Small Engine Dynamometer
Team # 14

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Executive Summary:

Within the contents of this report, the details regarding designing, manufacturing and testing an engine dynamometer are presented. Requested by the project sponsor, the dynamometer must be able to test up to 107 horsepower and 12,000 engine rpm. This ensures the system will be able to handle the currently used Yamaha R6 engine and allow the sponsor to test electric motors if the change to an electric drivetrain is made. The finished project must also be compact and fit within a 2'x4' envelope for easy storage within the team room on campus.

The entire project is divided into four main subsystems: frame, driveline, hydraulics and electronics. The frame is composed of 1" square tubing laid out to house all other subsystems. 14-gauge sheet also acts to protect the operator and creates a tabletop to house the hydraulic control valve and serves as a platform to lay tools, laptops and miscellaneous items. The frame sits on top of four castors which allow for easy transportation of the dynamometer.

The hydraulic system comprises a Hydreco 9WL pump, a Parker F2020s Hydraulic flow control valve and a Parker pressure relief valve. This system creates 58.88 gpm of flow at maximum operating conditions. The maximum pressure the pump can handle is 3000 psi. Therefore, hydraulic hose rated up to 3500 psi was chosen to connect the hydraulic system.

The driveline utilizes a 22-tooth, 5/8" pitch sprocket to transfer load from the engine to the hydraulic pump from the Yamaha R6's 520 roller chain. The pump has a 7/8" 13-tooth spline input shaft which is connected to a 1" keyed driveshaft through the use of a custom adaptor. 1" bearings appropriately house the driveshaft and allow for up to 7000 rpm.

The electronics feature an S-type load cell and hall effect sensor to measure rpm. An Arduino uno allows for the data measured on each dynamometer test to be read and recorded in real time. This data then can be further processed to graph horsepower and torque against rpm.

The entire system was successfully manufactured and tested. The testing was improvised with a Honda CRF 450r. Most notably, the values recorded by the dyno match up with sources providing dyno runs on the same dirt bike. The values recorded between test runs were found to be similar. Concluding that the end product is repeatable and accurate.

Acknowledgements:

Team 14 would like to acknowledge a few people that went out of their way to help during the manufacturing process. Specifically, Larry for help with the frame welding, Kyle for help with the driveline assembly welding. Dan for giving his time to turn the driveline bushing and offer his lathe for use. Clint for manufacturing design review and insightful conversation during the designing process. Brandon for thoughtful discussion in assembly design. The racing team for sponsoring this capstone project.

1 Project Overview

1.1 Background and Motivation

The world of motorsports and open-wheel racing aims to push the boundaries of what a vehicle is capable of. Racing teams all share one common goal; creating a faster racecar that outperforms every opponent. The concept of the perfect racecar can be broken down into many key parameters. From straight-line acceleration, cornering speeds, aerodynamics and its drag coefficient and many others. All these parameters play a role in improving a racing team's car.

The Society of Automotive Engineers (SAE) organize and host an open-wheel racing competition every year. This competition is designed for engineering students in their undergraduate degrees to create a team and compete in a two-thirds scale formula racer competition. The competition brings out creative and unique solutions in order to achieve better racecar performance. Teams attempt to improve their cars in a variety of ways: such as shaving excess weight off components on the car or optimizing suspension geometry to increase tire grip in corners. The competition itself declares a large rule set in order to maintain safety and to create an even competition among teams. This leads into the FSAE rule on engine size and displacement. All teams are limited to a maximum of 710 cubic centimeters (cc) displacement for their engines and all engines must use a 19mm air intake restrictor. This leads to relatively lower power numbers for these engines compared to what their stock forms are capable of.

Because of the restrictions regarding the racecar's power unit, it is valuable to extract as much power as possible. The main output parameters from an internal combustion engine are torque, horsepower and revolutions per minute. Horsepower being a function of the other two parameters.

$$Horsepower = \frac{Torque\ x\ RPM}{5252} \tag{1}$$

Where torque is measured in lb-ft and RPM is the revolutions per minute of the engine's crankshaft. The constant is derived from converting the power into horsepower.

Internal combustion engines can have their onboard computers modified in order to extract more power than originally available from the factory. This is a process known as tuning the engine and can be used in a variety of ways. Commuters can have their engines tuned to improve fuel economy and provide a smoother power output for their daily commutes to work. Meanwhile racers can use tuning in order to extract as much horsepower and torque that the engine can provide and improve throttle response.

In this capstone project's case, the tuning available to the sponsor is a process known as engine remapping. The process involves modifying fuel injector activation time. If the fuel injector is left on for a longer time, more fuel is delivered into the cylinder. The quantity of air and fuel in the engine leads to a ratio known as the Air-to-Fuel ratio (AFR). The AFR is the ratio of air mass to fuel mass that becomes mixed within the cylinder. And this leads to the main tuning technology that will be explored in this capstone. Car and Bike tuning shops modify a variety of engine parameters with their most important change being ignition timing. The Schulich Racing FSAE team does not attempt any ignition tuning as it can lead to devastating failures such as bent valves from cylinder to valve contact. The AFR tuning method can still lead to an increase in horsepower and torque in a much safer manner. The only potential issue to monitor is that there is no engine knock occurring when the AFR is raised. Engine knock is the pre-detonation of the air and fuel mix within the cylinder. This however is not as immediately devastating as contact between moving components and can be reversed if issues are found.

The following figure is one of the fuel maps used by the Schulich Racing team. This gives insight into the tuning method currently available for the team and has been used to guide ideas for the capstone dynamometer's control system.

Primary - Fuel Table - Injector Open Time (ms)

Enable Fuel | Setup Tracer | Clear Tracer

RPM

TPS (%)

	0	325	650	975	1300	1625	1950	2275	2600	2925	3250	3575	3900	4225	4550	4875	5200	5525	5850	6175	6500	6825	7150	7475	7800
100.0	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00
96.0	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81	5.81
92.0	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63	5.63
88.0	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44	5.44
84.0	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25	5.25
80.0	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06	5.06
76.0	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88	4.88
72.0	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69	4.69
68.0	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50
64.0	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31	4.31
60.0	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13	4.13
56.0	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94	3.94
52.0	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75	3.75
48.0	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56	3.56
44.0	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38	3.38
40.0	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19	3.19
36.0	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00
32.0	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81	2.81
28.0	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63	2.63
24.0	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44	2.44
20.0	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25	2.25
16.0	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06	2.06
12.0	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88	1.88
8.0	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69	1.69
4.0	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50
0.0	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31	1.31

Figure 1: Fuel Map for the Performance Electric's PE3 ECU

The values located within the array can be changed in order to change the AFR for specific engine rpms and throttle position. Which can be interpreted as different loading scenarios for the engine. Low RPM and Low TPS % would signify that the engine is in an idle state, while increasing RPM and TPS % indicates the engine is accelerating.

