Emissions, Fuel Economy, and Sound Reduction Improvements to the 2019 Polaris Indy XC featuring the ProStar 1000 Engine

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Innovations

The 2019 University of Minnesota Duluth Clean Snowmobile Team retained the same ProStar 1000 engine as the previous year, but almost all other components in this year's entry are different. This includes a new 2019 Polaris Indy XC chassis along with a heavy focus on simulations to improve on initial designs. The team has incorporated quite a few innovations to this year's entry including a rear-exiting exhaust system. To accommodate for this feature, a custom tunnel was created and analyzed to ensure it holds up to all load requirements. This change also involved utilizing an aftermarket tunnel cooler, which was previously built into the tunnel, to provide adequate cooling capacity for the 4-stroke ProStar engine. To complement the new exhaust system, a custom silencer, resonator, and tuned header were designed and constructed. Using GT Suite and ANSYS simulations, the designs were improved producing a silencer that targets 16 dominant frequencies and a resonator that provides attenuation for higher frequency levels. Additionally, a dry sump oiling system was installed because of the need for more room for the ProStar 1000 engine. Electronic oil pumps, a Yamaha Phazer oil reservoir, and custom designed oil pan were chosen to complete this design. Another modification that was integral to this year's design was the custom engine mounts that allowed the engine to be lowered further into the bulkhead, as well as a crankshaft extension that improved clutch alignment. GT Suite in conjunction with ANSYS Fluent was used to develop an intake design with tuned runner lengths that provides for better cylinder-to-cylinder volumetric efficiency . Finally, the team started the process of implementing a camshaft phaser in the ProStar 1000 by testing engine emissions by manually phasing the intake and exhaust camshafts.

Team Organization and Time Management

The UMD Clean Snowmobile Team first competed in the Clean Snowmobile Challenge in 2006. Since then, the team has competed 6 times, placing very well and always bringing a fun, marketable, and competitive sled. Coming off of a 2nd place finish last year, a majority of the core members returned for 2019. Presidency and team leads were handed out in April 2018, allowing for the 2019 competition preparations to formulate and start taking shape. These team roles are filled by: Jacob Krabbenhoft (President), Drew Glenna (Powertrain), Mason Busch (Electrical), Josh Dronen (Intake), Mason Todd (Exhaust), Nick Buerman (Drivetrain), and Cory Huot (Engine Packaging).

Regular video meetings were held each week over the summer with a good majority of the research and general design able to come together before the team was even back in school. Once at school, the team displayed the sled and recruited new members at activity fairs to help round out the team. To gain exposure, the team travels twice a year to an SAE Vehicle Night at our local SAE chapter meeting, making sure to engage with the public and other teams as much as possible. The remaining members and new incomers were assigned as helping hands to a system or the choice of smaller projects to help gain experience and pick up slack anywhere needed. Additionally, regular weekly meetings were held to keep the entire team in unison and solidify the general understanding of our goals and progress. A timeline chart, as seen in Figure 1, showed deadlines and project "to-do" lists that were made for each of the aforementioned team leads. This was followed closely to ensure that the project moved along in a timely manner.

Critical Design Reviews (CDRs) were incorporated into the design process this year to help enforce solid and well thought-out designs. This involved gathering the team together and listening to a team member present a major design for the build. The designs were analyzed and picked apart during each CDR to help find any flaws or unintended consequences that would arise from a faulty design. By having each team lead and group member present, this allowed for the design to be looked at from multiple views and determine how the part would affect each system of the sled and how they work together.

Timelines are never perfect and are always changing due to uncontrollable circumstances. The team faced these certain conditions at times throughout the year but worked together to face the problems from different aspects and angles. The team meshed well, each member bringing their own strengths to contribute which pushed the team to meet the goals set at the beginning of the year.

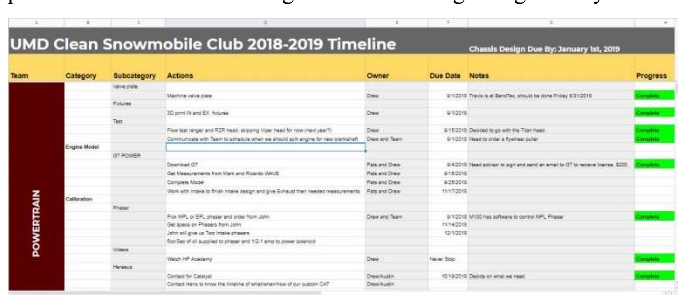

Figure 1. Organizational timeline for recognizing system, project, owner, and due date to track progress and project flow

Build Items of the Snowmobile

- 2019 Polaris Indy XC 129
- Polaris ProStar 1000 cc, 4-stroke, Gasoline, 88 Horsepower (GT Power)
- Camso Ice Attack XT
- Student designed rear-exiting exhaust with silencer, separate resonator tube, and tuned header
- Student designed intake manifold to evenly distribute intake air flow to each cylinder and meet packaging constraints
- Heraeus 3-way catalyst, 176.1x70 mm, 400 cpsi, 6:11:1 Pt:Pd:Rd
- Dry sump oiling system
- Electronic throttle control

Powertrain Design

Engine Improvements

The new engine package in 2018 is serving as a great foundation for the 2019 season. In the 2018 competition the Ranger ProStar 1000 engine package earned a 197.1 E score, which set a respectable bench mark for this year's powertrain design. GT Suite was a helpful addition for the 2019 engine design as it was used extensively in the design of the intake and exhaust systems. The first step was to use a flow bench to find the flow coefficients of the engine's cylinder head. The flow bench data was then inputted into the GT engine model shown in Figure 2. Certain parameters were set such as intake and exhaust runner lengths as well as plenum volume. These parameters were run using DOE experiments in GT to find desired runner lengths and plenum volumes. The plenum volume and runner lengths had to fit in the spaces provided in the sled which helped narrow down the possible options. The runner lengths and plenum volumes were the baseline for the intake and exhaust design where geometry was improved by using ANSYS. The intake and exhaust systems were continuously ran in GT throughout the design cycle to improve power output and reduce emitted noise of the engine. Figure 2 is the final GT engine model with the full exhaust and intake system imported through GEM 3D.

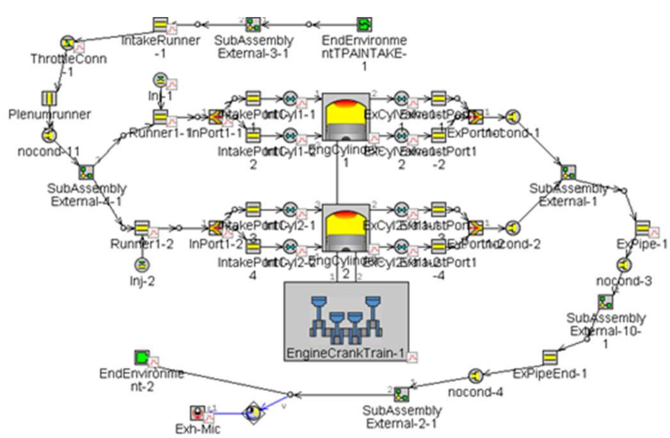

Figure 2. GT Power flowchart of the ProStar 1000 with custom intake and exhaust

Table 1. ProStar 1000 Engine Details

Engine	ProStar 1000
Displacement (cc)	999
Configuration	Inline Twin
Valve Layout	Dual Overhead Camshaft
Fueling	Full Sequential Port Fuel Injection
Compression Ratio	11:1
Bore x Stroke (mm)	93x73.5
Ignition Type	Coil near Plug
Block Material	Aluminum

