Emission Reduction of Turbocharged Diesel Utility Snowmobile

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Innovations

Damped Engine Mounting Design and Verification

This year, the team decided to make improvements on the engine mount design used the previous year, which was the first year the Clarkson Clean Snowmobile team participated in the Diesel Class. Due to the characteristics of the 3-cylinder Perkins Diesel engine, the snowmobile experienced a lot of vibration throughout the chassis. The reasoning for this was the lack of damping on the four mounting points. This extreme vibration was not only an annoyance that the rider had to deal with but also made it difficult to read the two gauges on the snowmobile.

In an effort to add damping to the system, the mounting design was completely redone. Originally, the front two mounting points for the engine were two quarter inch thick steel mounts, one end bolted to the engine, and the other mounted to a hollow aluminum bar on the chassis. This was a rigid system and transferred a lot of vibration to the chassis. For the new design, the aluminum bar was replaced with a solid steel bar with two vibrational dampers from AVproducts incorporated. To keep the mounting height the same, the team made cutouts into the solid steel bar to offset the height that the dampers created. The dampers were specifically engineered by AVproducts to dampen the unique vibrations of a small 3-cylinder diesel. This allowed for damped front mounts that transferred minimal vibration to the chassis. When remaking the front mount design, the design team noticed that the previous aluminum bar was slightly bent in one location due to the torqueing motion that the engine makes. To make sure this didn't happen on this year's design, the team decided to go with a solid steel bar that would not bend under the load and torque. Prior to fabricating the mounts, the design was created on SOLIDWORKS and then load simulated. An example of stress and deformation simulation for the bar is shown in Figures 3 and 4 and show that the values are well within the allowable limits.

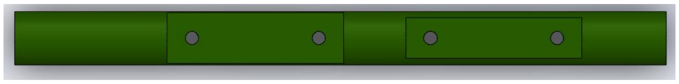

Figure 1. Front mount bar, top view.

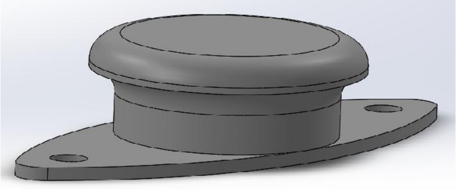

Figure 2. Vibration dampener from AV Products Inc.

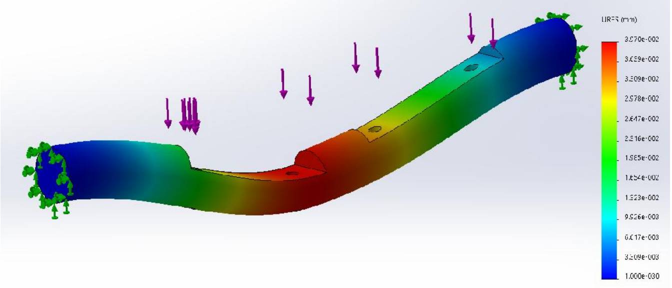

Figure 3. Sample SOLIDWORKS deformation simulation of steel bar.

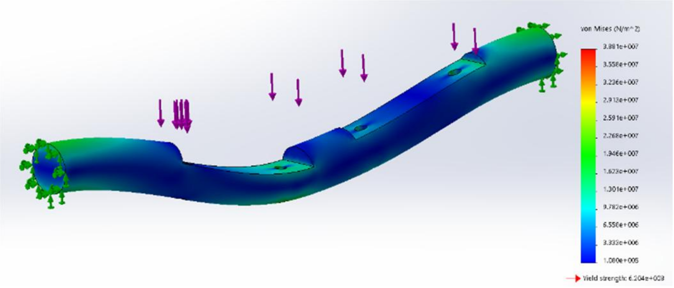

Figure 4. Sample SOLIDWORKS stress simulation.