1.2 Problem statement

Top teams in the formula student world spend countless hours testing and tuning their engines to get the highest performance they can with the restrictions set by the rules. These teams have either purchased dynamometers capable of testing their engines, or have student created setups through senior design courses. The sponsor of this project, the Schulich Racing FSAE team currently has no in-house method of tuning their engines. Presently, the team is required to travel to third party workshops with working dynamometer setups to do their tuning. This presents multiple problems for the team, with the most important issue pertaining to accessibility. In order to perform steady state tuning, dynamometers (dynos) must be run for long periods of time at several engine settings. Doing this at an outside shop requires not only a considerable payment to use their equipment, but also is a burden to move personnel and equipment from one shop to another.

This project aims to solve Schulich Racing's problem of not having a readily available and easily accessible dyno by creating a functioning dynamometer. This project will allow in-house engine tuning, and the justification of modifications done to increase the racecar's horsepower.

1.3 Overview of the project's scope

The project deliverables are defined for this project are:

1. A full design and CAD assembly of the dynamometer system. With as many components modelled and included to give a proper idea of what the actual in-person system will resemble.

2. A CAD drawing package, along with dimensioning and tolerancing for the systems that will be custom manufactured. This drawing package will also serve to allow an external source to create and assemble the system in the future; in the event that this capstone group is not able to create one due to the Covid-19 pandemic.

3. Proper documentation and protocols for the system's use. As this capstone project aims to be used for many years to come; the original capstone group will not be around to train every new Schulich Racing member. The documentation will allow new members of the team to understand and operate the system effectively and safely.

4. If possible, a working functional prototype that will be able to dyno test the Yamaha R6 engine (R6) that is used by the team.

5. Dyno tests that are repeatable and give values that are expected and similar in between dyno pulls.

1.4 Definition of Terms

FSAE: FSAE stands for Formula SAE, which is the competition that Schulich Racing competes in. SAE stands for society of automotive engineers and SAE is the governing body of the formula SAE competition.

Crankshaft: The crankshaft is the part of the engine that is rotated by the movement of the pistons and is where the rotational speed and torque of the engine are measured for the values reported in literature.

Idle: Idle is the point where the engine throttle is at its closed position, and the engine produces its minimum power. Idle conditions for the R6 are 1250 RPM and 12lbft of torque.

TPS %: The percentage that the throttle position sensor reads. A value of 100% indicates that the throttle is fully engaged.

Centistokes (cSt): A unit of kinematic viscosity equal to 0.01 St which is equivalent to 1mm^2/s.

RPM: Revolutions per minute.

Gpm: Gallons per minute.

2 Conceptual Design

2.1 Background research

The purpose of a dynamometer is to measure the power produced by an engine at a specified revolutions per minute [1]. A dynamometer can be used to verify changes made to the engine to increase performance. One way to measure the power output of the engine is by measuring the torque produced at a specific revolution per minute. Then equation (1) can be used to determine the power produced by the engine.

There are two types of tests that are done on dynamometers, steady state tuning and power pulls. Steady state tuning involves holding the engine at a specific speed and changing the air fuel ratio of the engine to increase power at that specific engine speed. Steady state tuning requires a dynamometer that is able to run for long periods of time while maintaining accuracy. Additionally, steady state tuning allows users to see results of tuning in real time. Power pulls involve running the engine through its complete RPM range and measuring the power at each RPM within the range. Power pulls require a fast response time from the dynamometer in order to get accurate readings. These runs allow the user to create power curves that show optimal running conditions for the engine.

There are two main categories of dynamometers, chassis dynamometers and engine dynamometers. Chassis dynamometers keep the engine mounted in the chassis of the vehicle and the dynamometer attaches to the wheel hub of the vehicle (Figure 2: MODEL : AED-300 engine dynamometer (left) [3] Figure 3: Render of a chassis dynamometer (right) [2]). Engine dynamometers remove the engine from the vehicle and attach directly to the engine output shaft or sprocket.

The advantage of a chassis dynamometer is that it measures power directly at the wheels and as a result, accounts for all mechanical losses through the vehicles transmission and drivetrain components. The disadvantage of chassis dynamometers is that they require more space, as the full vehicle needs to be present for testing. The advantage of engine dynamometers is that they are much more compact and mobile. The disadvantage is that there is typically power lost through the transmission and final drive of a vehicle that is not present in an engine dynamometer. However, this disadvantage is mitigated when testing motorcycle or other small engines as the transmission is typically internal to the engine as opposed to external gearboxes found in cars.

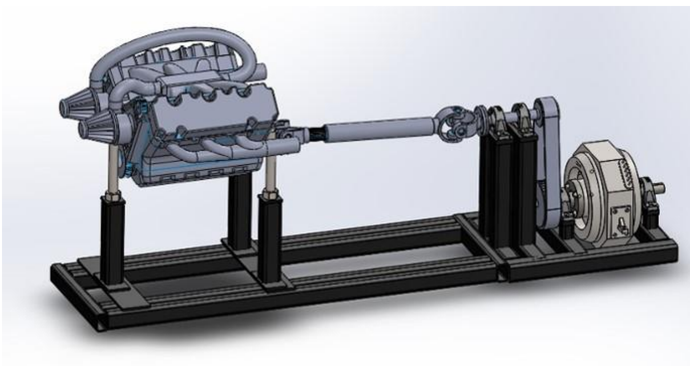

Figure 2: MODEL : AED-300 engine dynamometer

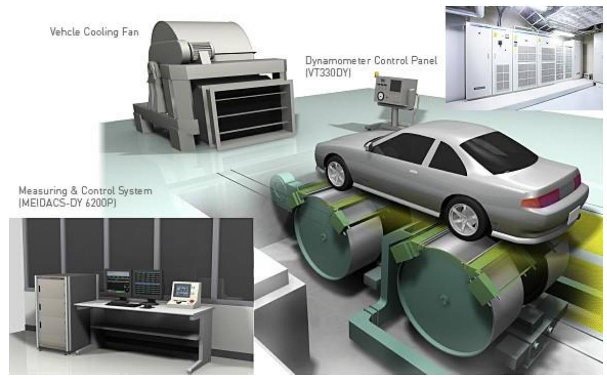

Figure 3: Render of a chassis dynamometer

For a dynamometer to work, it must have a system that is capable of applying a load on the engine. The four most popular ways of applying loads to the engine in modern dynamometers are: an eddy current brake, hydraulic brake, AC motor and inertial mass brakes [4].

