SAE Formula Powertrain

Initial Design Report

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DISCLAIMER

This report was prepared by students as part of a university course requirement. While considerable effort has been put into the project, it is not the work of licensed engineers and has not undergone the extensive verification that is common in the profession. The information, data, conclusions, and content of this report should not be relied on or utilized without thorough, independent testing and verification. University faculty members may have been associated with this project as advisors, sponsors, or course instructors, but as such they are not responsible for the accuracy of results or conclusions.

EXECUTIVE SUMMARY

Northern Arizona University's SAE Formula team is currently designing an open wheeled race car to compete in the SAE race in May 2026. Throughout the course of this semester, we aim to finish most of the design work and end with a full frame welded by the start of next semester.

The powertrain sub team has evaluated several different approaches to the car this year. Instead of using the Honda CBR600 4-cylinder motor, we have elected to pick a lighter 2-cylinder engine. While we are purchasing an engine, there are several other design challenges to consider. Per FSAE rules, there is a 20 mm intake restrictor diameter that we must hit, which presents a host of factors to consider with our intake design. We also must design an exhaust system, chain and final drive, cooling, and fuel delivery.

To this point, we have made progress in designing these components, but they are relatively rough estimates of what we will be doing. Without an engine to properly extract data from, we are limited by what has been put on specification sheets by manufacturers and mechanics. However, we have run simulations on components like the intake restrictor throat and chain drive, and have calculated engine modifications, exhaust, and fuel. While our designs are far from final, we have made important design decisions that should put us in the right direction.

We are purchasing a Kawasaki 650R engine, which will aid the 2026 car in weight reduction. From there, we will be able to determine important exhaust and intake parameters that would simply be guesses in the current state of the project. We will be manufacturing a bell nozzle restrictor throat, purchasing a Taylor Mk2 differential and the subsequent sprockets with a mid-range ratio. We will purchase pre-fab axles that are already cut to size. We will likely purchase high compression pistons (13.5 CR) for increased power. Going forward, there is a lot of research, calculations, and design work we must do before we can put the powertrain together.

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1 BACKGROUND

1.1 Project Description

The Society of Automotive Engineers Formula (FSAE) competition is a student-led event in which a team of mechanical and electrical engineering students design, build and race an open wheel car. The FSAE project gives students in-depth, hands-on experience with every level of automotive engineering. Per the rules of the FSAE competition, there are several design constraints that offer unique engineering challenges to be evaluated at the event.

The clients for this project include NAU's Sanghi College of Engineering, SAE, our sponsors, and our faculty advisors, David Willy and Perry Wood.

1.2 Deliverables

A short summary of Course, Customer, and Competition deliverables for the semester.

Deliverables:	Due:
Team Charter v1	4-Sep
Create Gantt Chart	30-Sep
Presentation 1	18-Sep
Peer Eval #1	19-Sep
Make Fundraising Flyer	19-Sep
Make Go Fund Me Post	22-Sep
Start Fundraising	Ongoing
Realis Picture, Email, and LinkedIn Post	26-Sep
Presentation #2 Calculations	7-Oct
Presentation #2	9-Oct
Competition Registration	24-Oct
Website Check #1	24-Oct
Buy Engine	31-Oct
Presentation #3	6-Nov
Prototype Demo #1	13-Nov
Peer Eval #3	14-Nov
Report #2	26-Nov
Prototype Demo #2	4-Dec
Final CAD and Final BOM	5-Dec
Project Management for 486C	6-Dec
Website Check #2	6-Dec
Peer Eval #4	7-Dec
Comp. Tech 26 Notice Form	8-Dec

1.3 Success Metrics

NAU FSAE's success in 2026 will be evaluated by, above all else, our ability to compete. If NAU is selected in the competition lottery, the main goal will be to pass the technical inspection. In the competition, important metrics will be acceleration, top-end speed, handling, and durability. If we can be within 10% of the best time for each race, we would consider ourselves successful.

A longer-term goal beyond the scope of this year's project will be to further the capabilities of NAU's FSAE for future iterations. This means to outfit the space in which we work to increase self-reliance and manufacturability right here in the NAU Formula shop.

2 REQUIREMENTS

To ensure success and contentment among customers, the project must be broken down into goals tailored towards the customer's needs. These goals are then defined by specific and quantifiable engineering requirements chosen to give structure to the design process, making it more efficient and giving the design the best chance at being exceptional.

2.1 Customer Requirements (CRs)

The 2026 FSAE took the advice from 2025 team members and advisors to establish a list of customer requirements. These were mainly based on the pitfalls of the previous team, which could not compete at the Michigan International speedway due to a few design oversights.

- Pass Tech Inspection
- Reliable/Durable
- Safe
- Limited slip rear power delivery
- Ease of maintenance
- Drivability
- Standardized parts
- More accessible fuel fill
- Lightweight

2.2 Engineering Requirements (ERs)

These same requirements can be into quantifiable targets, shown below.

- Idle exhaust noise under 98 dbC. (103 is passing)
- Reduce weight of muffler (kg)
- Remove mass from intake manifold (kg)
- Minimize mechanical loss testing (hp)
- Decrease Distance of fuel fill to outermost frame rail (mm)
- Robust throttle cable with 50mm of adjustment (mm)

2.2 House of Quality (HoQ)

Project:	QFD
Date:	09/14/25

System QFD: Powertrain

#	Technical Requirement	1	2	3	4	5	6
1	Engine noise less than 103 Dbc						
2	Reduce weight of muffler	+					
3	Remove mass from intake manifold						
4	Minimize mechanical loss testing	+		++			
5	Obtain LSD with smallest impact on budget						
6	Decrease distance from fuel fill to outermost frame rail						
7	Robust throttle cable with 50mm of adjustment				+		

Legend
- A 2001 CBR600 F4I
- B 2012 Kawi Ninja 650R
- C 2014 Yamaha YZ450f

#	Customer Needs	Customer Weights (1-5)	Engine noise less than 103 Dbc	Reduce weight of muffler	Remove mass from intake manifold	Minimize mechanical loss testing	Obtain LSD with smallest impact on budget	Decrease distance from fuel fill to outermost fra	Robust throttle cable with 50mm of adjustment	1 Poor	2	3 Acceptable	4	5 Excellent
1	Pass Tech Inspection	5	9	1	1	9	1	1	9					ABC
2	Reliable/Durable	4	1	1	3	9	3	1	9				B	AC
3	Safe	5	3	1	1	9	3	3	9			AB		C
5	Limited slip rear power delivery	3	1	1	1	9	9	1	1					ABC
6	Ease of maintenance	3	1	3	3	1	1	3	3			A	B	C
7	Drivability	3	1	3	3	3	9	1	9				AB	C
8	Standardized parts	4	3	1	1	1	3	1	3					ABC
10	Make filling up fuel more accessible	5	1	1	3	1	1	9	1			A		BC
11	Lightweight car	3	1	1	9	1	1	1	1		ABC			
	Technical Requirement Units		dBC	kg	kg	hp	$	mm closer	mm					
	Technical Requirement Targets		5	4	2	8	###	100	50					
	Absolute Technical Importance		93	47	89	177	109	91	185					
	Relative Technical Importance		11.8%	6%	11.3%	22%	14%	12%	23%					

Figure 2-1 - QFD

Based on the most important customer needs for the car, we applied engineering requirements that will ensure that each of the customers' requirements are satisfied. Each of the engineering requirements were then ranked with respect to the customer requirements to give a percentage of importance to each of the engineering requirements. The top two engineering requirements came out to robust throttle cable with 50mm of adjustment and minimized mechanical loss testing. This helps us emphasize our efforts on the more important aspects of design, which will help ensure that the car is finished on time, and finished with the adequacy of the customer.

3 Research Within Your Design Space

3.1 Benchmarking

Engine	2001 Honda CBR600 F4i [1]	2012 Kawi Ninja 650R [2]	2014 Yamaha YZ450f [3]
Stock Hp	*110 Hp @ 12,500 rpm*	*72.1 Hp @ 8,500 rpm*	*58 Hp @ 9,900 rpm*
Weight (lbs)	*Heaviest*	*Medium*	*Lightest*
Displacement (cc)	*599*	*649*	*449*
Peak Torque (lb-ft) (RPM)	*48 lb-ft @ 10,500 rpm*	*48.6 lb-ft @ 7,000 rpm*	*35 lb-ft @ 7,300 rpm*

Table 3-1 - Engine Specifications

Differential	Taylor Race Engineering V2 [4]	Drexler V2/V3 [5]	Modified ATV [6]
Cost	$2,450	$3,656	~$500
Weight (lbs)	8.9	5.9	11-21
Further Drivetrain Support (Y/N)	Y	Y	N

Table 3-2 - Differential Specifications

3.2 Literature Review

Aidan Willson

[1] SAE International, Formula SAE Rules 2025-v1, IC.1: “General Requirements”, VE.3: “Driver Equipment”,31 Aug, 2025

-General engine requirements for designing powertrain, max displacement, driver requirements for safety equipment.

[2] T. D. Gillespie, S. Taheri, C. Sandu, and B. L. Duprey, *Fundamentals of Vehicle Dynamics*. Warrendale, PA: SAE International, 2021.

Book used for calculating: Chapter 2: Theoretical Mechanical Top Speed based on gear ratios.

[3] "2015 Kawasaki Ninja 650 6," *Scribd*, 2015. https://www.scribd.com/document/776296988/2015-Kawasaki-Ninja-650-6 (accessed Sep. 14, 2025).

-Kawasaki Ninja 650 Service manual for obtaining gear ratios used in calculations. Specific ratios for each gear in the transmission and final drive ratio.

[4] K. Lutenbacher, B. Mayeaux, and J. Waller, "FSAE Engine Selection: Four or One Cylinder." Available: https://korilutenbacher.weebly.com/uploads/5/4/9/9/5499743/klutenbacher_bmayeaux_jwaller_me4633_fsae_report.pdf

-LSU Formula SAE engine selection from 2012, used as a reference for some basic calculations and their overall selection process for their drivetrain.

[5] MotorTrend Channel, "BIG Power from Small Block Engines! | Engine Masters | MotorTrend," *YouTube*, Jul. 15, 2023. https://www.youtube.com/watch?v=WAwxPMUe9JE (accessed Sep. 10, 2025)

-YouTube Video about generating more power from smaller blocks such as a 4-cylinder engine. Useful for future modifications to our engine and ideal components to swap out.

[6]"Formula SAE 2025 Overall Results." Accessed: Sep. 10, 2025. [Online]. Available: https://www.fsaeonline.com/CompResources/2025/c796a916-c2e8-4ce6-bae3-c79f36776556/FSAE_2025_MI5_results.pdf

-2025 formula results to benchmark (Not much out there), found some research from 2nd place Texas A&M but from 2005.

[7] B. Singh, "- Race Car Vehicle Dynamics Milliken Milliken," *Scribd*, 2025. https://www.scribd.com/document/858209436/Race-Car-Vehicle-Dynamics-Milliken-Milliken

-Powertrain dynamics, more information on differentials and the various kinds we might select from for our design.

Liam O'Connor

[8] J. B. Heywood, "Chapter 14: Modeling Real Engine Flow and Combustion Processes," in *Internal Combustion Engine Fundamentals*, 2nd ed, Mcgraw Hill Education, 2018

-Background on methods for modeling engine flow

[9] SAE International, Formula SAE Rules 2026-Draft, IC.2: "Air Intake System" , 11 Aug, 2025

- Formula rules governing restrictor diameter

[10] F. Leach, “The scope for improving the efficiency and environmental impact of internal combustion engines - sciencedirect,” ScienceDirect, https://www.sciencedirect.com/science/article/pii/S2666691X20300063 (accessed Sep. 18, 2025).

- Engine Thermal Efficiency

[11] “Mass flow choking,” NASA, https://www.grc.nasa.gov/www/k-12/airplane/mflchk.html (accessed Sep. 18, 2025).

- Max choked flow rate formula and calculation process

[12] R. Stone, “Chapter 6: Induction and Exhaust Processes,” in *Introduction to Internal Combustion Engines*, 2nd ed, SAE Inc., 1993, pp. 231–274

- Internal Combustion Engine Compressible Flow basics

[13] D. Willy, “Formula SAE Restrictor Power Calculations.” Northern Arizona University, Flagstaff, 2024

- Fundamentals for the max restrictor flow calculation

[14] R. Nakka, “Solid Rocket Motor Theory -- Nozzle Theory,” Richard Nakka’s Experimental Rocketry Site, https://www.nakka-rocketry.net/th_nozz.html (accessed Sep. 18, 2025).

- Theory for modeling ideal intake restrictor geometry based on a rocket nozzle.

Trent Greene

[15]“Figure 4: Drag and Lift coefficients of a FSAE car with different...,” *ResearchGate*, 2024. https://www.researchgate.net/figure/Drag-and-Lift-coefficients-of-a-FSAE-car-with-different-aerodynamic-packages_fig13_320556659

Research from University of Southampton about drag and lift approximations for non-specific FSAE geometries.

[16]Caleb's Engineering Projects, “Calculate Top Speed and 0-60 of an Electric Racecar (FSAE),” *YouTube*, Sep. 08, 2020. https://www.youtube.com/watch?v=pkwBeQO-0A8 (accessed Sep. 18, 2025).

Former FSAE Engineer series on the basics of design and calculations.

[17]Martin, “Engineering SPROCKET ENGINEERING DATA.” Available: https://www.martinsprocket.com/docs/catalogs/engineering/engineering%20catalog/sprocket-engineering-data.pdf

Manufacturer catalog with specifications and standards for varying sprocket designs.

[18]Tsubaki, “The Complete Guide to Chain.” Available: https://www.ustsubaki.com/wp-content/uploads/the-complete-guide-1.pdf

Manufacturer data on chain design, including force and standards specifications.

[19]E. Neumann, “Power Limited,” *hpwizard.com*. https://hpwizard.com/accelerating-power-limit.html

Basic power limit theory equations and explanation pertaining to tire traction and aerodynamic performance.

[20]FSAE, “FSAE Rulebook 2026 T5,” *Fsaeonline.com*, 2025. https://www.fsaeonline.com/cdsweb/gen/DownloadDocument.aspx?DocumentID=7ac1fb19-75d1-42d7-983d-ba9cd87fd092
FSAE rule specifications about sprocket, chain, and splash guards.

[21]E. Oberg, F. Jones, H. Horton, H. Ryffel, C. Mccauley, and N. York, “Machinery’s Handbook 29 th Edition INDUSTRIAL PRESS,” 2012. Available: https://dl.icdst.org/pdfs/files4/80364b03673ba30eb5ccf1e27e119ffc.pdf
Machinery handbook, used for chain and sprocket information.

[22] A. Schirn, “ASME B29.28-2015 (R2020): High-Strength Roller Chains - ANSI Blog,” *The ANSI Blog*, Sep. 08, 2023. https://blog.ansi.org/ansi/asme-b29-28-2015-r2020-high-strength-roller-chains (accessed Sep. 19, 2025).

ASME and ANSI chain standards.

[23] “All EPA Emission Standards | US EPA,” *US EPA*, Mar. 04, 2025. https://www.epa.gov/emission-standards-reference-guide/all-epa-emission-standards

[24] “J300_202405: Engine Oil Viscosity Classification - SAE International,” *Sae.org*, 2024. https://www.sae.org/standards/content/j300_202405/

[25]]Protolabs, “UL 94 Classification and Flame-Retardant Materials,” *Protolabs.com*, Apr. 14, 2017. https://www.protolabs.com/resources/blog/flame-retardant-thermoplastics-and-ul-classifications (accessed Sep. 19, 2025).

[26] ASTM International, ASTM D2700-22: Standard Test Method for Motor Octane Number of Spark-Ignition Engine Fuel, West Conshohocken, PA, USA: ASTM International, 2022

Marshall Fritz

[27] G. Blair, *Design and Simulation of Four-Stroke Engines*, SAE International, 1999.

Foundational textbook information on 4-stroke engine design. Provides a foundation in thermodynamics, combustion, and performance parameters to know the complete engine modeling for the NAU FSAE.

[28] A. Graham Bell, *Four-Stroke Performance Tuning*, 4th Edition, 2022.

Practical guide to tuning four stroke engines, intake and exhaust optimization strategies relevant to improving volumetric efficiency and torque in small racing engines.

[29] Y. Otobe, O. Goto, H. Miyano, M. Kawamoto, A. Aoki, and T. Ogawa, “Honda Formula One Turbocharged V6 1.5L Engine,” SAE Technical Papers, Feb. 1989, doi: 10.4271/890877.

Technical document concerning a turbocharged Formula One engine. Provides analysis of high-performance exhaust and intake design principles relevant to compact FSAE engines.

[30] V. Sharma, S. Hittalamane, P. Gunjal, and G. Edison, “Exhaust Header Designing for Formula SAE Car,” 2017.

Approaches for designing FSAE exhaust headers. Includes computations for runner diameter, length, and wave tuning. Applies directly to the NAU FSAE exhaust runner design.

[31] D. Kennedy and G. Woods, “Development of a New Air Intake and Exhaust System for a Single Seat Race Car,” ITRN, 2011.

Describes the development of intake and exhaust systems for a small race car, focusing on volumetric efficiency and eliminating back pressure. Provides information to guide design development of runner sizing.

[32] S. Mangukiya and R. Shah, “Enhancing the Performance of Single Cylinder Motorcycle Engine for Formula Student Vehicle by Optimizing Intake and Exhaust System,” JETIR, Vol. 5, Issue XX.

Discusses the optimization of intake and exhaust for a single-cylinder motorcycle engine and provides engineering techniques to size and balance airflow relative to the NAU FSAE exhaust design.

[33] A. Sayyed, “Air Flow Optimization through an Intake System for a Single Cylinder Formula Student FSAE Race Car,” IJERT, Vol. 6, Issue 01.

Focuses on airflow optimization relating to intake aspect of exhaust design process. Provides information to maximize engine performance and efficiency.

[34] P. H. Smith and J. C. Morrison, *Scientific Design of Exhaust and Intake Systems*, 3rd ed., R. Bentley, 1971.

Important work created on exhaust and intake system design. Provides a detailed methodology to calculate runner lengths, diameters, and tuning with pressure wave action. Useful for both diameter and length calculations in NAU FSAE exhaust and intake systems.

[35] B. J. McBride, M. J. Zehe, S. Gordon, and Glenn Research Center, Cleveland, Ohio, "NASA Glenn Coefficients for Calculating Thermodynamic Properties of Individual Species," Sep. 2002. [Online]. Available: https://ntrs.nasa.gov/api/citations/20020085330/downloads/20020085330.pdf

Reference for NASA polynomial coefficients with respect to determining specific heat, gas constant, and specific heat ratio of the exhaust gases. Critical to coefficient accuracy associated with the exhaust system when determining runner length and speed-of-sound based design specifications.

Jackson Nichols

[36]"Engine - fswiki.us," Fswiki.us, 2022. https://fswiki.us/Engine#Motorcycle_engines (accessed Sep. 18, 2025).

Used to increase familiarity in the world of FSAE. Two cylinders are smoother than one, and more simple than four

[37]M. Streeter, "The Suzuki RE-5 Was Supposed To Be The Future Of Motorcycles, Now It's An Example Of A Past Failure - The Autopian," The Autopian, May 04, 2023. https://www.theautopian.com/the-suzuki-re-5-was-supposed-to-be-the-future-of-motorcycles-now-its-an-example- of-a-past-failure/ (accessed Sep. 16, 2025).

