Adapting Diesel Technology for use in Recreational Vehicles, Snowmobiles

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ABSTRACT

For the 2012 SAE Clean Snowmobile Challenge, The University at Buffalo Clean Snowmobile Team has chosen to utilize a 2011 Polaris Turbo IQ chassis mated with a small 3 cylinder turbo diesel engine. The engine of choice is a Daihatsu made, Briggs and Stratton marketed DM950DT indirect injected engine. The base engine in its stock form meets 2012 CARB regulation. With the equipment selected, our team was able to produce a snowmobile with over 55 hp and 80 ft-lbs of torque all while keeping fuel economy well above 30 mpg. Emissions were also kept very low with the use of a diesel oxidation catalyst and a diesel particulate filter by Emitec.

INTRODUCTION

Due to the increasing regulations on exhaust and noise emissions on modern snowmobiles, there is an ever-growing need for further development of new technologies that assist in making snowmobiles cleaner and quieter. The Clean Snowmobile Challenge is a collegiate design competition for student members of the Society of Automotive Engineers (SAE) to accomplish these goals. "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, carbon monoxide and particulate matter than conventional snowmobiles, without significantly increasing oxides of nitrogen emissions." Furthermore 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. Guidelines for the 2012 SAE Clean Snowmobile Competition state that this year's entries into the competition must exceed the EPA 2012 emissions standards just as the manufacturers must do. Additionally, entries must pass the current Snowmobile Industry noise test minus two decibels. Our team has again chosen a diesel engine as the engine of choice for this competition as it has great potential not yet exploited in the recreational vehicle market. The diesel engine is vastly superior to gasoline two and four stroke engines with respect to efficiency, emissions, and torque output, all of which are very important design factors of modern snowmobiles.

CHASSIS SELECTION

With an aging 2005 Polaris chassis, The University at Buffalo's Clean Snowmobile Team began searching for a new chassis. Since the same motor would be used for the next coming year, a chassis with a similar bulkhead design and plentiful space was needed. Other considerations were cost effectiveness for the team, which was if discounts could be had and/or if old parts could be used again on the new chassis. Although many sleds were considered, the final choice was a 2011 Polaris Turbo IQ because of the total cost and the familiarity current team members already has with Polaris chassis'.

ENGINE SELECTION

Currently the industry standard for snowmobile engines includes only 2-stroke and 4-stroke gasoline engines. Our team feels that the manufacturers producing these snowmobiles have failed to consider the compression ignition engine as a viable option. The following diagram depicts these engine options with five important contributing design factors taken into account for engine selection. The chart displays each engine's scaled grade in each category, with 10 being most favorable and 0 being least favorable.

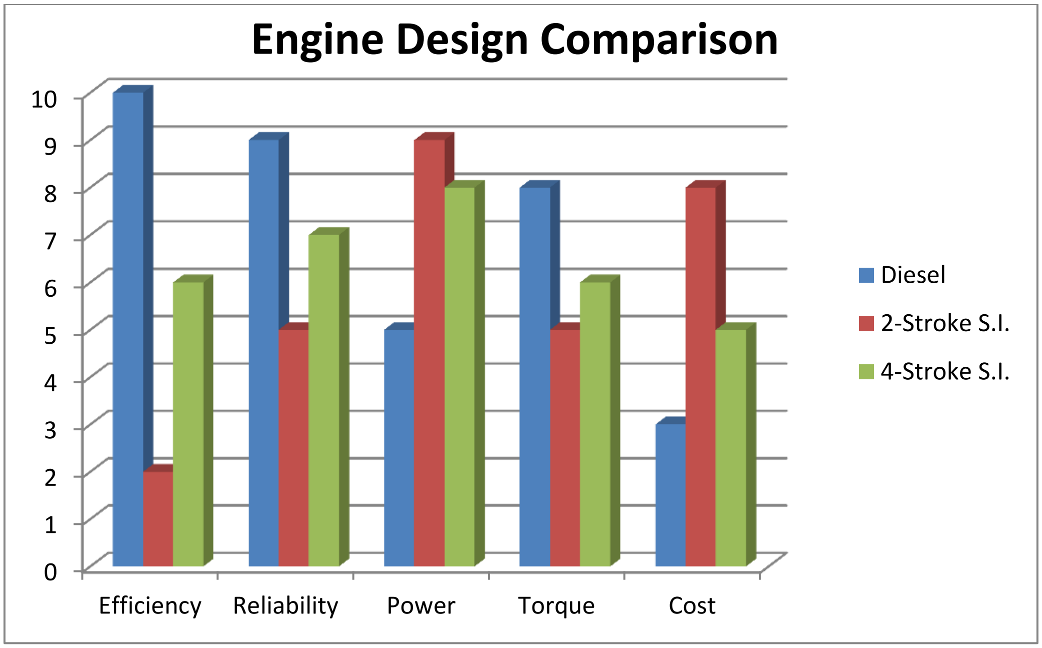

Figure 1: Scaled Comparison Chart of Viable Engine Options

The University at Buffalo Clean Snowmobile Team chose a diesel platform over a flex fuel ethanol platform, 2 or 4 stroke, for multiple reasons. One such reason is due to the "compromise" in the engine design for a flex fuel vehicle. As reported by the Department of Energy, a flex fuel vehicle operating on gasoline alone compared to diesel has a fuel economy of approximately 20 to 40% less. When E85 is run, the fuel economy drops from 40% to as much as 65%[5]. Another reason for our choice of a diesel is due to the lower total energy and carbon footprint of a biodiesel blend compared to an ethanol blend. As simulated in an Argonne National Laboratory GREET Well to Wheel analysis using default setting comparing E15 and BD5, biodiesel has a lower overall energy use and lower overall CO2 emissions[6]. Fuel flexibility is another factor that played into our decision. Bio-fuel and regular pump diesel are nearly identical reducing much, if not all concerns when switching between the two. With regards to the amount of fuel, no change is needed. Unlike a gasoline-fueled vehicle that needs to stay near to its stoichiometric ratio for operation and control its power output with a throttle body, a diesel changes its power output via the amount of fuel it injects every power cycle. Therefore, even if the energy content of the fuel changed, it would only hinder the full power operation, and have no effect at partial throttle.

The teams' largest concern when considering a diesel engine was for lack of a better term, power. The diesel engine excels in overall efficiency, emissions and maintenance, but for most snowmobilers, those concerns are not nearly as important as speed and handling. Considering the scope of this snowmobile competition and the points scheme, the team felt that even with these concerns the diesel provides the most potential.

To choose the actual model of engine to be used the team again mapped out criteria, rated each and made a final decision. The most basic requirement that helped to eliminate many options was size. The motor needed to fit in the bulkhead and not have to be pulled out each time service of any kind was required. Second, and equally important was the BSFC of the engine, as this attribute with directly effect fuel economy of the sled. Next, weight was considered. The team sought to find a lightweight engine without sacrificing reliability or toughness. Fourth, was stock power rating. As a basic guideline the team only considered engines that made over ~50 hp. Emissions were also considered but would ultimately be highly modified regardless.