Eddy current dynamometers use eddy currents produced by a rotating disk immersed in a magnetic field to apply a breaking torque to an engine [5]. The advantages of an eddy current brake are that they can withstand large amounts of braking torque, have a fast response to changing engine outputs and they are quite durable. The disadvantages of an eddy current brake are that they tend to heat up considerately during steady state tuning and often require additional cooling. Also, eddy current brakes are typically expensive to purchase or manufacture.

A water brake involves spinning an impeller submerged in water and using the energy transfer from the impeller to the fluid to apply a braking force to the engine [6]. The advantages of a water brake are that it can be used for long periods of time without heating up. They have a low inertia for the engine to overcome upon startup. Finally, there are no electrical components required to operate the brake. The disadvantages of water brakes are that they are quite expensive to purchase and manufacture.

AC motor brakes use a low inertia alternating current motor to apply a load to the engine. The advantages of AC brakes are that they have low inertia, and fast response times. The disadvantages for AC brakes are that they can only take a small range of engine horse powers before having to switch the motor.

Another way to apply a load to the engine is through a hydraulic cycle. In the hydraulic cycle, the engine powers a hydraulic pump and the amount of power required to drive the pump is dictated by the pressure in the system which is controlled using a flow control valve. In this system, the power needed to drive the pump is directly related to the hydraulic pressure and volumetric flow rate given in the following equation.

$$Horsepower = \frac{Volumetric\ Flowrate * Pressure}{1714} \tag{2}$$

Where the pressure is in pounds per square inch (psi) and the flow rate is in gallons per minute (gpm) and the constant is derived from converting into horsepower. The advantage of this kind of system are similar to those for the water brake, but with the additional advantage of being considerably cheaper to design and manufacture. The disadvantage of this kind of brake is that the selected pump must be able to handle the power range of the engine, which requires a large pump. Additionally, the hydraulic control valve can come in a variety of resolutions and must match the system requirements.

2.2 Major customer requirements / design functions / specifications / constraints

As laid out by the sponsor, Schulich Racing, the dynamometer must meet the following major constraints. These were decided by the design team and the sponsor during preliminary meetings. The most important requirement of the system is that all components can withstand a maximum horsepower output of 107 hp, engine speed of up to 12,000 rpm, and a torque output range of 12 – 47.7 lb-ft while maintaining functionality. These numbers were decided based upon the current engine setup used by the team, and the desired outputs of future electrical engines. This allows the team to safely use the dynamometer with the current Yamaha R6 engine and leaves room for potential higher output electrical engines. Accommodating if the team desires to alter their car's powertrain setup in the future. Continuing on this point, Schulich Racing is a university-based team with a roster that undergoes changes on a yearly basis. Senior members graduate and leave the roster, while brand new students are taken on by the team. The system must be simple enough that any member can familiarize themselves with it and operate it. This requires that the design team create a final product that is user-friendly to the point that new members can be trained to use it quickly and without prior knowledge of dynos. On top of this, there must be proper documentation so that team members have the understanding required to maintain or upgrade the system. The dyno must be mobile and compact, this will reduce space requirements for storing the system in the racing team's shop. Finally, the total cost of the project must be around $2,800 as approved by Schulich Racing.

2.3 Discussion of the final design idea

The final design idea chosen was the hydraulic brake system housed in a custom-built frame. This design was picked for a variety of reasons. The first being that the hydraulic system is the most cost-effective dyno. This allows our prototype to remain within the approved budget laid out by the sponsor, Schulich Racing. Another reason for this choice was the ease of manufacturing. A large majority of parts are purchasable online through suppliers such as McMaster-Carr, surplus stores and secondhand product websites such as eBay. The pieces that require custom manufacturing are procurable with the resources available to the team. Another reason for choosing the hydraulic setup is the low requirement of tooling for assembly. The assembly of the various subsystems, fasteners and connections are completable through the use of standard tooling.

Another reason for this choice revolves around the power requirements outlined by the project sponsor. This dyno must test up to 12,000 engine rpm and 107 horsepower. By selecting an appropriate hydraulic pump and flow control valve, the dyno will be able to test even higher horsepower. (Hydraulic design is further discussed in the following section) The hydraulic flow control valve also allows for ease of use. The entire system is straightforward with regards to operation, as the operator must only deal with throttle input and hydraulic oil flow restriction. Safety measures are easily implementable with only the need for a pressure-relief valve as a failsafe that will activate if the system pressure ever climbs too high.

The variability of the flow and subsequently the load on the engine allows for both forms of dyno tuning. As with all other designs, this choice of dyno allows for power pulls. Doing a full sweep of the rpm and load will provide insight into overall engine performance. Along with power pulls, steady state tuning is also possible with a hydraulic system choice. The ability to restrict flow by hand through the hydraulic flow control valve makes steady state tuning easy once the user is familiar with the dyno controls. By restricting flow and increasing load on the engine, the high load, low rpm sections of the fuel map can be accessed. The engine throttle can be kept at a constant value and the changing load dictates which fuel map section is currently under investigation. Along with live measurements from the dyno electronics, it is possible to make conclusions on how the horsepower and torque values changed. Saving the data from these dyno tests then creates the necessary framework to justify any engine tuning choices for the team. Justifications which can be presented at competition to the design judges. This allows the Schulich Racing team to score higher on their design presentation and the potential for a better overall finish.

2.4 Discussion of the performance metrics

The main goal of this project is to create a successful hydraulic brake dynamometer. Along with properly serving its purpose, there are a few more performance metrics to help gauge the final finished product.