Used when considering using rotary engine

[38]World Nuclear Association, "Heat Values of Various Fuels - World Nuclear Association," world-nuclear.org, 2020.https://world-nuclear.org/information-library/facts-and-figures/heat-values-of-various-fuels

Used in calculations of max hp for choked flow

[39]M. J. Moran, H. N. Shapiro, D. D. Boettner, and M. B. Bailey, Fundamentals of engineering thermodynamics, 8th ed. Hoboken, Nj: Wiley, 2014.

Thermodynamics text. Used for review and ultimately excluded from calculations.

[40] SAE International, Formula SAE Rules 2026-Draft, T.5: "Powertrain" , 29 Aug, 2025

Used for getting informed about restrictions in FSAE.

[41] F. M. White and H. Xue, Fluid Mechanics, 9th ed. New York, NY, USA: McGraw-Hill, 2021.

Used for review of compressible nozzle flow.

[42] R. G. Budynas and J. K. Nisbett, Shigley's Mechanical Engineering Design, 11th ed. New York, NY, USA: McGraw-Hill, 2019. ISBN 978-0-07-339821-1.

Used while discussing gear forces in the drivetrain.

3.3 Mathematical Modeling

The mathematical modeling for the FSAE powertrain was limited by our lack of currently available funds. We identified "independent" subsystems that could be modeled with basic inputs.

3.3.1 Theoretical Top Speed Calculations – Aidan Willson and Trent Greene

To set up further and more advanced calculations. We can take the gear ratios and peak engine performance characteristics of a specific engine such as the Yamaha YZ450F or Kawasaki Ninja 650R. The motorcycle owner's manual [10] and online engine specification tables were used to find specific gear ratios, peak torque and peak power values, as well as corresponding rpm for each peak value. With these specs obtained, I used these equations below from the Fundamentals of Vehicle Dynamics [9].

$$\omega_e = N_t * N_f * \omega_w \qquad \frac{\omega_e}{(N_t * N_f)} = \omega_w$$

$$V_x = \omega_w * r$$

Equation 3-1 - Speed Equations

Using equation (6) and manipulating to obtain the wheel speed based on engine speed (We), dividing by current gear ratio (Nt) multiplied by final drive ratio (Nf). Further using this value and a few conversions along the way to find the Velocity (8) in miles per hour for a specific engine and its gear/torque/power characteristics. These equations were all implemented into MATLAB for ease of modification if we need to compare engines. Additionally, you can edit the tire size in the code to test different selections for diameter. Below are the results of the program based on the inputs of rpm, front and rear sprocket teeth. These results can be further used to create a heatmap of various sprocket variations that we are considering.

```
Enter the max rpm value: 8500
Enter the number of teeth on the front sprocket: 17
Enter the number of teeth on the rear sprocket: 36
Top Speed in 1st gear: 78.37 mph
Top Speed in 2nd gear: 111.47 mph
Top Speed in 3rd gear: 143.33 mph
Top Speed in 4th gear: 171.97 mph
Top Speed in 5th gear: 197.79 mph
Top Speed in 6th gear: 224.25 mph
```

Figure 3-2

The two heatmaps display top speed based on peak torque rpm and peak power rpm. From these, we can obtain ideal ratios and peak top speed based on these ratios.

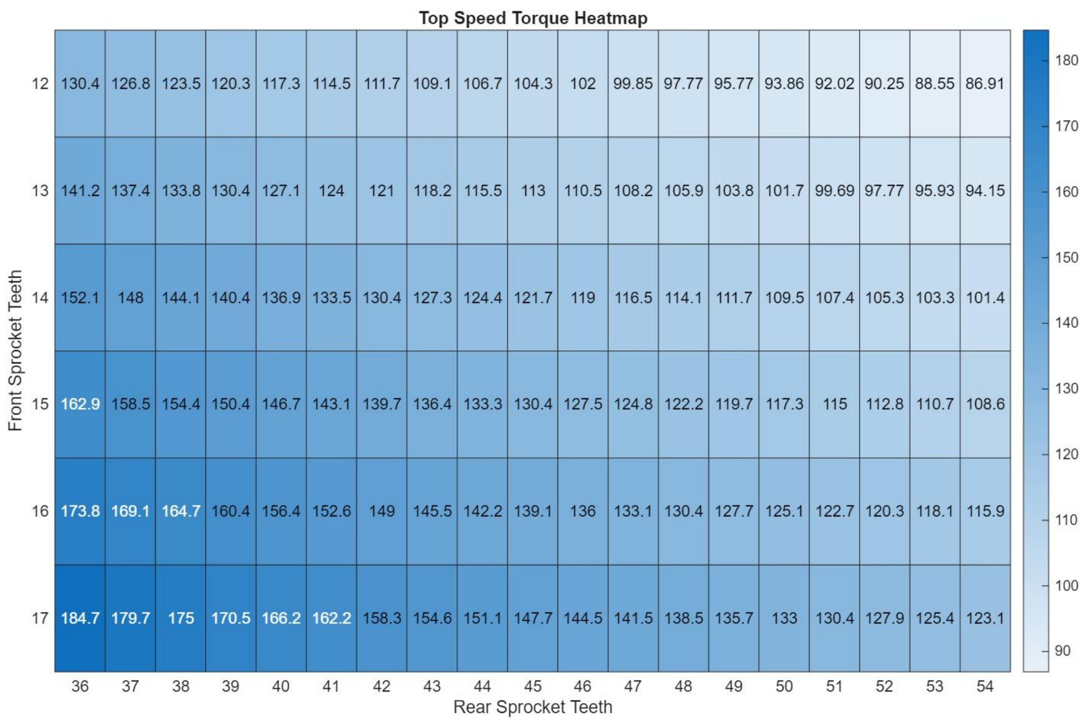
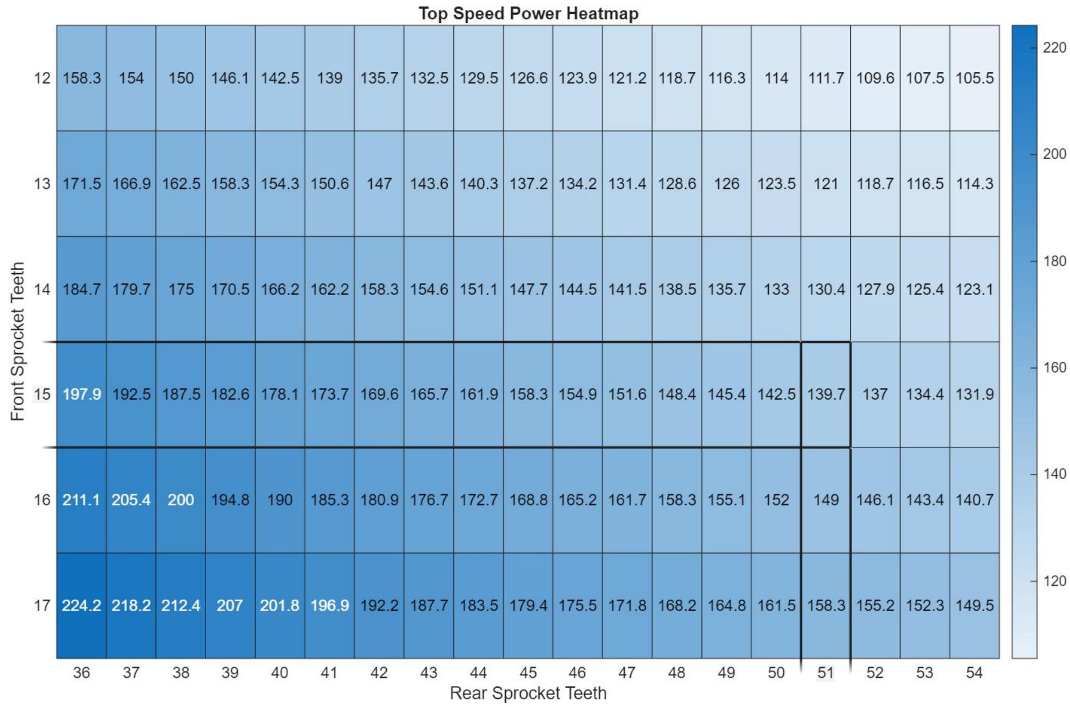


Figure 3-3-1: Theoretical Top Mechanical Speeds



Based on the heatmap data, we can see our highest top speed of 224.2 mph comes from a 17, 36 sprocket ratio and can plan accordingly to incorporate the results.

As a starting point for engine selection calculations, theoretical top speeds depending on engine power and gear ratios could be used to get estimates on basic expected performance and rule out certain gear ratios. The equation below was used, sourced from Erik Neumann of hpwizard.com. [33]

From equation (1) and **equation (1b) from the accelerating page**, we get: (more)

$$v_{max} = \sqrt[3]{\frac{q}{2} + \sqrt{\frac{q^2}{4} + \frac{p^3}{27}}} + \sqrt[3]{\frac{q}{2} - \sqrt{\frac{q^2}{4} + \frac{p^3}{27}}} \qquad (9)$$

Where:

$$q = \frac{P_t}{0.5\rho\left(C_D A - f_r C_L A\right)} \qquad (9.1)$$

$$p = \frac{f_r m g}{0.5\rho\left(C_D A - f_r C_L A\right)} \qquad (9.2)$$

Equation 3-2- Top Speed with Resistance

As the aero package has not yet been manufactured for the new car, rough estimates for drag and lift coefficients were made using the University of Southampton's guide based on non-specific aerodynamic package types [29]. This yielded a coefficient of drag estimate of 0.68 and a coefficient of lift of -2.68. Standard atmospheric air pressure was used, and a rolling resistance coefficient of 0.02 was assumed, as we are planning to use slick tires. For engine power inputs, the resulting graph below used a Kawaski 650R, which after a mechanical loss overestimate, makes around 64 horsepower. However, this could be changed to input any engine we would want to test for.

To parametrize this equation to account for final drive, we factored in an estimate on mechanical loss and varying gear ratios from Taylor Racing Engineering's sprocket catalog for FSAE vehicles. This includes a sprocket A count of twelve to seventeen, and a sprocket B count of thirty-six to fifty-four. The highest possible speed with resistance was divided by the highest possible mechanical speed to create a resistive coefficient (about 36.17%) that could be applied to all the mechanical speeds to estimate top speeds at each gear ratio. The resulting table is shown below.

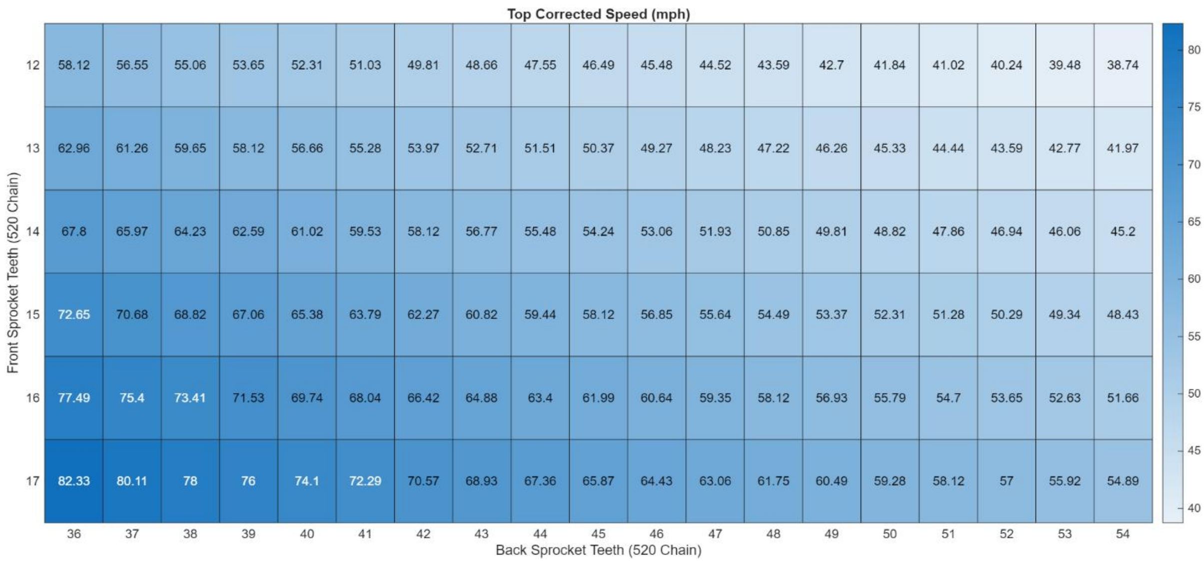


Figure 3-2: Top Corrected Speed in mph, with resistance

From this estimation, we see that in the lowest gearing, 82.3 miles per hour is the maximum possible speed. We can also rule out the highest gear ratios, where top speeds do not even hit 45 miles per hour.

3.3.2 Intake Restrictor Max Flow Rate – Liam O'Connor

Objective

This analysis is meant to calculate the maximum possible flow rate of air through the intake air restrictor for FSAE engines. This allows you to get a baseline of how much horsepower an engine can make when using the restrictor, before the engine's power is impacted by the restrictor.

Assumptions

- Atmospheric Pressure at inlet
- Isentropic-Compressible flow

- Ideal gas
- Velocity of Mach 1 at throat
- 20 mm diameter at throat (FSAE rules [5])

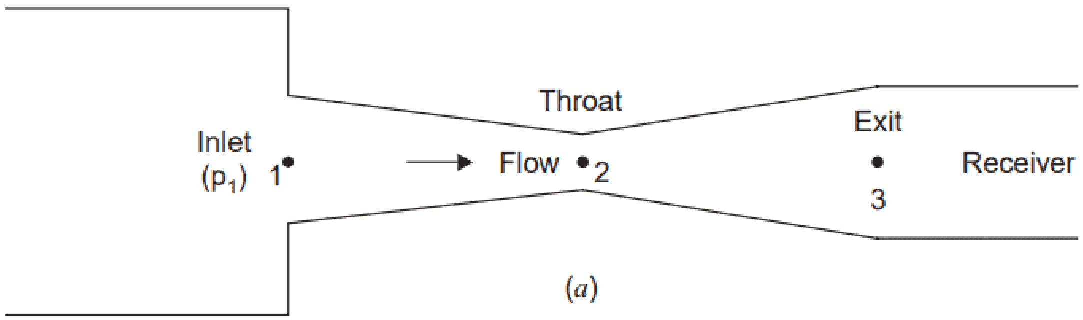


Figure 3-3 - Air Restriction Diagram

Based on the assumptions made above you can use equation 3-1 to solve for the maximum possible flow rate of air. This is based on the values used below.

- $A^* = 314.15\ mm^2\ (area\ at\ throat)$
- $T_0 = 300K\ (temp\ at\ inlet)$
- $\gamma = 1.4\ (specific\ heat\ ratio)$
- $R = 287\ \frac{J}{kgK} (gas\ constant)$
- $p_0 = 101{,}325\ Pa\ (atmospheric\ pressure)$

$$\dot{m} = \frac{p_o A^*}{\sqrt{T_o}} \sqrt{\frac{\gamma}{R}\left[\frac{2}{\gamma+1}\right]^{\left(\frac{\gamma+1}{\gamma-1}\right)}}$$

Equation 3-4 - Choked Flow Rate Formula

When plugging these values into equation 3-1 you get a maximum flow rate of 74.3 grams per second of air through the restrictor. This value is mostly useful for a baseline on a flow rate, which may not actually be achievable in real life, as this equation is only a function of the diameter, which means that the boundaries of the restrictor do not impact the final flow rate at all, which is not realistic for a real-life scenario. This number can be further modified by adding a coefficient of discharge, where a number is chosen by typical values associated with nozzle type. The goal for this is to design a restrictor that has the highest discharge coefficient, minimizing the boundary effects on the flow of the restrictor.

3.3.3 Intake Restrictor Geometry – Liam O'Connor

Objective

Develop a restrictor geometry that optimizes the air flow to the maximum possible given the FSAE diameter limit. This will subsequently minimize the power loss on the engine that is directly a result of the intake restrictor.

Assumptions

- Isentropic flow
- k=1.4
- Velocity of Mach 1 at throat
- Rocket nozzle geometry will perform optimally

By using formulas for rocket nozzle geometry, we know we'll be able to achieve the highest possible flow rate, as we know that rocket nozzles amplify velocity to the point of creating thrust, which should also work the best when used as an intake geometry.

$$\frac{A}{A^*} = \frac{1}{M}\left(\frac{1+\frac{k-1}{2}M^2}{1+\frac{k-1}{2}}\right)^{\frac{k+1}{2(k-1)}}$$

Equation 3-5 - Area Ratio for Rocket Nozzle

The above equation 3-2 calculates the area ratio with respect to Mach number and the specific heat ratio. This ratio is the cross-sectional area at any point x over the cross-sectional area at the throat of the nozzle. Because the throat diameter is known by FSAE rules (20mm), and the Mach number is assumed to be 1 at that point, we can use that to make a plot of Mach number with respect to radius of the nozzle. This gives you a plot that tells the radius of the nozzle based on the Mach number of the flow, which is important for figuring out exit sizes for the nozzle that will work the best.

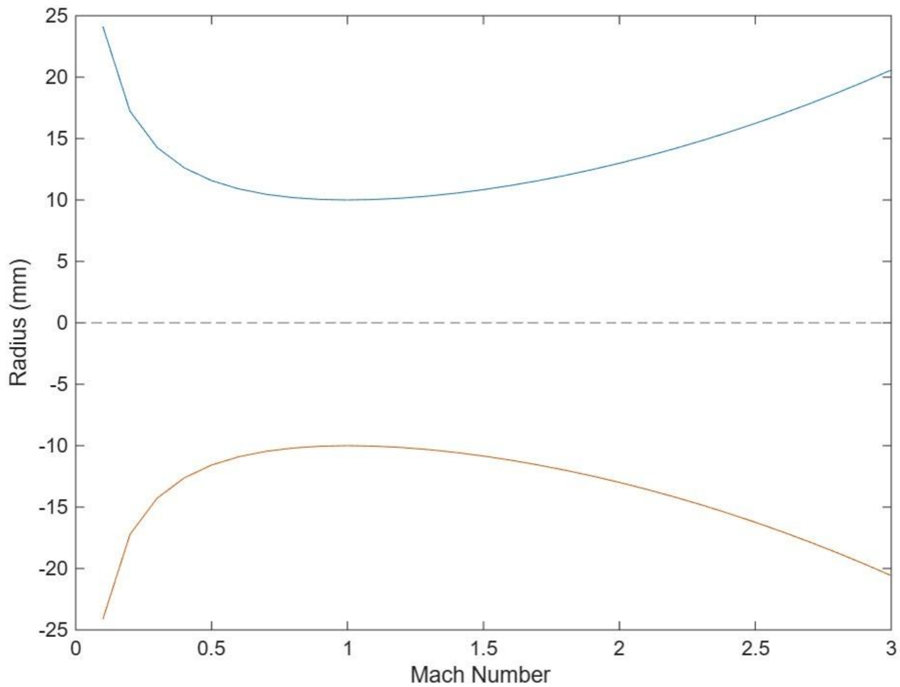


Figure 3-4 - Radius Vs. Mach Number

This plot shows the specific radius with exit Mach Number. With this you can figure out a suitable exit diameter for the restrictor and use that Mach number with the method of characteristics to get a suitable geometry.