The base Daihatsu DM950DT engine was chosen for its small size it fit within the frame rails and for its 2012 EPA compliance. The engine is a three cylinder, indirect injected turbocharged diesel engine. Other applications of this engine include 2003-current year Kawasaki Mule diesel utility vehicles and Husqvarna commercial lawn mowers.

Displaced volume	980 cc
Stroke	78 mm
Bore	73 mm
Connecting Rod	255 mm
Compression ratio	19.4:1
Number of Valves	2

Exhaust Valve Open	45° BBDC @ 0.15 mm lift
Exhaust Valve Close	10° ATDC @ 0.15 mm lift
Inlet Valve Open	10° BTDC @ 0.15 mm lift

Figure 2 – Current Engine Specifications

The internals of the motor in past years has been forged H-beam connecting rods manufactured by CP Carillo along with forged aluminum pistons by Arias. Upon further research it was clear that the motor's BSFC was much less than the factor rating. Also, soot production was higher than expected. It was noted that the compression ratio selected for the pistons was 17.4:1, down from a stock 24.8:1. The reason for this was solely to make more power. In the first years running this motor the team has serious issues keeping up with the rest of the sleds.

To optimize performance this year we investigated the validity of the choice to go with these lower compression pistons. When lowering compression you allow for a larger turbo charged to be fitted. The turbo charger can increase the mean effective pressure in the cylinder, which before was mainly controlled by the high compression ratio. Therefore decreasing the compression ratio enables the user to better control the amount of air going into the cylinders, thus the power. More fuel can then be added and the motor can be tuned to make more power depending on the user inputs. Disadvantages include severely decrease fuel efficiency and less efficient combustion when not under boost. Naturally the team sought to find a happy medium that would combine fuel efficiency with power.

Piston design was the next step forward. Previously the pistons used were a symmetrically dished style piston. This design is better suited for gasoline turbo charged engines. In our engine, being indirectly injected, there is a hot-spot on the piston where the majority of the air-fuel mixture is injected before combustion. In stock form the pistons had a domed design feature for this area to help optimize combustion efficiency. This design error along with the low compression yields a result far from optimal. To fix these issues a new piston design was initiated. Settling on a compression ratio of 20:5:1, and a dome design similar to stock, the motor would have the ability to make more power than in stock form while at the same time not losing as much efficiency. Due to matters out of our hands the pistons were never made and the team was forced to consider other options to combat the low compression and poor piston design.

Our team decided that the next best option was to replace the dual layered metal head gasket with a thinner gasket. Not being able to alter the design of the pistons, this was the only possibility in changing the compression ratio of the engine. After researching the different models of engines available, we decided to purchase the head gasket used in a 2010 Kawasaki Mule. This head gasket proved to be much thinner than the original, with a width of .020 inch compared to the original .040 inch gasket. This resulted in an increase in compression ratio from 17.4:1 to 19.4:1, much closer to what our team desired than before.

INTAKE AND INTERCOOLER SYSTEM

Efficiency of a turbo charging system is highly dependent on the design of the system after the air is pressurized and before it enters the engine as well as the design of the intake pulling air from the atmosphere both of which change the temperature, pressure, total flow and profile of that air. Each of these factors were considered when optimizing the current system used in the snowmobile from past years.

Within every turbo system there is a pressure drop due to the volume the turbo charger needs to fill with air between the compressor and the intake side of the cylinder head. This past year much consideration was put into reducing that total volume to increase the overall efficiency of the system. Previous years consisted of a long assembly of oversized aluminum piping with a standard air to air intercooler. This year, to intake sound, overall volume and cooling efficiency a water to air intercooler coupled with mainly silicone piping was used. The silicone piping absorbs the intake noise and contains it, whereas the aluminum piping amplified it. Total volume between the turbocharger and the motor was reduced from 285 cubic inches to 101 cubic inches, almost 65%. This decreases the "lag" felt by the turbocharger, giving much better throttle response. The new water to air intercooler utilizes the rear heat exchanger originally used for engine cooling to cool the charge air much more efficiently than the previous setup. The smaller core of the water to air intercooler also suffers from much less of a pressure drop across the intercooler than the previous air to air intercooler. This results in a higher manifold pressure, thus more air in the engine for combustion.

The stock intake manifold was originally designed to be used with a smaller turbo charger fitted by the Daihatsu. Naturally with an upgraded turbo a new less restrictive intake was needed. The solution was a aluminum tubular manifold. The plenum was large to allow for max airflow as well as the runners. Problems arose when the runner profile did not match the intake port on the head. The intake port was a rectangle and the piping was round. This caused turbulence in the flow and decreased performance. To solve this, the team looked back at the stock intake. The intake ports on the head had been slightly enlarged to increase flow but not more than what the intake manifold was capable of matching. The real problem areas were the geometry of the inlet, the size of the inlet and the plenum size. The solution was to use the runners and bottom section of the stock intake manifold, coupled with a large plenum on top made of aluminum tubing also allowing for the inlet to be situated in an appropriate location. Flow diagrams comparing the two manifold designs can be seen below.

For each intake manifold the pressure at the 3 outlets was defined as 15 PSI, the pressure at which the turbo will maximize at and also stay at for approximately 50% of the RPM range. Not included in the model is the change in properties of air from the new manifold to the next. The old manifold utilized the old air to air intercooling system with high pressure drops and inefficient cooling. The new water to air system cools the air much more efficiently, overall resulting in high pressure and cooler more dense air.

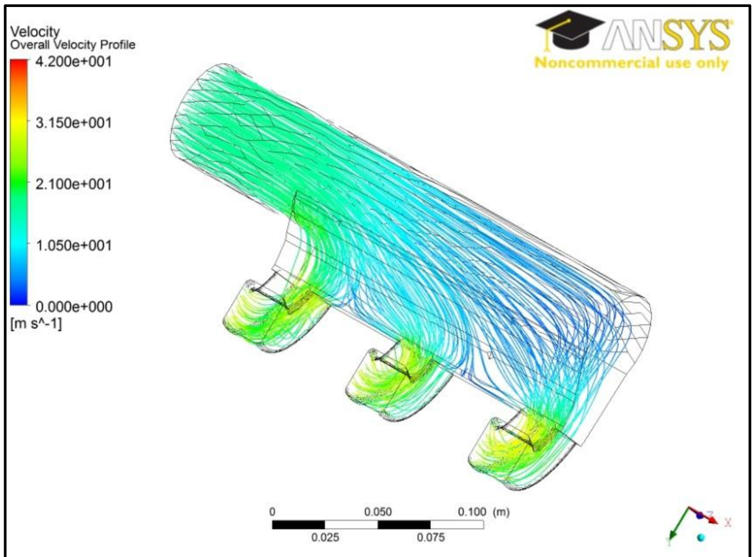

Figure 3 – 2012 Velocity Distribution – Intake Manifold

On the left is an image showing the velocity profile of the newly designed intake manifold for this year. The model was constructed using ANSYS CFX, a fluid flow design program. This manifold is what is currently on the sled and utilizes the stock runners from the OEM intake manifold but has an overall larger inlet and increase in the size of the plenum. Using the stock runners allows for a smooth transition into the cylinder head while still keeping a free flowing design. The larger plenum was designed to help constantly feed air to the motor. Although larger than stock the volume of the plenum is still 25% smaller than the previous design shown below. Main features of this model that should be noted are the smooth velocity lines throughout the length of the plenum all the way through to the runners and the gradual transition from lower velocity to higher velocity. Increased velocity does not necessarily inhibit performance. The profile of the transition does. The old manifold to the left shows how a rapid increase in velocity can happen.

