Engine Swap of a Polaris Rush 800 Pro-S with a Polaris RZR XP 1000

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Abstract

The St. Cloud State University Slick Cylinders Snowmobile Team has performed an engine swap for entry in the 2017 SAE International Clean Snowmobile Challenge. The engine swap was performed on a base model 2015 Polaris Rush 800 Pro-S chassis using a 2016 Polaris RZR XP 1000 engine equipped to operate efficiently on gasoline and ethanol fuel blends. The engine has been tuned using a Bullydog system and uses a mechanical throttle cable for simplicity. Custom engine mounting brackets, chassis reinforcement, wiring harness modifications, and the implementation of a catalytic converter and custom exhaust were designed to reduce emissions and noise levels. The lightweight chassis and powerful engine results in a powerful, environmentally-friendly vehicle that is still desirable to customers at a reasonable price.

Introduction

The Society of Automotive Engineers (SAE) created the Clean Snowmobile Challenge in 2000 when snowmobiles were banned from national parks due to their loud design and lack of emission control. The Clean Snowmobile Challenge is an engineering design competition among colleges and universities with the common goal of creating a clean, quiet, and practical snowmobile that is still desirable to customers. Teams will demonstrate their snowmobile improvements as well as reliability, efficiency, and cost effectiveness at Michigan’s Keweenaw Research Center from March 6th - 12th, 2017.

Sled Selection

One of the more difficult decisions for the team was choosing a base model sled. With having a Polaris sponsorship, the Slick Cylinders had a range of choices between the Indy, RMK, Rush, and Switchback models. Due to the intent of the competition, the RMK and Switchback were eliminated since the sled would not need a deep paddles or a long track to ride on groomed trails. Between the Indy and the Rush, the Rush was chosen due to the engine sitting under the hood and the intent of our engine swap. The Indy had more space for a replacement engine, but the team was concerned that the Indy models would not be able to hold up to the engine choice.

The team was donated a 2015 Polaris Rush 800 Pro-S. This chassis was strong enough to hold up to the horsepower and torque of the 2016 Polaris RZR XP 1000 engine which was the team's first choice for the engine swap. The Rush setup is very comfortable and user-friendly when it comes to adjusting the suspension, and it was known that stiffer suspension components could be added to the chassis with it being a similar model to the XCR race sled. With the 800 2-stroke engine out of the sled, the team determined that the 4-stroke engine would fit into its place.

Since the engine swap required a tougher chassis, the Rush was an averagely priced snowmobile that would fit the performance capabilities that the team anticipated. It also wasn’t too narrow or too wide to make it seem uncomfortable, and the rider seating allows hips to be above the knees to prevent any rider fatigue.

Engine Selection

When it came to choosing an engine, the team looked for a Polaris model 4-stroke that would squeeze into the Axys chassis and had similar torque and horsepower as the Rush. The best engine choice turned out to be a 2016 Polaris RZR XP 1000. This engine was just narrow enough to sit between frame members of the Rush with a few chassis wall reinforcements. This engine also only added 33 lbs to the nose of the sled (Rush engine @ 90 lbs and RZR engine @ 123 lbs). The chassis is no longer an “Axys” chassis because of this, but most sleds on the market are already nose-heavy and the team knew that stiffer front suspension components could be added.

The Rush 800 engine has 154 hp with 102 ft-lbs of torque, and the replacement RZR XP 1000 engine has potential of 110 hp with 72 ft-lbs of torque, plenty below the allowable limit of 130 hp for competition. A stock Rush 800 gets around 12-14 mpg, while the stock RZR XP 1000 in-unit gets around 10-11 mpg. With the XP 1000 engine in a lightweight snowmobile chassis, the fuel mileage is expected to rise.

Design Process

To help the Slick Cylinders design all of the components in the small space of the chassis, the team utilized 3-D modeling. Once the snowmobile was stripped of its stock engine, the chassis and RZR engine were scanned using a hand-held Artec Eva scanner and ran through Artec Studio Professional Version 10 software to produce a virtual model (*Figures 1 and 2*). With space being the team's main constraint, this CAD model made it simpler to engineer components that would need to closely fit next to another and still be operational.

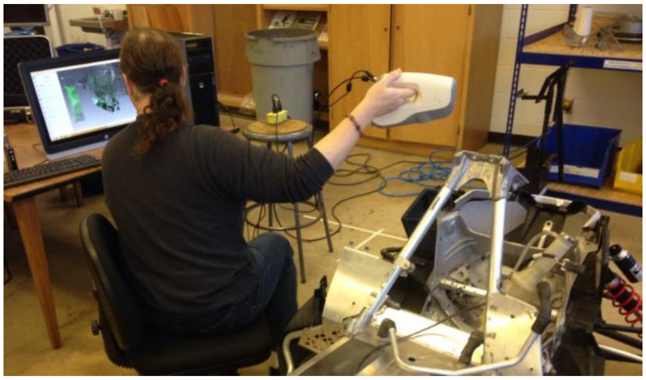


Figure 1: Utilizing the 3-D scanning package to create a model of the snowmobile interior.