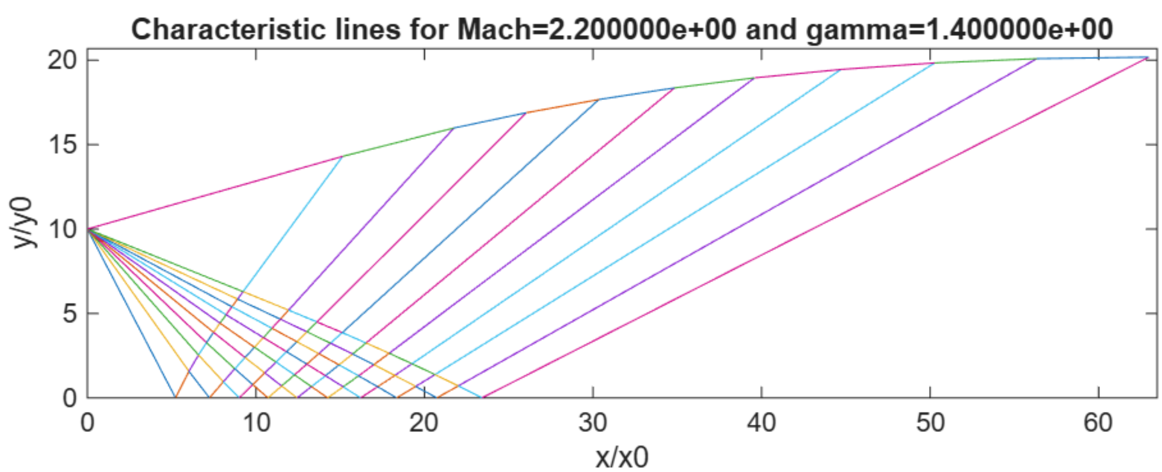


Figure 3-5 - Method of Characteristics Divergent Section

Using existing MATLAB code made by Shubham Maurya, a method of characteristics plot was created. This code takes user inputs of radius, exit Mach number, specific heat ratio, and number of characteristic lines to create this graph. With this you now have a full usable geometry that you can use to make CAD models for the diverging section of a bell nozzle. Theoretically this type of diverging section should be the most optimal for use as an intake restrictor.

3.3.4 Intake Restrictor CFD Testing – Trent Greene

Objective

This analysis sought to find the best intake throat shape using the above method of characteristics models (bell nozzles) and various de-Laval throat nozzles. Between the two styles, angles and lengths of the divergent section were changed to maintain the same overall shape. The goal was to achieve the lowest pressure drop possible and limit turbulence to achieve the best conditions for the plenum, as we are attempting to approach a steady-state, effectively infinite air supply.

Model Description

We used the described de-Laval and Bell nozzles in an ANSYS Discovery Simulations. Typically, de-Laval nozzles are the simplest to manufacture, but Bell nozzles tend to be more efficient. We used air gas as the volume and 6061-T6 aluminum for the structure. The inlet velocity was set at 25 m/s with no induced pressure (0 Pa) at the exit. This was a rough approximation of being at cruising speed in the vehicle. The main assessment criteria was finding which had the lowest pressure drop, not actual pressure values, at the 25 m/s inlet is an ideal case. All convergent geometry was the same across all models. We can, however, normalize the conditions across all models to assess which is the best performing. The model characteristics for each are shown below, as are the tested pressure drops from the simulations.

Table 3-3

Name	Length of Divergent section (mm)	Type	Max Pressure Drop (Pa)
Strt_throat_1	104	De-Laval	466
Strt_throat_2	77	De-Laval	569
Strt_throat_3	89	De-Laval	532
Strt_throat_4	153	De-Laval	444
Strt_throat_5	208	De-Laval	471
MOC_throat_1	63	Bell	786
MOC_throat_2	153	Bell	412

We first modeled one Bell type option and five options for de-Laval type restrictor. After running these simulations, we found that the 63mm was efficient for its short length, but in general, the longer de-Laval lengths did better. However, the 208 mm throat was too long, as pressure drop began to increase again after the 153 mm throat. Using this information, we created a second Bell nozzle using method of characteristics that was limited to a 153 mm length. This was the best performing nozzle. Creating more Bell nozzles to test would simply be too time consuming.

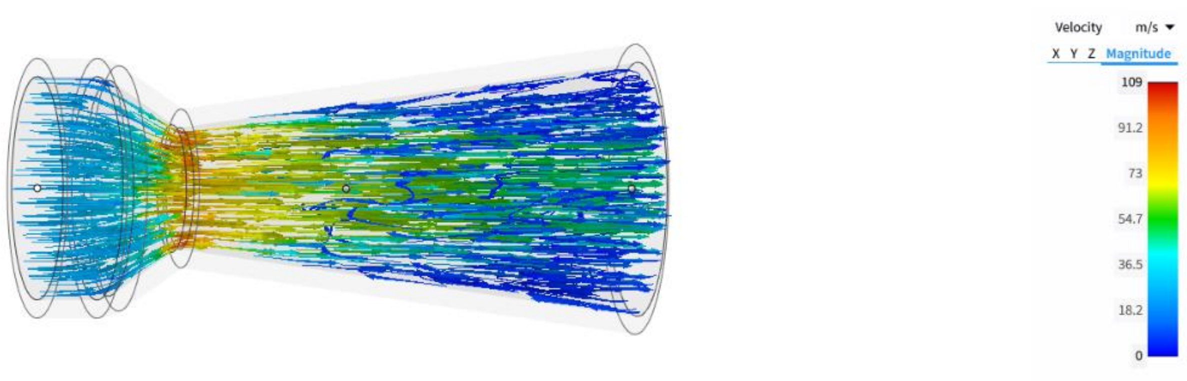


Figure 3-3-6; 77 mm de Laval Nozzle

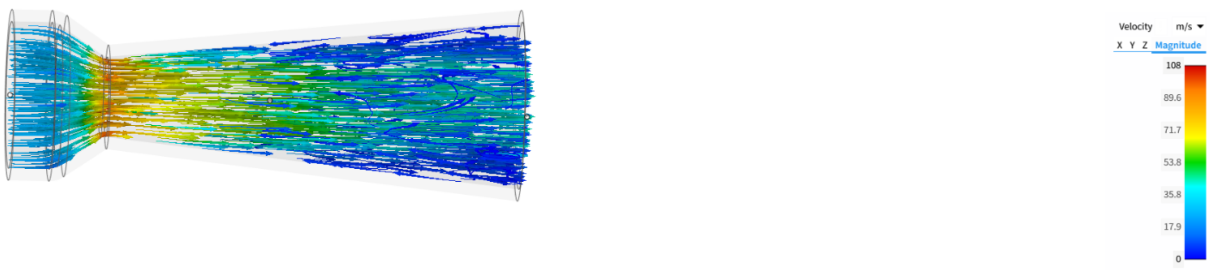


Figure 3-7; 89 mm de-Laval Nozzle

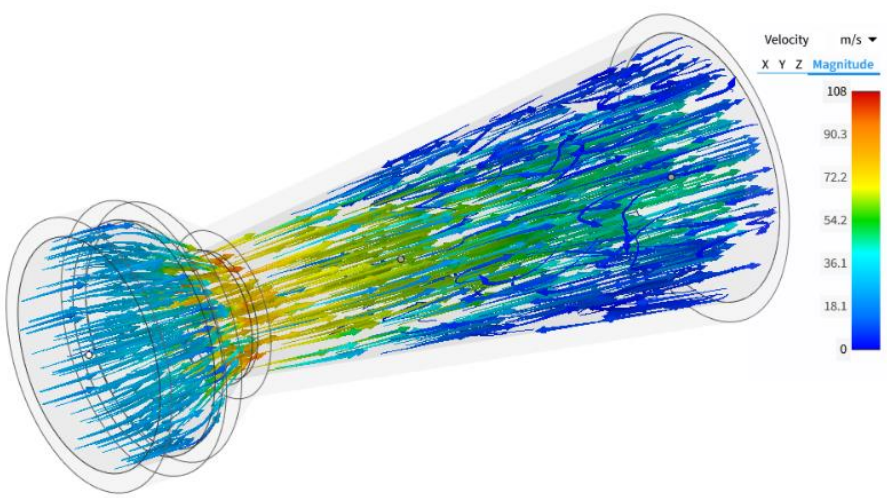


Figure 3-3-8; 104 mm de-Laval Nozzle

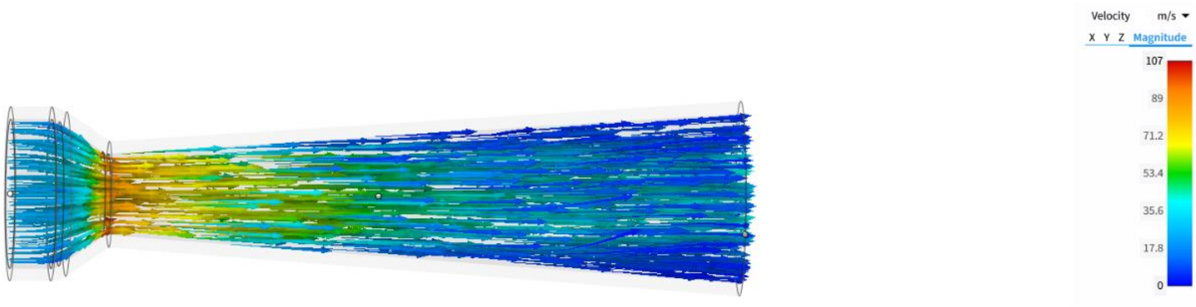


Figure 3-3-9; 153 mm de-Laval Nozzle

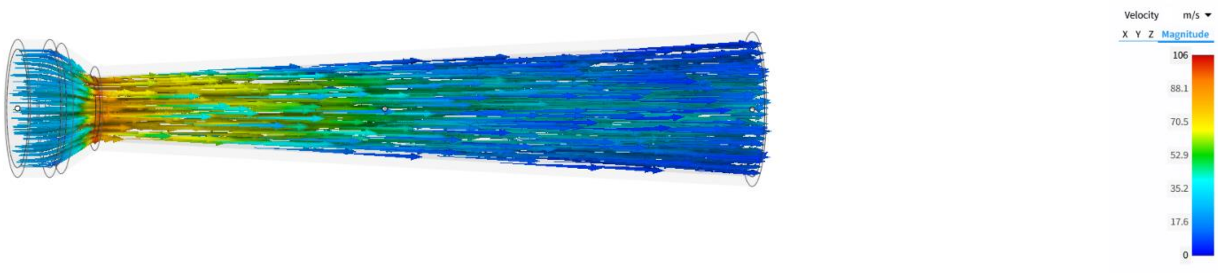


Figure 3-10; 208 mm de-Laval Nozzle

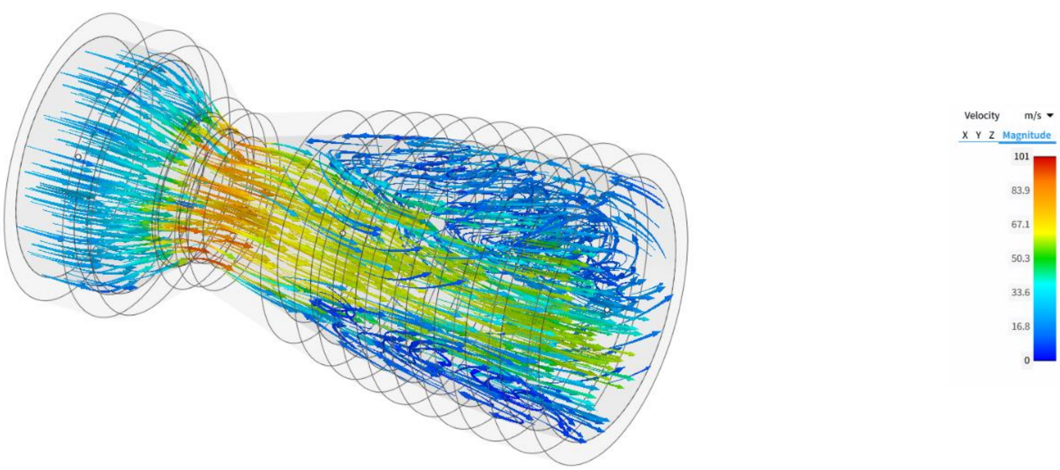


Figure 3-11; 63mm Bell Nozzle

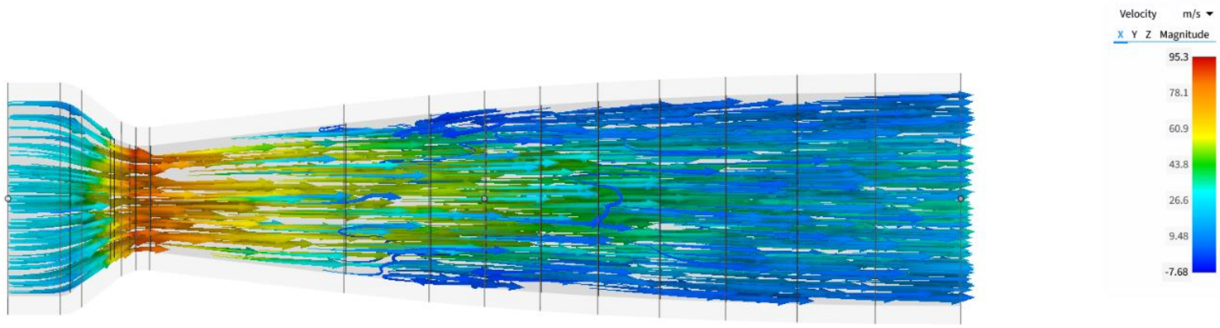


Figure 3-12; 153 mm Bell Nozzle

3.3.5 Engine Horsepower Restriction – Liam O'Connor

Objective

Calculate the threshold, in terms of horsepower, in which the engine will be choked off by the intake restrictor. This will help in choosing an engine, because it will help find an engine that will be affected minimally by the restrictor, strictly in terms of horsepower.

Assumptions

- Ideal Gas
- $\eta = 0.25\ (thermal\ efficiency)$
- $AFR = 14.7\ (air\ fuel\ ratio)$
- $LHV = 44\frac{Mj}{kg}\ (low\ heating\ value)$

Using a thermal efficiency of 0.25 is meant to underestimate horsepower possibilities, as lots of engines hover around 30% thermal efficiency. Using an air fuel ratio of 14.7 represents the ideal stoichiometric ratio for an internal combustion engine. It's likely that the real engine won't be at stoichiometric ratio, it helps to use it because it limits calculation errors associated with using a richer AFR, which will overestimate an engine's possible power output, when using the formula from this section. An LHV of 44 is a standard number based on 93 octane, which is a standard fuel likely to be used with the chosen engine.

$$\dot{m}_{fuel} = \frac{Power}{LHV * \eta_{thermal}}$$

Equation 3-6 - Power Formula

The above equation 3-6 uses power, LHV, and thermal efficiency to calculate the mass flow rate of fuel. Switching this equation around, you can use a known mass flow rate of fuel to calculate the power instead. Based on the max mass flow rate of air going through the restrictor from section 3.3.2, you can calculate the mass flow rate of fuel by dividing the max air flow rate by the AFR. With that you can then calculate the horsepower threshold.

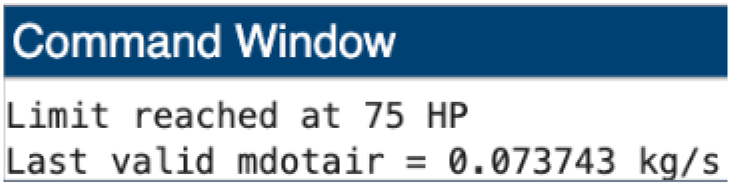


Figure 3-13 - Horsepower Threshold

Using a MATLAB code to find the threshold, the threshold came out to 75 horsepower. This helps us by giving us a metric to compare stock engines to find one that's as close as possible to the threshold. This will give us an engine that theoretically shouldn't be affected by the intake restrictor very much, meaning power loss will be minimal. Then we can effectively choose an engine that gets as close as possible to the threshold, while also being the lightest possible, which will give us a clear winner for the optimal engine.

3.3.6 Exhaust System – Marshall Fritz

Runner / Collector Diameter

Objective

This analysis aimed to find the most appropriate exhaust runner and collector diameters for the Formula SAE powertrain based on flow assumptions created from the Ideal Gas Law. The motivation for this calculation was to evaluate that the velocity of the exhaust gases remains suitable for design purposes, high enough to scavenge the cylinders, but low enough to reduce the amount of backpressure.

Model Description

A computational MATLAB model was developed to find exhaust dimensions based upon the function of engine parameters and gas properties. The model assumes:

- Steady, compressible flow with negligible losses across small distances using short exhaust runners.
- Ideal gas behavior for intake and exhaust flow conditions.
- Constant pressure and temperature at the exhaust outlet for each engine cycle.

Key user-defined inputs include:

- Engine displacement V_T [cc]
- Engine speed N [rpm]
- Volumetric efficiency V_E
- Cylinder count n_{cyl}
- Intake and exhaust temperatures T_i, T_e [°C]

- Atmospheric pressure P_{atm}[atm]
- Exhaust port diameter D_{port} [mm]
- Target mean exhaust velocity v [m/s]

Governing Equations

Analysis begins with the volumetric flow rate of air per intake stroke of a four-stroke engine:

$$Q_{t,intake} = \frac{V_T \cdot N}{2 \cdot 60} \cdot V_E$$

Equation 3-3 - Intake Volumetric Flow Rate3-7

From ideal gas law, air density is taken to be defined by (11) and the total mass flow rate of the engine is:

$$\rho_i = \frac{P}{R \cdot T_i}$$

Equation 3-8-4 – Intake Air Density

$$\dot{m}_{air} = \rho_i \cdot Q_{t,intake}$$

Equation 3-5 – Intake Mass Flow Rate

At the exhaust, assuming negatable fuel mass from combustion and ideal gas characteristics, the total exhaust volumetric flow rate is denoted by:

$$Q_{t,exhaust} = \frac{\dot{m}_{air} \cdot R \cdot T_e}{P}$$

Equation 3-6 – Exhaust Volumetric Flow Rate

Flow rate is then divided by the number of cylinders (14) and the area of the runner is calculated by dividing by the target exhaust velocity, giving the area which can be easily converted into a diameter:

$$Q_{c,exhaust} = \frac{Q_{t,exhaust}}{n_{cyl}}$$

Equation 3-7 – Volumetric Flow Rate per Cylinder

$$A_r = \frac{Q_{c,exhaust}}{v_{target}}$$

Equation 3-8 – Runner Area

The MATLAB outputs for runner diameter, collector diameter, and total exhaust flow rate. The outcome allows correct exhaust geometric sizing referenced against thermodynamic and fluid flow constraints. Aiming for an exhaust gas velocity of approximately 50–70m/s is typically a good range for optimizing scavenging and reducing backpressure for small displacement FSAE engines. Likewise, the model could be used parametrically to model and evaluate alternative design conditions, engine sizes, or target performance. Further refinement might involve:

- Incorporating modeling of exhaust pulse timing and wave reflections.
- Using CFD to validate flow uniformity and pressure losses through the collector.

At the target engine speed of 7000 rpm and a volumetric efficiency of 0.9, the analysis of the exhaust flow yielded a total volumetric flow rate for the 650cc twin-cylinder engine of 0.113 m^3/s. The design mean exhaust velocity of 60 m/s dictated an approximate runner diameter of 34.7 mm, which is closely aligned with the 35 mm exhaust port diameter, showing that the flow should maintain consistent continuity. The collector diameter of 49.0 mm provides sufficient cross-sectional area to combine both runner streams together without excessive backpressure. These outcomes indicate that the exhaust geometry is a satisfactory balance between flow and back pressure to promote efficient gas removal and maintain performance at target engine speeds.

Header Runner Length

Objective

The purpose of this model is to evaluate the ideal exhaust runner length for the FSAE engine based on its exhaust gases' thermodynamic properties and pressure wave propagation. A proper runner length can help maximize scavenging, yields more torque, and improves volumetric efficiency.