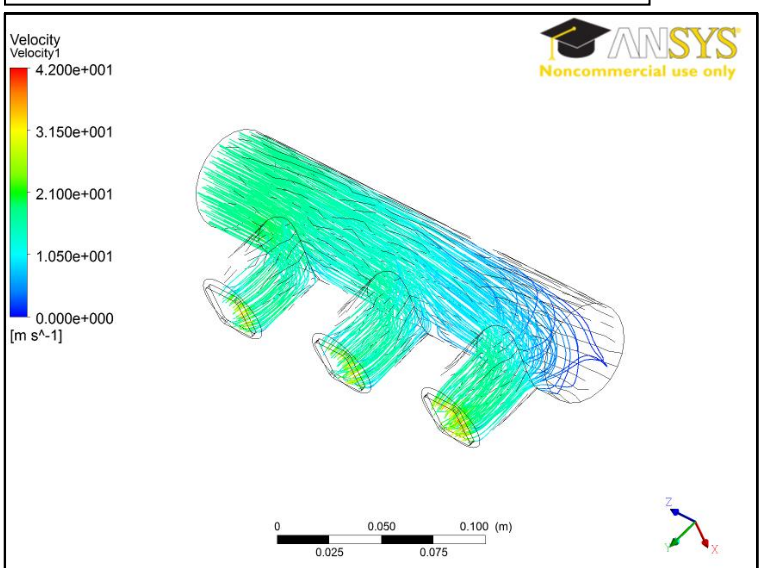

Figure 4 – 2011 Velocity Distribution – Intake Manifold

To the left is a model created using the same software above to show the flow through the old intake manifold. This was a tubular design using 2.25 inch pipe for the plenum and individual 1.5 inch pieces of pipe for the runners. Although better performing than stock this manifolds crude design warranted its need for change. The large empty space seen on the right side of the model shows how a symmetrical design is not idea, as well as the circular shaped runners mating to rectangular intake ports on the head. The latter is the cause for turbulence right before the intake valves. Although turbulence helps with the mixture of the fuel, the fuel has nothing to mix with if the turbulence is high enough to limit the air flow overall. Another point of concern is the uneven distribution of air to all three cylinders. The runner furthest away is getting the least amount of air while the one on the end has the most. Optimal conditions would be for all three to get the same. This trend can also be seen in the new intake manifold design as well but in a less drastic fashion.

On the next page are two images showing the pressure distribution across the plenum and the runners. The top picture is the newly designed manifold for 2012 and the image below is the old manifold used in 2011. The pressure distribution inside the plenum is similar but the runners are drastically different. The runners in the top pictured show a trend of decreasing pressure as the height decreases. This is simply because the air flowing on the outside of the runners has to travel over a larger circumference and therefore has an increased speed when compared to the air flowing on the inside, smaller diameter corner. Each cylinder appears to have relatively even pressure and a extremely similar pressure distribution. The image below shows the old intake. Each runner has very few similarities and the pressure distribution is random. This makes for changing conditions inside the motor, increasing the difficulty for proper tuning and repeatability.

TURBOCHARGER

The addition of a Garrett GT15 variable nozzle turbocharger and intercooler is a major design change for this year. By matching our engine flow rate and desired intake pressure ratio to the compressor maps of various turbochargers, the GT15 VNT was chosen as the best match. The turbocharger, used on the Volkswagen 1.9L TDI engine, is Garrett's fastest spinning turbocharger which allows for flexibility when running it at different RPM ranges and with different displacements making it desirable for use on our diesel engine. The variable nozzle design allows for a more dynamic operating range which can spool quickly when the vanes are closed and flow a greater amount of air at fully open. The ability to vary the amount of air flowing through the turbine gives greater efficiency at both low and high compressor flow rates.

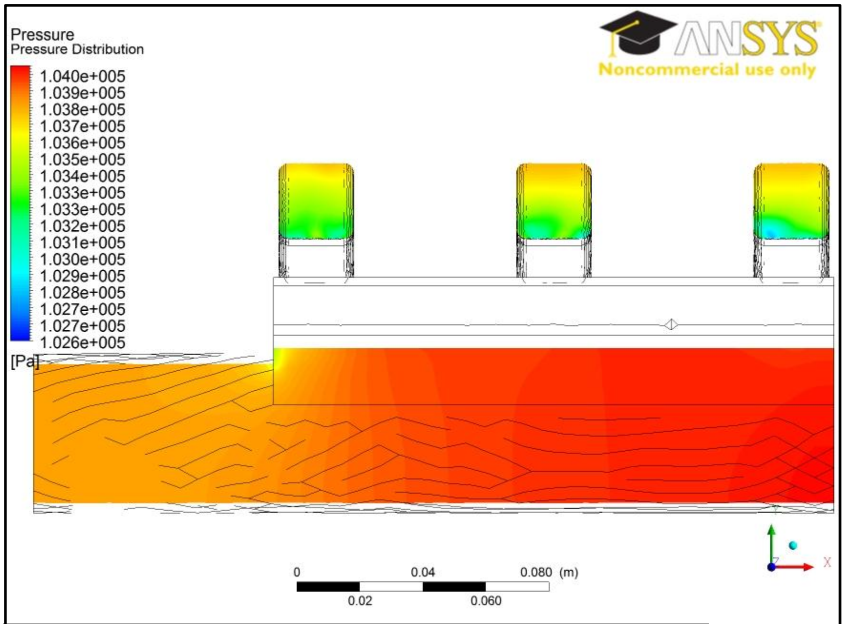

Figure 5 – 2012 Pressure Distribution – Intake Manifold

EXHAUST SYSTEM

Exhaust flow is essential to turbine efficiency. How the exhaust pulses reach the turbine affects both the efficiency and the spool properties of the turbine. Past years' sleds have seen equal-length style manifolds, tubular log style manifolds, as well as cast style manifolds. Past years' manifolds have run into issues with turbulence created by the transition from rectangular exhaust ports to tubular runners, creating turbulence, which greatly affected turbine efficiency. Another major downfall of the tubular manifolds of past was very poor heat retention. Expansion cooling was also an issue when using runners of larger cross sectional area rather than the exhaust ports on the cylinder head. In order for high turbine efficiency as well as optimal catalyst performance, as much heat as possible must be retained.