The rear engine mount design saw even more significant change from the prior competition year, again to improve strength and added damping. In the 2017-2018 year, the rear mounts consisted of a damper bolted to the chassis of the 2018 Polaris Switchback with a bolt that ran through the damper and into the rear mounts of the engine. These mounts had mixed effectiveness. Although the dampers damped some vibration, they were also designed in a shear forced fashion. Through research and analysis, the team realized the boosted effectiveness of having the dampers in compression rather than a shear application. The reasoning behind this is the damper can then hold more weight, is stronger, and still damps vibration to the chassis. The resulting idea that resulted from this was a basket idea, where the mount would be bolted to the chassis through existing mount holes. Then there is a basket that contains a sandwich damper which the engine brackets sit on. Also, there are damping pads in front and on the sides of the engine bracket to damp the side to side motion. Then the engine brackets are bolted to the sandwich damper. This allowed for damping in all directions that the motor moves and cradles it within a basket shape. It also effectively puts the main dampers in compression, rather than in a shear force situation, maximizing their effectiveness. Images below show the two rear mounts along with the subsequent engine bracket. The yellow shown in Figure 3 is the vibration dampener, the brown is the damping pads, and the grey is the aluminum mount itself. The left side is the same except flipped. Figure 4 is a mock-up of how the mount attaches to the motor for the opposite side.

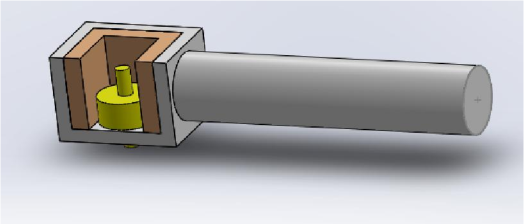

Figure 5. Right rear mount, front view.

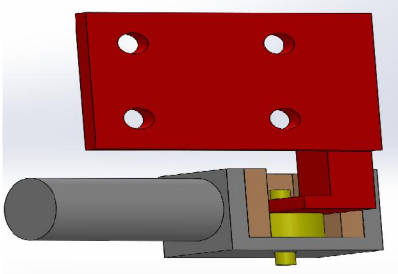

Figure 6. Left rear mount, front view with motor attachment design.

Similarly, to the front two mounts, the designers created the basket mounts in SOLIDWORKS and then load simulated them. The results from the load simulation are tabulated in Table 1.

Table 1. Tabulated stress and deformation simulation results.

	Expected max deflection (mm)	Expected max stress (kPa)
Front bar	0.0397	22.68
Right rear mount	0.1971	18.38
Left rear mount	0.0631	17.32

A final picture of both the front and rear mounts is shown below. The other rear mount not shown is very similar to the other rear mount, just shorter in length.

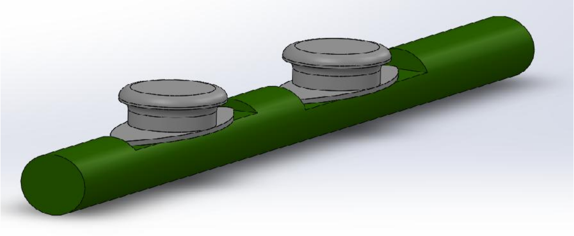

Figure 7. Front mount assembly.

Figure 8. Rear mount assembly.

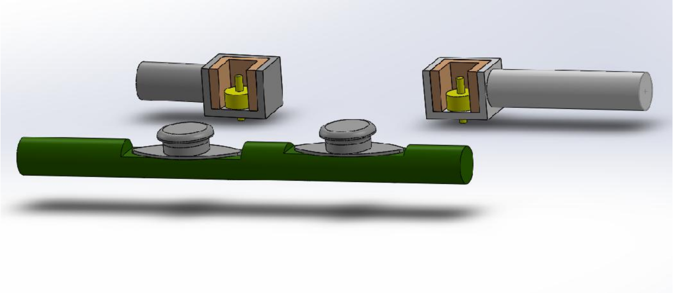

Figure 9. Full engine mount assembly as in chassis.

The above image is a replication of the actual positioning of the mounts within the Polaris Chassis, looking from an angled front view. This engine mounting design is unique to this Perkins engine within this Polaris Axys chassis and does not exist anywhere in industry. That being said, these engine mounts are an effective engineering design that is backed by Engineering analysis, careful thought and preparation, efficient machining techniques, and effectively holds the motor in place while also damping vibration transfer to the other parts of the snowmobile.