Model Description

The model utilizes a quasi-one-dimensional organ pipe inspired analysis of exhaust gas dynamics, which includes air-fuel ratio, exhaust temperature, and exhaust composition. The following assumptions have been made:

- Exhaust gas is an ideal gas mixture.
- Combustion products are limited to CO_2, H_2O, N_2, O_2, and CO.
- The runner length is adjusted for pressure wave reflections and the operational RPM of the engine to allow for resonance at the desired speed.

Key user-defined inputs include:

- Exhaust temperature T_C [°C]
- Air–fuel ratio and equivalence ratio ϕ
- Engine speed N [RPM]
- Fuel stoichiometry and chemical composition for 93-octane gasoline

Governing Equations

Analysis starts with defining the equivalency ratio from the defined air fuel ratio (AFR):

$$\phi = \frac{AFR_{stoich}}{AFR}$$

Equation 3-9 – AFR Equivalency Ratio

Mole fractions are computed for each combustion product and specific heat at constant pressure is obtained through NASA Glenn polynomial coefficients:

$$y_i = \frac{n_i}{\sum n_i}, \quad i = CO_2, H_2O, O_2, N_2, CO$$

Equation 3-10 – Mole Fraction of Exhaust

$$C_{p,mix} = \sum_i y_i \cdot C_{p,i}(T)$$

Equation 3-11 – NASA Glenn's Polynomials

The gas constant for the exhaust is required to find the specific heat ratio both are calculated using the following equations:

$$R_{mix} = \frac{R_u}{MW_{mix}}$$

Equation 3-12 – Gas Constant

$$\gamma = \frac{C_{p,mix}}{C_{p,mix} - R_{mix}}$$

Equation 3-13 – Specific Heat Ratio

Speed of sound is a function of the gas constant, specific heat ratio, and temperature. This speed changes the rate of pressure wave propagation and is a major part in determining the desired length. The final runner length is also dependent on the revolutions and exhaust valve duration:

$$a = \sqrt{\gamma \cdot R_{mix} \cdot T}$$

Equation 3-14

$$L = \frac{a \cdot 120}{N}$$

Equation 3-15

Results

The MATLAB program provides:

- Speed of sound a in the exhaust gas [m/s]
- Exhaust runner length L [m]

These are first order approximations that may be useful in designing runners in general to tune pressure waves and improve engine performance.

The runner length is an important variable for engine performance in high-speed racing applications. Runners that are properly sized:

- Improve low to mid-range torque via pressure wave scavenging.

- Reduce exhaust backpressure, thus allowing for more volumetric efficiency.
- Can be tuned to happen in certain RPM ranges based on engine mapping.

At an exhaust gas temperature of 700 °C, an AFR of 13.2, and an engine speed of 7000 RPM, the speed of sound in the exhaust was calculated to be 607.2 m/s, resulting in the optimal runner length of 0.8675 m (2.85 ft).

This runner length is configured so that the exhaust pressure waves are aligned perfectly with the engine valve timing at 7000 RPM, ultimately placing the power peak exactly where intended. By tuning the runner to this length, the exhaust pulses arrive back at the exhaust port, which assists with scavenging and increasing fill, at the correct time. This gives a length that is mathematically sound for the intended target RPM and ensures that the engine will produce peak power at the torque peak.

3.3.7 High Compression Piston Comparison – Aidan Willson

Objective:
Validate whether it will be useful to incorporate a high compression piston kit into the power train design. Using MATLAB Simulink modeling to obtain velocity vs time and RPM vs time curves that compare the compression.

Model Description:
A Simulink model using powertrain components was created with driveline blocks provided in the Simulink software. It uses component models such as engine, transmission, chain drive, differential and wheels to output the Velocity vs RPM. Modify the components to simulate a specific engine performance curve and use this data to confirm the performance increase if a higher compression ratio is used.

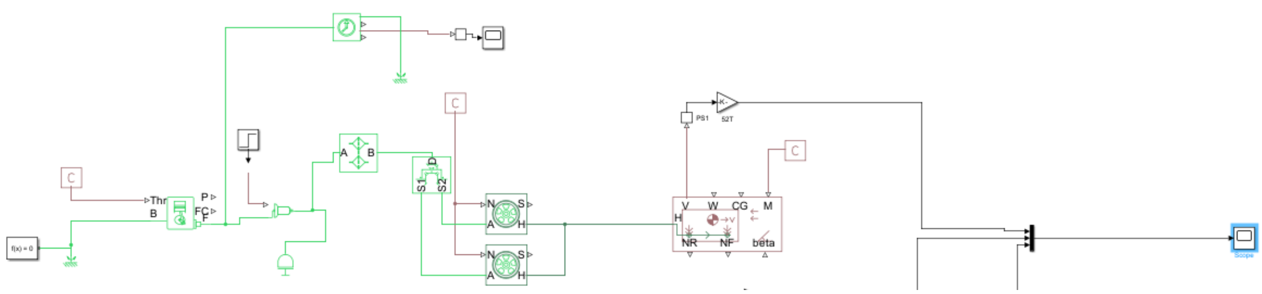


Figure 3-14 Vehicle Model in Simulink

The stock Kawasaki ninja 650R has a compression ratio of 10.8:1 and a peak power of 53.7 kW. Using a higher compression ratio increases the volume size between the top dead center and bottom dead center of the cylinder. This model will be analyzing a new compression ratio of 13.5:1. Based on another model that Liam created, we obtained a potential power increase of ~1kW gained based on the new ratio. But there is High chance the estimation is flawed and does not give accurate results. Most likely a higher power gain from the higher compression ratio is obtained. To offset this error, we will use a theoretical 10% power gain to accurately estimate and compare the power when using high compression pistons. The resulting power increase would be ~5.37 kW gained, and this gives a new peak power of 59.07 kW.

For this model we will compare the stock compression ratio power, the calculated higher compression ratio yielding ~1 kW of power gained, and a theoretical higher compression ratio yielding a 10% power gain. All three can be viewed on the Velocity vs Time and RPM vs Time graphs below.

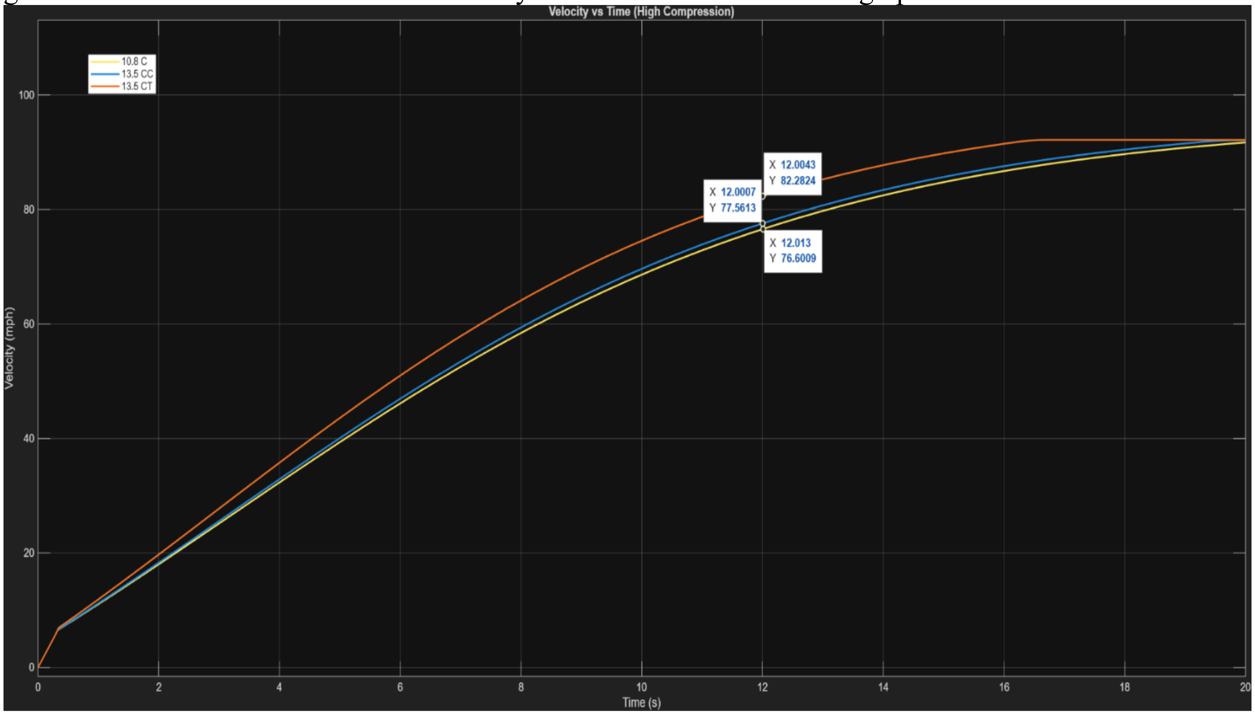


Figure 3-15 Velocity vs Time for Various Compression ratios

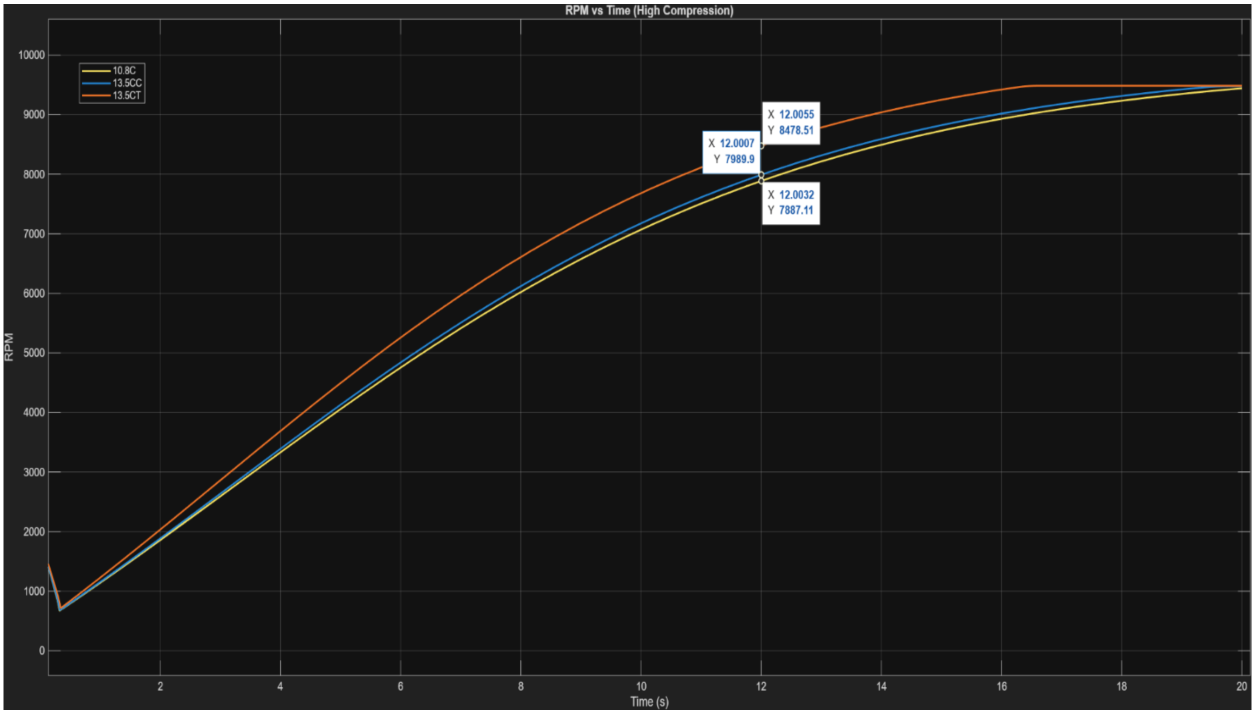


Figure 3-16 RPM vs Time for Various Compression ratios

Results:
Looking closer at the graphs of each compression ratio variation, we can observe the minimal increase in performance if the actual power gain is only ~1 kW. Only increasing top speed from 76.6 mph at 12 seconds to 77.56 at the 12 second mark. We can see the theoretical higher compression ratio that gives a power increase of 10% results in a much more promising velocity and rpm curve. A potential increase in top speed from 76.6 mph to 82.28 mph at the same 12 second mark. A large potential increase in RPM at 12 seconds from 7887 rev/min to almost 8500 rev/min. An overall increase of 5.68 mph and the rpm curve show a significant increase as well.

Discussion:
Our compression ratio can be fairly easily modified using a high compression piston kit. The question is if the increased compression ratio would be a valuable and effective design purchase. If the power increase is in fact only a ~1 kW increase, then based on our results the high compression kit would not be effective or worth the money. Based on the results of a theoretical 10% power increase using 13.5:1 high compression piston kit, the increase in velocity and peak rpm would be very beneficial for maximizing the performance of our engine. If we do decide to go that route, JE pistons make an affordable 13.5:1 compression piston kit for the Kawasaki 650R at a price of $525 [36] and is easily accessible via their website. The increased compression ratio would also mean that we need to increase the fuels' octane. With additional resources from an online forum [37], the chart below can be used to verify the correct octane to be used.

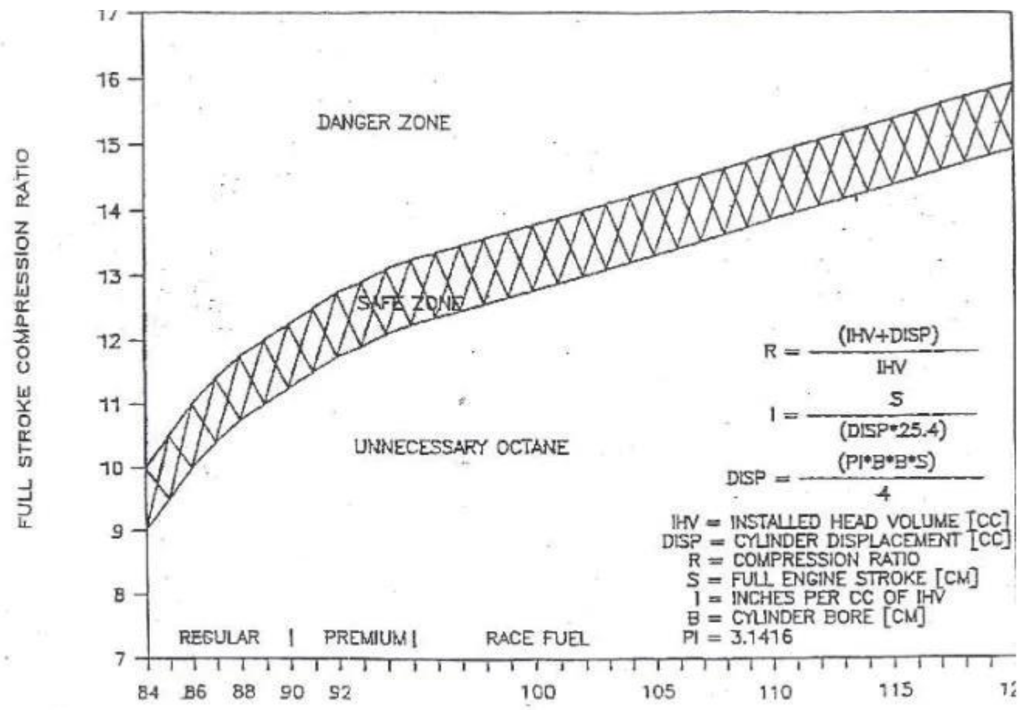


Figure 3-10 - Octane Vs. Compression Ratio

Based on the chart, a compression ratio of 13.5:1 would fall into the 100-octane range. This works out well because the fuel order form for Formula Competition provides fuel of up to 100-octane.

3.3.8 Axle Stress Calculations – Jackson Nichols

To calculate the stress the team's half shafts will endure, combined stress analysis was performed given the chosen engine's torque.
The von Mises equation,

$$\sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

Equation 3-16

was selected for this purpose as half shafts are mostly made of SAE 4130 or SAE 4140 ductile steels. Comparing the allowable stress to the actual stress experienced, the minimum diameter of the shafts then converged to 15.9 mm.

3.3.9 Fuel Tank Capacity – Jackson Nichols

As shown in section 3.3.5, the maximum possible horsepower given our intake restrictor was 75 horsepower. Pluging this value back into the Power Formula (equation 33), the mass flow rate of fuel can be found. At 0.0051 Kg/s, the mass fuel rate is rated for fuel usage at the rpm of the engine, where it makes the most power. For our selected engine this is about 8,000 RPM.

Fuel tank capacity was then calculated based off of the endurance loop race time of SAE Formula 2025. This is the longest race in the event and so is used as the most fuel necessary for the 2026 car. Also included was a factor of safety of 1.1. The team determined this to be sufficient as the car will cycle through max power RPMs opposed to doing the entire race at this engine speed.

Fuel tank capacity then, was determined to be 2.7 gallons. This calculation is checked by the average FSAE tank being 1.3-2.6 gallons [43]. Purchase of the engine and load testing will enable a more precise measurement of AFRs and temperatures which will enable more precise calculations in this area.

4 Design Concepts

4.1 Functional Decomposition

For the functional decomposition, we solely put subsystems that we must make design decisions for. For example, the extent to which we are making calculated decisions for with the engine are strictly air management and pistons. Beyond those two subsystems of the engine, we are likely keeping the engine we purchase stock.