For this year's design there was an attempt to improve the dyno emissions score by implementing BorgWarner's Cam Torque Actuated (CTA) phaser on the intake camshaft of our ProStar 1000. The phaser changes the valve opening and closing times relative to the engines cycle. The original plan was to run a phaser on both camshafts to greatly improve emissions and torque. Advancing the intake and retarding the exhaust camshafts under lower engine speed and part loadings would supply a large benefit to the emissions and power output. Finding the number of crank angle degrees that our intake valve could advance, and the exhaust valve could retard, was essential in determining the maximum valve overlap. After measuring the cold piston-to-valve clearance, it was found that the intake valve could be advanced 38 crank angle degrees, and the exhaust valve could be retarded 49 crank angle degrees, each with 1.5 mm of clearance. Collecting dynamometer emissions data was the next step, done by phasing the intake and exhaust camshafts manually, one cam sprocket tooth at a time. The graphs below illustrate the large improvements of HC, CO and NOx emissions compared to the stock valve timing.

Figures 3, 4, and 5 demonstrate the value of phasing camshafts with the ProStar 1000 engine. The emissions test conducted was the RMC 5 mode emissions test. One tooth on the cam sprocket was equal to 19 crank angle degrees. To ensure valves would not contact the piston, the intake camshaft was only advanced one tooth, while the exhaust camshaft was retarded two teeth. The exhaust valve closing later in the cycle draws more exhaust gases into the cylinder while another fresh charge of fuel and air is mixed into the cylinder from the intake stroke. This is called internal exhaust gas recirculation, which reduces the need for an external exhaust gas recirculation loop. The unburned hydrocarbons that are left in the cylinder from late exhaust valve closing are set to endure another combustion cycle which reduces HC emissions. The late exhaust valve closing also reduces NOx emissions by cooling combustion temperatures allowing for less nitrogen to atomize with oxygen. HC and NOx were reduced significantly through modes 2, 3 and 4 while retarding the exhaust valve timing. Retarding the exhaust camshaft also decreased CO for modes 2, 3 and 4 compared to the stock camshaft timing. As shown in Figures 3, 4, and 5, retarding the exhaust cam one tooth or more produces high CO as well as resulting in a decrease in power output of the engine significantly. When the intake camshaft advances, with the exhaust camshaft timing in the stock position, it decreases NOx, CO and HC in modes 3 and 4. Intake advancement increases the engines volumetric efficiency during mid and low RPM ranges. Modes 1 and 2 are areas that show the stock intake camshaft timing is better than advancing. The intake air inertia effect is evident for the intake camshaft timing to be at or close to the stock cam timing. It is important to take into consideration the power output of the engine in each of the modes where the intake advancement will increase power through lower and mid-range engine speed at part load. Once the engine reaches a certain engine speed, it begins to produce more power as the reduction of valve overlap takes place. The engine ran the best emissions in modes 2, 3, and 4 when the exhaust camshaft was retarded one or two teeth and the intake camshaft was advanced one tooth (highest valve overlap). As seen in Figures 3, 4, and 5, some of the valvetrain configurations were not able to complete mode 5, as the excessive valve overlap created unsustainable engine idle.

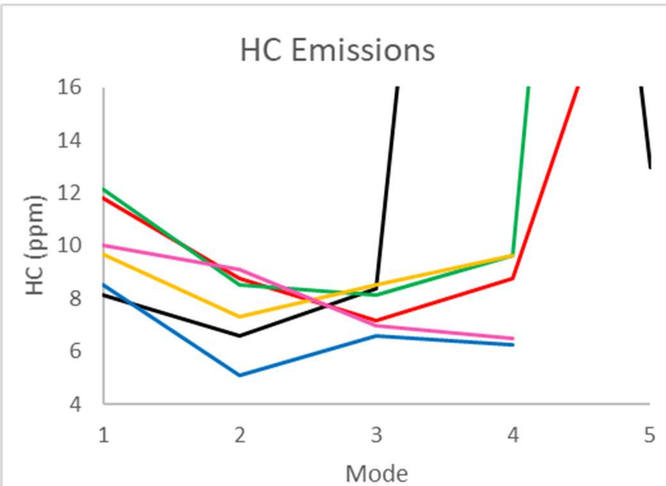

Figure 3. HC emissions data collected from various cam phasing positions. Refer to Table 2 for legend.

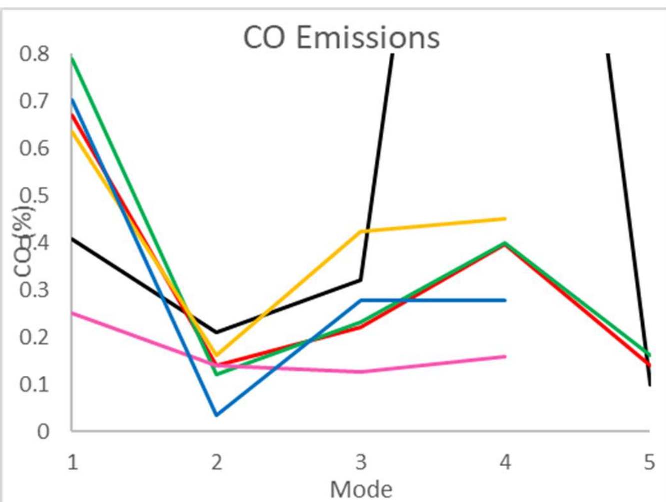

Figure 4. CO emissions data collected from various cam phasing positions. Refer to Table 2 for legend.

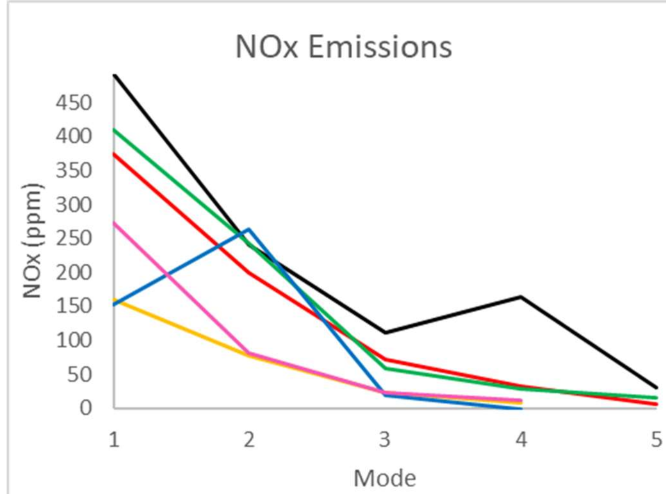

Figure 5. NOx emissions data collected from various cam phasing positions. Refer to Table 2 for legend.

Table 2. Legend for Figures 3, 4, and 5.

Color	Configuration
Black	Stock Intake and Stock Exhaust
Red	Advanced one tooth intake stock exhaust
Green	Stock intake one tooth retarded exhaust
Yellow	Stock intake two teeth retarded exhaust
Pink	Advanced one tooth intake retarded one tooth exhaust
Blue	Advanced one tooth intake retarded two teeth exhaust

Although the original plan was to phase both intake and exhaust camshafts, the complexity of fitting both phasers in the engine and calibrating them properly became evident. Advancing only the intake camshaft would still increase the dyno emissions score greatly, as shown in Figures 3, 4, and 5. Phasing only one camshaft would decrease the amount of cylinder head and camshaft modifications as well as make the engine slightly simpler to calibrate. BorgWarner aided in the design of the custom camshaft and head modifications for the engine. Figure 6 is an image of the head modifications that the engine required to fit the phaser and solenoid.