Dry Sump Oil System & Oil Lines

The fitment of a diesel engine into a Polaris Switchback SP chassis was difficult, particularly due to the vertical limitations of the chassis' design. The simplest way to shorten the height of the engine was by decreasing the vertical height of the oil pan and integrating a dry sump oil system. The use of a dry sump oil system was the best answer to this problem and allows for the appropriate amount of oil to safely lubricate the engine and turbocharger. A block diagram of the operation of our dry sump system is shown below in Figure 10.

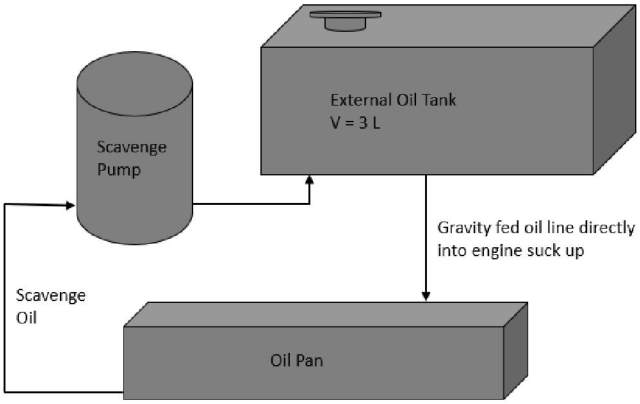

Figure 10. Dry sump oil system block diagram to show the flow of oil from engine oil pan, to the oil pump, then returned to the engine oil tank.

In order to properly and reliably run a dry sump oil system, a new oil pan was designed using SOLIDWORKS and machined on a Haas Super Mini Mill using Mastercam. Reducing the height of the pan from 10 centimeters to 4.5 centimeters gave the team enough space to comfortably mount the engine in the chassis without having to alter the structural integrity of the chassis.

The oil pan was designed so that oil will gravity feed from an external oil tank into a small slot directly attached to the engine suck up. This slot was sealed with an O-ring such that the engine oil pump will directly pull oil from the external oil tank, guaranteeing proper lubrication of all engine components, assuming the external oil tank will have sufficient oil.

To guarantee that the external oil tank will have a sufficient oil supply, the bottom of the tank slopes towards a trough which leads into the oil line where a scavenge pump with a capability for 8 liters per minute pulls the excess oil back into the tank. This pump has the ability to pull oil back into the pan with a factor of safety of 2 to guarantee all excess oil will be pulled into the external tank. The design of this oil pan is shown in Figure 11 below.

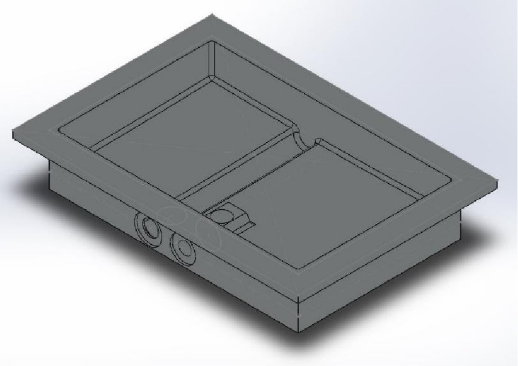

Figure 11. Oil Pan Design.

The original oil pan housed enough space for 3 liters of oil, the recommended oil capacity for the engine. However, the reduction in height did not allow for that capacity, especially considering the additional oil requirement for the turbocharger. Therefore, the main purpose of the external oil tank design is to hold a sufficient volume of oil while satisfying the physical constraints of the chassis. A minimum volume of 3 liters of oil was determined to be the need for the external tank, as well as a tank no more than 14” in width for chassis fitment.

The final component for the design of a reliable dry sump oil system is the size of the oil lines. The minimum required internal diameter (ID) for each oil line was calculated using Equation 1 below [2].

$$D_P = \sqrt{\frac{4Q}{\pi V}} \tag{1}$$

- D_P = Pipe Diameter (mm)
- Q = volumetric flow rate (mm^3/s)
- V = velocity (mm/s)

According to The Engineering Toolbox, flow velocities for suction lines should be under 0.5 m/s, and should be between 1.0 and 2.0 m/s for lines with booster pumps [1]. Therefore, a conservative desired velocity of 0.4 m/s in suction and 1.5 m/s in boost was used in Equation 1. Furthermore, the volumetric flow rates were measured by filling a graduated cylinder and calculating volume divided by time. The results of these calculations are tabulated in the following table, along with the size of the actual lines used.