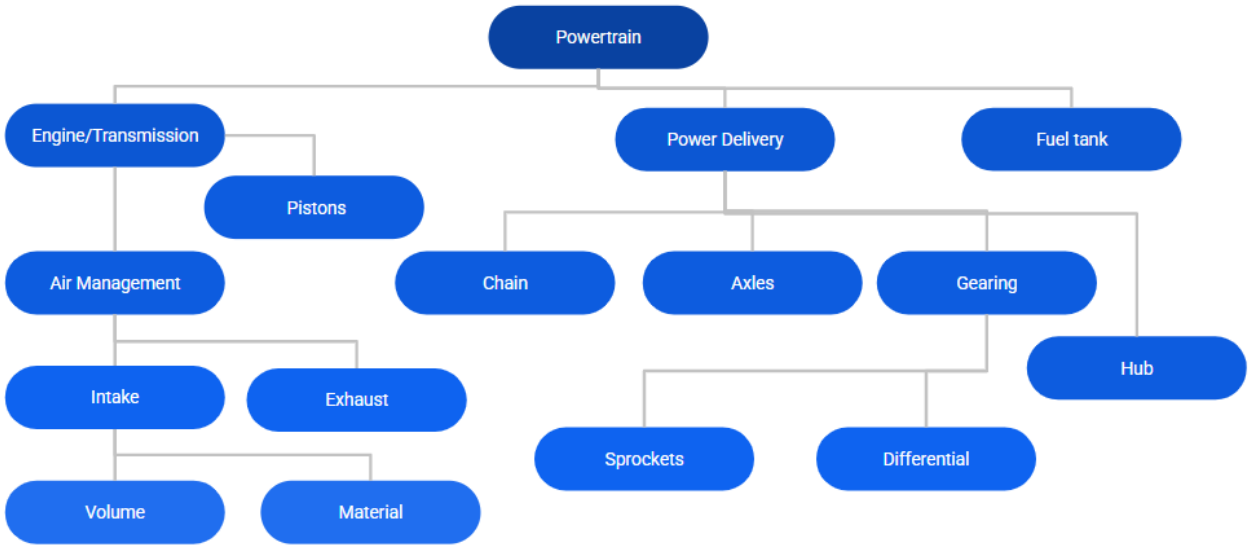


Figure 4-1

4.2 Concept Generation

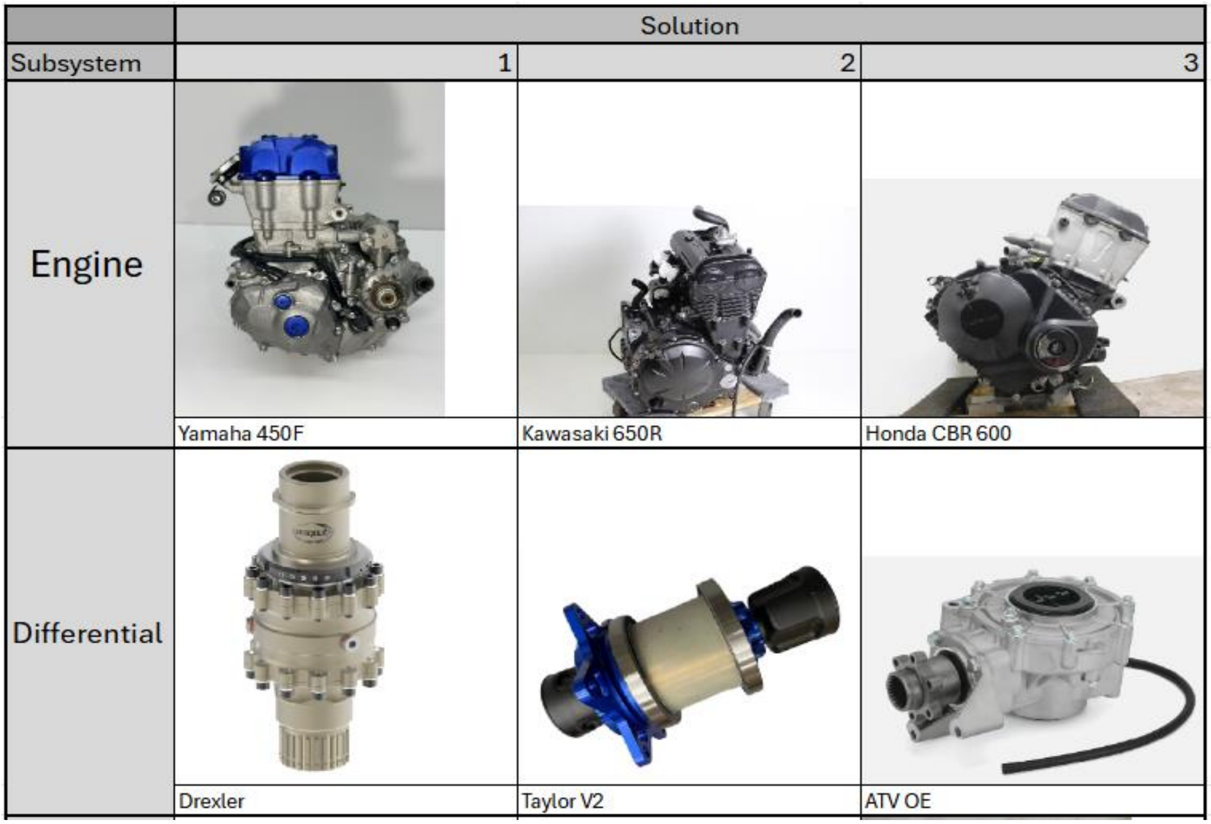


Figure 4-2: Morphological matrix of Engine and Differential Options

Engine

- Yamaha 450F
    - Pros :
        - Lightest of our options
        - Good power to weight
        - Compact
    - Cons :
        - Low power, comparatively (58 hp)
        - Would require very large intake
        - Acoustic harmonics are not ideal
- Kawasaki 650R
    - Pros:
        - Good power to weight

        - More power than Yamaha
        - Intake does not need to be as large; takes in more air per cycle
        - Lowest powerband
    - Cons:
        - Unbalanced
        - Air intake is still not near steady state
        - Not as much power as CBR (72 hp)
- Honda CBR 600
    - Pros:
        - Most power (110 hp)
        - Closest to steady state air intake
        - Balanced
    - Cons:
        - Very heavy
        - High powerband

Differential

- Taylor Mk2
    - Pros
        - Cheap ($2400)
        - Lightweight
        - Works with 520 chain and expected loads, high quality LSD
    - Cons
        - Must use Taylor specific sprockets
- Drexler
    - Pros
        - High quality LSD
        - Light
        - Commonly used in formula
    - Cons
        - Expensive ($3500)
- ATV OEM
    - Pros

    - Durable
- Cons
    - Heavy
- Difficult to integrate

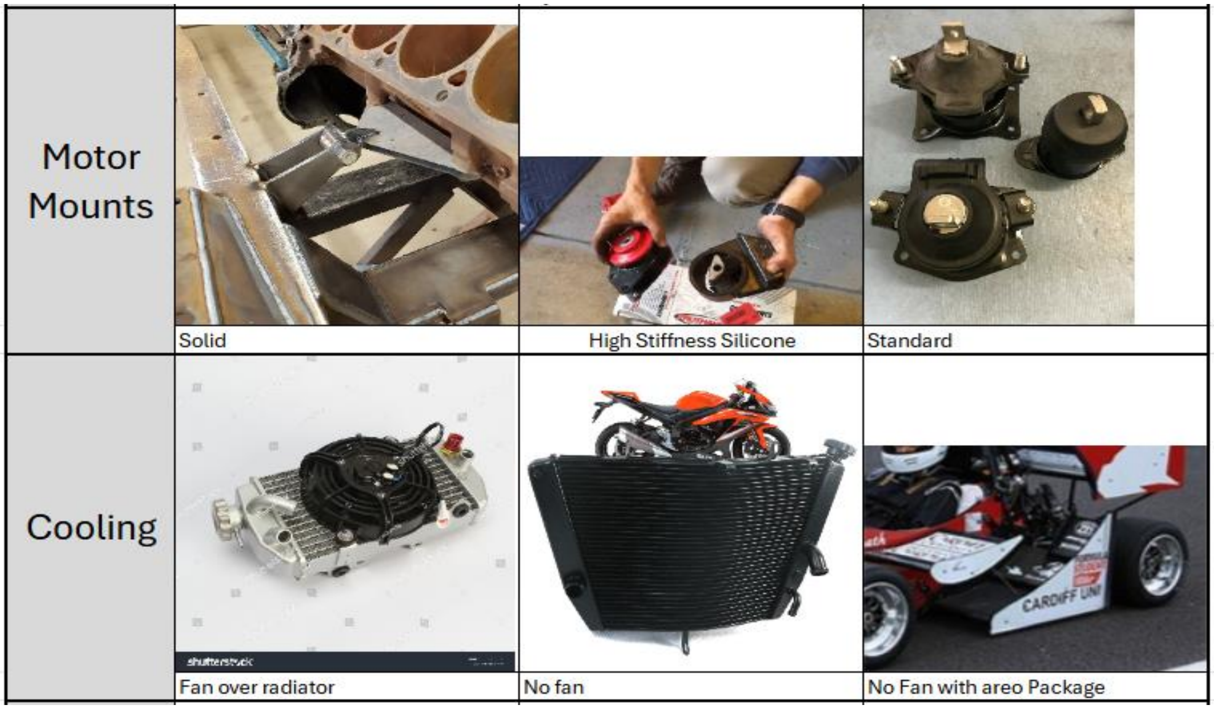


Figure 4-1showing radiator and motor mount options

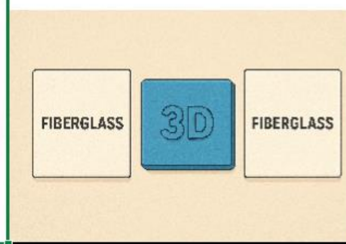
Intake Material	 Aluminum	 Plastic	 Compostite Printed Plastic w/ Strengthening
Fuel Tank	 Stainless	 Plastic	 Aluminum

Figure 4-4; Intake and Fuel Tank Morphological Matrix

Intake Material

- Aluminum
  - Pros
    - High quality surface (low roughness)
    - Good heat resistance
  - Cons
    - Heavy
    - Expensive, for both materials and manufacturing
- Plastic
  - Pros
    - Light
    - Cheap
    - Easy to manufacture

- Cons
    - Poor heat resistance and durability
    - Surface roughness is not as good
- Composite
  - Pros
    - Good surface roughness is achievable
    - Super lightweight
    - Good heat resistance
  - Cons
    - Very expensive and difficult to manufacture

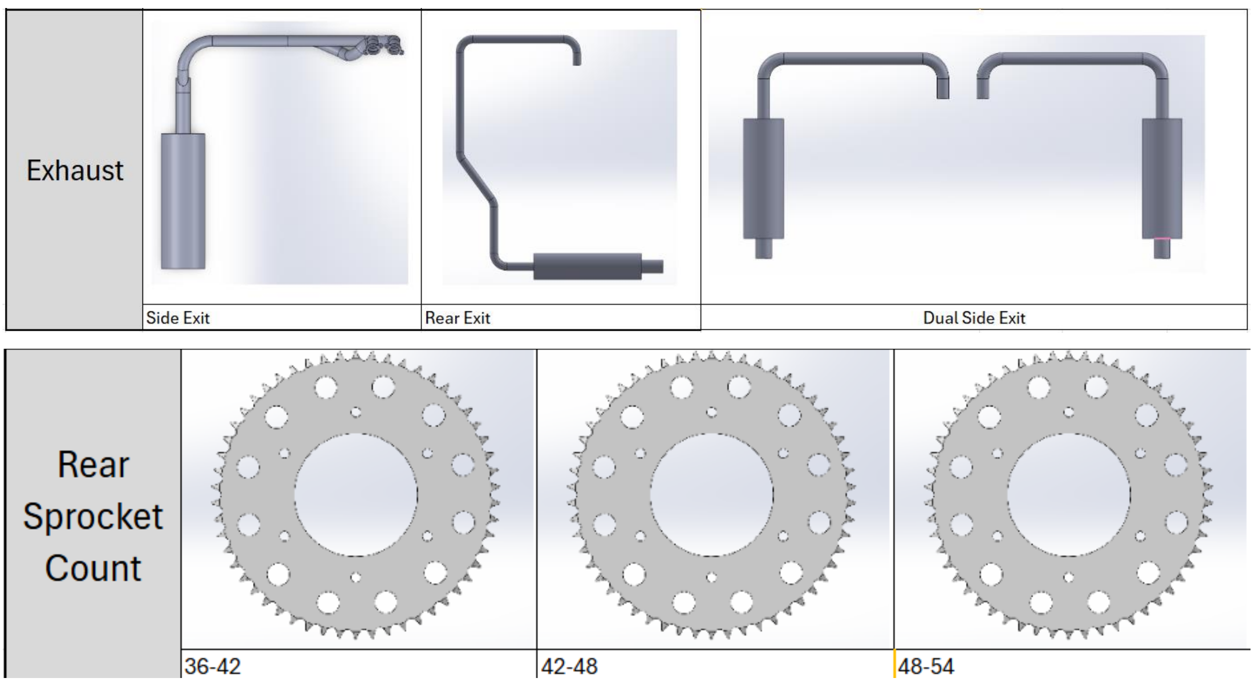


Figure 4-5; Morphological Matrix with Every Part Choice

4.3 Selection Criteria

Selection criteria differed for each component of the power train. In general, methods that prioritized weight and cost reduction were selected. In this thinking, the powerhouse of the powertrain, our engine, must be able to reach the maximum possible horsepower given the intake restrictions while being as light as possible. Based on the horsepower threshold calculation in section 3.3.5, the Kawasaki is the closest in horsepower to the calculated 75 horsepower threshold, indicating that it makes the most horsepower while still being low enough to have little horsepower loss. Even though the Honda CBR has quite a bit more power, we know that it will be severely limited by the restrictor, likely dropping it down to similar numbers as the Kawasaki. Since the Kawasaki is only 2 cylinders, it is much lighter than the Honda, making it a better overall choice for both acceleration and top speed. Compared to the Yamaha, the Kawasaki is heavier than the Yamaha, but the power and torque discrepancy between them makes the Yamaha a poor choice. The Kawasaki also has better opportunity to make horsepower through high compression pistons, where the Yamaha and Honda are already high compression engines, with little to gain in that respect. Lastly, the torque is a major deciding factor in which engine is best for Formula. Speeds are fairly low in each event, meaning that the emphasis should be on having the best acceleration. Of all three engines, the Kawasaki makes its torque lowest in the rpm range, which will yield better acceleration than the other engines. The Kawasaki also makes the most peak torque, further highlighting it as the best choice.

For the differential, the cheapest option that provided limited slip power delivery was to be selected. For the material used in motor mounts, comfort and performance were to be balanced, since the price difference between high quality aftermarket and OEM options is not substantial enough to be a selection criterion. The selection criteria for the cooling system were again to minimize cost. Furthermore, simplicity was valued for reliability here. In the case of intake, selection for size and shape must maximize the engine's ability to receive air for any given stage of the power cycle. The material of the intake was to be selected to maximize weight loss and strength. Again, the price difference between methods and materials was not substantial enough to be considered. Axles had the selection criteria for being made of SAE 4130 or SAE 4140 steel. They also must've withstood the calculated stresses specific to the engine chosen. The method of chain tensioning was determined based on simplicity, measured by number of moving parts, and ease of maintenance. The exhaust system was to be selected based off of its ability to flow and properly scavenge exhaust gases. This was determined to use simulation software. Finally, the size of the rear sprocket was selected to balance top speed and acceleration. It was shown in speed and acceleration curves that it would be optimal to have a mid-range gear ratio using a small front sprocket to reduce the envelope on the transmission

4.4 Concept Selection

The parallel twin Kawasaki engine and Taylor Race Engineering Mk 2 differential proved to the team to be the cheapest while still hitting our horsepower and stress requirements. Thus, these two components were selected by the process of elimination. For the exhaust, the rear exit design was selected for it's increased scavenging, lower sound output, and simpler fabrication. These properties were determined by CAD software and this layout had the best of these properties out of all three exhaust systems modeled. The cooling system was picked to be a water cooled engine with aero components to aid in it's cooling. After a conversation with the aero group, this was determined to be possible, and also fulfilled the requirements for simplicity on the powertrain side of things. Further components selected for the design of the car were high stiffness silicone motor mounts, aluminum intake material with a large internal volume, prefabricated axels cut to size (Taylor Race Engineering), an aluminum fuel tank, and a solid chain guide/ tensioner.

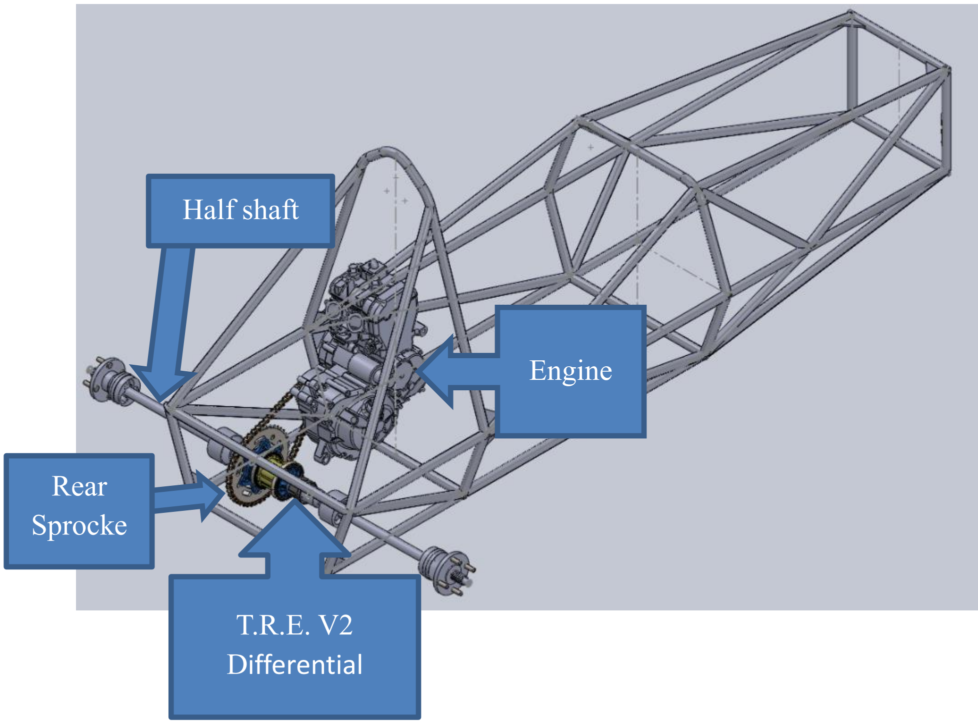


Figure 4-6; CAD Model as of now

5 CONCLUSIONS

Northern Arizona University's SAE Formula team is currently designing an open wheeled race car to compete in the SAE race in May 2026. Throughout the course of this semester, we aim to finish most of the design work as well as have a full frame welded and ready for next semester.

The powertrain subteam has evaluated several different approaches to the car this year. Instead of using the Honda CBR600 4-cylinder motor, as in the NAU 2024 and 2025 cars, we have elected to pick a lighter 2-cylinder engine. While we are purchasing an engine, there are several other design challenges to consider. Per FSAE rules, we must impose a 20 mm diameter intake restrictor, which presents a host of factors to consider with our intake design. We also must design an exhaust system, chain and final drive, cooling, and fuel delivery.

To this point, we have made progress in designing these components, but most of what we have are estimates of what we will be doing in the future. Without an engine to properly extract data from, we are limited by the meager details on specification sheets by manufacturers and mechanics. However, we have run simulations on components like the intake restrictor throat and chain drive, and have calculated engine modifications, exhaust, and fuel. While our designs are far from final, we have made important decisions that should put us in the right direction, backed by calculations that can be easily adapted once more data is available to us.

We are purchasing a Kawasaki 650R engine, which will aid the 2026 car in weight reduction. While the intake design will be more involved because of the less frequent airflow and restrictor diameter, the power to weight and lower powerband should outweigh the negatives. From there, we will be able to determine important exhaust and intake parameters that would simply be guesses in the current state of the project. We will be manufacturing a long bell nozzle restrictor throat, purchasing a Taylor Mk2 differential and the subsequent sprockets with a mid-range ratio. We will purchase pre-fab axles that we will cut to size. We will likely purchase high compression pistons (13.5 CR) for increased power.

Going forward, there is a lot of research, calculations, and design work we must do before we can put the powertrain together. Once we can purchase these items and begin assembly and manufacturing, we can begin testing our initial designs and making any necessary modifications.