Figure 6. Camshaft phaser and solenoid housing installed on an adapted ProStar 1000 cylinder head

Due to the cam phaser mounting strategy, a custom camshaft needed to be designed. The camshaft was unable to be machined before this year's competition, which ultimately pushed the project to the 2020-year design. A GT model of the camshaft phaser is in the process of being designed which will aid in knowing when and where the intake camshaft should be phased. The emissions results gave the team a great indication, but GT will provide more accurate knowledge of what phase angle the engine wants at each engine speed and load.

Due to the time constraints, phasing the intake camshaft was out of reach for this year, however a RZR cylinder head was incorporated as the alternative. The RZR cylinder head was flow tested, and a model was designed in GT Power. The higher compression ratio of the RZR engine would prove to increase power. A goal for this year was to increase power from last year's engine package which had a peak power of 45kW and 68 N-m of torque. Figures 7 and 8 show the power curves and volumetric efficiency graphs of each cylinder for the GT model of the RZR 1000 engine. The peak power and torque of the RZR engine was 65.5 kW and 95 N-m, respectively. This equated to a 45 percent increase in peak power and a 40 percent increase in peak torque. The power and torque increases were mostly due to the cylinder head change as the RZR cylinder head was able to flow a greater amount of air compared to the Ranger cylinder head. The Ranger camshafts were installed on the RZR head to avoid the emissions downfall that the RZR camshafts could invoke with more aggressive lobe profiles.

The exhaust header length along with the intake runner lengths and plenum volume were designed using a DOE within the GT engine model. The runner lengths and plenum volume were set as parameters with bounds that would fit inside the bulkhead. With less room on the intake side of the engine, smaller intake runners and plenum volumes were going to have to work where the exhaust runners were able to make up for the power loss. The experiment yielded the exhaust to be 26.5 inches and the intake runners to be four inches. Longer intake runners would have been better to smooth out the volumetric efficiency curves (Figure 8), but these lengths simply could not fit in the allotted space. The plenum volume was best at 200 cubic inches or greater, which gave a great starting point for the intake geometry to be modified through ANSYS simulations.

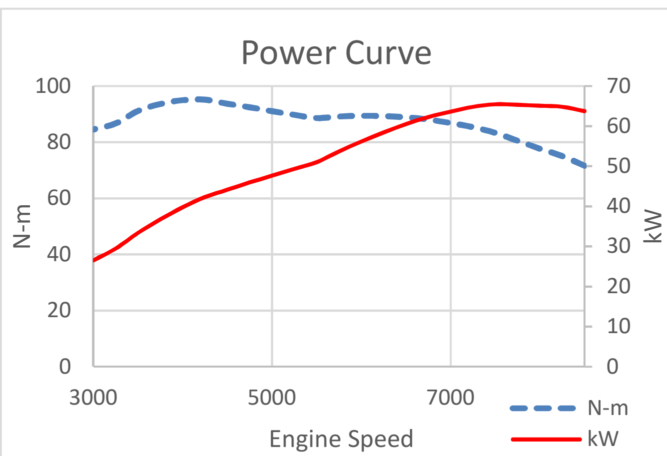

Figure 7. Power and torque curves of engine package estimated by GT Power

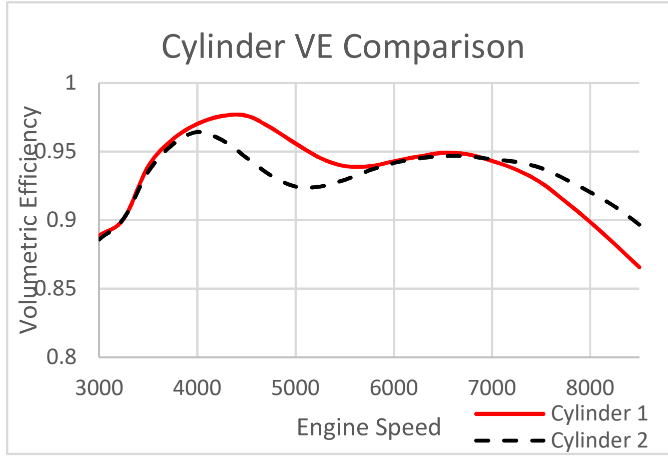

Figure 8. Volumetric efficiency of each cylinder predicted by GT Power

Calibration

The engine was calibrated with a MoTeC M130 standalone ECU. The tuning strategy involved the use of a manifold air pressure sensor, which applies the speed density principle and monitors the pressure changes inside the intake plenum. Dual exhaust gas temperature (EGT) sensors, as well as dual oxygen sensors were implemented into this year's exhaust system to allow for closer control of in cylinder temperatures as well as air-fuel ratios. The EGT sensors provided the ability to correctly trim the ignition timing and fuel delivery to each cylinder. Once the exhaust gases were similar in temperature throughout the operating range, the dual oxygen sensors were able to correct more effectively when enabling the closed loop fueling control. This is a large improvement from last year's calibration where there was one oxygen sensor and one EGT sensor. Running only one of each caused differences in the engines cylinder temperatures and fuel delivery. Ignition timing and fuel trimming could have been calibrated by trimming each cylinder and watching for decreases in emissions which becomes an iterative process. The addition of the second oxygen and EGT sensors sped up the process of correctly tuning each cylinder with the redesigned exhaust and intake system. The engine was calibrated at a stoichiometric mixture throughout the operating range.

Emissions

Figures 9, 10, and 11 display emissions data collected from three different catalytic converters tested. The three catalysts are equal in overall length (176.1mm) and diameter (70mm). The variables being tested were different cell densities, washcoat densities and washcoating elements. Catalyst 1 had a cell density of 400 cpsi and a washcoat of Pt:Pd:Rh, 6:11:1 with a washcoat density of 75g/ft^3. Catalyst 2 had a cell density of 6:11:1 with 400 cpsi and 75g/ft^3. It was also paired with a 400 cpsi, 1:0:1, 33g/ft^3 catalyst. The third and final catalyst was a 600 cpsi, 6:11:1 and 75 g/ft^3 paired with a 400 cpsi, 6:11:1 and 75 g/ft^3. The dyno emissions test was again the 5 mode RMC test, collecting data with a Horiba 5 gas emissions analyzer. The three emissions targeted were hydrocarbons, nitrogen oxides and carbon monoxide. The catalyst was selected by weighing each mode: mode 1 (12%), mode 2 (27%), mode 3 (25%), mode 4 (31%), and mode 5 (5%). Catalyst 2 was chosen from this weighted analysis, which proved to be the preferred option with the 5-mode emissions test. The 2019 engine package reduced HC by 82% , CO by 85% and NOx by 94%, compared to the 2018 engine package.