Table 2. Tabulated results of oil line sizing calculations.

	Calculated ID (mm)	Actual ID used (mm)
Engine to turbo	7.318	4.57
Turbo drain	12.675	15.875
Oil pan suck up	15.524	14.1
External oil tank to oil pan	14.452	14.1
Engine to rocker shaft	5.175	4.57

From the calculated ID’s shown in the middle column above, the actual ID of each line was chosen based on the lines available and slight adjustments in the velocities used.

Team Organization and Time Management

The team leaders for the duration of this project have led the 2019 Clarkson Diesel Sled project from March 2018 – March 2019. These executives were nominated and elected as team leaders immediately following the conclusion of the 2018 Society of Automotive Engineers (SAE) Clean Snowmobile Challenge (CSC) in 2018 by all of the active team members. Their names, positions, and roles are defined below.

Name: Shawn Schneider
Position: Co-President
Role: Running weekly meetings, maintaining schedule, handling sponsorships and sponsor relations, executive decision making

Name: Billy Windsor
Position: Co-President
Role: Project manager, component purchasing and management, executive decision making

Name: Ryan Phillips
Position: Treasurer
Role: Manage budget, track spending, MSRP

Name: Clayton F. Lyon
Position: Secretary
Role: Manage sponsor list, manage communications with team members

Name: Addison Hall
Position: Safety Manager
Role: Ensure shop safety at all times, weekly safety audits

Name: Ryan Alberts
Position: Social Media
Role: Manage Clarkson Diesel Sled social media pages

Name: Brad Cyr
Role: Head of Manufacturing
Role: CNC machining, SOLIDWORKS & Mastercam design

Proper time management is essential when working on a large project. The Clarkson Diesel Sled team schedule planned following the SAE CSC challenge in 2018 is shown in Gantt chart form in Appendix A. However, when problems occur, major schedule delays can be experienced. This can be an issue when it comes to hard deadlines like the CSC competition. This year, the Clarkson Diesel Sled team was actually ahead of schedule until January, when some issues became present that required fixing, causing the test schedule to be set back. A good example of this was when the emissions analyzer turned faulty and had to be sent out for repair. Another problematic delay in the project was when the diesel particulate filter clogged. This happened when it was running on the dyno stand after various adjustments to the engine. In order to unclog the DPF it was sent to FlowMax DPF, which took a weekend to bake and unclog. However, the team needed to get speed data for its clutch tuning. In order to solve this dilemma, a straight pipe exhaust was fabricated so that speed testing could occur temporarily until the DPF was ready and more accurate results could be found, helping the team to remain on schedule. Other issues that caused testing setbacks include leaks in the aluminum oil pan, engine fuel injection tuning issues, and engine timing issues.

Specifications

The following is an inclusive list of the 2019 Clarkson Diesel Sled specifications and main components comprising the construction of a turbo-powered diesel snowmobile:

Table 3. Specifications.

Base Chassis	2018 Polaris Switchback SP
Engine Model	Perkins 403D-07
Displacement	760cc
Stroke	72.0 mm
Bore	67.0 mm
Compression Ratio	23.5 : 1
Cylinders	3 inline
Cycle	4-stroke
Combustion System	Indirect Injection
Turbocharger	IHI AS15
DOC/DPF	Gladstone DOC + DPF
Maximum Boost pressure	40 kPa (gage)
Track	Camso Ripsaw 144" x 15" x 1.25", 2.52" pitch, 2-ply
Studs	114, 1.325" height
Skid	TKI Offset 2 Wheel Axle Kit with 9" Composite Big Wheels, Hiperfax slide rails
Skis	Curve XS, Woody's 6" Carbides
Battery	Shorai Xtreme-Rate 12-Volt LifePo4 LFX Lithium Battery
Primary Clutch	Customized TEAM Industries clutch
Secondary Clutch	Stock
Intake	Use of stock air box

Design Content

Emission Reduction

The main component of the emissions system is an oversized diesel emissions component designed for a 2.5L Volvo C30, V40, and XC90. The system features a diesel oxidation catalyst (DOC) for oxidizing nitrous oxides (NOx), hydrocarbons (HC), and carbon monoxide (CO), as well as a diesel particulate filter (DPF) designed to catch and trap soot particulates.