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7 APPENDICES

7.1 Appendix A: Figures

System QFD: Powertrain

Project:	QFD
Date:	09/14/25

Legend

A	2001 CBR600 F4I
B	2012 Kawi Ninja 650R
C	2014 Yamaha YZ450f

		1	2	3	4	5	6
1	Engine noise less than 103 Dbc						
2	Reduce weight of muffler	+					
3	Remove mass from intake manifold						
4	Minimize mechanical loss testing	+		++			
5	Obtain LSD with smallest impact on budget						
6	Decrease distance from fuel fill to outermost frame rail						
7	Robust throttle cable with 50mm of adjustment				+		

			Technical Requirements							Customer Opinion Survey				
	Customer Needs	Customer Weights (1-5)	Engine noise less than 103 Dbc	Reduce weight of muffler	Remove mass from intake manifold	Minimize mechanical loss testing	Obtain LSD with smallest impact on budget	Decrease distance from fuel fill to outermost fra	Robust throttle cable with 50mm of adjustment	1 Poor	2	3 Acceptable	4	5 Excellent
1	Pass Tech Inspection	5	9	1	1	9	1	1	9					ABC
2	Reliable/Durable	4	1	1	3	9	3	1	9				B	AC
3	Safe	5	3	1	1	9	3	3	9			AB		C
5	Limited slip rear power delivery	3	1	1	1	9	9	1	1					ABC
6	Ease of maintenance	3	1	3	3	1	1	3	3			A	B	C
7	Drivability	3	1	3	3	3	9	1	9				AB	C
8	Standardized parts	4	3	1	1	1	3	1	3					ABC
10	Make filling up fuel more accessible	5	1	1	3	1	1	9	1			A		BC
11	Lightweight car	3	1	1	9	1	1	1	1		ABC			
	Technical Requirement Units		dBC	kg	kg	hp	$	mm closer	mm					
	Technical Requirement Targets		5	4	2	8	###	100	50					
	Absolute Technical Importance		93	47	89	177	109	91	185					
	Relative Technical Importance		11.8%	6%	11.3%	22%	14%	12%	23%					

Figure 7-1 - QFD

```
Enter the max rpm value: 8500
Enter the number of teeth on the front sprocket: 17
Enter the number of teeth on the rear sprocket: 36
Top Speed in 1st gear: 78.37 mph
Top Speed in 2nd gear: 111.47 mph
Top Speed in 3rd gear: 143.33 mph
Top Speed in 4th gear: 171.97 mph
Top Speed in 5th gear: 197.79 mph
Top Speed in 6th gear: 224.25 mph
```

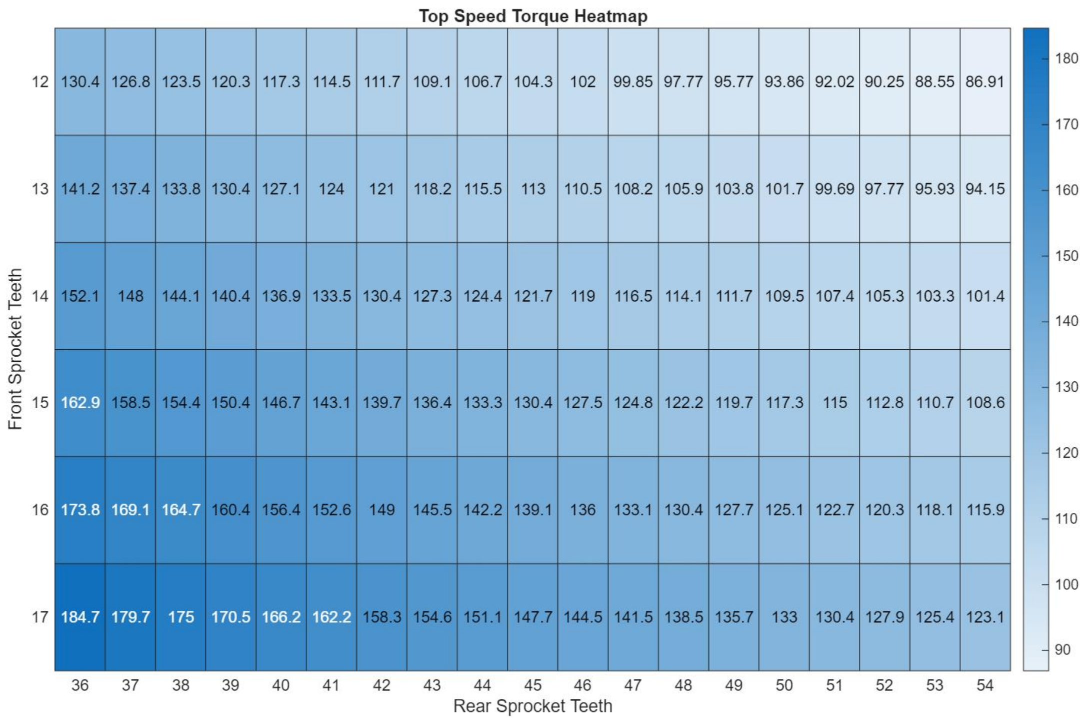


Figure 3-7-2: Theoretical Top Mechanical Speeds

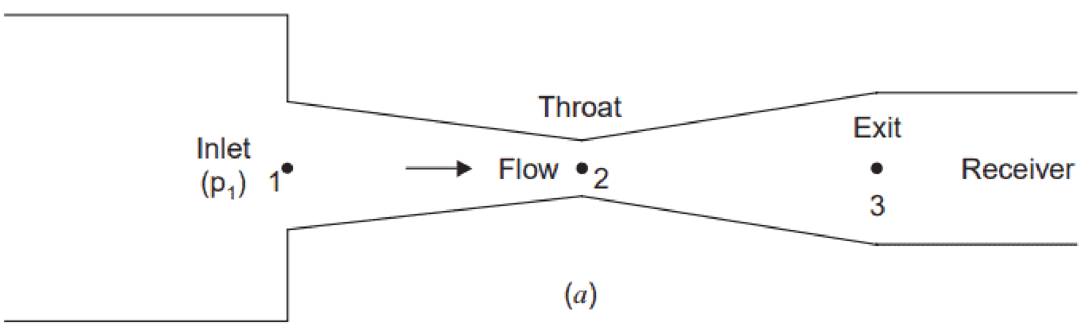


Figure 7-3 - Air Restriction Diagram

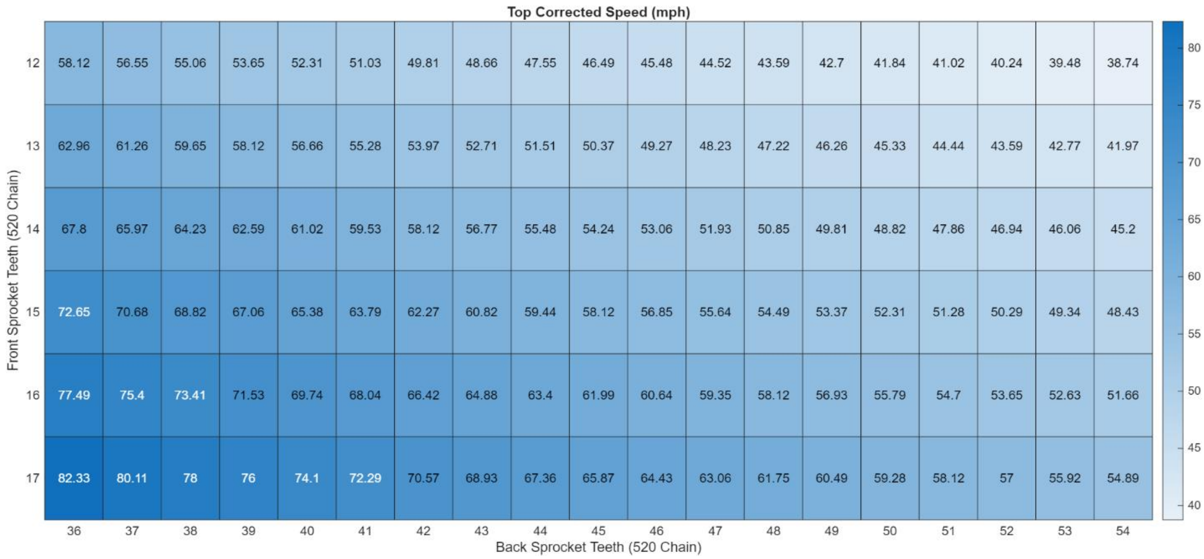


Figure 7-4: Top Corrected Speed in mph, with resistance

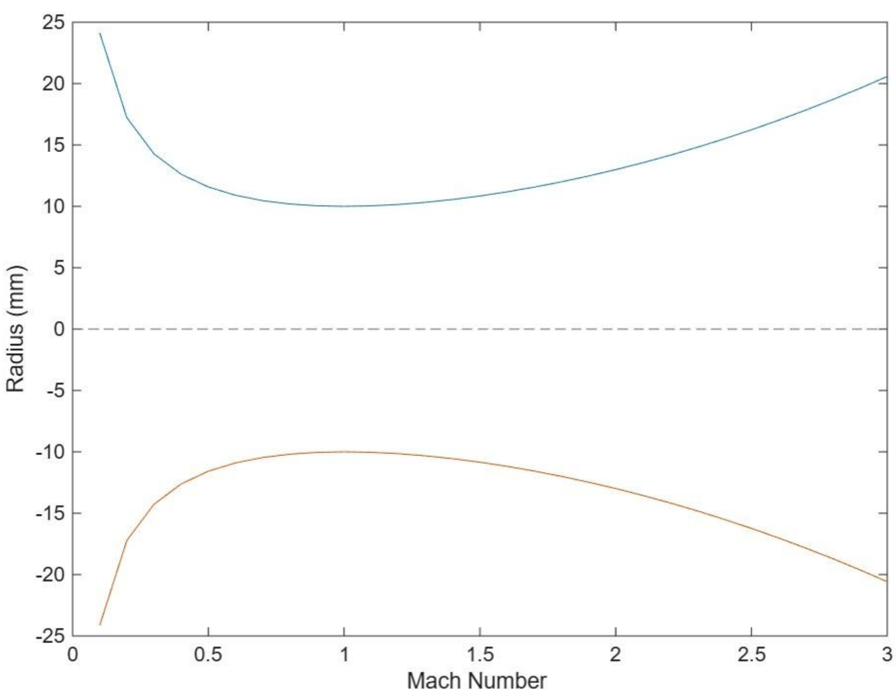


Figure 7-5 - Radius Vs. Mach Number

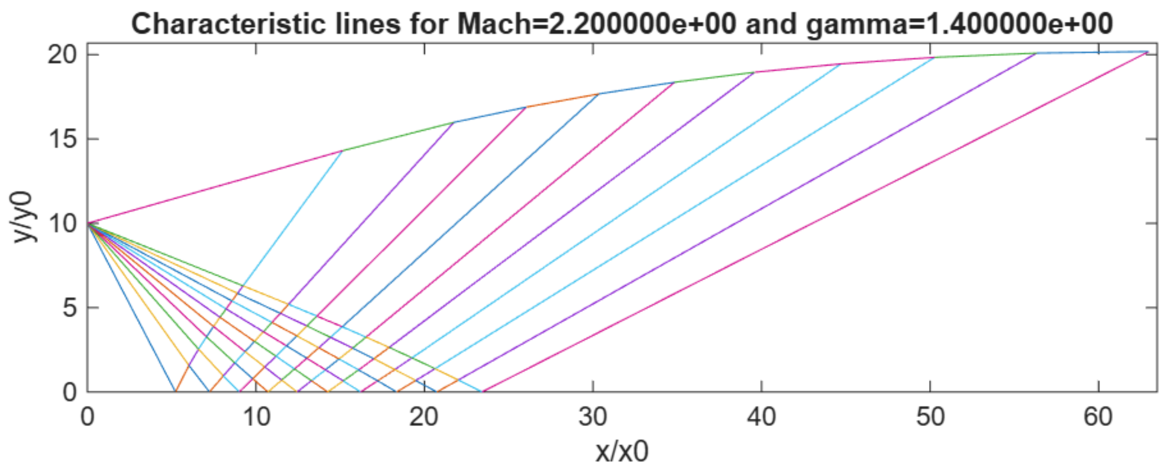


Figure 7-6 - Method of Characteristics Divergent Section

Aluminum alloy, wrought, 6061, T6

User defined name	Aluminum alloy, wrought, 6061, T6
State	Solid
Density	2710 kg/m³
Viscosity	0 Pa·s
Thermal expansion coefficient	2.28e-5 1/°C
Isotropic thermal conductivity	155 W/m·K
Specific heat	0.916 kJ/kg.C
Description	Aluminum, 6061, T6, wrought Data compiled by Ansys Granta, incorporating various sources including JAHM and MagWeb. ANSYS, Inc. provides no warranty for this data.
Class	Metals - non-ferrous
Subclass	Aluminum alloys

Figure 7; Aluminum Parameters for Intake Simulation

User defined name	Air
State	Gas
Density	1.16 kg/m³
Viscosity	1.83e-5 Pa·s
Thermal expansion coefficient	0.00333 1/°C
Isotropic thermal conductivity	0.0258 W/m·K
Specific heat	1.02 kJ/kg.C
Description	Gas, Air Data compiled by Ansys Granta, incorporating various sources including JAHM and MagWeb. ANSYS, Inc. provides no warranty for this data.
Class	Fluids
Subclass	Gases

Figure 8: Air parameters for intake simulations

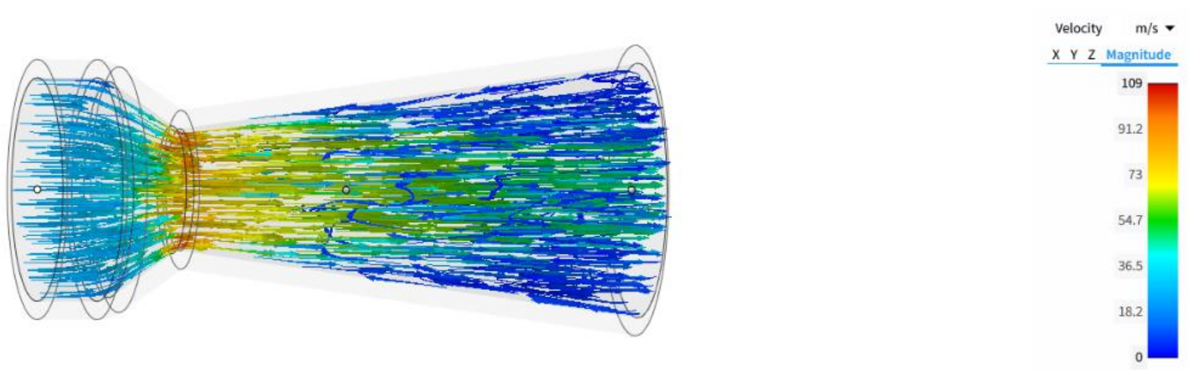


Figure 3-7-9; 77 mm de Laval Nozzle

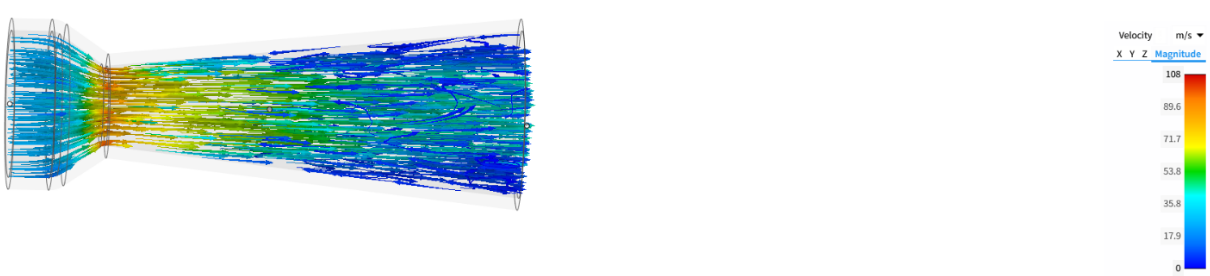


Figure 7-10; 89 mm de-Laval Nozzle

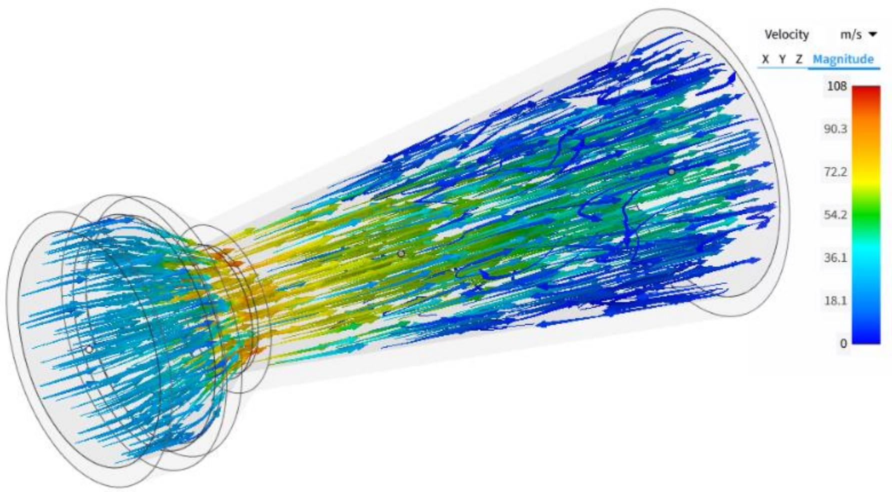


Figure 3-7-11; 104 mm de-Laval Nozzle

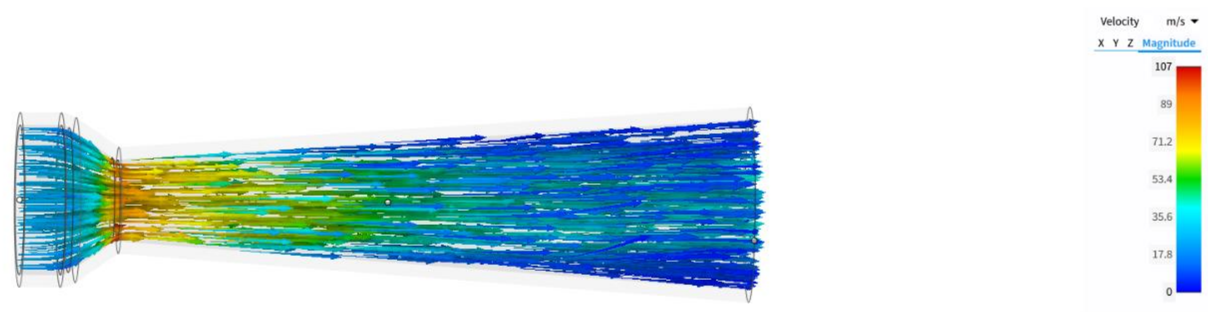


Figure 3-7-12; 153 mm de-Laval Nozzle

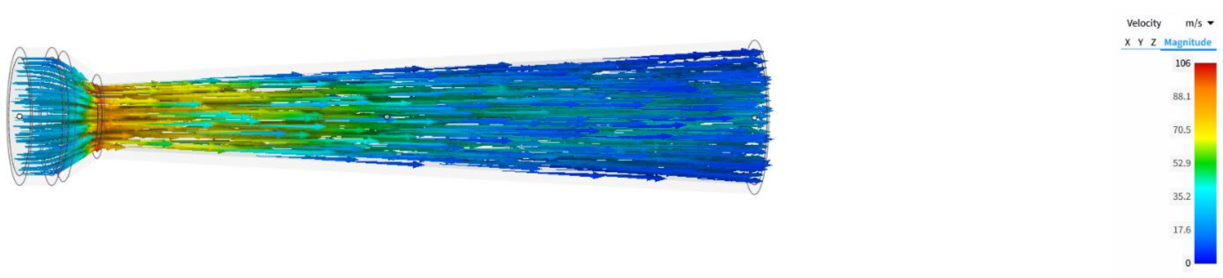


Figure 3-10; 208 mm de-Laval Nozzle

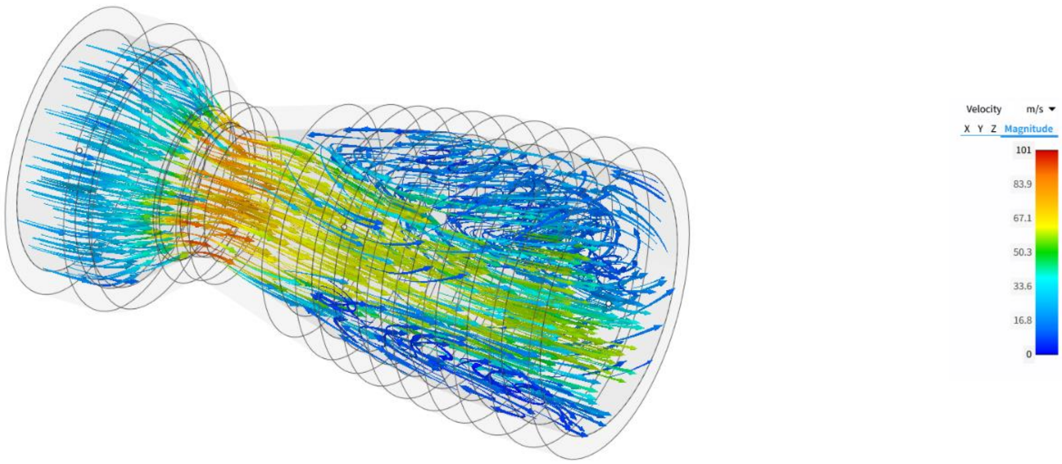


Figure 3-11; 63mm Bell Nozzle

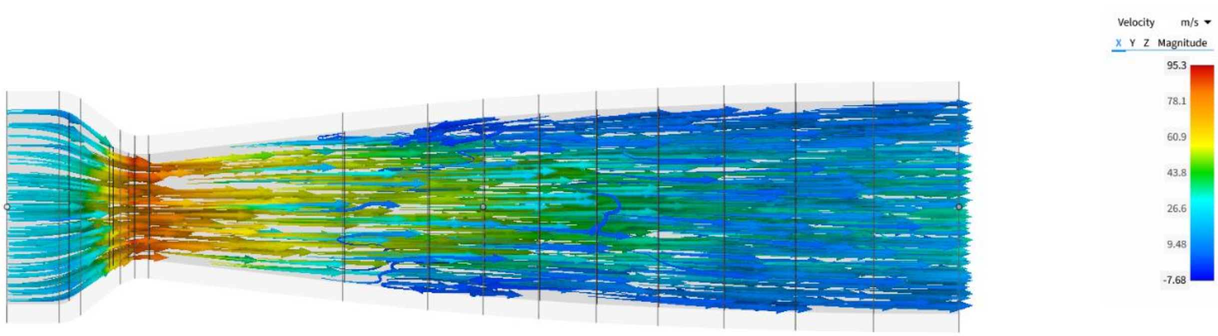


Figure 3-12; 153 mm Bell Nozzle

Command Window

```
Limit reached at 75 HP
Last valid mdotair = 0.073743 kg/s
```