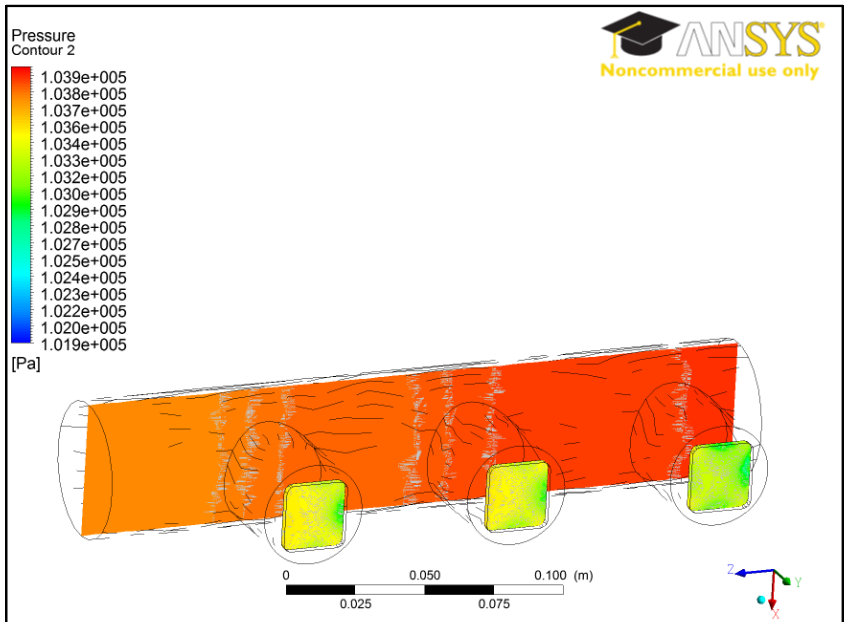

Figure 5 – 2011 Pressure Distribution – Intake Manifold

The turbocharging technique chosen for our snowmobile was the constant pressure turbo charging theory. This particular method was optimal for our sled's application because of ease of manufacture, minimal loss of heat, and higher efficiency for constant rpm applications. The stock manifold was chosen for our constant pressure setup due to the cost to performance ratio. In order to optimize flow through the stock manifold, the outlet and all of the inlets were ported and polished. The outlet was bored out in order to match the turbine inlet size. The rectangular inlets match the exhaust ports on the cylinder head, which was also key advantage of previous years' subsystems.

Constant pressure systems gain their efficiency due, in part, to their low heat transfer. If a cast manifold is used, it is typically made from cast iron, which has a low thermal conductivity of approximately 27-46 btu/hr*F [1]. The use of a low thermal conductive material will ensure that more energy is available to the turbine, according to the energy balance at the turbine. The cast manifold has significantly less surface area than a pulse turbocharged manifold, which uses individual exhaust runners to run exhaust gasses to the turbocharger. These properties help the cast manifold retain heat, optimizing turbine efficiency.

For our application of a diesel engine and a constantly varied transmission, the relative change in rpm's is rather insignificant. This is ideal for the constant pressure application, as a change in pressure is damped inside the manifold and will increase the time that the turbocharger increases speed. Constant pressure systems are ideal for use in constant speed applications with respectable efficiency.

The post turbo section of the exhaust contains a two-part catalyst to reduce emissions. Post turbo, pre catalyst piping matches the turbine outlet to maximize flow, while retaining as much heat as possible for the catalyst to be efficient. All post turbine sections are wrapped with fiberglass heat wrap to retain heat, and in turn to optimize catalyst performance. The post cat section utilizes a slightly larger pipe diameter to reduce backpressure.

COOLING SYSTEM

During last year's competition, one of the biggest issues the University at Buffalo Clean Snowmobile Team faced was having their motor overheat during dynamometer testing and field competition. This year we took a good look into why the motor overheated in the first place and have made significant changes to the motors cooling system to prevent this.

The first thing that should come to mind is the fact that we are using a DM950DT diesel engine in our snowmobile, which runs at higher air-to-fuel ratios than gasoline engines, creating hotter combustion and exhaust gas temperature. Although these are good for the collection of hazardous exhaust gases by the catalytic converter they also increase our engine oil temperature and temperatures under the hood. We have also modified out diesel engine for performance and doubled the amount of power output, increasing running temperature and output of heat significantly. In addition, the chassis of our Polaris Turbo IQ sled, containing two inline heat exchangers, was designed for use with a gasoline engine and not the turbo-diesel engine we are implementing into it.

After reviewing these factors, we decided to bypass the current dual heat exchanger cooling system on our Polaris chassis and replace it with a radiator out of a Polaris Sportsman 800 EFI. The use of heat exchangers on the Turbo IQ chassis is heavily dependent on the existence of snow to cool the fluid passing through its system. Unfortunately snow is not always abundantly existent, last year's Clean Snowmobile competition being an example where even the ambient air was lacking in coldness. In these cases running a hotter motor than the factory cooling system is designed for can easily become a large problem. With the substitution of a radiator, we are no longer dependent on snow (or cold temperatures) to facilitate the cooling of our engine, and now only require simple airflow. The very fine fins which make up a radiator can transfer heat with ease to the outside air. Our chosen radiator was designed for a 760cc gasoline engine that gets use in much warmer climates than we expect to see.

To produce the best cooling for our engine we decided to mount the radiator close in the front of the sled. This will provide the most airflow to assist in cooling our engine during all external temperatures and conditions. Ducts within the hood will be made to facilitate cooling and allow hot to escape. Due to these changes, we believe that our snowmobile will not only have a better cooling system for our specific motor, but a better cooling system overall.

Radiator kits are becoming increasingly popular in the sport of snowmobiling, especially in events such as grass drags and water skipping. Aftermarket companies are beginning to recognize the demand for radiator kits as additional cooling or complete replacements to stock cooling systems.

EMISSIONS CONTROL

With the increase in power and fueling, the stock emissions strategy no longer works with our engine. To keep the emissions level down, we had to utilize a two part catalyst system including a Diesel Oxidation Catalyst, a PM-Metalit diesel particulate filter from EMITEC, and a variable boost program to control diesel soot, carbon monoxide, nitrogen dioxide, and un-burnt hydrocarbons. The function of the diesel oxidation catalyst is to convert un-burnt hydrocarbons, carbon monoxide, and excess oxygen into H_2O, carbon dioxide, and nitrogen dioxide. While nitrogen dioxide is a measured pollutant, it is used in the filter to burn the diesel soot. The PM-Metalit Diesel Particulate Filter is unique in many ways. Firstly, it is a partial flow technology meaning that it will never clog with ashes like wall filters. In addition, this system has the advantage of not needing to use excess fuel for filter burn-offs due to the continual burn-offs. The particulate filter also works with upstream NO_2 from the oxidation catalyst to reduce separated particles. To optimize the conversion ratios of NO_2 and the burn-off of the soot, an electronic linear actuator was mounted on the turbocharger to vary the boost pressure. This actuator was programmed using varying loads from a water brake dynamometer. The O_2 and NO_x were read using a NTK NO_x Sensor. With a constant load applied, the boost pressure was varied to have the least amount of NO_x coming out of the treated exhaust.

DIGITAL DISPLAY AND ELECTRICAL SYSTEM

The electrical system of the snowmobile consists of a starting battery, starter, alternator, and chassis wiring. The battery consists of a 1164 cranking ampere Absorbed Glass Mat battery supplied by Braille Battery. The battery is located in the seat box of the snowmobile. To conform with rule 4.8.52 the battery is located in a non-conducting Lexan box located under the seat. To facilitate the work on the electrical system in the front of the snowmobile and to protect the power wire running under the seat, a battery disconnect switch is locating in the back of the snowmobile too. For the shutdown of the engine and the fuel pump, all of the chassis power is run through a relay driven by the kill switches.