Diesel Oxidation Catalyst (DOC)

The first component for emissions reduction is the diesel oxidation catalyst. For the diesel emissions application, the DOC is lined with alumina, rare earth oxides, zeolites, and precious metals to aid in the conversion of harmful gases to neutral entities. Alumina provides a large surface area per unit volume for the other oxidizers to occupy, increasing the likelihood of oxidation for the harmful gases. Rare earth oxides, zeolites, and precious metals then line the surface area provided by the alumina and act as the oxidizing agents in the oxidation of harmful gases. However, this oxidation is not uniform with temperature, which is shown in the figure below.

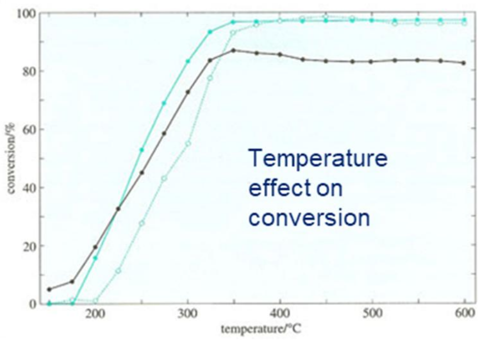

Figure 12. Conversion Percentage vs Temperature for NOx, HC, and CO.

Design in regard to temperature will have a major impact on the efficiency of the system. Ideally, the DOC should be located as close to the engine as possible to prevent heat loss through the exhaust system. For this reason, the DOC was mounted in the front of the engine bay just eight inches from the turbo. However, due to dimensional constraints, the DPF had to be detached and welded adjacent to the DOC, rather than directly inline. Therefore, less heat will be held within the DOC and DPF. In addition, the volume of the DOC relative to the size of the 760cc engine will take a longer period of time to heat up the DOC to a point where it can even begin oxidizing the harmful pollutants. To decrease the time to fully heat up the DOC and allow it to maintain a higher temperature, the entire exhaust system was wrapped with titanium heat wrap to hold the necessary heat in and shield vulnerable components nearby.

In order to validate the design of the exhaust system and the components utilized, a test plan to measure pollutants and convert the readings from parts per million (ppm) to grams per kilowatt hour (g/kWh) was established. First, the engine was allowed to warm up before a slow power sweep was conducted to establish max torque and engine speed values for modal testing. Since the ability for a ramped modal cycle is not possible with the in-house testing equipment available, the pollutants were measured at each mode for the appropriate amount of time. These modes are shown in Table 4 below.

Table 4. Discrete 5 mode duty cycle.

Mode	Speed %	Torque %	Weighting factor	Time (min)
1	100	100	0.12	2
2	85	51	0.27	3
3	75	33	0.25	3
4	65	19	0.31	3
5	idle	0	0.05	3

In order to convert ppm to g/kWh, a MATLAB program was written to find the average value of each pollutant measured from the output of a Snap-On 5 gas analyzer and put the values through a number of equations. First, the mass flow rate of the exhaust is estimated based on engine displacement, density of air, the fact that the engine is a four stroke, engine speed, and approximate air to fuel ratio. The density of air was interpolated from published charts based on intake air temperature and boost pressure. The equation is shown below. Note that omega is used to represent angular speed, but the unit is revolutions per minute.

$$\dot{m} = \frac{\rho V_d \omega}{2 * \left(60 \frac{s}{min}\right)} \left(1 + \frac{1}{AFR}\right) \tag{2}$$

- $\dot{m}$ = mass flow rate (kg/s)
- ρ = air density (kg/m^3)
- V_d = Engine displacement volume (m^3)
- ω = engine speed (rpm)
- AFR = air to fuel ratio (unitless)

Once the mass flow rate was estimated, it could be used in Equation 3 shown below along with the average value of the pollutant ppm and the appropriate value of molar mass from Table 5.