Figure 7-13 - Horsepower Threshold

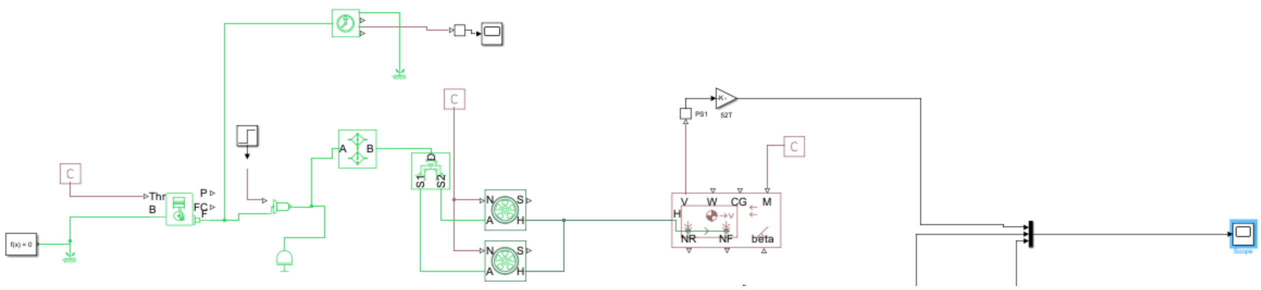


Figure 3-14 Vehicle Model in Simulink

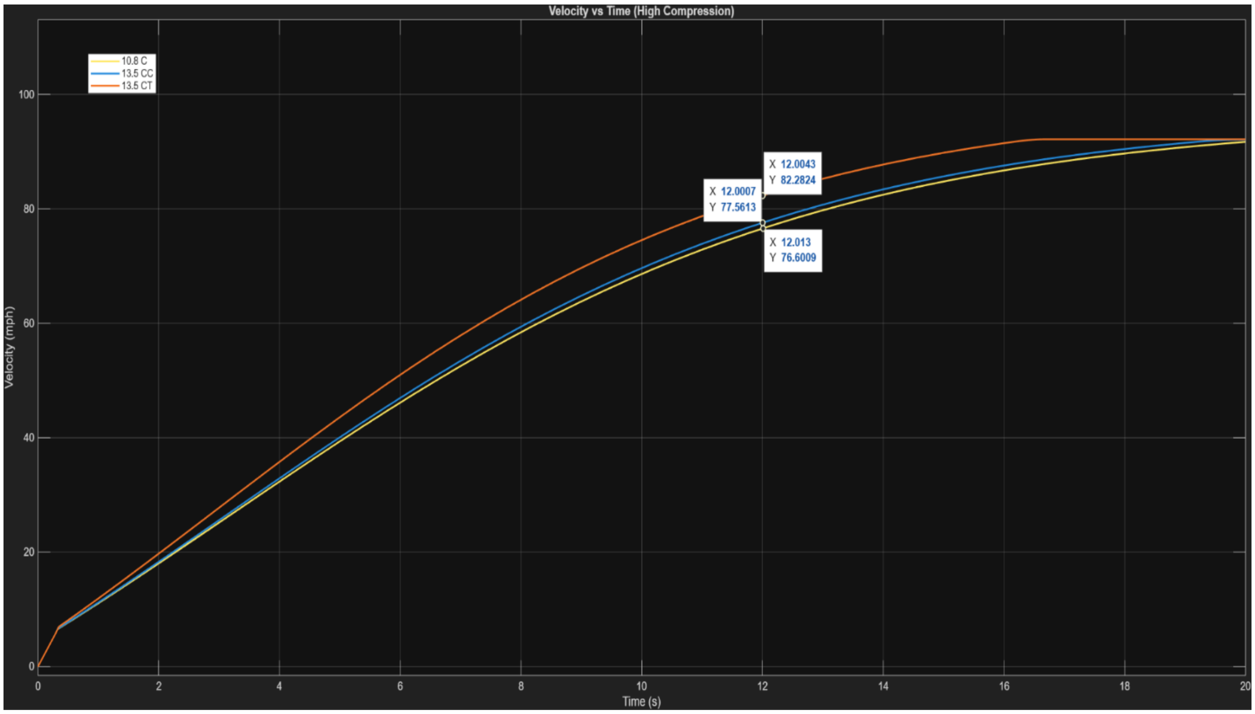


Figure 3-15 Velocity vs Time for Various Compression ratios

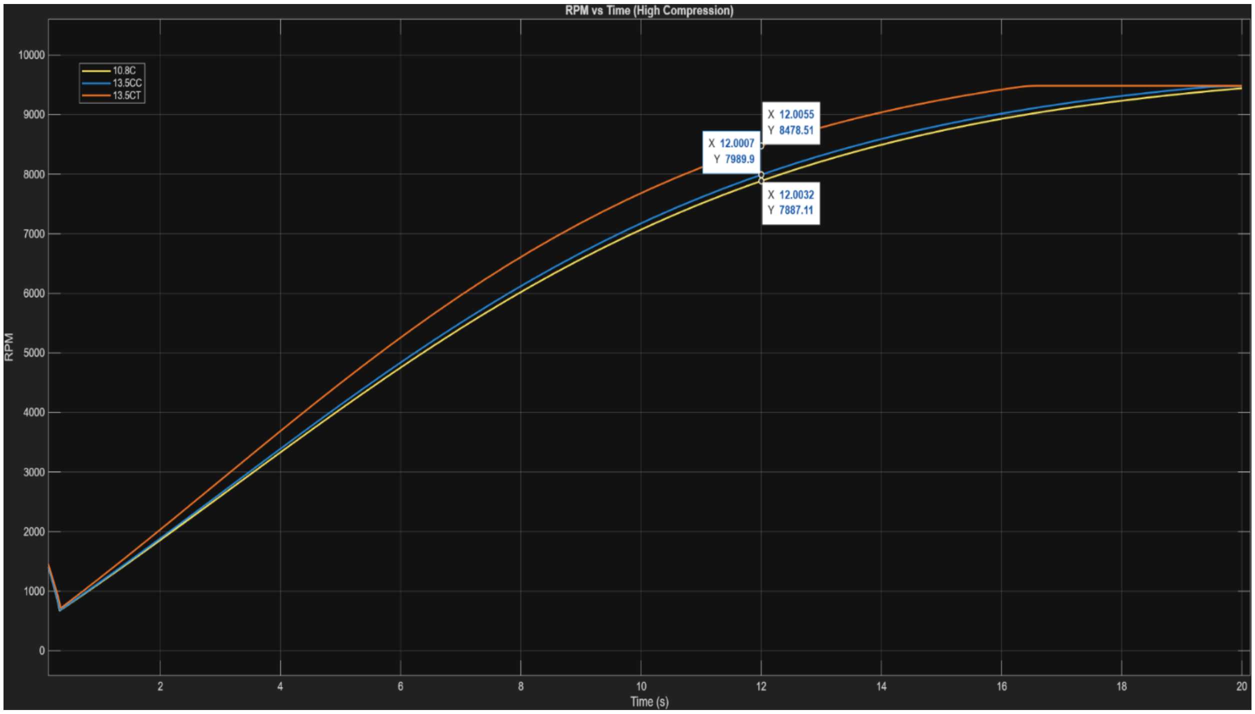


Figure 3-16 RPM vs Time for Various Compression ratios

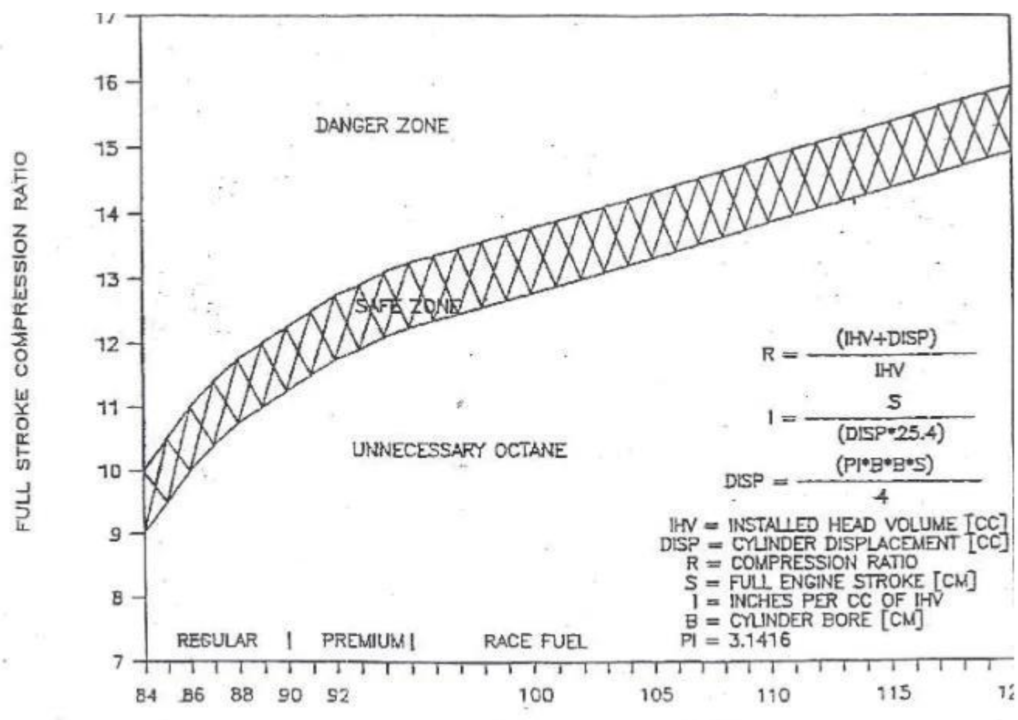


Figure 7-17 - Octane Vs. Compression Ratio

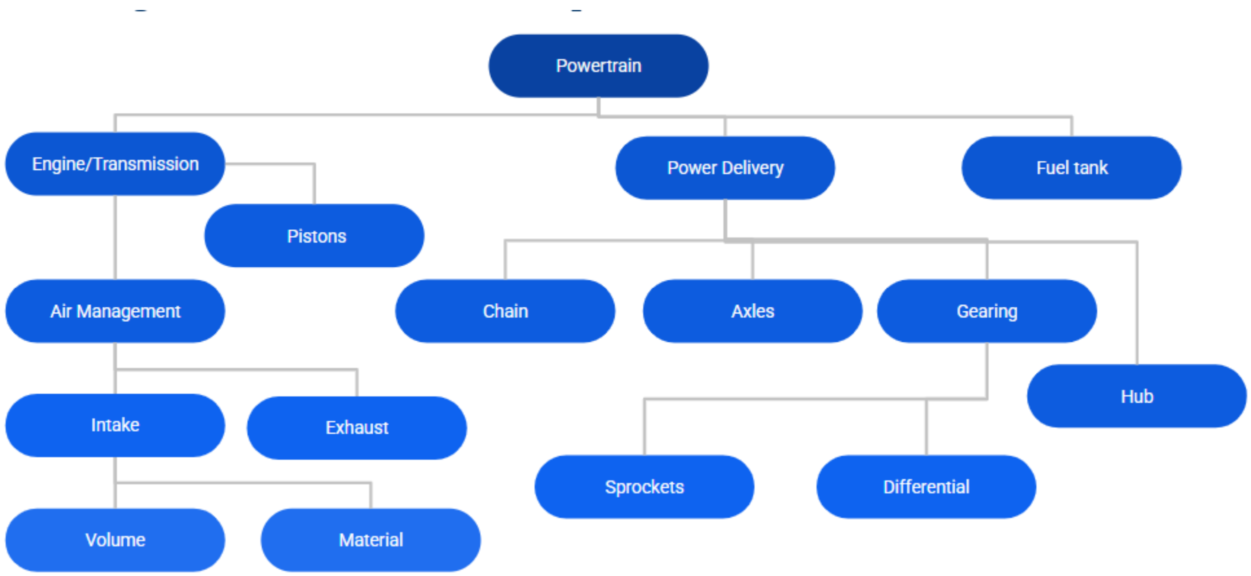


Figure 4-1

	Solution		
Subsystem	1	2	3
Engine	 Yamaha 450F	 Kawasaki 650R	 Honda CBR 600
Differential	 Drexler	 Taylor V2	 ATV OE

Figure 4-2: Morphological matrix of Engine and Differential Options

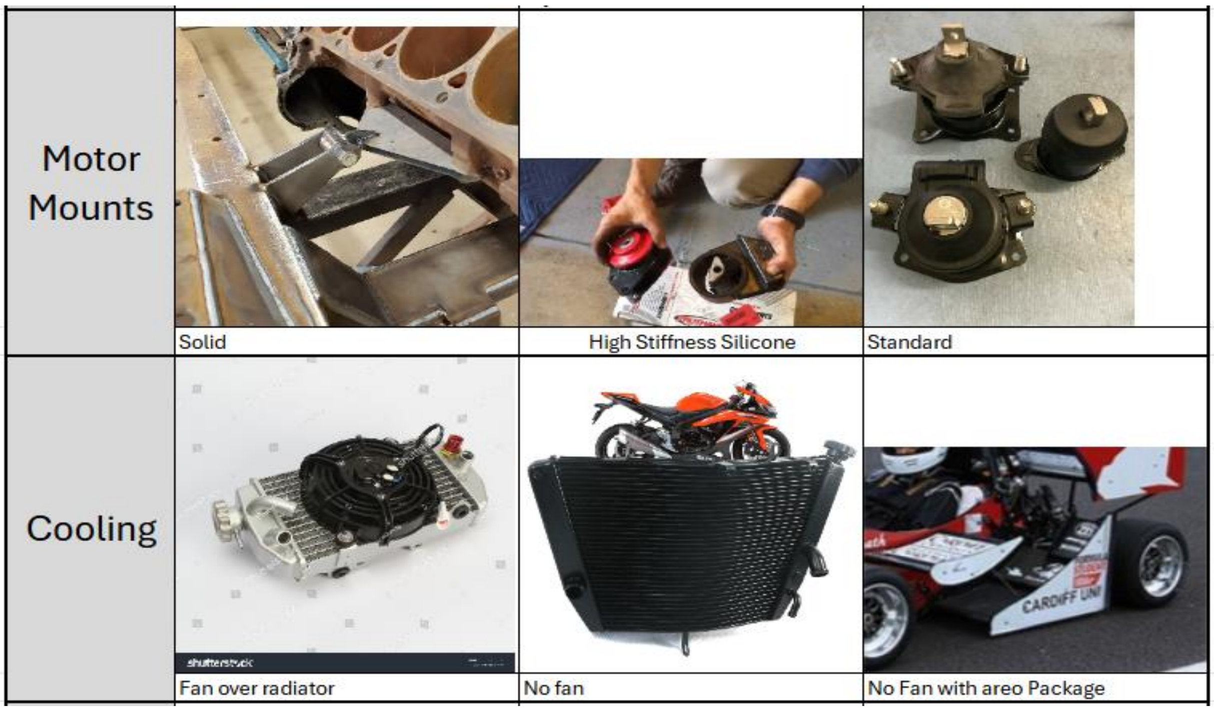


Figure 7-13showing radiator and motor mount options

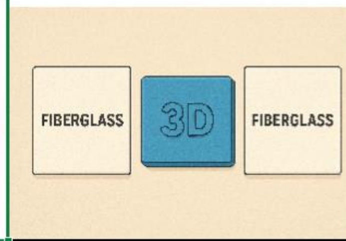
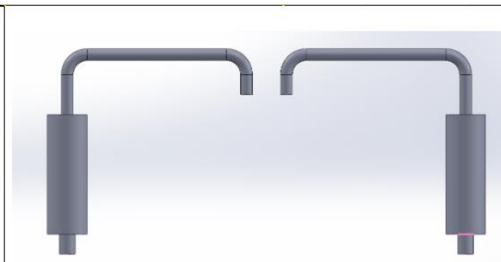
Intake Material			
	Aluminum	Plastic	Compostite Printed Plastic w/ Strengthening
Fuel Tank			
	Stainless	Plastic	Aluminum

Figure 4-4; Intake and Fuel Tank Morphological Matrix

Exhaust			
	Side Exit	Rear Exit	Dual Side Exit

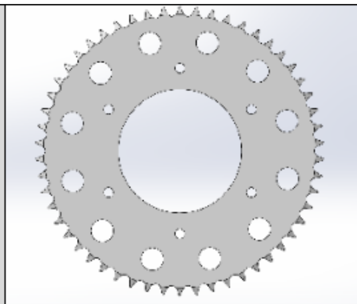
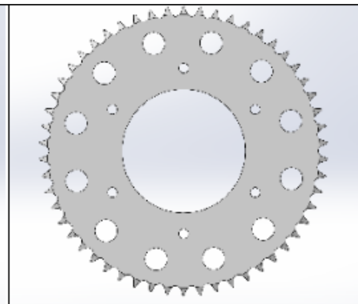
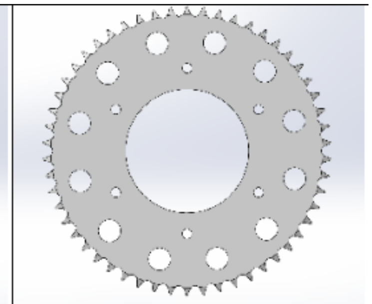
Rear Sprocket Count	 36-42	 42-48	 48-54