The snowmobile is equipped with an LCD display and small computer for sensory feedback to the rider about the state of the snowmobile. It uses an ARM 9 powered TS-7400 single board processor (SBC) to gather information about all sensors and relay desired information to the display and controllers. The controller has a 200 MHz processor and can read up to 20 digital inputs and outputs, while maintaining operation at temperatures as low as -40 ^{0}F. The controller is also able to record data on an SD card for testing or diagnostic purposes. This will allow a dealership or repair center to use it to extract information about what the engine was doing before a crash or to diagnose what is wrong with the snowmobile.

The display used replaces the standard gauges and can be implemented by a manufacturer onto any snowmobile. It can be changed by the manufacturers to show different sensor readings or warnings based on the model that it is being implemented in. With a size of 4.3” and a high contrast it will be easily readable to the riders during any operation time. The brightness of 5000 nit is at least 3 times higher than displays used on current smartphones and other devices used outside. A quick glance at the display will be enough for a rider to see the speed they are currently travelling at or gather any information they are interested in.

BELT DRIVE SYSTEM

The characteristics of the diesel engine used in the SUNY at Buffalo’s snowmobile has plagued the team for years with drivability problems. This is mainly due to a gear reduction that was not suitable to properly use the diesel motor’s strengths. Basically a more drastic gear reduction was needed in order to produce a more optimal system.

Each concept only considers the modification of either the chain case or the CVT or both. This is mainly because of the simple two part system that the drive train is composed of. To add another component to this system would further decrease the machine’s overall efficiency. This would not be acceptable considering that the current overall efficiency of internal combustion snowmobiles is very low.

With a scope so concentrated, it was difficult to have designs that are vastly different. Because of this in total there are 6 designs that were considered and evaluated further.

1. Belt Drive with Cogged Belt and stock CVT – For this concept the belt used would either be a standard cog belt design, or a synchronous high torque drive, HTD belt design. A cogged belt is the best mix of characteristics of each a V-belt and a chain driven system. The HTD belts are typically thinner than regular belts but are also stronger and have a broad operating range. This would utilize the stock CVT clutch setup.

2. Belt Drive with Cogged Belt and modified CVT - This is the same concept as stated above except the CVT setup would be drastically modified to change its initial and final drive ratios.

3. Belt Drive with V-Belt with stock CVT - Incorporated in this design would be either a classic V belt design, ribbed V-belt with a truncated V profile or full V profile. The advantage of the ribbed V belt design is that even though the belt is thinner and more flexible, its flat shape allows it to bear a greater load than a regular non ribbed belt design. Advantages and disadvantages of this system mainly fall in the same category. The ability of the two pulleys to operate at different speeds by letting the belt slip and stretch. This is an advantage because this will help to prevent damage to the components when stresses get too high. In the same way there are circumstances when under full acceleration and heavy load the belt might slip when not needed. This will decrease the efficiency of the whole system.
4. Belt Drive with V-Belt with modified CVT. See above options.

For both of the belt drive systems a brake rotor can be directly attached to jackshaft or driveshaft, allowing for a compact design eliminating other components under the hood completely, therefore also reducing overall weight. Also ease of adjustability for both designs is very high. That is, with tension taken off the belt and the belt removed, a new pulley with a different diameter can be easily swapped on. This will help with tuning of the entire machine, because as the characteristics of the engine change, and the dynamics of the clutches change, changes to the final drive will have to be made to fine tune the top speed, acceleration and finally and most importantly fuel efficiency.

5. Spur gear case with stock CVT – Standard to almost every snowmobile produced today this spur gear chain combo is efficient and very durable. The parts are plentiful and can be bought from many different suppliers. The disadvantages of

such a system are mostly weight, noise and ease of service/adjustability. Using a chain case will require a "case" filled with gear oil, heavy gears and chain and finally very noisy considering how quiet the engine is. When making adjustments, which will constantly be made, the gear case has to be drained and nearly everything disassembled. Changing the final drive with require a different gear, but for optimal conditions for our motor a gear is not currently available, so either on will have to be custom made or retrofitted to work in our snowmobile. This option will utilize the stock CVT and rely solely on the modification to the gearing to provide a proper overall gear ratio.

6. Spur gear case with modified CVT – Using the changed gear ratios of each the CVT and gear case in order to properly match the engine.

	Metric	**Value**	**1**	**2**	**3**	**4**	**5**	**6**
1	Weight	Decreased combine weight of original assembly by >30%	3	3	3	3	1	1
2	Efficiency	>95% overall	3	3	1	1	2	2
3	Sound Level	Decrease noise level by >20% (need baseline from last year)	3	3	3	3	1	1
4	Serviceability	Complete belt/chain and gear/pulley change in <1 hour	3	3	3	3	2	2
5	Adjustability	Multiple gear/pulley options for tuning	3	3	3	3	2	2
6	Braking	At par or better than original system	3	3	3	3	2	2
7	Top speed and acceleration	Top Speed >70mph 0-60 mph in <6 sec	2	3	2	3	2	3
8	Cost	Less costly than original setup	3	1	3	1	2	0
9	Service Life	3 seasons	2	1	2	1	3	2
		Total	**25**	**23**	**23**	**21**	**17**	**15**

With all the factors involved it resulted in the final design incorporating a belt drive system utilizing a cogged HTD belt design with a stock CVT setup. This will be incorporating the brake rotor and caliper into the system as well. The reasons behind this decision were rating how well each system fulfilled the technical specifications. Each technical specification was evaluated according to each system. This was done using a rating system whose values are shown below.

- 3 – Above and beyond the requirements
- 2 – Satisfactorily fills the requirement
- 1 – Partially fills the requirement
- 0- Does not fill the requirement

After a quick inspection of the chart it is clear that option #1 was selected.

Power Evaluation

1. Basic Relationship:

$$P_{track} = P_{flywheel} - P_{drivetrain\ losses}$$

Flywheel Power

The power at the flywheel is provided by dyno testing and the assumption that as a result of this year's improvements the power of the snowmobile will increase by 15%. With an original power of 51.32 HP and 76 ft-lbs of torque, the new expected values will be 59.02 HP and 87.40 ft-lbs.

CVT Efficiency

For our system, instead of using variable efficiencies we took a range of 81% to 90%. In our analysis utilizing worst and best case scenario for each respectively.

CVT to Belt Drive Efficiency

The connection from the secondary clutch to the drive sprocket of the belt drive is a direct connection riding on a ball bearing. Because of this there is very minimal power loss, < 1%.

Belt Drive Efficiency

This value is easily calculated by the equation:

$$Efficiency\ \% = \frac{(Driven\ gear\ RPM) * (Driven\ gear\ torque)}{(Driven\ gear\ RPM) * (Drive\ gear\ torque)} * 100 \qquad (1)$$

For the Gates Polychain system the efficiency ranges from 97% to 98%. This is mainly due to the no slip characteristics of the cogged belt.