$$EP_i = (EV_i * 10^{-6}) \frac{M_i}{M_{exh}} \frac{\dot{m}_{exh}}{P_{eff}} \tag{3}$$

- EP_i = Pollutant mass, i, referenced to Peff (g/kWh)
- EV_i = Exhaust emission value of components, i, as volume share (ppm)
- M_i = Molar mass of the components, i, (kg/kmol)
- M_{exh} = Molar mass of the exhaust (kg/kmol)
- $\dot{m}_{exh}$ = Exhaust mass flow (kg/h)
- P_{eff} = Power output (kW)

Table 5. Molecular masses used to calculate the pollutant mass in g/kWh. [3]

Component	Molecular Mass [kg/kmol]	Comments
NO_X	46.006	Treated as NO_2
CO	28.0104	
HC	13.876	
SO_2	64.061	
Exhaust Dry	30.21/29.84	5% O_2 / 9.6% O_2
Exhaust Wet	28.84/28.82	5% O_2 / 9.6% O_2

With the g/kWh value for each pollutant calculated, the E-score could then be calculated using Equation 4. After calculating E-score at each mode, the weighting factors were applied and to get one overall estimated E-score.

$$E = \left[1 - \frac{(HC + NOx) - 15}{150}\right] * 100 + \left[1 - \left(\frac{CO}{400}\right)\right] * 100 \tag{4}$$

- E = E-score (unitless)
- HC = Hydrocarbons (g/kWh)
- NO_x = Nitrous Oxides (g/kWh)
- CO = Carbon Monoxide (g/kWh)

Despite having a MATLAB program written, all appropriate testing equipment, and a test plan, this data was unable to be calculated for the design paper deadline due to difficulties related to engine timing and fuel injection tuning. The team found it would be more appropriate to tune the engine properly before publishing emissions results. This data is to be expected in the design presentation.

Diesel Particulate Filter (DPF)

Downstream from the DOC and directly connected is the DPF. The system inside the DPF provides a large surface area for soot to collect, which approaches 99% efficiency once the initial soot begins to collect. This collection will continue to occur until either the system clogs or the active regeneration phase occurs. The exact makeup of the DPF is unknown due to information that the company was willing to give out but is likely made of Silicon monocarbon (SiC) or titanium aluminum alloy (TiAl). These compounds have the ideal thermal gradients to handle the excessive exhaust temperatures that they will be exposed to.

At the current time, the team is pursuing potential options to integrate an active regeneration cycle, although the DPF can be run as a partially regenerating DPF. For partial regeneration, the exhaust heats up to temperatures above 350C so that NO_x will be reduced to N_2O in the catalyst. The N_2O will then react with soot in the DPF, partially oxidizing some soot and releasing non-harmful gases into the air. Although the DPF clogged up earlier in product testing, the engine was re-tuned to run leaner and produce less soot, so there should be a much longer period of time before the DPF would clog again.

In order to test the amount of soot escaping the DPF, a testing system was designed and fabricated. The system samples the exhaust gas, pulling a controlled amount of mass through a 2-micron Pall filter. The flow is induced by a pump and controlled by a critical orifice at 2.2 liters per minute. The critical orifice was selected due to availability in the research labs, although the desired critical orifice was 5 liters per minute to match the pull of the AVL Micro Soot Sensor used at the competition. The testing schematic described above is shown in Figure 13.

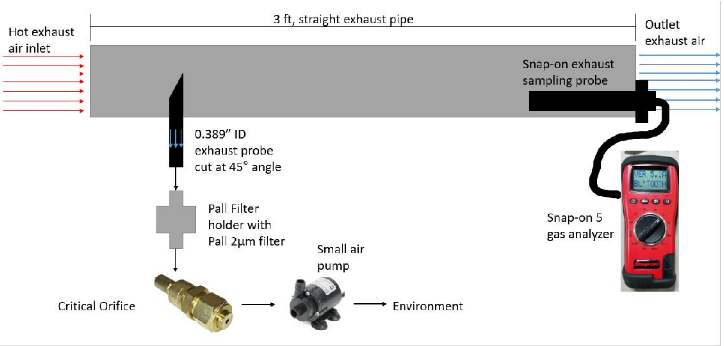

Figure 13. Emissions testing schematic.