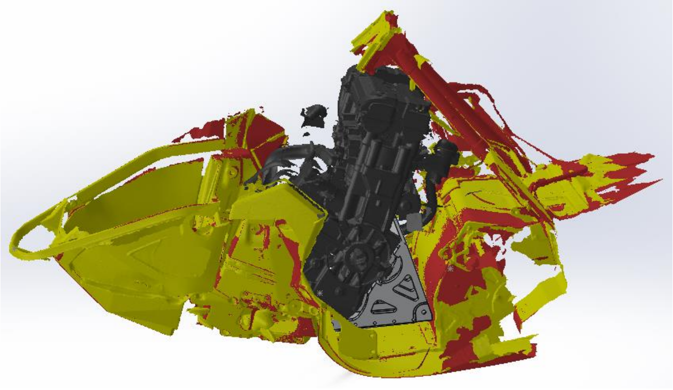


Figure 2: Assembled chassis and engine scans in SolidWorks.

Engine Support

The rear engine bracket is made from a series of ½" 5052 T32 Aluminum plates bolted together using grade 5 ¼-20 bolts. All of the plates were designed in SolidWorks and cut using a water jet. The ½" thick plates were necessary for threading the 1/4-20 bolts and added to the overall rigidity of the support. The team chose the thread depth by calculating the number of threads required to shear the bolt, shown in *Equation 1*.

$$A_S = \pi \cdot D_{major} \cdot w_o \cdot p \qquad (1)$$

Where A_s = Shear Area, calculated to be 0.033 in^2
D_{major} = Major diameter of threads
w_o = Outer thread factor
p = Thread pitch

For which the yield strength of the aluminum threads was calculated below in *Equation 2*:

$$S_{ys} = 0.577 \cdot A_s \qquad (2)$$

Where S_{ys} = Yield strength of the engine mount, calculated to be 545.67 lbf.

And the force to break the stock bolt is show by *Equation 3:*

$$F_{Break} = T \cdot \pi \cdot r^2 \qquad (3)$$

Where T = Tensile strength x Shear Area, calculated to be 5400 lbf from 5052 H32 material properties.

Using this design method eliminated any concern of stripped threads. The bracket is bolted directly to the engine through six 10 mm allen head cap screws and then attached to the chassis through the 4 stock vibration isolators.

A force calculated from a snowmobile drop of 6' and an average force to stop suspension travel in 6" was used to estimate the largest vertical force on the engine mount. The largest expected vertical force as shown by *Equations 4 and 5* give 492 lbf.

$$Velocity = \sqrt{2 \cdot g \cdot h} \qquad (4)$$

$$Force = m_{engine} \cdot a \qquad (5)$$

Other forces applied to the bracket are the engine torque and belt tension. The worst-case scenario for this setup is when the primary clutch first engages with the belt. When the clutch first engages, the belt rests on the primary clutch shaft at radius of 1.125" and the torque output from the RZR engine at takeoff is about 60 lbf, shown in *Figure 3*.

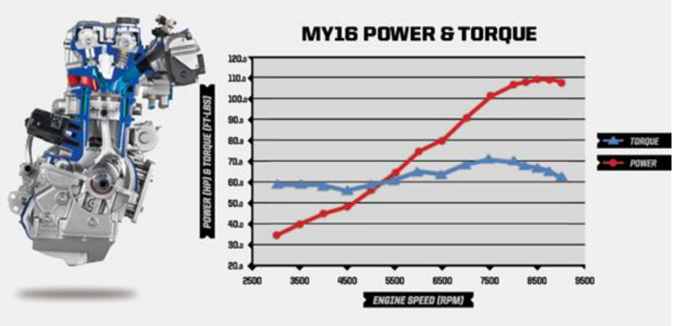


Figure 3: 2016 Polaris RZR XP 1000 power and torque graph. Refer specifically to the takeoff RPM torque.

As shown in *Figure 4*, this resultant of 1,236 lbf (shown in *Equation 6*) is applied to the bracket through the engine.

$$PTO_{Force} = \frac{Torque \cdot Tension_{max}}{r_{min}} \cdot \cos(belt\ angle) \tag{6}$$

Where Torque = Takeoff torque at 60 lbf
$Tension_{max}$ = 640 lbf
Belt Angle = 15°

When all of these loads are combined, the maximum von Mises stress is 1.632×10^8 N/m^2 with a safety factor of 1.19.

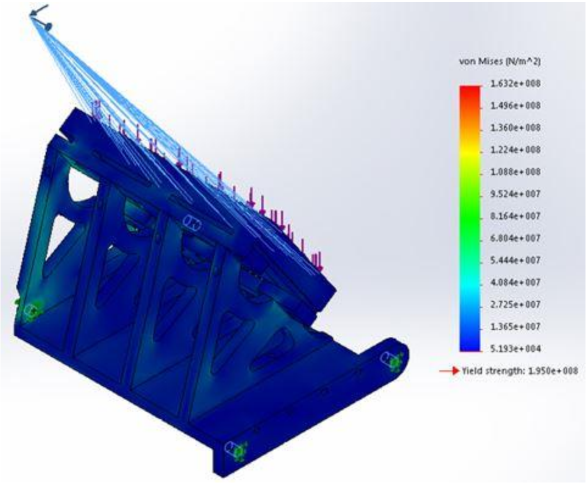


Figure 4: Von Mises stresses of the engine mounting bracket loaded with 1,236 lbf in SolidWorks.