The first performance metric involves prototype build quality. This is measured through considerations such as dimensional tolerancing, and final part inspections. All manufactured components must satisfy appropriate tolerances in order to ensure a good quality end product.

Another performance metric involves durability. The final design and prototype need to be durable enough to last years of wear and use. This involves a product that can withstand dings and hits that are bound to happen in a busy team room, once the covid-19 pandemic is over of course. The design and prototype must also be able to handle some misuse when brand new operators are trained on the system.

The final performance metric involves sponsor satisfaction. This is an ongoing metric that is confirmed through continuous sponsor project updates and feedback. Throughout the entire project, the sponsor was kept in continuous feedback and information on developments and updates about the project.

3 Design Development

3.1 Discussion of the overall design

A hydraulic brake dynamometer was chosen for the final design because it was the most cost-effective option for the range of power input from the engine. Additionally, the design of this kind of dynamometer can be easily designed to be compact and would easily fit inside the footprint of the frame. One of the major challenges with this design is how to have the pump both supported and free to rotate at the same time. Another challenge with this design is sourcing a pump that can handle the required power from the engine. The main components of the final design are the frame, hydraulic system, driveline, and data acquisition system.

3.2 Description of key design components

3.2.1 Frame

The frame of the dyno is the base of the whole system, all sub-systems must be able to function properly and have enough room to be modified or maintained. The frame design is compact enough to be stored in the sponsor's workshop while also accommodating every system.

1" Mild steel square tubing was used for the frame's structural components, welded together using TIG welding with ER-70 filler rod. The steel tubing used is a familiar material to the design team, as it is used in almost every Schulich Racing vehicle. On top of this, the mild steel is cheap, readily available and its material properties will be more than adequate for the purposes of our prototype. In contrast to the usual racing goals of the sponsor, large amounts of weight are not a concern. This makes the decision to use steel much easier, as it has desirable strength-to-weight ratios. All expected loads were simulated using this material type, and it was found suitable for our purposes. Finally, a large advantage of using steel is its weldability. Steel is relatively simple to weld and the design team has experience working with it. The full cost of the steel on the frame was $333.10 CAD and its final dimensions are 2 feet wide, 4 feet long and 3 and a half feet high.

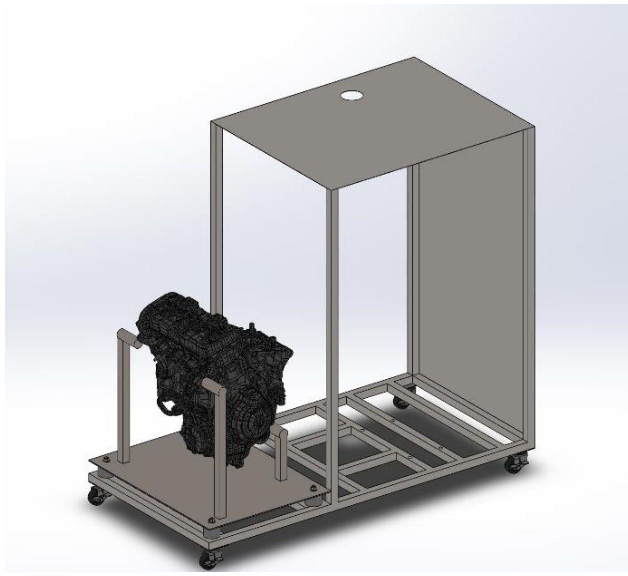

Figure 4: Final Frame Design

Another major requirement of the frame is modularity. It cannot only handle the current engine used by the team. It needs to be able to accommodate any reasonably higher output engines in the future. With the racing world moving further towards carbon neutral or fully electrical powertrain setups, the need for a modular setup was prioritized. To satisfy this, a hands-off approach was decided to be the best course of action for the scope of this design project. This setup involves having the sponsor create engine mounts for any future engines that can be mounted straight to the frame of the dyno. This solution was acceptable to the sponsor, as this is already a standard practice for the team when they perform initial tests of a new engine. An engine mounting area was designed on the frame along with the rest of the dyno. This area is 2 feet wide and 1 and a half feet in length and can fit any engine type the sponsor may run. To mount the engine to the frame, a custom-made mount must be used. The R6 engine currently in use by the team has a stand already made.

As this design involves using an engine with speeds up to 12000 rpm, the consideration of vibration damping was necessary. The design team has decided that the dyno frame must deflect no more than 5 mm in order to ensure the chain attached to the drivetrain and engine stays aligned and keeps proper tension. To minimize the effect of the vibrations created by the engine, the engine mounts are fastened to the dyno frame using Neoprene vibration-damping sandwich mounts. These mounts allow for 1/8 inch of deflection under a 175 lb single load, they also provide 5/8 inch of M10 x 1.5 mm thread to bolt the engine mount to.

All sub-systems of the frame are attached with various fasteners. The driveshaft is mounted via nut and bolt, the hydraulics are mounted with bolts and hex screws, and the electronic housings were mounted with 3d printed components attached to the frame. The decision to use fasteners as opposed to welding components is to ensure removing any part for maintenance or modification is simple and does not require breaking components. The final design of the frame leaves enough room for every subsystem to function properly while also allowing connections to be close. Another benefit of the size of this frame is that it is easy to remove and add systems to the frame if needed. The final design of the frame and all of its subsystems is shown below in figure 5:

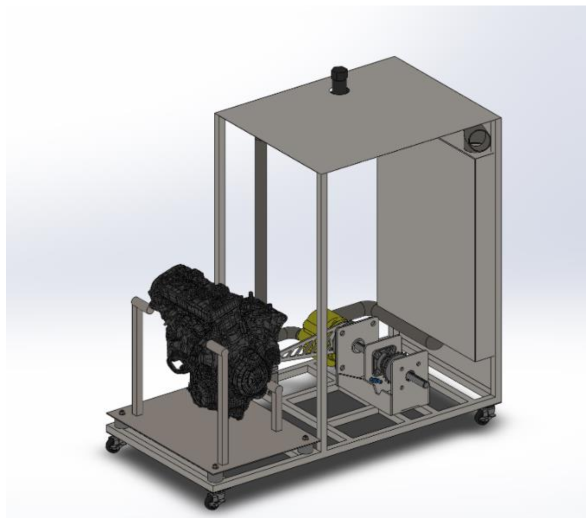

Figure 5: Full CAD Assembly

3.2.2 Hydraulics

The hydraulic system is made up of the following key components: the hydraulic pump, the hydraulic flow control valve, the pressure relief valve, and the hydraulic reservoir. The hydraulic pump will initiate flow into the control valve which will restrict the flow downstream and create back pressure. The back pressure will cause the engine to have to work harder for the system to maintain its pressure and flowrate. From the control valve, the fluid will flow into the reservoir which will be sized appropriately to reduce temperature buildup in the oil. The pressure relief valve's purpose is to relieve pressure should the system pressure exceed the 3000psi limit on the control valve. Figure 6 below shows the plumbing schematic for the hydraulic system.