Figure 4-2; Morphological Matrix with Every Part Choice

7.2 Appendix B: Tables

Engine	**2001 Honda CBR600 F4i [1]**	**2012 Kawi Ninja 650R [2]**	**2014 Yamaha YZ450f [3]**
Stock Hp	*110 Hp @ 12,500 rpm*	*72.1 Hp @ 8,500 rpm*	*58 Hp @ 9,900 rpm*
Weight (lbs)	*Heaviest*	*Medium*	*Lightest*
Displacement (cc)	*599*	*649*	*449*
Peak Torque (lb-ft) (RPM)	*48 lb-ft @ 10,500 rpm*	*48.6 lb-ft @ 7,000 rpm*	*35 lb-ft @ 7,300 rpm*

Table 7-1 - Engine Specifications

Differential	**Taylor Race Engineering V2 [4]**	**Drexler V2/V3 [5]**	**Modified ATV [6]**
Cost	$2,450	$3,656	~$500
Weight (lbs)	8.9	5.9	11-21
Further Drivetrain Support (Y/N)	Y	Y	N

Table 7-2 - Differential Specifications

Name	Length of Divergent section (mm)	Type	Max Pressure Drop (Pa)
Strt_throat_1	104	De-Laval	466
Strt_throat_2	77	De-Laval	569
Strt_throat_3	89	De-Laval	532
Strt_throat_4	153	De-Laval	444
Strt_throat_5	208	De-Laval	471
MOC_throat_1	63	Bell	786
MOC_throat_2	153	Bell	412

Table 7-3

7.3 Appendix C: Equations

$$\omega_e = N_t * N_f * \omega_w \quad \frac{\omega_e}{(N_t * N_f)} = \omega_w$$

$$V_x = \omega_w * r$$

Equation 37-1 - Speed Equations

From equation (1) and **equation (1b) from the accelerating page**, we get: (more)

$$v_{max} = \sqrt[3]{\frac{q}{2} + \sqrt{\frac{q^2}{4} + \frac{p^3}{27}}} + \sqrt[3]{\frac{q}{2} - \sqrt{\frac{q^2}{4} + \frac{p^3}{27}}} \qquad (9)$$

Where:

$$q = \frac{P_t}{0.5\rho\,(C_D A - f_r C_L A)} \qquad (9.1)$$

$$p = \frac{f_r m g}{0.5\rho\,(C_D A - f_r C_L A)} \qquad (9.2)$$

Equation 37-2- Top Speed with Resistance

$$\dot{m} = \frac{p_o A^*}{\sqrt{T_o}} \sqrt{\frac{\gamma}{R}\left[\frac{2}{\gamma+1}\right]^{\left(\frac{\gamma+1}{\gamma-1}\right)}}$$

Equation 73-4 - Choked Flow Rate Formula

$$\frac{A}{A^*} = \frac{1}{M}\left(\frac{1 + \frac{k-1}{2} M^2}{1 + \frac{k-1}{2}}\right)^{\frac{k+1}{2(k-1)}}$$

Equation 37--53 - Area Ratio for Rocket Nozzle

$$\dot{m}_{fuel} = \frac{Power}{LHV * \eta_{thermal}}$$

Equation 7-4 - Power Formula

$$Q_{t,intake} = \frac{V_T \cdot N}{2 \cdot 60} \cdot V_E$$

Equation 3-7 - Intake Volumetric Flow Rate7-5

$$\rho_i = \frac{P}{R \cdot T_i}$$

Equation 7-68 – Intake Air Density

$$\dot{m}_{air} = \rho_i \cdot Q_{t,intake}$$

Equation 3-9 – Intake Mass Flow Rate

$$Q_{t,exhaust} = \frac{\dot{m}_{air} \cdot R \cdot T_e}{P}$$

Equation 3-10 – Exhaust Volumetric Flow Rate

$$Q_{c,exhaust} = \frac{Q_{t,exhaust}}{n_{cyl}}$$

$$A_r = \frac{Q_{c,exhaust}}{v_{target}}$$

Equation 3-11 – Volumetric Flow Rate per Cylinder *Equation 3-12 –*
Runner Area

$$\phi = \frac{AFR_{stoich}}{AFR}$$

Equation 3-13 – AFR Equivalency Ratio

$$y_i = \frac{n_i}{\sum n_i}, \quad i = CO_2, H_2O, O_2, N_2, CO$$

Equation 3-14 – Mole Fraction of Exhaust

$$C_{p,mix} = \sum_i y_i \cdot C_{p,i}(T)$$

Equation 3-15 – NASA Glenn's polynomials

$$R_{mix} = \frac{R_u}{MW_{mix}}$$

Equation 3-16 – Gas Constant

$$\gamma = \frac{C_{p,mix}}{C_{p,mix} - R_{mix}}$$

Equation 3-17 – Specific Heat Ratio

$$a = \sqrt{\gamma \cdot R_{mix} \cdot T}$$

Equation 3-18

$$L = \frac{a \cdot 120}{N}$$

Equation 3-19

$$\sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

Equation 3-20

7.4 Appendix D: Matlab

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%% Written by: Liam O'Connor %%%%%%%%
%%% Title: Max Restrictor Flow calculation and Max horsepower before Choked
%%% Flow
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%5
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%            Restrictor maximum flow rate calulation
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
clc
A=0.00031415; % m^2  x-section at throat
y=1.4; % specific heat at 294K
t=300; % Kelvin temp at inlet
p=101325; % Pascal Pressure at inlet
r=287; % gas constant
m=(p*A)/(sqrt(t));
n=(y/r)*((2/(y+1))^((y+1)/(y-1)));
m_max=m*(sqrt(n)); %maximum mass flow rate through restrictor (kg/s)
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%% Max Horsepower Before Choked Flow %%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
LHV=44; % low heating value for 87-93 octane 44-46 Mj/kg
https://world-nuclear.org/information-library/facts-and-figures/heat-values-of
-various-fuels
AFR=14.7; %% Air Fuel ratio
eth=0.25; %thermal efficiency
mdotair=NaN; %initializing mdotair
last_mdotair = NaN; %initializing last_mdotair
for HP=30:1:120 %Horsepower
   power=HP*0.0007457;%conversion from horsepower to MegaWatt
   mdotfuel=power/(LHV*eth); %calc mdotfuel
   mdotair=mdotfuel*AFR; % mass flow rate of air

   if mdotair>m_max
         fprintf('Limit reached at %d HP\n',HP);
       fprintf('Last valid mdotair = %.6f kg/s\n', last_mdotair);
       break
   end
   last_mdotair=mdotair;
end
```

```matlab
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Written by: Marshall Fritz
% Title: Exhaust Diameter Calculations Based on Ideal Gas Law
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%             Exhaust Diameter Calculation
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
clc; clearvars; close all;
% USER INPUTS
V_T_cc   = input('Enter total engine displacement [cc]: ');
N        = input('Enter target engine speed [rpm]: ');
VE       = input('Enter volumetric efficiency (0-1): ');
v_target = input('Enter target mean runner velocity [m/s]: ');
n_cyl    = input('Enter number of cylinders: ');
T_i      = input('Enter Intake temp [C]:') + 273;    % Convert to K
T_e      = input('Enter Exhaust temp [C]:') + 273;   % Convert to K
P_atm    = input('Enter Air Pressure [atm]:');
% CONSTANTS
R = 287;                                      % J/kg·K
P = P_atm * 101325;                           % Convert pressure from atm to Pa
% CALCULATIONS
V_T = V_T_cc * 1e-6;                          % Total engine displacement [m^3]
Qt_intake = (V_T * N / (2*60)) * VE;          % Total intake flow [m^3/s]
rho_i = P / (R * T_i);                        % Density of Intake Air [kg/m^3]
m_dot_air = rho_i * Qt_intake;                % Mass Flow Rate of Air [kg/s]
Qt_exhaust = (m_dot_air * R * T_e)/P;         % Total exhaust volumetric flow [m^3/s]
Qc_exhaust = Qt_exhaust / n_cyl;              % Per-cylinder exhaust volumetric flow [m^3/s]
A_r = Qc_exhaust / v_target;                  % Runner area [m^2]
d_r = sqrt(4*A_r/pi)*1000;                    % Runner diameter [mm]
A_col = n_cyl * A_r;                          % Collector area [m^2]
d_col = sqrt(4*A_col/pi)*1000;                % Collector diamater [mm]
% OUTPUT
fprintf('\nExhaust Sizing Results\n');
fprintf('Runner Diameter: %.1f mm\n', d_r);
fprintf('Collector Diameter: %.1f mm\n', d_col);
fprintf('Total Exhaust Volumetric Flow Rate: %.3f m^3/s\n', Qt_exhaust);
```

```
% Kawasaki Ninja 650
% peak torque occurs at 7000 rpm
% peak power occurs at 8500 rpm
% input for new bikes
RPM = input('Enter the max rpm value: '); % rev/min
% input for sprocket size
fsprocket = input('Enter the number of teeth on the front sprocket: ');
rsprocket = input('Enter the number of teeth on the rear sprocket: ');
FR = rsprocket/fsprocket; %final drive ratio
g1 = 2.438;
g2 = 1.714;
g3 = 1.333;
g4 = 1.111;
g5 = 0.966;
g6 = 0.852;
rear_r = 16/2; % rear radius in inches (Wheel+tire)
% Calculate the wheel rotational speed for each gear
% first gear
g1wheel = ((RPM*2*pi)/60)/(g1*FR);
g2wheel = ((RPM*2*pi)/60)/(g2*FR);
g3wheel = ((RPM*2*pi)/60)/(g3*FR);
g4wheel = ((RPM*2*pi)/60)/(g4*FR);
g5wheel = ((RPM*2*pi)/60)/(g5*FR);
g6wheel = ((RPM*2*pi)/60)/(g6*FR);

%convert wheel speed to ground speed using wheel radius and convert units to
% mph
speed1 = ((g1wheel * rear_r)/12)*(3600/5280);
speed2 = ((g2wheel * rear_r)/12)*(3600/5280);
speed3 = ((g3wheel * rear_r)/12)*(3600/5280);
speed4 = ((g4wheel * rear_r)/12)*(3600/5280);
speed5 = ((g5wheel * rear_r)/12)*(3600/5280);
speed6 = ((g6wheel * rear_r)/12)*(3600/5280);

% Display the calculated speeds for each gear
fprintf('Top Speed in 1st gear: %.2f mph\n', speed1);
fprintf('Top Speed in 2nd gear: %.2f mph\n', speed2);
fprintf('Top Speed in 3rd gear: %.2f mph\n', speed3);
fprintf('Top Speed in 4th gear: %.2f mph\n', speed4);
fprintf('Top Speed in 5th gear: %.2f mph\n', speed5);
fprintf('Top Speed in 6th gear: %.2f mph\n', speed6);
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%PROGRAM: TOPSPEED
%%%PROGRAMMER: TRENT GREENE
%%%%%%%%%%%%%%%%%%%%%%%%%%
clear all ; clc;
%PARAMETERS
output_torque = 48.679; %lb-ft, dependent on motor
max_torque_rpm = 7000; %rpm
engine_horsepower = 72.1;% dependent on motor
engine_kW = engine_horsepower / 1.34102; %in kW
engine_torque_standard = 47.2 %lb-ft
```

```
engine_torque_standard =
47.2000
```

```
engine_torque_metric = engine_torque_standard / 1.34102;
max_horsepower_rpm = 8500; %rpm
front_sprocket_teeth =  [12:1:17]; %cycles through available sprocket sized from
taylor fsae catalog
back_sprocket_teeth = 36:1:54; %same as above, this goes on diff
tire_diameter = 16; %inches, this is up for debate; from Hoosier catalog for FSAE
trans_gear_ratio = 0.852; %in final gear (6th for 6 speed)
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%EQUATIONS
tire_circumference =  pi * tire_diameter / 12; %tire circumference in feet
topspeed_torque = zeros(length(front_sprocket_teeth), length(back_sprocket_teeth));
topspeed_power = zeros(length(front_sprocket_teeth), length(back_sprocket_teeth));
```

```
% Theoretical top mechanical speed @ max hp and torque rpms
for i = 1:length(front_sprocket_teeth)
    for j = 1:length(back_sprocket_teeth)
        topspeed_torque(i,j) = max_torque_rpm / [(back_sprocket_teeth(j) /
front_sprocket_teeth(i)) * trans_gear_ratio] * tire_circumference * 60 / 5280;
%miles per hour
        topspeed_power(i,j) = max_horsepower_rpm / [(back_sprocket_teeth(j) /
front_sprocket_teeth(i)) * trans_gear_ratio] * tire_circumference * 60 / 5280;
    end
end
figure(1)
heatmap(back_sprocket_teeth, front_sprocket_teeth, topspeed_torque);
```

```
g = -9.801; %m/s^2
vehicle_mass = 250; %kg, rough ballpark
rolling_resistance = 0.02 * vehicle_mass * g * 4; %ballpark standard for fsae
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%top speed calculation
for i = 1:length(front_sprocket_teeth)
    for j = 1:length(back_sprocket_teeth)
        torque_tires(i,j) = engine_torque_metric * (back_sprocket_teeth(j) /
front_sprocket_teeth(i)) * trans_gear_ratio;
    end
end
q = tire_power_kW/ (0.5 * air_density * (drag_coeff - rolling_resistance *
lift_coeff));
p = rolling_resistance * vehicle_mass * g / (0.5 * air_density * (drag_coeff -
rolling_resistance * lift_coeff));
topspeed_friction = ((q / 2 + (q^2 / 4 + p^3 / 27)^(1/2))^(1/3) + (q / 2 - (q^2 / 4
+ p^3 / 27)^(1/2))^(1/3)) * 2.236936; %converted to hp at end
correction_factor = topspeed_friction ./ max(topspeed_power,[],'all');
topspeed_corrected = correction_factor .* topspeed_power;
figure(3)
heatmap(back_sprocket_teeth, front_sprocket_teeth, topspeed_corrected);
```

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%             Restrictor maximum flow rate calulation
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
clc
A=0.00031415; % m^2  x-section at throat
y=1.4; % specific heat at 294K
t=300; % Kelvin temp at inlet
p=101325; % Pascal Pressure at inlet
r=287; % gas constant
m=(p*A)/(sqrt(t));
n=(y/r)*((2/(y+1))^((y+1)/(y-1)));
m_max=m*(sqrt(n)); %maximum mass flow rate through restrictor (kg/s)
%%%%%%%%%%%%% Hp calcs
LHV=44; % low heating value for 93 octane 44-46 Mj/kg
AFR=14.7; %% Air Fuel ratio
eth=0.25; %thermal efficiency
mdotair=NaN; %initializing mdotair
last_mdotair = NaN; %initializing last_mdotair
for HP=30:1:100 %Horsepower
   power=HP*0.0007457;%conversion from horsepower to MegaWatt
   mdotfuel=power/(LHV*eth); %calc mdotfuel
   mdotair=mdotfuel*AFR; % mass flow rate of air

   if mdotair>m_max
         fprintf('Limit reached at %d HP\n',HP);
       fprintf('Last valid mdotair = %.6f kg/s\n', last_mdotair);
       break
   end
   last_mdotair=mdotair;
end
%amount of fuel
fprintf('\n Amount of Fuel\n')
power=75*0.0007457;%conversion from horsepower to MegaWatt
mdotfuel=power/(LHV*eth) %calc mdotfuel [kg/s]
fprintf('Kg/s\n\n')
FOS = 1.1;
Time = 23.15 * 60 * FOS; %1st place time convert to sec * Factor of slowness
fuelInKgs = Time * mdotfuel;
fuelInGal = fuelInKgs * 2.2 /6.3 %gallons
fprintf('Gallons\n')
%{
FSAE fuel tanks typically range from about 5 to 10 liters
(approximately 1.3 to 2.6 US gallons)
%}
```

```
%%%%%%%%%%% Solid Shaft %%%%%%%%
%initialize variables
iter = 1;
d =2;%mm
%Constants
%material: medium steel
FOS = 1.5; % Factor of safety
Sy = 450e6; % Yield strength of medium steel in MPa
T= 66;%MNm @6,900 RPM
M = T/.3302 * 0.508; %Tangent force in tire (Torque/ 13" overall tire) * lever
arm(20" axle with 3deg shaft misalignment)
%Calcs
StressAllow = Sy/FOS;
dm = d*1e-3; %convert d to m
J = pi()*dm^4/32; %Polar moment of intertia
I = pi()*dm^4/64; %Area moment of intertia
c = dm/2; % Outer radius m
ShearS = T*c/J;
BendingS = M*c/I;
Stress = sqrt(3*ShearS^2+BendingS^2); % Calc Von Mises Stress
while (Stress > StressAllow)
d = d+0.1;
dm= d*1e-3;   %m
iter = 1+iter;
J = pi()*dm^4/32; %Polar moment of intertia
I = pi()*dm^4/64; %Area moment of intertia
c = dm/2; % Outer radius m
ShearS = T*c/J;
BendingS = M*c/I;
Stress = sqrt(3*ShearS^2+BendingS^2);
end
```

```matlab
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Written by: Marshall Fritz                                              %
%    Exhaust Runner Length Based on Specific Heat Ratio of Exhaust Gas    %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%                        Runner Length Calculation                        %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Inputs
T_C = input('Enter exhaust gas temperature [°C] (625-825°C): ');
AFR = input('Enter air-fuel ratio: ');
rpm = input('Enter engine speed [RPM]: ');
T = T_C + 273.15;  % Convert to Kelvin
% Stoichiometry
AFR_stoich = 14.7;       % Stoichiometric AFR
phi = AFR_stoich / AFR;  % Equivalence ratio
% 93 Octain Gasoline Composition
nC = 8; nH = 18;
nO2_stoich = nC + nH/4;
nO2_actual = nO2_stoich / phi;
nN2 = nO2_actual * 3.76;
% Determine combustion products
if phi <= 1
    n_species = [nC, nH/2, nO2_actual-nO2_stoich, nN2, 0];  % [CO2, H2O, O2, N2, CO]
else
    n_species = [nC/phi, nH/2, 0, nN2, nC - nC/phi];        % [CO2, H2O, O2, N2, CO]
end
species = {'CO2','H2O','O2','N2','CO'};
MW = [44.01, 18.02, 32.00, 28.01, 28.01];  % g/mol
% Mole fractions
y = n_species / sum(n_species);
% NASA TP-2002 High-Temp Coefficients
a.CO2 = [3.85746  0.004414 -0.000002215  5.2349e-10 -4.7208e-14];
a.H2O = [3.034    0.002177 -1.641e-07    -9.7042e-11  1.682e-14];
a.O2  = [3.283    0.001483 -7.580e-07    2.095e-10   -2.167e-14];
a.N2  = [2.953    0.001397 -4.926e-07    7.860e-11   -4.608e-15];
a.CO  = [3.048    0.001352 -4.858e-07    7.885e-11   -4.698e-15];
R_u = 8.3145;  % J/mol·K
% Compute Cp for each species
Cp = zeros(1,5);
for i = 1:5
    coeffs = a.(species{i});
    Cp(i) = R_u * (coeffs(1) + coeffs(2)*T + coeffs(3)*T^2 + coeffs(4)*T^3 + coeffs(5)*T^4);
end
% Mixture Properties
Cp_mix_mol = sum(y .* Cp);
MW_mix = sum(y .* MW);
R_mix = R_u / (MW_mix / 1000);          % J/kg·K
Cp_mix_mass = Cp_mix_mol / (MW_mix / 1000);
gamma = Cp_mix_mass / (Cp_mix_mass - R_mix);
% Speed of Sound
a_sound = sqrt(gamma * R_mix * T);      % m/s
% Runner Length
a_ft = a_sound * 3.28084;               % convert to ft/s
L_in = (a_ft * 120)/rpm;                % inches
L_m = L_in * 0.0254;                    % meters
% Results
fprintf('Speed of sound:    %.1f m/s\n', a_sound);
fprintf('Runner Length:     %.4f m\n', L_m);
```