A modal analysis in SolidWorks also confirms that the natural frequency at the first bracket assembly node is 425 Hz (*Figure 5*). With redline frequency calculated to be at 150 Hz, this mode 1 is well above the maximum engine natural frequency.

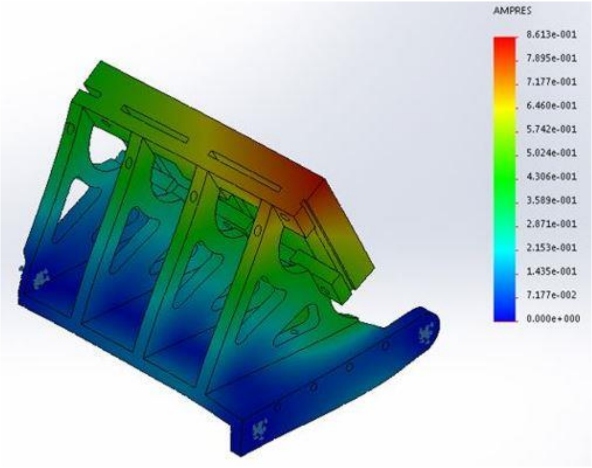


Figure 5: Modal analysis of the engine mounting bracket showing the first natural frequency.

Although the engine mounting bracket was designed to fully support the engine, additional front engine brackets were designed for further support as seen in *Figure 6*.

Figure 6: Final design of the front engine mounting bracket.

The brackets were chosen to be mounted from the upper chassis support bar and contour to two mounting locations on the engine. The upper chassis support bar will be taking some vibration load and therefore needed vibration isolators. These isolators were chosen to have a durometer rating of 40 which is the same rating as the RZR XP 1000 vibration isolators.

Using modal analysis in ANSYS, the front engine mounts were analyzed to determine how they would react to the vibrations of the engine under normal operating rpm. As mentioned preciously, the expected idle to redline frequencies expected are 50 Hz to 150 Hz.

After modal analysis seen in *Figures 7 and 8*, it was found that the frequency of the first mode is 0 Hz and will not be encountered. Mode 2 is encountered at 1318.4 Hz, which is extremely higher than what the mount will see. From these results, it can be concluded that both front engine brackets will provide additional support to the engine and effectively reduce vibrations because of their high and forward mounted locations.

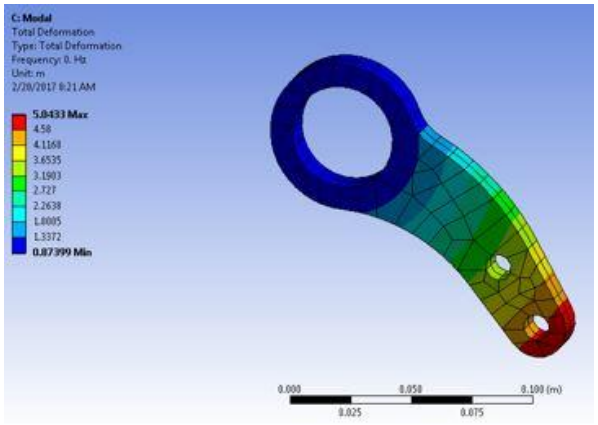


Figure 7: Mode 1 analysis of the front engine mounting bracket at 0 Hz.

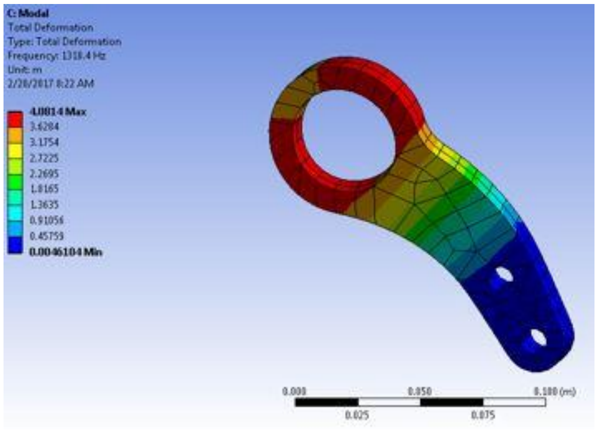


Figure 8: Mode 2 analysis of the front engine mounting bracket at 172 Hz, above the engine's natural frequencies.

Chassis Support

With the RZR XP 1000 engine being twice as tall and heavier than the Rush 800 engine, some chassis reinforcements had to be designed.