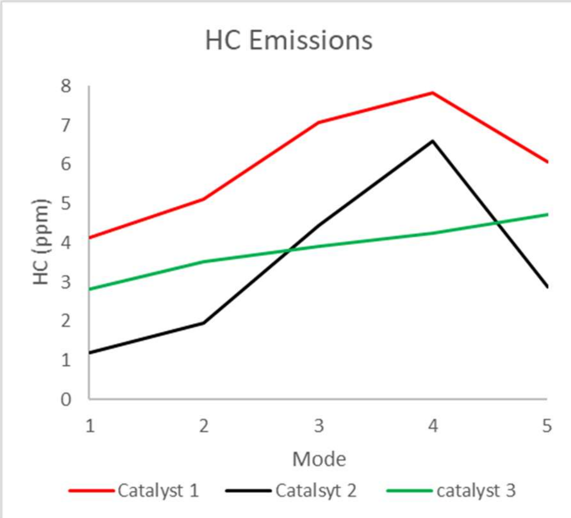

Figure 9. HC emissions during the 5-mode test for each catalyst tested

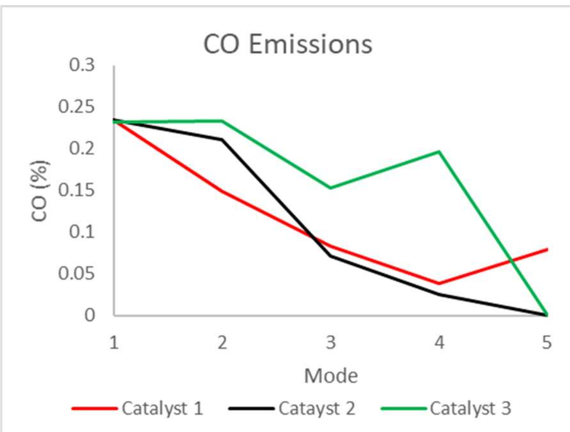

Figure 10. CO emissions during the 5-mode test for each catalyst tested

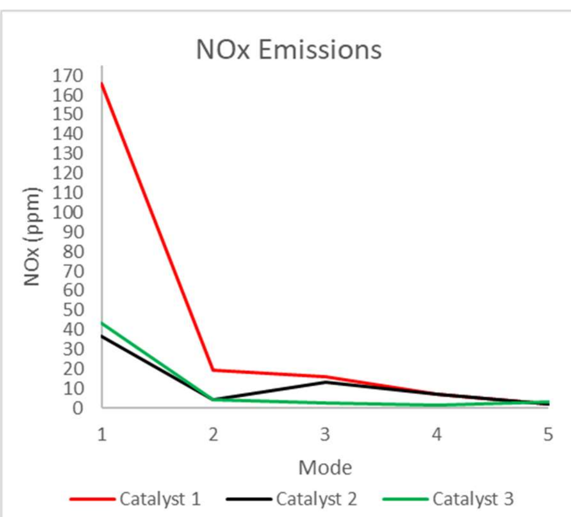

Figure 11. NOx emissions during the 5-mode test for each catalyst tested

Dry Sump Oiling System

Learning from last year's design, the team decided that the ProStar 1000 needed to be lowered further into the bulkhead to improve the clutch alignment and the handling of the machine. However, with the factory oiling system, this was not possible. The wet sump oil pan was simply too deep to allow the engine to be lowered further into the bulkhead or moved to better align the primary and secondary clutches. To solve this issue, the team decided to implement a dry sump oiling system to replace the original wet sump system found on the ProStar 1000.The main concerns when designing the new engine oiling system was supplying an adequate amount of unaerated oil into the engine and being able to adequately scavenge that oil out. To ensure that these concerns were accounted for in the team's new design, the ProStar's stock engine oiling system needed to be characterized. To do this, the team employed a sandwich plate between the oil filter and the engine to allow for a testing loop to be added to the system. This loop contained a flow meter to measure the amount of oil flowing through the engine throughout its operating range. It was found that the stock oil system produces a peak flow rate of 9.75 LPM at 520 kPa of oil pressure, which can be seen in Figure 12.

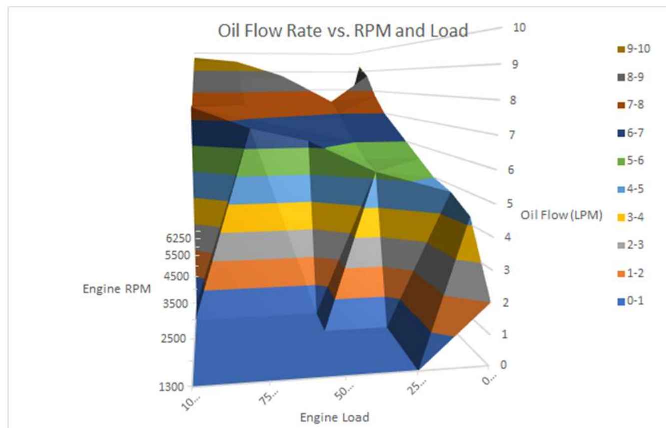

Figure 12. Stock ProStar 1000 oil flow rate compared to RPM and load throughout the operating range

It was decided that employing an electronic oil pump to circulate oil into the engine would be difficult due to packaging constraints and the inability to effectively power an electric oil pump of that size. With this in mind, the team chose to utilize the engine's stock oil pump to supply pressurized oil into the engine and scavenge oil out of a newly designed oil pan using two small electronic scavenge pumps. Two Turbowerx Exa-Pump Mini scavenge pumps, which are each capable of moving 7.6 LPM with a low amp draw, were chosen. These pumps were selected as they would be able to draw more oil out of the engine than what was entering, which is needed for an effective dry sump system. While testing in the dynamometer, it was discovered that pressure was building inside the crankcase as the scavenge pumps were not as effective at scavenging crankcase gasses. To solve this issue, a new positive crankcase ventilator with a check valve was installed where the original PCV was located. In order to minimize oil blow-by, custom stainless-steel baffles were used to partition the PCV away from as much oil as possible, shown in Figure 13. These baffles were placed in such a way that the flow of oil particles is impeded, but pressure from the crankcase can still be vented through the valve and into the intake tract.

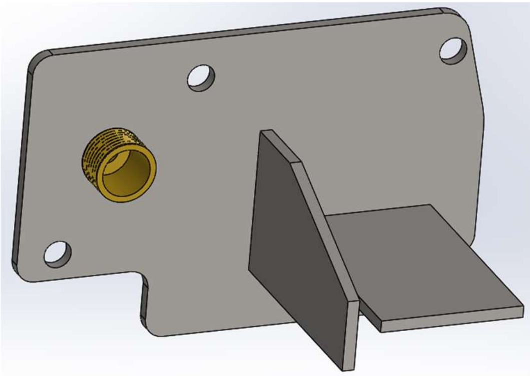

Figure 13. Positive crankcase pressure relief valve plate with designed baffles

Like most dry sump oiling systems, these scavenging pumps needed to feed into an external oil reservoir. After extensive research, the team decided to not manufacture a custom reservoir due to the complex design of baffles to remove aeration while still allowing for oil to flow through the reservoir efficiently. It was determined that an OEM equivalent would be a better option, and a dry sump oil reservoir was chosen from a 2007 Yamaha Phazer. This tank was chosen because it is a readily available OEM part that was specifically designed for dry sump oiling. The capacity of the tank is 2.5 quarts, which is equal to the capacity of the wet sump oiling system of the ProStar 1000. In addition to the 2.5 quarts within the reservoir, another 0.5 quart of oil is added to fill all oil lines. This allowed for an oil capacity which was greater than that of the ProStar 1000 and Yamaha Phazer, which allows the oil reservoir to effectively remove aeration in the oil.