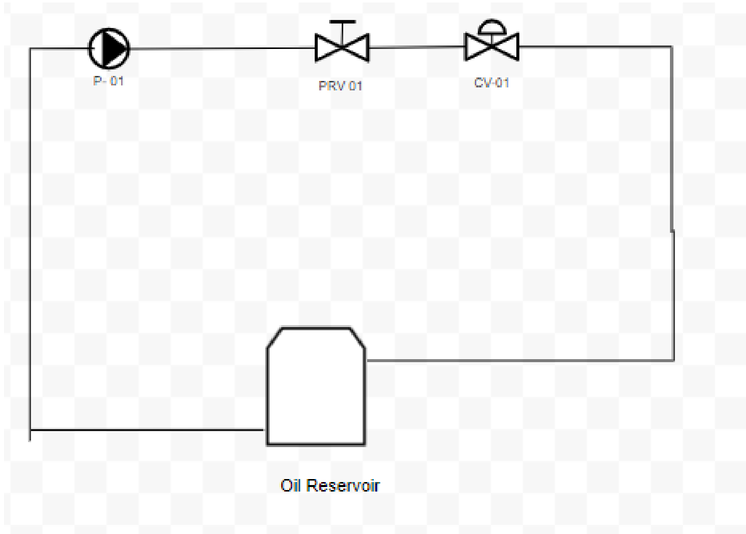

Figure 6: Hydraulic system schematic

For this system, the Hydreco 9WL-190503 hydraulic pump was chosen. This pump was chosen for its mix of specifications which allow it to meet the system's requirements. The pump features a 4.53 cubic inch displacement, 3000 rpm maximum speed and a pressure limit of 3000 psi. Using the following equations, the maximum horsepower the pump can withstand is calculated, assuming an efficiency of 80%.

$$GPM = \frac{Displacement\,(cu.in)\ x\ RPM}{231} \tag{3}$$

$$HP = \frac{GPM\ x\ Pressure}{1714\ \ x\,Efficiency} \tag{4}$$

The maximum flowrate for the pump is 58.88 gallons per minute, this will be important later on in selecting the hydraulic flow control valve. And the maximum horsepower is 121.43HP needed to drive the pump to maximum capacity. This meets the requirement of our system being able to dyno up to 107 hp specified previously.

This pump was also the most reasonably priced with a total cost of $491.02. This allows the capstone group to remain within the budget that has been approved by the sponsor. The pump also features SAE 16 inlet and outlet threads which can be fitted with standard off-the-shelf fittings for connections.

Figure 7: The Hydreco 9WL-190503 Hydraulic Pump. [9]

The main requirement for the hydraulic control valve is that it must be able to withstand the flow rate and pressure produced by the pump. Cracking pressure in a control valve is the minimum upstream pressure that the control valve requires to be able to control the flow. In order for the low-end power of the engine to be tested, the control valve will require a low cracking pressure. For this system, the Parker F2020S control valve was selected. This control valve is able to withstand 70gpm at 3000psi [10]. As a result, this control valve can withstand the 58.88gpm flow rate and 3000psi pressure produced by the pump. Additionally, the cracking pressure for this control valve is 5psi. Also, this control valve can operate in temperatures between -40°C and 121°C without the nitrile seal breaking. The valve allows flow in one direction only. The control knob includes a set screw to prevent the knob from drifting while the dynamometer is doing steady state tuning. Also, the coloured rings around the stem as shown in figure 7, will allow for a chart to be made correlating the colour to the flowrate, allowing for ease of use for steady state tuning. The manufacturer states that the first three turns of the knob are to make fine adjustments for lower flowrates and the next three turns open the valve completely. This allows for a higher resolution at the low to mid end of the power range and a lower resolution at the high end of the power range. However, during the FSAE competition the formula car operates in the lower to mid-range of the engines power due to rule restrictions on the air intake of the engine. This allows the operator to accurately and finely control the dyno powered by the R6. The total cost of the control valve was $174.17 including shipping costs and import fees.

Figure 8: Parker F2020s control valve [11]

The final major component in the hydraulic system is the pressure relief valve. The primary purpose of this valve is to ensure the system remains in a safe operating pressure. The control valve is essentially a three-way valve that allows flow only in one direction until the pressure in the system exceeds a set cracking pressure. Once the cracking pressure is exceeded in the system, the pressure relief valve then opens, and flow is diverted into the other channel allowing the oil to bypass the control valve and flow directly into the reservoir. Allowing the system to release the excess pressure without damaging any of the pressure sensitive components such as the control valve. For this project, the pressure relief valve selected was the Parker RPL-16-A, which has a range of cracking pressure from 2500 – 5000 psi. The pressure relief valve is set to a cracking pressure of 2500 psi ensuring that while the system is operational, no damage can be done to the control valve.