One of the main issues with the engine swap was that the oil pan of the XP 1000 sits directly in the way of one of the chassis wall reinforcement bars. This bar sat in the lower portion of the frame and was used to hold the frame together between the rear (from driver view) arm of the upper a-arms. Under certain circumstances, the a-arms will both pull and push on the walls of the chassis, and this aluminum bar prevented the frame from cracking.

The most difficult part of this reinforcement design was predicting loads that the support bar would see. Calculations show that the support bar would fail in tension by stripping the threads at 714.3 lbf. Since the original design already included a safety factor, the replacement did not need to be designed for higher loads.

To route the loads acting on the support bar and to make space for the engine, a steel rib made of ⅜" 1020 steel was designed to extend between the upper a-arm and skirt along the chassis walls and base. This rib had to be both strong in compression, yet elastic enough to survive tension loading.

Unfortunately, the ideal rib location interfered with the steering linkages, so the rib had to be set behind the ideal location just under 1". To get the rib to sit properly, custom chassis inserts had to be cast and machined to hold the plate in place. These inserts only provide support in compression which is why 5052 T32 aluminum was used.

Utilizing the 3-D scan of the chassis and a 3-D printer, prototypes were made to precisely follow the contour of the chassis. Due to the lack of time and the small size and complexity of these parts, sand casting was our quickest option since it could be done in-house. The team was able to use their final prototype as a mold for casting and slots could easily be machined for the plate to slide into. (*Figure 9*)

Figure 9: The 3-D printed parts and the sand casting used to pour the aluminum.

After compression loads were considered, the major tensile loads needed to be accounted for. Using 1020 steel, the team was able to design brackets that could be threaded into the plate and survive loads of 715 lbf before yielding (Using *Equation 2*). The brackets were plasma cut and bent to 90 degrees, and then support ribs were welded on. (See *Figure 10* for final model and *Figure 11* for FEA)

Grade 8 bolts were chosen to hold the brackets on the plate and were threaded directly into the plate. They were torqued down to 8 ft-lbs and Loctite was used to ensure they would be held in place.

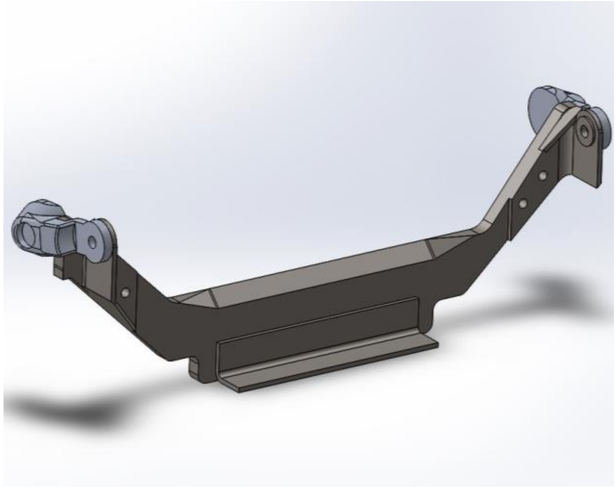


Figure 10: SolidWorks model of the chassis support bracket and the cast aluminum pieces.

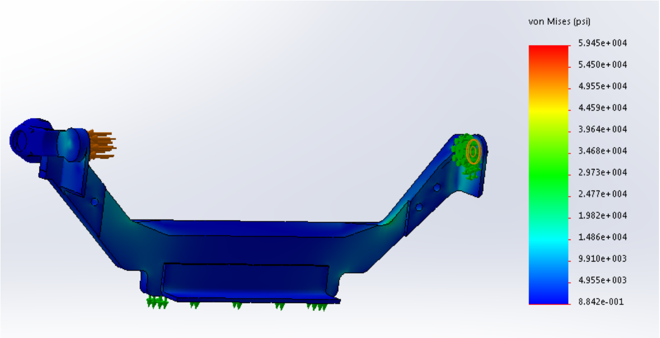


Figure 11: Chassis support bracket loaded in tension at 715 lbf. The maximum von Mises stress shown is below the yield strength of 1020 steel.

Air Box and Intake

A key issue with the engine swap was making room for the throttle body. Without extending the chassis or making any extreme modifications, the engine sat so close to the upright handlebar supports that the throttle body could not fit properly. The left (rider view) upright support had to be changed from an aluminum straight pipe to a ⅛” steel walled tube that bends around the throttle body, just missing the fuel pressure regulator.

Just bending the upright support was not enough to make room for the throttle body and an even more daunting issue came into view. The throttle body completely prevented the fuel tank from resting against the uprights. After looking into many options, the team decided to make a custom fuel tank that would avoid this contact.

The plan for the fuel tank was to create a cavity extending inwards of the tank to make room for the throttle body to extend past the uprights to allow airflow. Initially, the cavity was to be made of aluminum, inserted into the cut-out section of the tank (*Figure 12*), and then bonded and sealed to prevent leaks. However, it was clear that there would not be a way for the tank to seal properly, and safety was at high risk.

