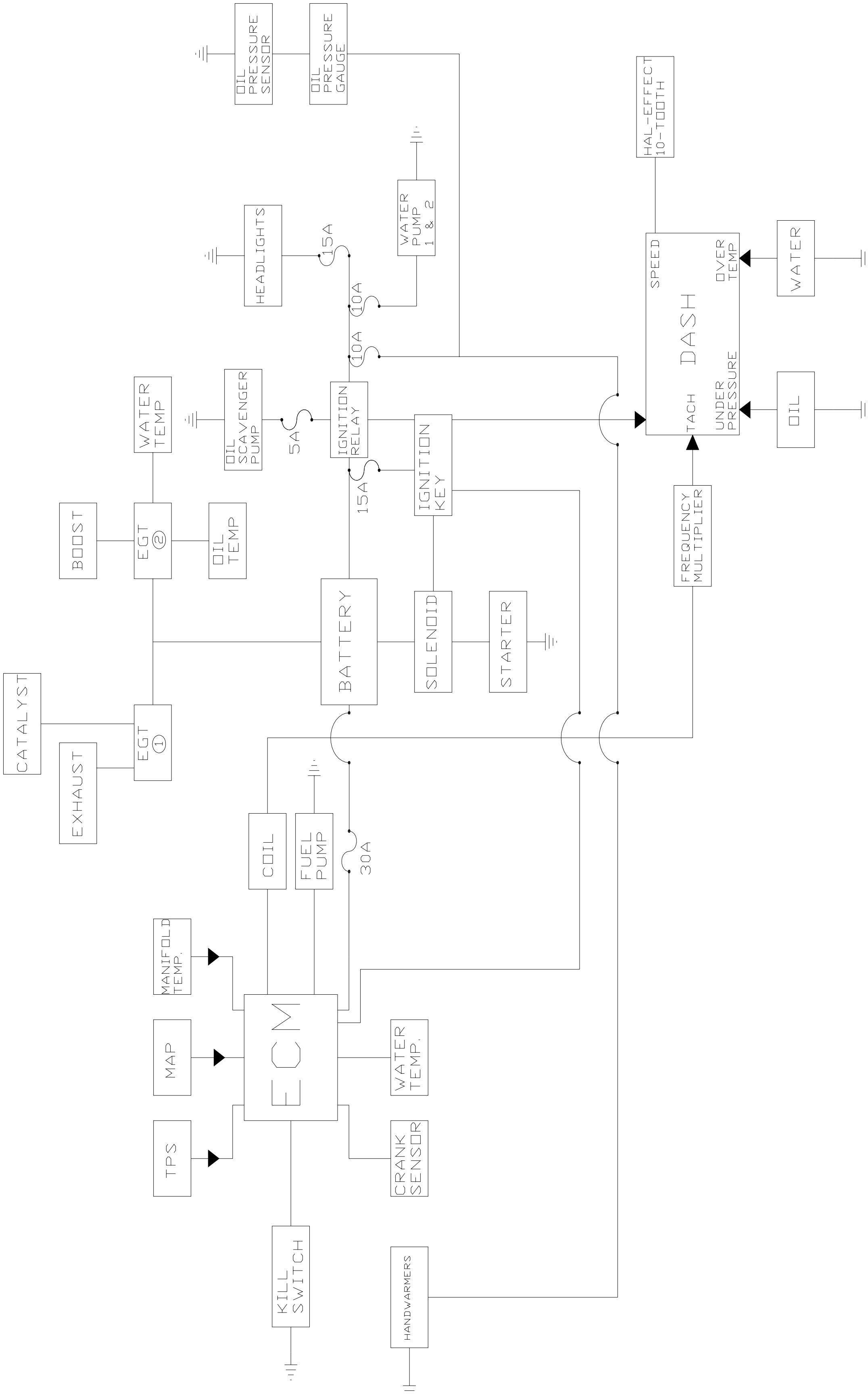
APPENDIX A



OIL PRESSURE SENSOR
OIL PRESSURE GAUGE
HAL-EFFECT 10-TOOTH
HEADLIGHTS
15A
WATER PUMP 1 & 2
10A
10A
SPEED
OVER TEMP
WATER
DASH
TACH
UNDER PRESSURE
OIL
WATER TEMP
OIL SCAVENGER PUMP
5A
IGNITION RELAY
IGNITION KEY
15A
FREQUENCY MULTIPLIER
BOOST
EGT ②
OIL TEMP
BATTERY
SOLENOID
STARTER
CATALYST
EXHAUST
EGT ①
COIL
FUEL PUMP
30A
MANIFOLD TEMP.
MAP
TPS
ECM
WATER TEMP.
CRANK SENSOR
KILL SWITCH
HANDWARMERS