The fluid flowing through the hydraulic system was determined based on the aforementioned components. The main properties of the fluid of importance are the fluid's density, specific heat capacity and its viscosity. These properties determine the heat transfer, and the flow properties of the fluid. Typical hydraulic systems use ISO 32, ISO 46 and ISO 68 hydraulic oils. The different grades of oil have different viscosities. For example, at 40 degrees ISO 32 has a kinematic viscosity of 32 centistokes and ISO 46 has a kinematic viscosity of 46 centistokes. The density and specific heat capacity of the hydraulic fluid depends on the make-up of the fluid. For example, a water-glycol mixture has a higher specific heat capacity and density than a synthetic hydraulic oil. A typical hydraulic gear pump requires a minimum kinematic viscosity of 25 centistokes in order to produce the flow and pressure required of the pump [12]. The average mineral oil has a specific heat capacity of 1.87-1.94 kg/kjK [13][14]. The hydraulic fluid that has been selected for this project is Certified Hydraulic Oil AW 32. This oil was chosen because it is available at Canadian Tire in large quantities for an economical price, it has a kinematic viscosity of 32 centistokes and a density of 869 kg/m^3 [15]

The piping will be flexible tubing rated for 3500psi internal pressure and will be 3/4-inch inner diameter to match the inner diameter of the ports on the pump. This kind of hose can be found at Greenline Hose in Calgary for $10 per foot. The pressure relief valve needs to open when the system pressure reaches 2500psi and will allow the flow of the hydraulic oil to go directly into the reservoir. Additionally, the fittings that will be used in the system must be schedule 40 steel or black iron so that the system pressure remains below the maximum operating pressure for these fittings.

3.2.3 Driveline

The driveline's purpose serves to transfer load and rotation from the engine to the hydraulic pump. The Yamaha R6 engine uses a #520 roller chain which equates to a 5/8" pitch chain. Therefore, an appropriate sprocket may be sized that will fit the chain and sized appropriately to reduce revolutions coming from the engine. Noting that the max engine rpm will be 12,000 rpm, a 22-tooth sprocket was chosen. This sprocket is created for a 1" keyed driveshaft. A 0.25x0.25" key being the standard size for such a configuration. The keyed shaft, key and sprocket are all readily available from McMaster and allow for a quick turnaround time. In order to guide the driveshaft, accompanying 1" bearings are used. These bearings are rated for up to 7000 rpm; therefore, they will be able to withstand the speeds for the hydraulic pump.

In order to connect the hydraulic pump to the driveshaft, a custom coupler was fabricated. The hydraulic pump features a 7/8" 13-tooth spline drive, while the driveshaft is a 1" keyed shaft. Conventional love-joy connectors were too big and cumbersome for the setup. The solution to this problem was to bore an existing coupler. A 7/8" 13-tooth coupler can be modified to mate a 1" driveshaft through the use of a lathe and boring bar. Following this, the coupler and driveshaft can be welded together in order to prevent any separation.

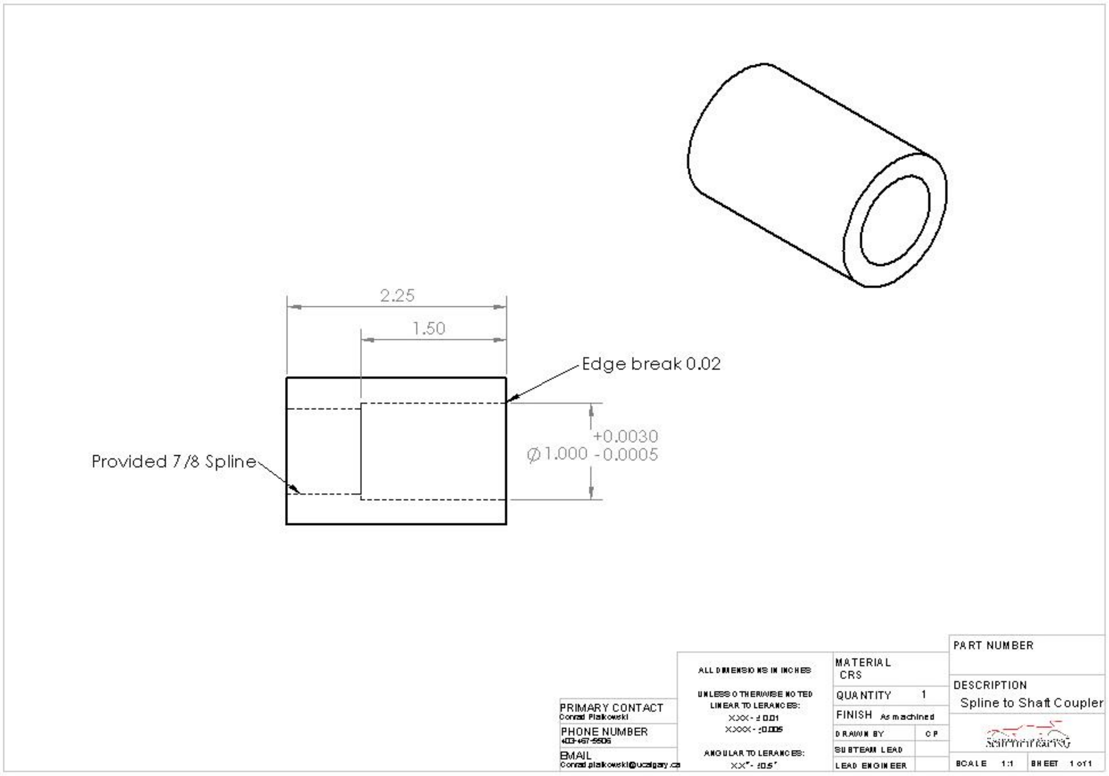

Figure 9 Spline to Shaft Coupler Machine Drawing

The next problem the driveshaft solves is the requirement of independent rotation of the pump casing. As the hydraulic pump is spun, the casing must rotate freely in order to register the load created by the engine. Without free rotation, the pump will seize and will not pump hydraulic oil. To solve this problem the moment arm incorporates a bushing. The bushing will then utilize another bearing to allow for independent rotation. The moment arm also has a welded bolt at the end in order to thread the load cell onto it. The moment arm incorporates the same bolt hole pattern that is seen with the hydraulic pump. Thus, nuts and bolts can hold the pump to the moment arm, which will be welded to the bushing. With a tight enough fit, the bushing inside the bearing can cantilever the entire hydraulic pump and moment arm. With a bored-out interior, the driveline and accompanying connector can pass through to the pump and rotate freely.