Figure 12: Fuel tank scan with initial air box, air box cover, and filter. The final design only uses the air box cavity.

With safety in mind, the team reached out to plastic manufacturers to thermal form the part. Sportech agreed to help with this issue and make a custom plastic insert of the initial aluminum cavity so that it could be plastic welded into the fuel tank and prevent any possible leaks. With the thermal formed custom part, it is the same design as the aluminum piece but was formed to fit the stock tank (*Figure 13*).

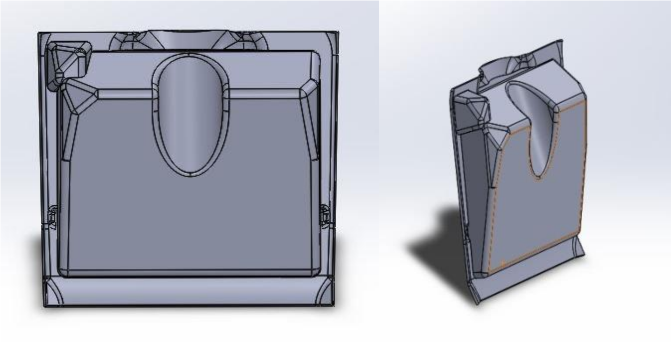


Figure 13: Final air box design that was thermal-formed by Sportech so that it would conform to the tank to prevent any leads when plastic welded together.

Making the cavity into an air box had to be scrapped to allow time for testing and to finish up the rest of the snowmobile before getting to a dynamometer. This idea would be an improvement for a following team to make on the sled.

While there is no air box and no air filter, the intake is still located in a protected space that would not have any debris inflow, so the design was concluded safe.

Battery and Radiator Mount

When the 2016 RZR XP 1000 engine is sitting in the RZR chassis, it requires a minimum of 410 cold crank amperage (CCA) from the battery to fire up. However, in cold weather conditions, a battery with 575 CCA is recommended. The RZR battery was much too large and heavy to fit into the nose of the Rush, so to stay competitive for the cold-start challenge and keep weight low, the team selected a Shorai LFX Lithium Ion battery with 540 CCA and weighing in just under 5 lbs and was half the size.

It is well-known that the RZR XP 1000 engine is prone to overheating, so the team was worried about making it through dyno testing. Even in cold weather and with both a front and rear heat exchanger, the team decided to add a radiator. The leftover space in the nose of the sled was already reduced from battery placement, so it was necessary to find a small radiator that could take on a large amount of coolant inflow.

The 2007 Yamaha Attak has a 1000 cc 4-stroke engine with both a front and rear heat exchanger that flow into its small radiator located just in front of the right foot. Since the snowmobile was comparable to the Rush/RZR engine swap, the team decided to choose the small radiator at just 8.25” long by 5.5” tall and 1.5” deep, just small enough to squeeze into the nose next to the battery.

The fan from the 2007 Yamaha Attak was also implemented, but instead of pulling warm air off of the radiator like in the Yamaha Attak, the team decided to pull cold air in from cooling slots cut into the nose of the sled and blow onto the radiator. This design is meant to keep the sled running cooler as it sits at idle or when there is not enough snow to cool down the stock heat exchangers.

With the battery being right next to the radiator, the mount holding the radiator and the battery was implemented into one piece (*Figure 14*). To ensure that the battery would not get too hot, an aluminum shielding was placed between the battery and radiator. This mount was made from ⅛” 5052 H32 aluminum which is soft enough to allow 90 degree bends with very tight radii. It was made from three separate pieces cut from a plasma table and then bent to shape. Both the base and the center upright were slotted so they slid together for easy assembly. These slots added strength and held the two pieces into position for welding.

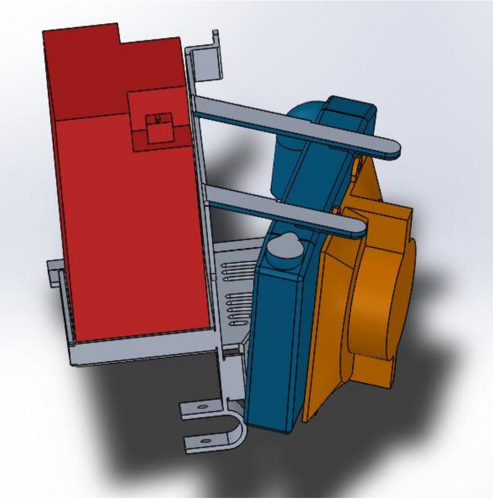


Figure 14: Nose view of battery, radiator, and radiator fan sitting in the designed bracket.

Exhaust Design

In order for the exhaust to be compatible with the 4-stroke engine and stock muffler, the team designed a custom exhaust system. The new exhaust routing also needed to fit within the chassis and allow for a catalytic converter. The exhaust header and exhaust adapter plates were machined out of ¼” 1020 steel and properly seal and to ensure no exhaust leakage.