In order to calculate the mass flow rate of soot escaping the system, the weight of the filter is measured to the thousandth of a gram before and after the test to get the change in weight. This, along with the amount of time the pump is run, will provide the mass flow rate of soot entering the sample probe. Then, assuming the soot mass flow rate through the probe is proportional to the soot mass flow rate through the entire exhaust, equation 5 shows how the overall soot escaping the DPF can be calculated. Note that the mass flow rate of the exhaust was estimated using Equation 2.

$$\dot{m}_{soot,total} = \dot{m}_{soot,probe} * \frac{\dot{m}_{exhaust}}{\dot{m}_{probe}}$$

(5)

The test is easy to time because no flow will enter the filter when the pump is not operating because no air will escape the pump. The pump will pull enough air such that the critical orifice will achieve sonic speed. Sonic speed through the critical orifice was verified by hooking the pump and critical orifice up to a flow meter and measuring the volumetric flow rate.

Again, the soot data should be expected in the design presentation at the competition because the engine tune at the current time is not finalized for the competition.

Clutching

In order to optimize the CVT for the Perkins diesel engine, a new drive clutch had to be utilized in order to achieve a sheave-to-sheave ratio in which the snowmobile would reach the minimum speed requirement of 35mph. Centrifugal force is the force that the flyweights generate, allowing them to rotate angularly outward from their pin connection. Centrifugal force increases as a function of mass, radius, and velocity, which can be seen in Equation 6.

$$F = MRV^2$$

(7)

- F = Centrifugal force (N)
- M = Mass of flyweight (kg)
- R = Radius from the drive clutch rotating axis to the centroid of the flyweight (m)
- V = Angular velocity (rad/s)

After falling short of the minimum speed requirement during the previous competition, it was determined that more speed could be gained by increasing flyweight or decreasing the spring force in the primary. A plot showing a trend of vehicle speed vs. flyweight can be seen below, justifying our assessment.

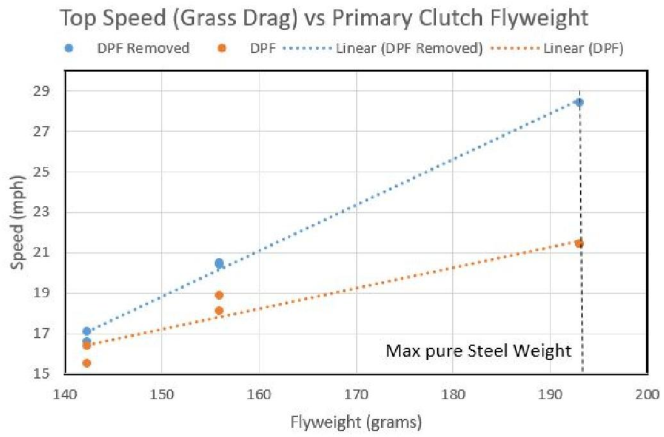

Figure 14. Vehicle top speed utilizing 143g, 156g, and 193g flyweights.

When contacting TEAM Industries Inc. for larger flyweights, it was mentioned that the 193g flyweights that were utilized are in fact the heaviest weights the manufacturer makes. After learning this, it was determined that pursuing a lower spring rate could be beneficial.

Table 7. Drive clutch spring comparison.

Spring (color):	Rate @ 2 9/16 in.	Rate @ 1 5/16 in.
Baseline spring (Gold)	75 lb	275 lb
New spring (Red)	74 lb	228 lb

Noise Reduction

The Clarkson Diesel Snowmobile includes several modifications with the intent of noise reduction. Although sound pressure level combines on a logarithmic scale as shown by Equation 7, every amount of sound pressure that escapes the snowmobile chassis has an additive effect [4].

$$L = 10 * \log_{10}\left(\sum_{1}^{n} 10^{\frac{L_n}{10}}\right)$$

(7)

- L = Total sound pressure level (dB)
- L_n = Source specific sound pressure level (dB)
- n = Sound source number

In order to design the quietest system possible, both ambient engine bay and combustion noise were considered. To quiet combustion noise through the exhaust system, in addition to the sound dampening effects of the DPF, an AP Exhaust 6525 conventional muffler was added into the system downstream of the emission components. On the other end of the system, the intake was connected to the stock airbox in order to quiet potential noise from the turbocharger pulling air into the system. However, due to project delays, the sound level of the system driving by at 35 mph, 50 feet away for comparison between straight-piped, DPF and DOC equipped, and DPF and DOC and muffler equipped was not measured as it was not prioritized over other issues and emissions testing. Even so, results are expected for the design presentation at competition.