Drive Cog Efficiency

Of all the efficiencies this is the hardest to evaluate. The drive cogs are mounted on a drive shaft, and as they rotate the cogs interlock with a ridge pattern on the inside of the track. The track is made of a dense rubber material. If conditions were optimal each surface would be dry, the two components would interlock completely and there would be no slippage. In the field there is snow, both surfaces are most likely wet/icy and there is slippage mainly because of the incomplete interlocking of drive cogs and track. Furthermore the actual surfaces are not precisely manufactured and loosely fit together. All of these things cause the efficiency to decrease. Taking into consideration all of these the worst and best case efficiencies are 75% and 85% respectively.

Combining all of the above relationships the following equation can be produced:

$$P_{track} = P_{fw} - \left[P_{fw}\left(1 - \left(eff_{CVT} * eff_{CVT\ to\ BD} * eff_{belt\ drive} * eff_{drive\ cogs}\right)\right)\right] \qquad (2)$$

2. Power to maintain various constant speeds:

$$P_{min} = P_{aero\ drag} + P_{track\ losses} + P_{trail\ resistance\ losses} \qquad (3)$$

Aerodynamic Drag

When any vehicle is in motion it fights to travel through the air. The basic relationship to find the power to overcome aerodynamic drag is:

$$P_{aero} = \left(\frac{1}{2}\rho V^2 C_d A * V_{snowmobile}\right) \qquad (4)$$

The only variables left are C_d and A. C_d is the coefficient of drag. For a snowmobile the coefficient of drag varies, but is typically around .3. Therefore, a worst case of .45 and best case of .27 were used in the analysis. A, is the frontal area. Similar to the C_d this also varies. To estimate our frontal area, no specifications are given by the manufacturer, a simple diagram was used.

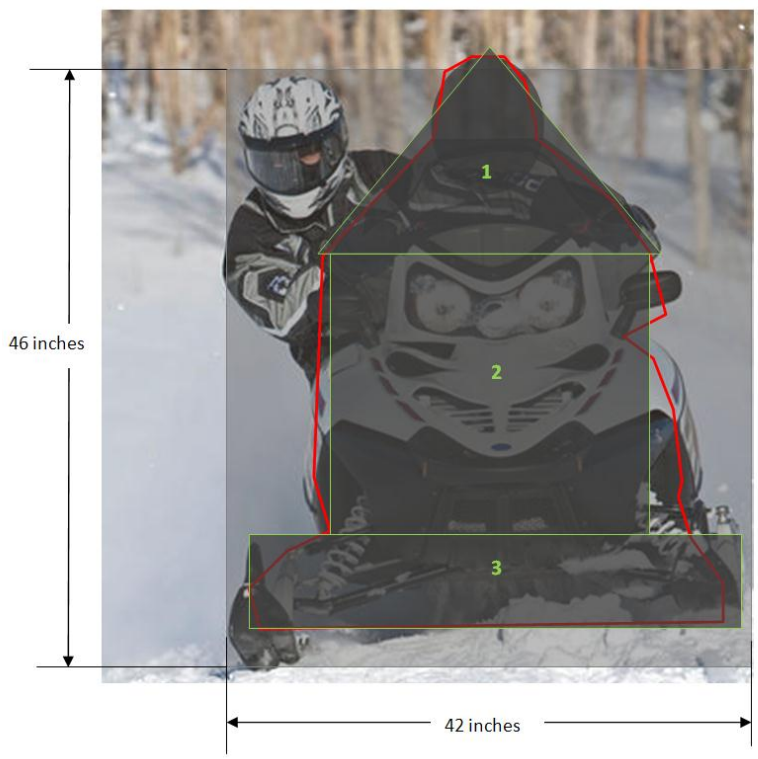

Figure 6 – Cross sectional area of a snowmobile

The red out line is the approximate shape of the rider and snowmobile combined when in an upright position. Another approximation is made to convert the area taken up by the sled and rider to conventional polygons. From there referencing the total frame size dimensions of each of the shapes can be found.

1. $Dimensions \rightarrow base = 26.0\ in,\ height = 15.9\ in$

   $\boldsymbol{Area} = 206.7\ in^2 = 1.44\ ft^2$
2. $Dimensions \rightarrow\ 24.0\ in\ x\ 20.44\ in$

   $\boldsymbol{Area} = 490.56\ in^2 = 3.41\ ft^2$
3. $Dimensions \rightarrow\ 38.0\ in\ x\ 7.67\ in$

   $\boldsymbol{Area} = 291.33\ in^2 = 2.02\ ft^2$

$$\boldsymbol{Total\ frontal\ area} = 1.44\ ft^2 + 3.41\ ft^2 + 2.02\ ft^2 = 6.87\ ft^2 \qquad (5)$$

Track Losses

When the track is in motion it is constantly sliding over and under the rear suspension. This causes a frictional force to be produced reducing the total power output the snowmobile can put to the ground. With the two surfaces being steel and polyethylene, and the dynamic coefficient of friction between them being .2 the force of friction can be calculated, once the load that it will see is known.

With no rider the weight of our sled is 650 lbs. The distribution is solved for using 3 different scales and is as follows:

$$Track\ \% = 42.3\ \% \rightarrow\ 275\ lbs$$

$$Each\ front\ ski\ \% =\ 28.8\ \% \rightarrow\ 187.5\ lbs$$

Adding in the weight of a rider that is 180 lbs:

$$80\%\ of\ weight\ on\ track \rightarrow\ 144\ lbs$$

$$10\%\ on\ each\ front\ ski \rightarrow\ \ 18\ lbs$$

This yields a total weight on the track of 419 lbs. From this the force equation is:

$$Force_{fric} = f * F_{normal} \qquad (6)$$

$$F_{fric.} = (.2) * (419\ lbs) = 83.8\ lbs$$

The power equation that relates this equation is:

$$P_{track\ losses} = V_{track} * F_{fric} =\ V_{track} * 83.8\ lbs \qquad (7)$$

Trail Resistance

The trail resistance is due to the friction between the front skis and the snow. The track is assumed to have no trail resistance because of two reasons. First we are traveling on groomed trails therefore they are relatively smooth, and secondly because the track is traveling at the same speed as the snowmobile. Using similar equations from above:

$$Total\ Load\ on\ front\ skis = 2 * (187.5\ lbs + 18\ lbs) = 411\ lbs \qquad (8)$$

The coefficient of friction between the skis and snow is .1 for static and .05 for kinetic. Only the case for kinetic will be evaluated, static is irrelevant.

$$F_{fric.} = (.05) * (411\ lbs) = 20.55\ lbs \qquad (9)$$

$$P_{trail\ resistance} = V_{ski} * F_{fric} =\ V_{ski} * 20.55\ lbs \qquad (10)$$

Once again with the above relationships:

$$P_{min} = \left(\frac{1}{2}\rho V^2 C_d A * V_{snowmobile}\right) +\ \left(V_{track} * F_{fric}\right) + \left(V_{ski} * F_{fric}\right) \qquad (11)$$

Knowing that at any given time :

$$V_{snowmobile} = V_{track} = V_{ski} \qquad (12)$$

The equation can be simplified to:

$$P_{min} = V * \left[\left(\frac{1}{2}\rho V^2 C_d A\right) + \left(F_{fric}\right) + \left(F_{fric}\right)\right] \qquad (13)$$

3. In order to maintain speed: $P_{track} \geq P_{min}$ (14)

All of these equations were all used together to get a baseline top speed. This was done using an excel file that took into account changing speed, best and worst case values and different power levels at changing locations in the power band. At top speed many of the variables cancel out which makes for the initial analysis to be simpler. Once a top speed is established we can then tune the clutches to maximize the acceleration.