For the engine to header connection, a steel-core graphite gasket was used. For the header to catalytic converter interface, a steel core laminate donut gasket was used. The three bolts at this connection point were also loaded with springs to reduce vibration stress on the welds throughout the exhaust. The back side of the catalytic converter was attached with standard bolts and a custom flange using the Chevy Aveo standard steel catalytic converter gasket. Where the exhaust connects to the muffler, the pipe tapers into a donut gasket which fits into the Rush 800 muffler and is attached with three springs.

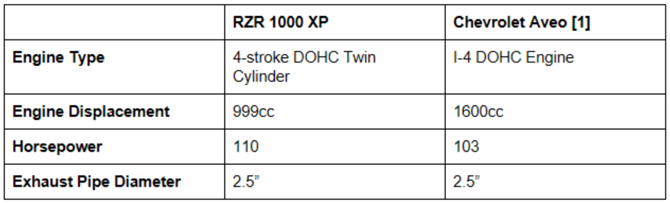
The new exhaust system was intended to minimize any additional back pressure to ensure original performance. The decrease in diameter from 2.5” to 2.25” was placed directly before the muffler to keep flow through the exhaust uniform and maintain similar back pressure as seen in the stock RZR platform.

High temperatures would become a large factor in the design. The team found that it was not easy to keep the exhaust away from all areas where heat would play a factor. A titanium cloth heat shield was chosen to decrease the temperature up to 50% and aluminum tape was applied to hoses near the exhaust.

Catalytic Converter Selection

The 2008 Chevrolet Aveo catalytic converter was chosen because it has 103 hp, similar to the RZR at 110 hp, with an engine size of 1600 cc which was one of the smaller engines on the market. It also has a pipe diameter of 2.5” which matches the RZR exhaust, which is ideal for proper backpressure (See *Table 1* for comparison). For this particular model of car, the catalytic converter was attached to the exhaust directly after the pipes converged from the engine. This was an ideal situation for the team, considering there was a shortage of space in the sled and the temperature range would be able to be reached.

Table 1: RZR engine comparison to a 2008 Chevrolet Aveo.



	RZR 1000 XP	**Chevrolet Aveo [1]**
Engine Type	4-stroke DOHC Twin Cylinder	I-4 DOHC Engine
Engine Displacement	999cc	1600cc
Horsepower	110	103
Exhaust Pipe Diameter	2.5”	2.5”

This catalytic converter features a ribbed body that minimizes expansion and distortion when the converter heats up. These ribs form a channel that protects the cushioning mat from direct exposure to exhaust gases, and they hold the ceramic catalyst in proper alignment. The converters use a monolithic honeycomb catalyst which is designed for maximum flow and surface area. The manufacturer of the catalytic converter states that it will begin reacting at 500°F. For the new RZR 1000 engine the temperature at peak RPM will be 763°F leaving an adequate range of temperature for the converter to fully operate.

Exhaust Support Design

In order to support the custom exhaust system, the team performed structural analysis and fatigue calculations on the support bracket. The bracket would not only see a downward force due to the weight of the catalytic converter, but would also add vibration in the parallel direction of the converter. A vertical bracket made of 5052 T32 aluminum (*Figure* 15) was designed to support the catalytic converter and attach to the frame for rigid support, with all mounting locations isolated. It was found to have a safety factor of 20.3 with anticipated loads varying from engine vibrations (*Figure 15*).

In addition to this rigid support for the catalytic converter, the system is suspended using springs to provide vibration suppression and flexibility.

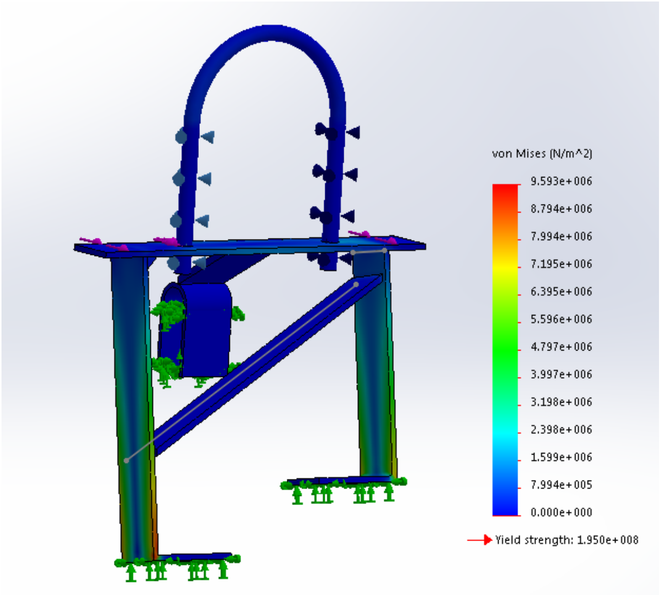


Figure 15: Loading analysis of the catalytic converter support that sees forces well below the yield strength of the material.