To produce a high-quality oil pan, the team set forth criteria for the oil pan to meet. It would need to be a single piece to avoid the potential of an oil leak, allow for oil to be plumbed to the stock oil pump, feature multiple scavenge points, and be as thin as possible to allow for the engine to be lowered as deep into the bulkhead as possible. Due to the oil pan needing to be a single piece, it was determined that it would be machined from 6061-T6 aluminum. This material was chosen as it would easily be able to withstand the harsh conditions that the pan will face, its machinability, and it is readily available. The new oil pan would also need to allow for oil to flow through it, because the engine would be supplied oil using the stock pump located at the bottom of the crankcase. To allow for this, the oil pan was designed with a feature that would press-fit into the inlet of the stock oil pump and could be tapped so that oil would flow through the pan and into the pump.

It was decided that oil could simply be gravity fed into the bottom of the pan up into the inlet port of the pump, because the original pump was designed to pull oil from a wet sump. Multiple scavenge points were also necessary for removing oil from the crankcase due to the large volume of oil that flows through the engine. The orientation of the engine tilts back towards the tunnel which aids in the pooling of oil in the back of the pan, which is where two scavenge points were placed. To promote oil pooling in this location, multiple components were designed into the pan to direct oil towards the scavenge points. These components include sloped features and rounded edges. To maintain minimal thickness, the pan was split into multiple different sections so that each individual section could be as thin as possible, while still directing oil towards the pickups. This was beneficial for both engine mounting and accommodation of the lower steering assembly in the front of the snowmobile. The final design is shown in Figure 14. After many hours inside of the team's dyno cell with a sight glass installed inline before the inlet of the engine's oil pump, it has been proven that the design meets the team's criteria as it is able to effectively supply unaerated oil to the engine and scavenge it out of the oil pan into the external reservoir.

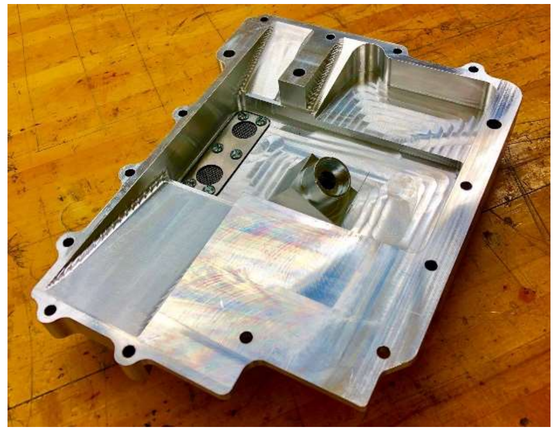

Figure 14. Dry sump oiling pan with oil inlet shown at center and oil scavenge ports located on the left

Cooling

The team chose to employ a rear-exiting exhaust and a custom tunnel, and as such, the snowmobile's original tunnel coolers were removed. To supplement this loss, an aftermarket 8" x 36" tunnel cooler was installed in the new tunnel, as seen in Figure 15. In addition to the new heat exchanger, a radiator was mounted in the nose of the snowmobile. The radiator chosen was a 1.5" x 5.5" x 13" aluminum radiator with cooling fans to allow for adequate airflow through the radiator. This helps to provide extra capacity to cool the large displacement, 4-stroke ProStar 1000, especially at low speeds, idle, and minimal snow conditions.

Figure 15. Rear-exiting exhaust (resonator) and tunnel cooler mounted in newly designed tunnel

Exhaust

Silencer and Resonator Design

The UMD Clean Snowmobile Team takes pride in developing and designing a complete exhaust system. This year's entry features a tuned header, sound absorbing resonator tube, and a multi-pass reactive silencer. Limited by the packaging constraints of the engine bay, the exhaust system is routed into the modified tunnel towards the rear of the snowmobile; this is referred to as a rear-exiting exhaust system. This innovation has proven to be a practical solution to the packaging problem and overall, allowed the team to design an exhaust system with low sound output, and proven durability.

The primary source of sound emitted from a snowmobile comes the engine's exhaust. The alternating exhaust strokes of a 4-stroke engine generates large pressure pulsations. These alternating pressure levels create the harsh noise emitted from the exhaust tailpipe. A silencer serves as a component to counteract these sound pressure imbalances and allows the exhaust system to emit a quality sound.

In an effort to design the silencer around all specific characteristics of the engine, a series of simulations were conducted in GT Suite to determine the dominant frequencies of the powertrain package. These results are shown in Figure 16. Dominant frequencies are defined as the sound frequencies with the greatest sound pressure levels. Setting up a microphone at the tailpipe of the GT Power model without any sound limiting devices, the team acquired a handful of data plots including 1/3, 1/10, and full octave sound pressure levels throughout the operating range of the engine package.

The dominant frequencies from the 1D simulations were utilized to determine the specific sound wavelengths to target for the silencer. A Microsoft Excel spreadsheet was used to calculate the quarter wavelength for each of the dominant frequencies. The baffles and expansion chambers of the silencer were designed using this quarter wavelength information by way of destructive wave interference. Overall, the silencer was designed to eliminate certain frequencies by targeting the incoming pressure waves with the highest sound intensity.

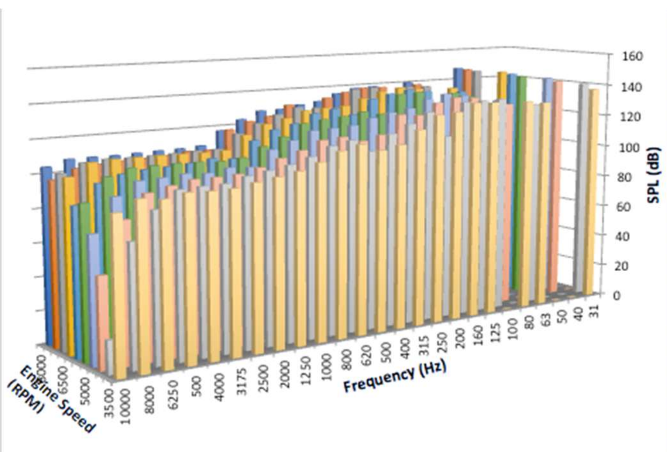

Figure 16. Sound intensity vs. RPM and frequency created from GT Power with an open header. These were used to determine dominant frequencies.

This silencer is a 5-stage multi-pass, reactive, combination expansion chamber design targeting 16 dominant frequencies from 31.25 Hz to 800 Hz. The first chamber is reactive and reflective, designed around the principles of destructive wave interference. Exhaust pressure waves enter the first chamber of the silencer, which is composed of 11 rigid baffles. Each baffle is placed using the quarter wavelengths of the 16 dominant frequencies. When the initial sound waves enter, they are reflected to create destructive interference with incoming sound waves. The baffles were manufactured from two sheets of stainless steel with ceramic packing material placed between them. This chamber was also designed to act as an expansion chamber. The expansion chambers throughout the silencer are over 1.5 times larger than the cross-sectional area of the exhaust pipe. The second chamber is connected to the first chamber with multiple transfer tubes to give the silencer a relatively low backpressure. The second chamber is the first of four total tuned expansion chambers. Each expansion chamber was designed by using the Microsoft Excel program, which tuned the length and width of the chambers to attenuate a specific dominant frequency.

GT Suite was used to improve the design of the silencer. Three designs were constructed and converted into GT subassemblies using GEM 3D. These assemblies were then compared, and the best was chosen as an initial design. This design was then modified to lower sound and maintain horsepower before final assembly. The construction of the silencer is shown in Figure 17.