In addition to the noise from combustion, ambient noise in the engine bay, such as from the oil pump, fuel pump, component vibration, and so on required damping as well. To counter this, all panels were lined with engine foam rated for 60% sound absorption on McMaster Carr. With the engine straight-piped and the decibel meter placed on the non-exhaust side, 3 feet high and 5 feet from the engine bay, the engine bay sound level was monitored with side panels on and off. The sound level without side panels off was 85.1 dB, and 84.7 dB with the side panels on. This test shows a slight drop in overall sound level, but it is not significant due to the amount of noise cause by the straight-piped exhaust that was aimed directly at the ground. Further results are pending and can be expected in the Clarkson Diesel Sled design presentation.

Safety Features

Some engine safety features provided on the snowmobile include gauges to make sure everything is running properly. The essential gauges for engine health monitoring are RPM, coolant temperature, oil pressure, and exhaust gas temperature (EGT). Oil pressure is important to monitor, making sure the pressure stays between 30 and 60 psi. If the pressure drops below 30 psi, significant damage can be caused to the engine due to lack of lubrication. This will also show if the dry sump oil system fails for any reason and needs to be fixed before causing damage. Additionally, knowing the RPM of the engine is crucial, making sure we don't rev above 3500 RPM, so the engine doesn't "redline" and nothing internally is damaged. Coolant temperature helps to gauge how hot the engine is, making sure nothing is overheating. Lastly, the EGT sensor helps tell how hot the exhaust is, which is another measure of how hot the engine is running. Knowing this temperature is important because it can reveal if the in-cylinder temperature is too hot which can lead to internal damage, piston scoring. The maximum allowable EGT temperature reading is 1300F.

Under the hood of the sled, heat reflecting sound absorbing polyurethane foam sheets are attached to all the panels. These sheets provide sound reduction up to 60% and a heat reflection of up to 225F. Additional heat protection components are heat sheaths around the fuel and coolant lines, to protect them from melting due to proximity to the exhaust. Also seen in the engine bay are covers for the clutch, flywheel, and pulley. These covers protect from failure of rotating parts that could launch at the rider or bystanders at high rates of speed. Furthermore, stud retorqueing is recommended annually.

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[4] Sengpiel, Eberhard. RT60 Calculator Wallace C. Sabine Calculation Reverb Time Reverberation Time Sabin Formula Online Sound Pressure Sound Level - Sengpielaudio Sengpiel Berlin. Accessed February 18, 2019. http://www.sengpielaudio.com/calculator-spl.htm

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- BorgWarner
- BTD Management
- Camso
- Cintas
- Curve Industries
- D&D Racing
- Dynojet Research
- Faurecia
- FlowMax DPF
- Hygear Suspension
- Ingles Performance
- Klim Technical Riding Gear
- Land and Sea (DYNOmite)
- Lucyk Contracting
- Monster Tool Company
- New York State Snowmobile Association
- Norton
- Perkins
- Pink Ribbon Riders
- Polaris
- Ricardo
- Snap-On Tools
- SOLIDWORKS
- TEAM Industries
- Windsor Landscaping
- Woody's

Without them, this project would not have been possible

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Definitions/Abbreviations

CO	carbon monoxide
CSC	clean snowmobile challenge
CVT	continuously variable transmission
DOC	diesel oxidation catalyst
DPF	diesel particulate filter
EGT	exhaust gas temperature
G/KWH	grams per kilowatt hour
HC	hydrocarbons
ID	internal diameter
MG/H	milligrams per hour
NO_x	nitrous oxides
PPM	parts per million
SAE	society of automotive engineers
SiC	silicon monocarbon
TiAl	titanium aluminum alloy

Appendix A: Schedule Gantt Chart