Steering Post Design

A steering post had to be designed because of limited space in the engine bay. The team decided to change the steering post from a single bent shaft to a series of two u-joints for compactness made of 1020 steel and with a total torque reduction of 13.1% (*Figure 16*). The original steering shaft was modeled to find the maximum torque to design for, which yielded at 150 Nm. Given that the angle change of the handlebars was only 9 degrees, the handling effects were considered minimal (See *Equations 7, 8, and 9*).

$$Torque\ loss = Torque\ in * \cos(angle)^2 \qquad (7)$$

$$Total\ torque\ loss = Torque\ in - Final\ Torque \qquad (8)$$

$$\%\ Loss = \frac{Total\ torque\ loss}{Torque\ in} * 100 \qquad (9)$$

Additional mounting points were added to keep the shaft maximum deflection under ¼". A support plate was made out of 1020 steel and bolted to the upper support frame to accommodate the necessary angle change and ensure minimum torque loss.

See *Equation 10* below for the maximum fatigue strength of the support plate.

$$Fatigue\ Strenth\ = UTS' \times C_{surf} \times C_{temperature} \times C_{reliability} \times C_{load} \times C_{size} \qquad (10)$$

Where Cs_{urf} = Surface scaling factor
$C_{temperature}$ = Temperature scaling factor
$C_{reliability}$ = Reliability scaling factor
C_{load} = Loading scaling factor
C_{size} = Size scaling factor

Mounted to the new support plate are two sleeve bearings that can withstand temperatures from the engine or outside conditions, attached with grade 5 bolts.

A secondary mounting location was installed along the chassis support plate at the nose of the sled, and used a ball joint to hold the post in place. (*Figure 16*)

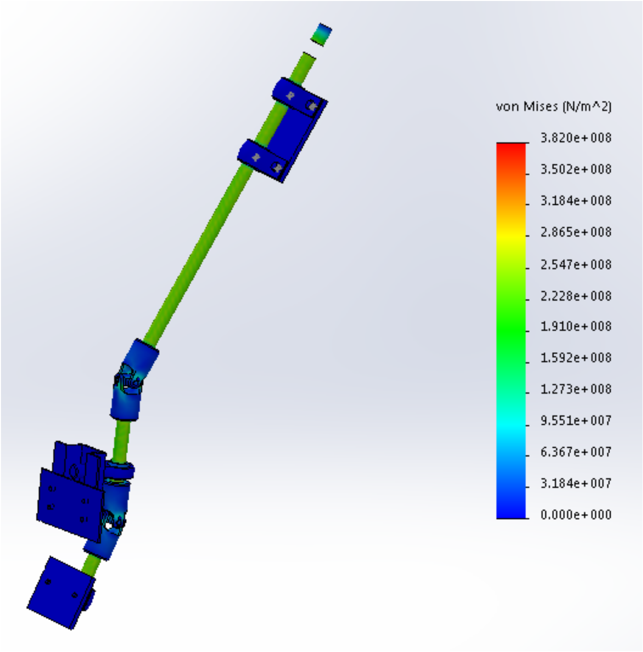


Figure 16: Loading analysis of the steering post. Both u-joints can be seen, as well as the ball joint and all three mounting plates.

Engine Tune

Along with the above components, the engine swap required a wiring harness swap as well. Most of the major components from the RZR engine required to run the engine connected easily to their matching counterparts (i.e. IAC, MAQS, TPS, PTO, MAG, instrument cluster, etc.) and the sensors required a simple connector swap (i.e. headlights, tail lights, fan, etc.) or a small calibration (i.e. speed sensor) to operate normally.

A Bullydog system was implemented to run the unit, and a flex fuel sensor was added to improve performance capabilities from any fuel blend.

Clutch Cover Design

Per requirement 8.4.7, a snowmobile with modified engine would need a clutch cover to protect the driver and bystanders from clutch failure (*Figure 17*). The clutch cover on the snowmobile is designed to withstand the operating range of the engine in order to not fail due to vibrations or cause excessive noise during sled operation. Using ANSYS, the team was able to design a clutch cover that would travel through zero modes of vibration during normal operation.