A sound-absorbing resonator tube was also placed before the silencer along the length of the tunnel. The resonator provides attenuation for the higher frequency levels ranging from 1000 Hz to 20000 Hz. Perforated pipe with an open area of 20% is surrounded by ceramic high-density packing material. This prevents the packing material from being dislodged from the resonator and allows the sound pressure waves to escape the intended exhaust flow path. The packing material absorbs the sound pressure waves passing through the open area of the perforated pipe, and the fibers of the packing material dissipate the sound energy as thermal energy. Figure 19 shows the resonator prior to final assembly.

Final engine sound intensity was predicted using GT Suite to validate the entire exhaust system. These results depict sound intensity throughout the full range of engine speeds at full load and are shown in Figure 18.

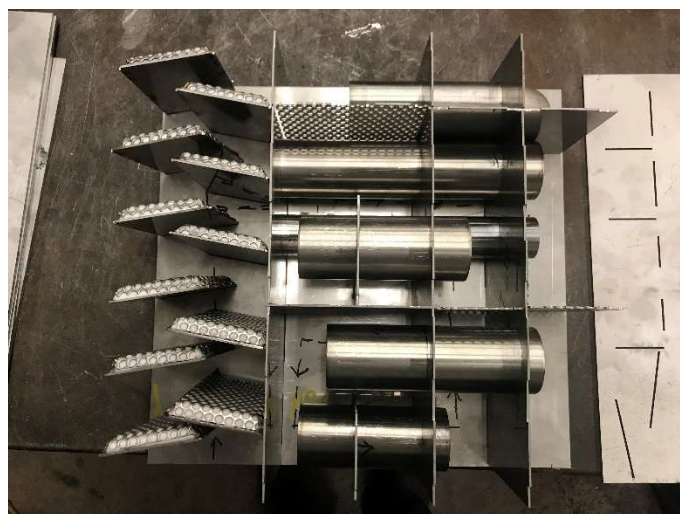
Figure 17. Silencer during manufacturing highlighting the layout of the reflective first chamber and multi-pass expansion chambers

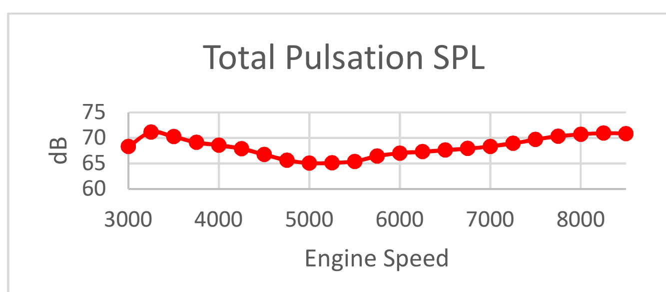

Figure 18. GT Power prediction of sound intensity from engine package vs. engine speed at full load.

Figure 19. Resonator tube prior to final assembly

Header Design

In order to preserve the performance of the team's engine package, the entire exhaust system was designed with a heavy emphasis on analysis. For this reason, GT Suite was the primary source for determining the exhaust header runner lengths to promote scavenging. The team was able to successfully simulate and manufacture a tuned performance header with equal runner lengths of 26.5 inches. The custom header is fabricated from 1 ¾" 304 stainless-steel tubing and features separate oxygen and EGT sensors for each cylinder. A photo of the completed header is shown in Figure 20.

Figure 20. Tuned exhaust header mounted in the snowmobile's bulkhead

Intake

This year's intake system was completely redesigned to improve volumetric efficiency, noise, and packaging constraints. The primary function of the intake system is to supply a uniform air charge to each cylinder while minimizing noise. GT Suite was used in conjunction with ANSYS Fluent simulations to improve the intake design. Over 30 different plenum designs were simulated to create this year's design.

Intake Packaging

The intake is on the rear-facing side of the engine, so packaging became a high priority. As the overstructure and gas tank are very close to the intake, it had to be a compact design that fit the space available. The plenum was designed to meet these restrictions while still meeting the geometry requirements from the 1D and 3D simulations. The GT Power model was used to confirm that the plenum and intake runners would work harmoniously to deliver equal intake charge to each cylinder.

The electronic throttle body mounts directly to the plenum in the same orientation as the factory engine. A large diameter silicon intake tube, utilized to decrease pressure drop through the intake system, extends from the throttle body to the upper plenum. An elbow was also designed to attach this tubing directly to the stock upper plenum and is shown in Figure 21.

Figure 21. Upper intake elbow designed to route intake air from the stock upper plenum.

Intake Analysis

In the early stages of development, runner lengths and plenum volume for the intended horsepower and torque were determined in GT Suite. Once these were determined, the shape of the plenum was then fine-tuned using ANSYS Fluent.

Many different plenum configurations were simulated using ANSYS Fluent CFD simulations to improve the flow characteristics. For the boundary conditions, intake port pressure traces were calculated in GT Power. These pressure waveforms were set as the outlet pressures for each runner, and the inlet was set to atmospheric pressure. A k-epsilon turbulence model was used to improve the accuracy of the results.

At higher engine speeds, the shape of the plenum became critical to supplying equal amounts of air to each cylinder. In initial iterations of the intake design, one cylinder would typically get starved above 5000 RPM resulting in the volumetric efficiency of one cylinder to oftentimes be over 20% of the other cylinder. Typically, the path with the highest pressure drop would be the starved cylinder.

To unify the flow to each cylinder, changes were made to the shape of the plenum. The main modification was an L-shaped section cut through the plenum. A parametric study was conducted in ANSYS Fluent where the dimensions to the plenum were set as an input parameter. Pressure drop from the inlet to each runner and mass flow rate through each runner were set as the output parameters to ensure the design would provide uniform flow. Multiple design points were simulated in Fluent. Figure 22 shows the velocity vectors from runner with the final design, which visualizes the uniform flow at peak power engine speed. The configurations with the lowest difference in cylinder-to-cylinder pressure drop and mass flow rate were then confirmed and adjusted with GT Suite.

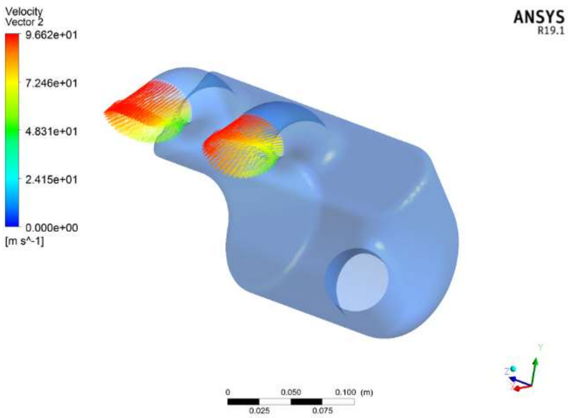

Figure 22. Velocity vector cross section of plenum from ANSYS Fluent at peak power engine speed

The final intake plenum design was manufactured using selective laser sintering (SLS) of carbon fiber nylon 11 and is shown in Figure 23. This material was chosen for it's superior thermal and mechanical properties compared to other additive manufacturing materials. As cylinder-to-cylinder volumetric efficiency distribution was a large focus for this design, it was compared to last year's plenum design to showcase the improvement. This comparison is shown in Figure 24.