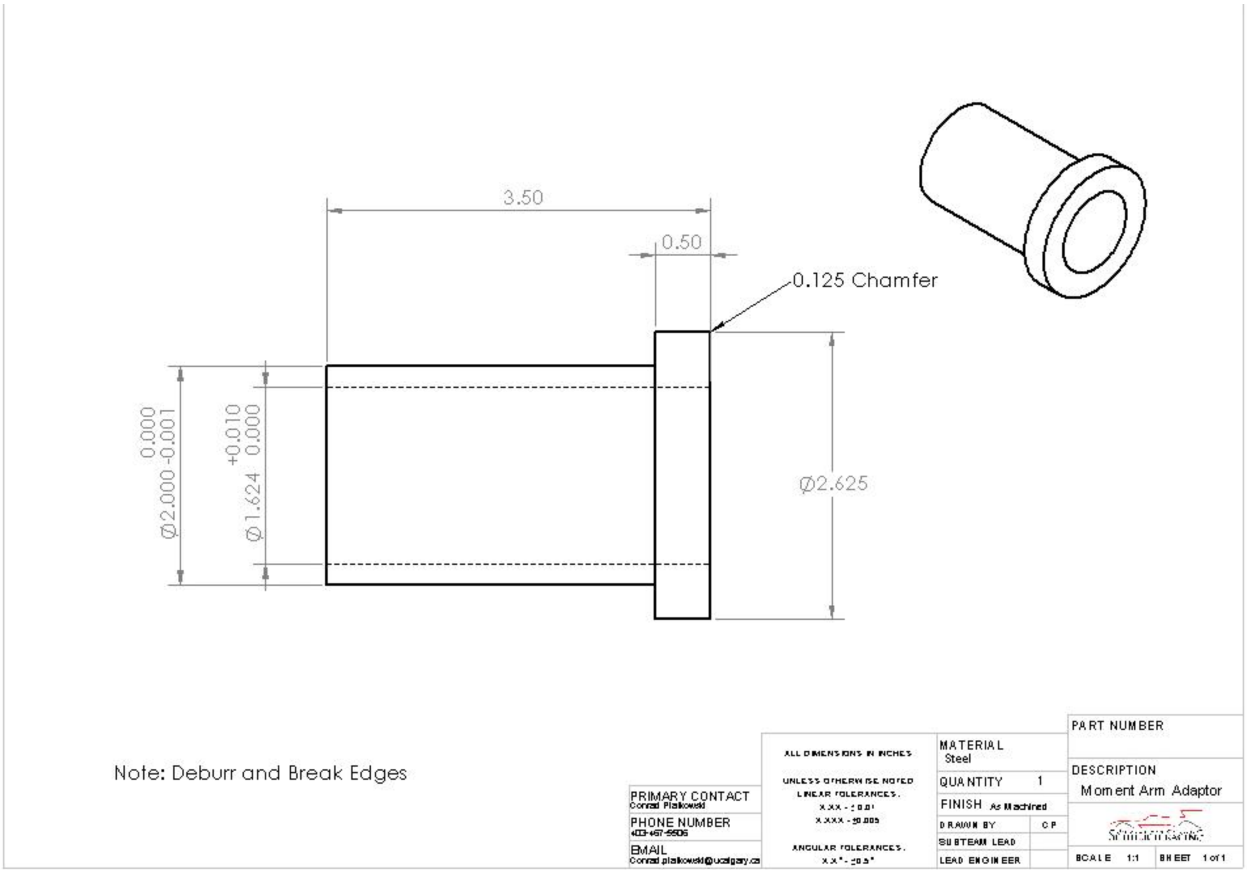

Figure 10 Bushing Machine Drawing

The bushing incorporates a tightly toleranced bearing fit in order to match a 2” bearing. The opposite side sized to 2.625” matches the diameter on the hydraulic pump face. The tight fit between the bushing and the bearing allows for the cantilever of the hydraulic pump. A 1.624” inside bore allows for the driveshaft and coupler to pass through with additional clearance.

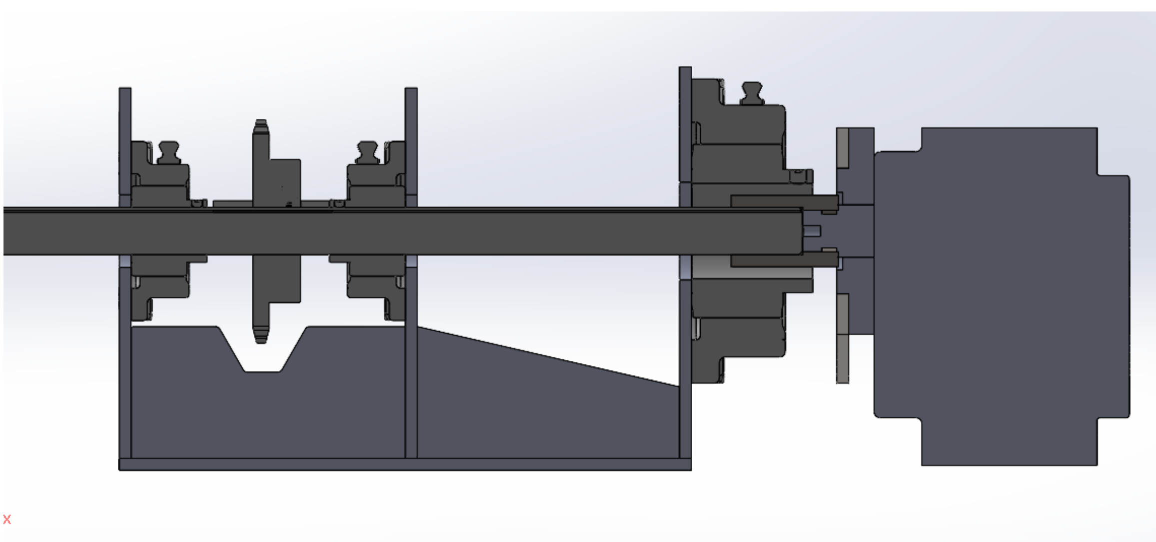

Figure 11 Section View of the Driveline Assembly

To house the entire driveline. 0.25” mild steel is selected to create the structure. Mild steel offers excellent weldability which will serve as the connection method between all the plates. A vertical plate with the bearing hole pattern was created, adding 0.05” to the diameters for additional room to be used for alignment during assembly. The plates feature a prong design in order to slot together and allow for easy identification during assembly. Once again, additional clearance of 0.015” was added to the prongs in order to compensate for any manufacturing error. This way, additional material will not have to be grinded, filed or cut in order to fit the components together prior to welding. Side plates have also been designed to help keep the entire assembly square. A cut out is also featured in order to avoid contact with the chain rolling on the sprocket.