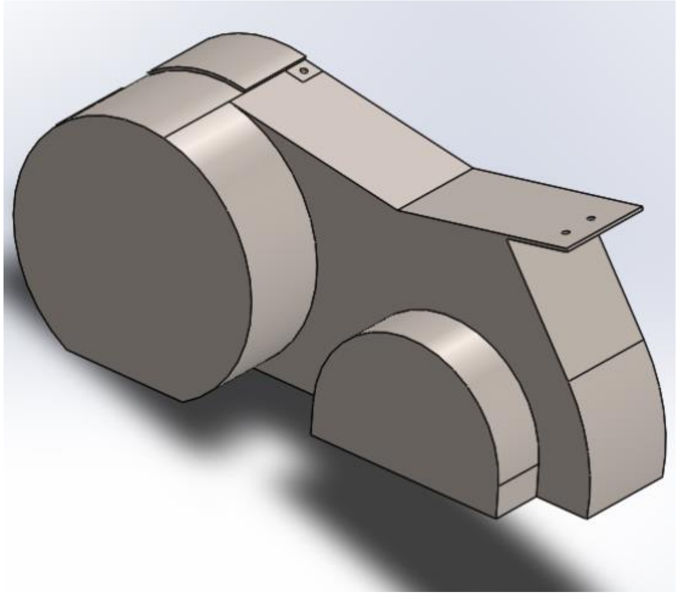
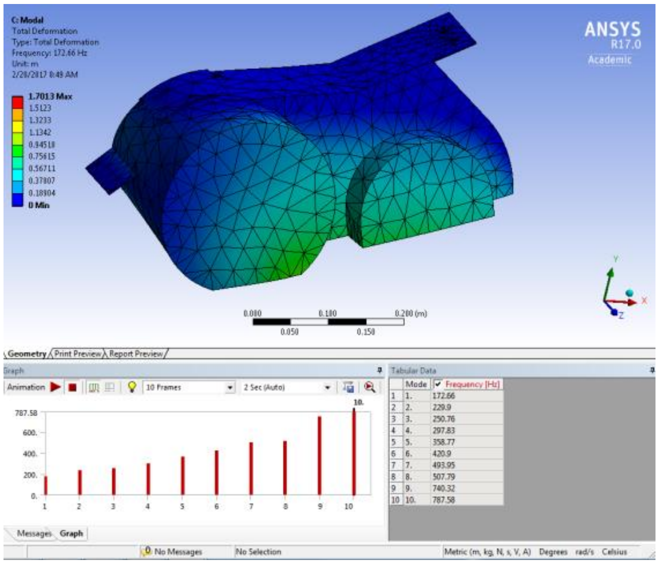


Figure 17: SolidWorks model of the clutch cover and the two main mounting locations, one of which mounts to the Rush stock location and the other mounting to one of the chassis bars.

It can be expected that the engine will range from 50 Hz at idle to 150 Hz at red line. In order to account for this, the clutch cover encounters mode 1 at 172.66 Hz and mode 2 at 229.9 Hz. Mode 1 is above the redline engine expected frequency of 150 Hz and therefore will never reach its natural modes of vibration. Reference *Figure 17* for the clutch cover design in SolidWorks and *Figure 18 and 19* for the modal analysis in ANSYS.



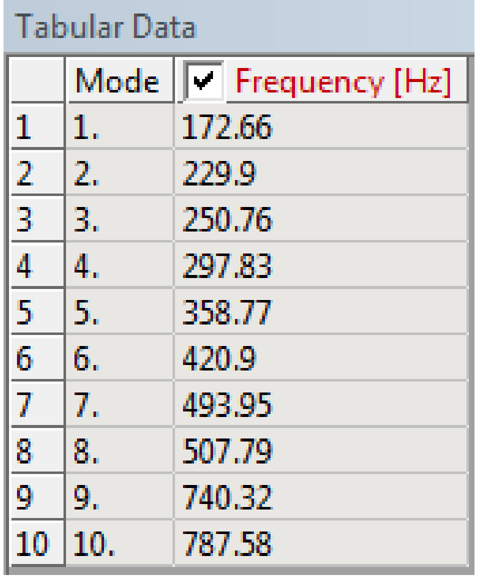
C: Modal
Total Deformation
Type: Total Deformation
Frequency: 172.66 Hz
Unit: m
2/28/2017 8:49 AM

Tabular Data

	Mode	✓ Frequency [Hz]
1	1.	172.66
2	2.	229.9
3	3.	250.76
4	4.	297.83
5	5.	358.77
6	6.	420.9
7	7.	493.95
8	8.	507.79
9	9.	740.32
10	10.	787.58

No Messages
No Selection
Metric (m, kg, N, s, V, A) Degrees rad/s Celsius

Figure 18: Modal analysis of the clutch cover. All vibrations are well above the natural frequency of the RZR engine, as can be seen in Figure 19.



Tabular Data

	Mode	✓ Frequency [Hz]
1	1.	172.66
2	2.	229.9
3	3.	250.76
4	4.	297.83
5	5.	358.77
6	6.	420.9
7	7.	493.95
8	8.	507.79
9	9.	740.32
10	10.	787.58

Figure 19: Modal analysis of the clutch cover as performed in ANSYS for 10 different nodes.

MSRP Estimate

After a full engine swap and custom mounts, the overall MSRP value of the Rush/RZR snowmobile is $13,801, just $1000 more than the stock Rush at $12,799. This price takes into account the RZR engine, tow hitch, tachometer, air box, throttle body, custom exhaust, radiator, custom mounts, battery, and all of the other smaller components.

Conclusion

The overall design of the Slick Cylinder's sled is one that inspires creativity and challenges the status quo of current snowmobile design. The team has successfully performed an engine swap and placed a 4-stroke engine from a RZR into a Rush chassis, worked around throttle body clearances, engineered a new chassis support that fits to the sled, and designed and manufactured a fully custom exhaust including a catalytic converter.

All of these components offered extreme challenges, however, with passion, good leadership, and a well understanding of engineering concepts combined, the team accomplished many tasks that had not been attempted before.

Acknowledgements

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