Figure 23. Final intake plenum designed to achieve even air flow distribution to each cylinder

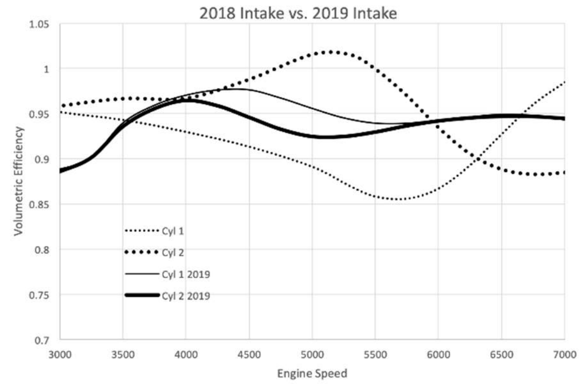

Figure 24. Comparison of volumetric efficiency distribution throughout the RPM range at full load between 2018 and 2019 entries (GT Power)

Chassis Modifications

Tunnel Modification

One of the team's major goals this year was to implement a rear tunnel-exiting exhaust. When deciding on exhaust routing through the tunnel, it was realized that it would be difficult to avoid placing the exhaust near the stock tunnel coolers. Options were explored to avoid this, and it was decided that it would be best to remove the stock tunnel coolers and replace them. In the Polaris AXYS chassis, the tunnel coolers were built in as a structural member of the tunnel and as such, care was taken to analyze a new design. A full tunnel model was created in SolidWorks, and static structural and buckling analyses were completed using ANSYS.

The new tunnel is manufactured out of a sheet of 1/8" 5052 aluminum and includes 1/8" angle aluminum on the inside corners to add stiffness to the design which is shown in Figure 25. Through FEA analysis, the new tunnel design was proven to be equivalent to the stock component. More details on this analysis were discussed in the team's chassis paper entry. Ultimately, this design allowed for a more optimal mounting location of the rear-exiting exhaust and provided adequate space for the new tunnel cooler.

Figure 25. Tunnel modification on the 2019 Polaris Indy XC 129

Engine Mounts

As Polaris currently does not offer a production 4-stroke snowmobile, changes in the bulkhead had to be made to mount the larger ProStar 1000 engine. To improve on the overall packaging of the snowmobile, it was decided that the engine needed to be lower than the previous design to lower the center of gravity of the machine, so new engine mounts needed to be designed. The goal for these new mounts was to bridge the gap between the stock engine mounts on the bulkhead and the rear engine mounts on the engine. This worked well last year, and as the team worked to get the engine lower and more centered in the bulkhead, the design had to be modified. Newly designed mounts, shown in Figure 26, allow the engine to be more vertical while simultaneously lowering it into the bulkhead. The ProStar 1000 also has engine mounts on the front of the engine that are tied in with the new modified lower crossbar in the front of the bulkhead. To ensure the engine would not move excessively under clutch engagement, a torque stop was designed utilizing the stock location on the bulkhead and engine. ANSYS FEA was used to validate these designs as an entire system. More details about these simulations and the mounts were discussed in the team's chassis modification paper.

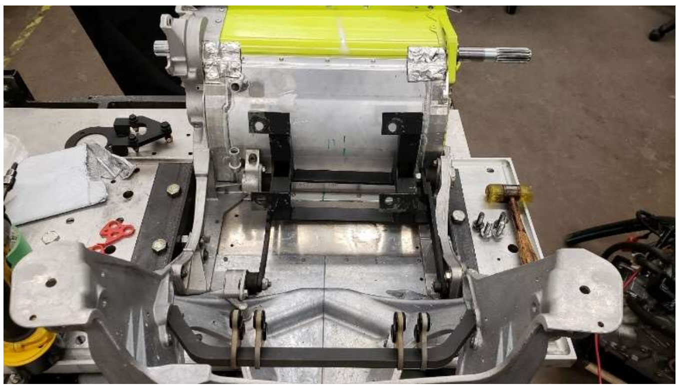

Figure 26. Newly designed front and rear engine mounts

Crankshaft Extension

One of the difficulties encountered with last year's design was that the output on the ProStar 1000 was very short, and required that the engine to be as far to the clutch side of the bulkhead as possible. This made it challenging to mount, design, and place other systems around it. As the engine was moved towards the center of the bulkhead this year, it was clear that a longer output shaft was needed. Two options were explored including a fully custom crankshaft or an addition to the existing crankshaft to extend the clutch out 2.3 inches. Without the experience or machinery in house, custom fabrication shops were contacted to assist the team in finding a solution. Ultimately, the solution consists of an extension that mounts onto the existing output shaft to move the clutch mounting point outwards 2.3 inches, shown in Figure 27. Along with this, the engine's crankshaft and balancing shaft were rebalanced to account for this addition. This also required the fabrication of a custom bolt for mounting the primary clutch to the output shaft. This was achieved by designing a grade 10.9 bolt that mimicked the geometry of the stock hardware but increased in overall length.

Figure 27. Custom output shaft extension utilized to extend clutch location 2.3 inches away from the engine

Over-Structure

Another major chassis modification included the rear spars on the over-structure which was needed to fit the intake plenum and fuel rail. Ultimately, the same rear spars were used from last year's design and are made from steel rather than the stock aluminum. These spars have been bent outwards to allow more space to be utilized in between them. Another modification to the over-structure included modifying the sidebar running from the rear of the bulkhead to the front of the over-structure, as the new engine mounting location resulted in an interference issue. To correct this, a new sidebar was designed that goes over the magneto cover. This modification also allowed for mounting of the dry sump components and electronic water pump to the sidebar. Both modifications were analyzed using ANSYS and are shown in Figure 28. Detailed results from these studies are further discussed in the team's chassis paper entry.

Figure 28. Rear overstructure modified to fit the intake plenum and fuel rail

Clutch Cover

A custom clutch cover was designed to comply with competition rules. Keeping safety in the forefront, the part was manufactured from 6061-T6 aluminum 0.090" thick and has small slots in it for keeping the belt and clutches cool. Due to the crankshaft extension modification, the clutch cover extends down below the center-line of the clutches. Also, the cover spans over the front of the clutches to completely encase the clutches and belt. A photo of the final design is shown in Figure 29.

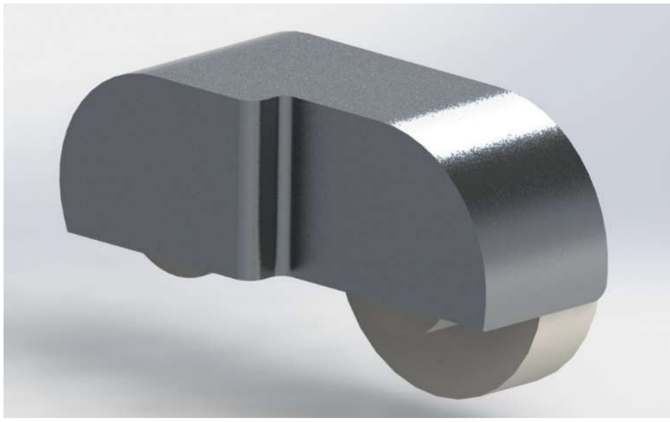

Figure 29. Custom clutch cover design

Suspension Modifications

Due to the added weight of the ProStar 1000 over the stock powertrain package, it was decided that the suspension needed to be changed to improve handling over last year's design. To account for the increased engine weight, the front suspension coil over spring rates were increased. Along with this, all four shocks on the snowmobile had the valve stacks adjusted to increase suspension stiffness. With the addition of a rear-exiting exhaust, a reduction in suspension travel was required to maintain clearance when at full compression. To achieve this, aluminum blocks were manufactured that moved the contact point upward 1.75 inches and rearward .5 inches. These changes combine to make the sled handle better in addition to allowing for the rear-exiting exhaust.