With our excel sheet we calculated that at 80 mph there would be a total power loss for everything of 57.97 HP using all of the worst case scenarios and for best case the total power loss was 44.14 HP. This speed is to be used as a maximum and will not be the final top speed, it will be used more as a limiting factor. Now that we had a goal top speed we then moved onto the gearing system to analyze and choose our best option for sprocket sizing.

Clutch Ratios

For our clutch setup, Team Dynamics Tied, the initial and final drive ratios are 5.19 and .83 respectively.

Gear Ratio Selection

With a top speed designated based on our conceptual power evaluation, the actual gear ratio to be applied was determined. The approach for choosing an appropriate gear ratio was based on requirements placed on the machine during certain competition challenges, a give-and-take relationship between top speed and acceleration, fuel economy, and the important requirement of crisp throttle response and overall ride ability.

As it is the longest and most important dynamic portion of the competition, the endurance event was chosen as the most significant factor in regards to gear ratio selection. During endurance, the sled travels 100 miles in a 2 1/2 hour time frame; experiencing diverse terrain, changing trial conditions, and highly variable engine demands. Competition snowmobiles are to follow a event judge riding a lead machine, which will purposely travel at varying speeds averaging 45 mph. With this in mind, it was recognized that a vehicle speed of 45 mph would be an important design parameter. It should also be noted that designing with the endurance event in mind will be very relevant to real-world riding considerations.

Another important consideration is the fact that the engine will be making maximum torque at approximately 3000 rpm. After engine torque reaches this maximum, its value drops by only 17% as the engine reaches peak horsepower output at 4400 rpm. Internal combustion engines operate at maximum efficiency while full throttle is being applied at the RPM point of maximum torque. With this in mind, the CVT is configured to maintain the engine at or near 3000 rpm throughout shifting and to experience max shift at 45 mph. This will ensure that maximum fuel efficiency will be maintained throughout the majority of acceleration times up to 45 mph. As the vehicle surpasses 45 mph, the CVT will stop shifting, and engine rpm will begin to climb. Once the CVT has completed its shift, acceleration will be based solely upon available engine output and characteristics. With this configuration, the engine can apply ample torque to continue accelerating the vehicle as well as rpms. Also, horsepower is 82% of its maximum value at this point, and can quickly increase to its full potential with the available torque.

An excel spreadsheet was developed to allow for sprocket sizing and engine rpms to be quickly changed to produce the associated angular velocities and vehicle speed. Conversely, an additional module allows for engine rpms and vehicle speed to be varied to produce a desired gear ratio. Efficiencies were not taken into account for this evaluation, as it was assumed that angular velocities at the flywheel could be directly related to those at the track due to the application of a direct drive system; belt slippage at the CVT was considered negligible. Further explanation of the content of this spreadsheet and the equations used can be found in Appendix A. In conjunction with our design approach gear ratios are evaluated at 45 mph and also at maximum vehicle velocity, both of which correspond to maximum shift of the CVT. Choosing the best gear ratio came down to balancing the dynamics of the engine to maintain adequate top speed while not sacrificing acceleration. Favorable acceleration behavior was characterized by reaching the intermediate speed goal of 45 mph by a relatively low engine rpm point.

As the desired sprockets are relatively inexpensive and easy to install, it was decided that two different drive sprockets and one driven sprocket would be ordered. This allows for more freedom in applying the appropriate ratio to our system and real-world testing. Through testing it was determined that the lower gear ratio provided sufficient acceleration and returned high fuel mileage and therefore would be utilized.

Gear Ratio	Target RPM (@45 MPH)	Maximum Velocity (@4800 RPM)
1.6	2875	75.4 MPH
1.714	3065	70.4 MPH

Figure 7 – Gear Ratio Selection

Detailed System Design

With our major evaluation finished, and all components selected, the process moves forward onto the selection of sprockets, bearings etc. along with the configuration of the system. Finally after the system is configured a final costing analysis is done.

Sprocket and Belt Selection

The first step in calculating the belt sprocket combo is to choose a belt pitch. To do this the design HP was found to be 82.63. From here using the max rotational speed of 6032 rpms, found in the gearing excel worksheet (Appendix G) and the HP the pitch could be found. This was done using the chart provided by Gates. The belt with a pitch of 8 mm was selected. Appendix [H]

With a belt size selected, center to center distance and desired gear ratios known a belt and sprocket sizes were chosen. Appendix [I]

Belt : 8MGT-800
Drive Sprocket (1.6) : 30 teeth
Drive Sprocket (1.71) : 28 teeth
Driven Sprocket : 48 teeth

With this information a safety check was needed to make sure that the speed of the drive gear did not exceed 6500 fpm. Next the belt width was determined first by checking to see what load the belt could handle with the corresponding sprocket choices. At first the belt width of 21 mm was chosen, but after calculations that was not sufficient. The actual width was found to be 36 mm.

With the information above the final sprocket and belt selection were made.

Belt : 8MGT-800-36
Drive Sprocket (1.6) : PB8MX-30S-36
Drive Sprocket (1.71) : PB8MX-28S-36
Driven Sprocket : 8MX-48S-36

Multi-Point Drive Configuration

For various reasons, it is not advisable to have a fixed drive system were the center to center distance between the pulleys is non-adjustable; such is the case in this configuration as the position of the shafts cannot change. Belts wear over time and need periodic adjustment to maintain safe and effective operation. This system will therefore be a multi-drive setup that incorporates an idler wheel to provide proper tension to the belt. This will also allow for large variances in belt slack when changing between the drive sprockets. The idler wheel will be a flat cylinder, located outboard of the belt on its "slack" side. It should be positioned as close to the center of the belt span as possible, with minimal arc length contact. Although it mounts in a slot and is effectively infinitely adjustable, its position in operation is fixed. It is therefore important that tension be monitored and adjusted regularly. Proper orientation was determined with the help of Solidworks and this data was also used in the force calculations which follow.

The layout of the sprockets, the idler wheel, and the belt where configured using Solidworks. This allowed for the effective belt length (due to a difference in center to center differences) to be compared to the actual belt pitch length; with the expectation that any slack would have to be rectified by the idler wheel. With some quick trial and error, an idler position was determined that satisfied the actual belt pitch length, and the unit vector values were determined using Cartesian coordinates and the Solidworks sketch.

$$s = r * \theta \qquad |\vec{v}| = \sqrt{\hat{\imath}^2 + \hat{\jmath}^2} \qquad \vec{u}\ (unit\ vector) = \frac{\vec{v}}{|\vec{v}|} \qquad (15)$$

Proper tension of the belt and the necessary force to be applied by the idler wheel are easily determined given the applied power, sprocket speed, and other geometries. The following calculations correspond to the smaller drive sprocket diameter of 2.807 inches.