Sound Improvements

Chassis

A critical component of this competition, and improving a snowmobile, is reducing sound. For this season, proper justification was needed for changing from a Polaris Rush Pro-S with a PRO-XC rear suspension to the new Indy XC's PRO-CC rear suspension.

The process for chassis sound testing is the same as in previous years. Utilizing a ten-horsepower electric acting as the primary clutch, the power was transferred through the entire drivetrain down to the track. The track was set on a stainless-steel table that was built to sit in the bottom of a large metal tank. This allows the weight to rest on the track and suspension, much like when it sits on the ground. With the table acting as the boundary condition, load was placed on the rear suspension system, drive sprockets, idler wheels, and axle wheels, all in an effort to accurately measure the simulated sound levels of riding on snow. In order to keep the track and skid properly lubricated, an external water pump was used to provide a constant layer of soapy water on the table. The system was improved this year by adding two walls of sound dampening material, one to the front and one to the rear of the sled. The complete set-up has noise reduction walls on three sides.

The room that housed the chassis sound dynamometer was a closed room that had no other sound inputs during testing. While it wasn't as ideal as a full quiet room, it was very consistent for a university level club. The dyno was placed along one wall, and the sound meter was placed in a position six feet away. The electric motor was run at three distinct speeds: 30, 50, and 70 hertz while sound levels were recorded at each position. Testing began with a baseline, which was evaluated with the acoustic imaging method developed last year along with the implementation of the new dynamometer improvements. These results produced the contour map shown in Figure 30 which displays that a major source of sound originated from the front of the skid and the tunnel sides.

Figure 30. Baseline chassis sound imaging of the 2019 Polaris Indy XC with the chassis sound dynamometer improvements (Measurements in dBA)

Multiple sound reduction strategies were tested, and their respective performances were analyzed. As shown in Table 3, Test 1 was a baseline of the stock drivetrain. Test 2 was a study of track tension versus sound intensity, in which, the track tension was set loose and tightened in small increments until sound levels rose higher than the starting level. As shown in Figure 32, an ideal track tension for sound reduction was found to be a tensioning bolt length of 2.0625 inches. Test 3 was the addition of one layer of Lizard Skin, a sound deadening material, to the underside of the tunnel along with another layer of a high-temperature rated Lizard Skin product to withstand the increased temperatures of the rear-exiting exhaust. Test 4 was the removal of the idler wheels from the skid and the addition of DuPont Vespel graphite insert hyfax. For Test 5, an anti-stab kit was added that features a set of four wheels which mount in the front of the rails. These wheels serve to reduce track slap at the front of the skid. Finally, Test 6 was the addition of tunnel shrouding to block and deflect sound from inside the tunnel. Detailed sound levels from each of these tests are shown in Figure 31. The final acoustic imaging map after these modifications are shown in Figure 33.

Table 3. Overview of the sound reduction strategies shown in Figure 31

Test 1	Stock Baseline
Test 2	Track Tension Adjustments
Test 3	Lizard Skin
Test 4	Remove of Idler Wheels
Test 5	Addition of Anti Stab Wheel Kit
Test 6	Installing Shrouding

Through the various improvements, the emitted sound was found to have decreased overall by 3 dBA, resulting in a final sound level of 78 dBA. This is lower than the sound level of 81 dBA achieved from last year's entry with the Pro-S drivetrain. These results are all independent of the engine package and its sound properties.

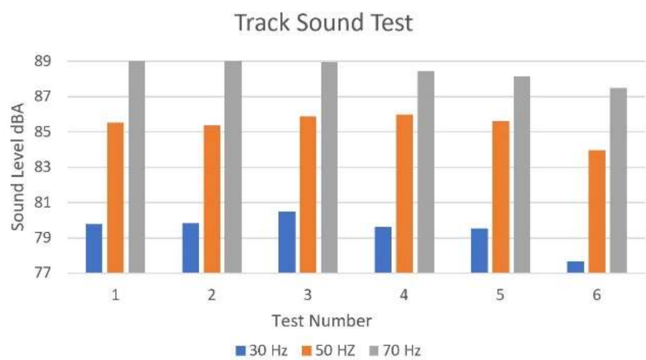

Figure 31. Sound testing on the chassis sound dynamometer of six different sound reduction strategies.

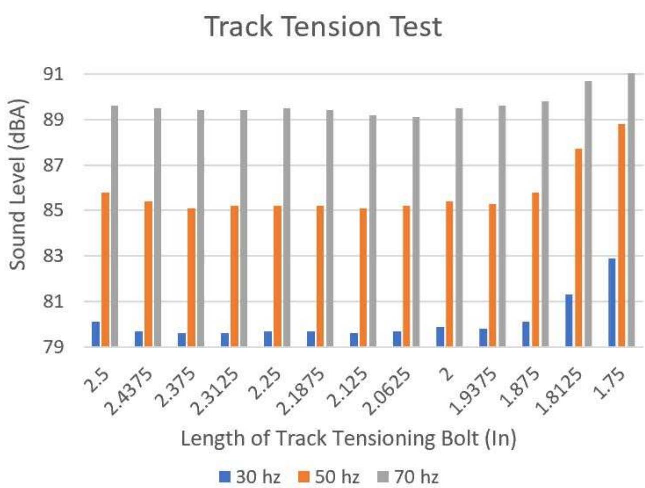

Figure 32. Detailed analysis of the track tension's impact on total chassis sound output.

Figure 33. Final sound imaging of the 2019 Polaris Indy XC with chassis sound improvements. (Measurements in dBA)

Conclusion

The 2019 UMD Clean Snowmobile Team has made significant improvements to the Polaris Indy XC chassis and the ProStar 1000 engine package. A heavy focus on design analysis was used with GT-Suite and ANSYS to validate and improve on initial designs. The team's rear-exiting exhaust, dry-sump oiling system, custom designed intake plenum, and handpicked catalyst all compliment the engine package to create a system that is practical and quiet with drastically reduced emissions and a competitive MSRP of $11,499.

References

1. Hatti, K., Shah, S., Thombare, D., "A Practical Approach towards Muffler Design, of 10 Development, and Prototype Validation," SAE International Journal of Engines. 2010. doi:10.4271/2010-32-0021. https://saemobilus.sae.org/content/2010-32-0021.
2. Heywood, John B. "Internal Combustion Engine Fundamentals" McGraw Hill Inc. 1988.
3. Kane, Jack. "- Exhaust System Technology-." EPI, Inc. April 29, 2015. Dec 2018. http://www.epi-eng.com/piston_engine_technology/exhaust_system_technology.htm.
4. Lee, Ming-Ran, et al. "Exhaust System Design for Sound Quality." SAE Technical Paper Series, 2003, doi:10.4271/2003-01-1645. https://saemobilus.sae.org/content/2003-01-1645.
5. Munjal, M. L. Acoustics of ducts and mufflers. West Sussex: John Wiley & Sons Ltd. 2014.

Definitions/Abbreviations

CO	Carbon monoxide
DOE	Design of experiment
ECU	Engine control unit
EGT	Exhaust gas temperature
GT Suite	1D engine modeling analysis software
HC	Hydrocarbons
NOx	Oxides of nitrogen
PCV	Positive crankcase ventilation
RMC	Ramped modal cycle
RPM	Revolutions per minute