Static Belt Tension

$$T_s = \frac{20 * P}{S} + MS^2 = \mathbf{289\ lb} \quad (16)$$

$$M = Manufacturer's\ Constant$$

$$P = Applied\ Power\ (hp)$$

$$S = \frac{D_{sprocket} \ * RPM}{3820} \qquad (17)$$

Applied Deflection Force to Maintain Tension

$$F_d = \frac{1.2 * T_s + \left(\frac{t}{L}\right) Y}{16} = \mathbf{25.4\ lb} \qquad (18)$$

$$t\ (span\ length) = \sqrt{C^2 - (\frac{D_{large} - D_{small}}{2})^2} \quad (19)$$

$$C = center\ to\ center\ distance$$

$$L = Belt\ Pitch\ Length$$

$$Y = Manufacturer's\ Constant$$

Shaft and Bearing Load Calculations

To determine the load on the shafts and bearings, the forces due to the belt were determined. A simple expression resulted in two tensions, each corresponding the vectors aligned with the direction of the belt in the multi-drive layout. These magnitudes were applied as vectors using the unit vectors calculated above, and their vector sum is the magnitude and direction of the applied force on the shaft.

$$T_{taught} = \frac{144067 * P}{D_{sprocket} * RPM} = \mathbf{509\ lb} \qquad (20)$$

$$T_{slack} = \frac{18008 * P}{D_{sprocket} * RPM} = \mathbf{63.6\ lb} \quad (21)$$

$$\vec{F}_{shaft,total} = T_{taught} * \vec{u}_{taught} + T_{slack} * \vec{u}_{slack} = < -66.9, -568.7 > lb \qquad (22)$$

$$|\vec{F}| = \sqrt{\hat{\imath}^2 + \hat{\jmath}^2} = \mathbf{572.6\ lb} \qquad (23)$$

Using a simple moment equation, the loads on the bearings can be determined. The bearing of most concern is the jackshaft bearing (JB), as we will be utilizing a bearing of our own choice to be press-fit into the belt drive plate. Also, two bearings will need to be selected to be located within the idler wheel. A desired life for these components of 4000 hours is appropriate, providing over 30 yrs of service for individuals who ride 4 hours a day, 100 days a year. The following equations produced a catalog rating of **99.6 lb** for the idler bearings and **5132 lb** for the jackshaft bearing.

$$F_{JB} = \frac{F_{shaft} * (a + b)}{a} = 654\ lb \quad (24)$$

$$C = F * (\frac{L*\omega*60}{10^6})^{3/10} \qquad (25)$$

$$C = Catalog\ Rating, \qquad F = Radial\ Load\ , L = Desired\ Life, \qquad \omega = Desired\ Speed$$

COLD START PERFORMANCE

Cold staring has historically been a perceived disadvantage of diesel engines. However, we have made changed to enable the engine to start easier in cold weather. The first and foremost difference is the change from shielded metal glow plugs to ceramic self-regulating glow plugs. The advantage of these glow plugs is the ability to heat to 1100°C within 3 seconds, then self-regulate to 1000°C after. They also have the advantage of no needed external current control4. Another problem inherent to diesel engines is the possibility of the fuel gelling due to cold temperature. If the fuel were to gel, the injector pump has a fuel return line that returns warm fuel from the pump back to the fuel tank to warm up the fuel. Once the engine is running, the turbocharger will work as an exhaust brake to speed up the engine warm-up. With the ECU sensing cold coolant and no throttle applied, it will signal the linear actuator on the turbo to close the vanes all the way to produce a parasitic load on the engine. This load will in turn cause more heat to be rejected to the coolant jacket and warm the block. Once the computer sees that the coolant is up to temperature, it will signal the LEA to move the turbocharger vanes to a less restrictive position.

MAINTENANCE AND SERVICEABILITY

Maintenance and service on a diesel snowmobile is as easy as that of a conventional gas powered snowmobile. Engine oil needs to be changed approximately every 7500 miles or every year at minimum. Long drain periods are possible due to the robustness of the oil used for compression ignition engines and the synthetic properties needed for cold temperature operation. Engine oil changes are aided by the addition of a remote oil drain hose and a remote oil filter located in the space under the clutches. The fuel and ignition system needs very little maintenance due to the face that fueling and ignition are both controlled by the injection pump. The injectors and injection pump only need to be checked if there is a noticeable change in engine performance. Also, if a problem did arise, the injectors and injection pump can be serviced at any reputable diesel engine shop. One thing that needs to be checked regularly is the fuel filter/ water separator. Due to the very close tolerances in the injection pump and injectors, the fuel must be clear of any debris or water. Changing the fuel filter is facilitated with the filter being located in the open area above the chain case. Bleeding the fuel system is facilitated with a fuel lift pump and a bleed screw on top of the fuel filter. Glow plugs that aid in cold start up only need to be replaced if they have malfunctioned and there is noticeable white/blue exhaust smoke at start up. Obviously, the absence of spark plugs removes the hassle of worn and fuel fouled plugs and reduces maintenance costs and labor. The robustness of the diesel engine increases the time before rebuild or major engine overhaul is necessary, in stark contrast to the conventional gas powered snowmobile. All of the above benefits for ease of maintenance and serviceability highlight the attractiveness of a diesel powered snowmobile for the consumer market.

SUSPENSION AND TRACTION

The rear suspension has been upgraded this year with an M-10 Airwave skid. The suspension is superior in both allowed travel and weight. The suspension allows a more aggressive rider-forward stance and also a higher viewing position to allow proper navigation of snowmobile trails. The front suspension used this year will be a pair of FOX FLOAT Airshox. They eliminate the need for standard shock springs by using an internal floating piston coupled with high-pressure nitrogen gas. There is also a weight savings of about 6 lbs. due to the lack of an external spring. FLOAT Airshox also have a great amount of adjustability via a miniature air pump that changes the internal pressure of the shock. This allows for adjustability from race to trail conditions as well as rider weight in a matter of minutes.

Last year it was noted in the various noise tests that the track and more specifically the studs contributed a large amount to the overall noise the sled produced. To combat this, the team sourced a track with smaller built in studs. In this way performance would not be decreased and at the same time sound would be. Compared to a track without studs, only ¾ of a pound is added. Furthermore the track footprint changed from the previously used Camoplast Hacksaw track to the new track, the Camoplast Ice Attak, one that is known for its tranquility on the trail.

SKIS

C&A Pro Trail XT skis are used on the 2011 sled. These skis are lightweight and offer a great amount of durability and prolonged wear. The patented design on the bottom of the ski allows for reduced darting and improved cornering because of the big footprint it makes when cutting through the snow. More responsive steering is imperative in this design since the diesel engine adds a significant amount of weight to the front of the snowmobile. Additionally a pair of Woody's carbides will be used to give the snowmobile better traction in unfavorable conditions.

CONCLUSION

The University at Buffalo 2011 SAE team has successfully designed and implemented a high performance clean burning turbo-diesel engine for use in a snowmobile chassis. The engine is a highly efficient alternative to a traditional gasoline powered snowmobile. With fuel mileage at least twice that of the competition and the performance that snowmobilers come to expect, our entry for this year exceeds the expectations of the CSC. With currently available technology, the diesel snowmobile can meet the demands of snowmobilers, the EPA, and manufacturers.

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