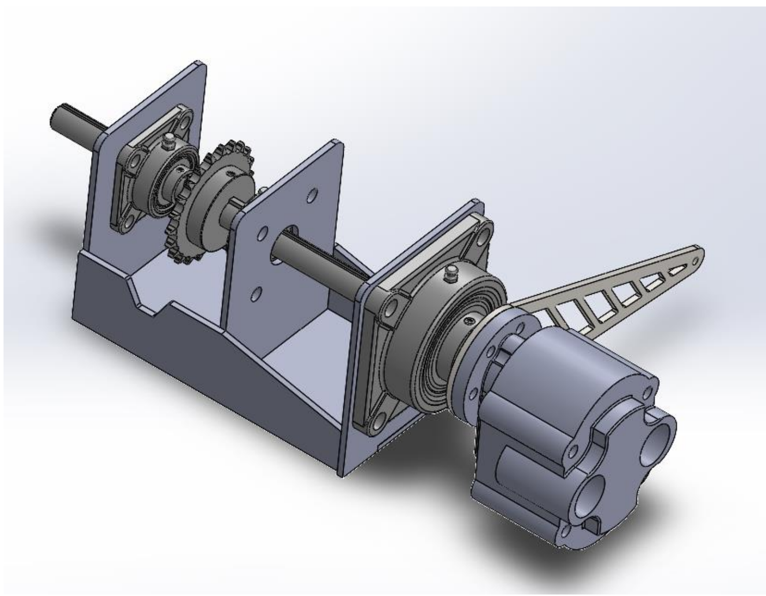

Figure 12 Isometric View of the Driveline Assembly

3.2.4 Electronics

To create a HP and Torque graph versus RPM, there would need to be two physical parameters recorded. The first one being torque and the second being RPM. Having these values as standalone with the dyno makes sure that it can generate a HP/Torque versus RPM graph for whatever is hooked up to it. Schulich Racing wanted to make sure that the dyno would be capable of bigger and better engines in the future, which could include an electric engine. By having the Torque and RPM readings separate from whatever engine system is on the dyno, the team should be able to record consistent data readings from the dyno and tune the engine based off that.

As seen from the driveline, the hydraulic pump is “free floating”. This means that it only has one degree of freedom, which is about the driveline axis. The distance from the pump’s axis of rotation to the load cell is 12 inches. The following figure shows the distance between the axis of rotation for the pump, and the linear axis which the torque will push on.

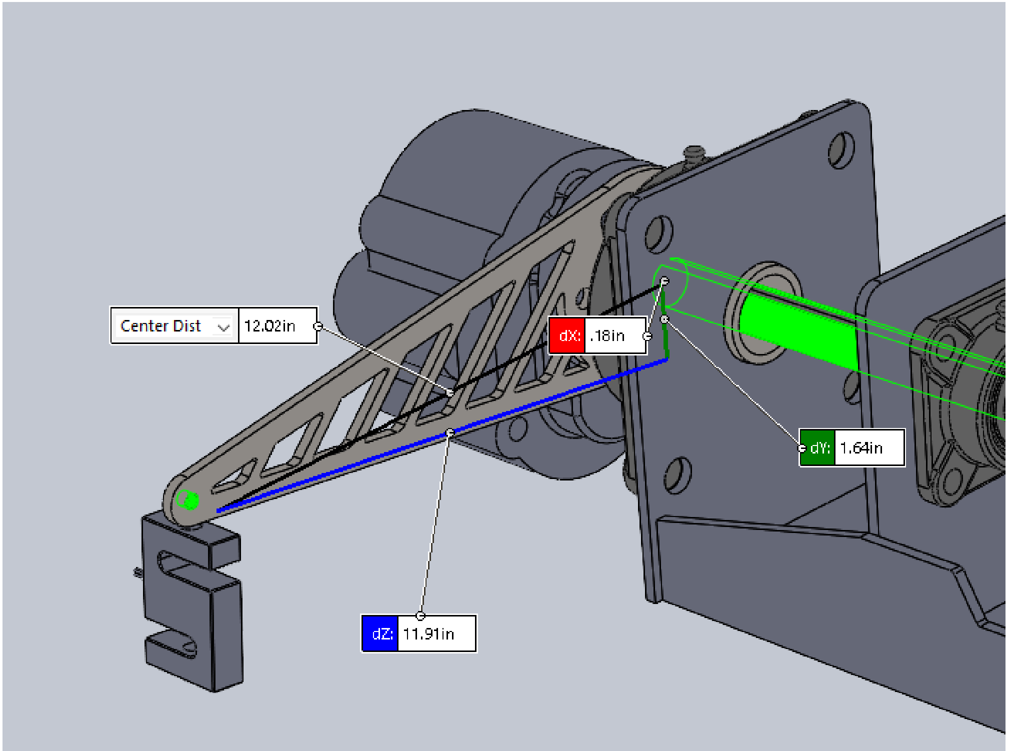

Figure 13: Load Cell Distance

The S type load cell pictured in Figure 12 above, shows how the load cell will be configured when attached to the moment arm. A typical load cell contains a Wheatstone bridge within it. A Wheatstone bridge is designed to change linearly with a compressional or tensional load. The following illustration shows how a Wheatstone bridge works.

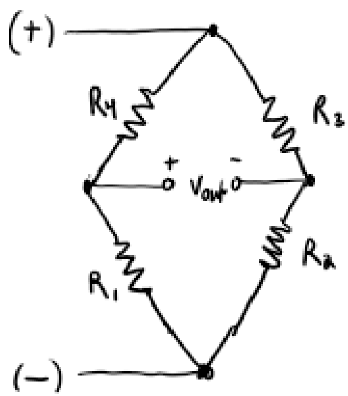

Figure 14: Wheatstone Bridge

In most load cell applications, an amplifier is needed to amplify the signal from the load cell. As the changes generated from it are extremely small. Using an amplifier makes these signals larger and easier to generate meaningful data from. The following amplifier will be connected to the load cell:

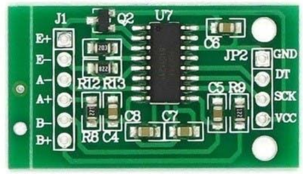

Figure 15: HX711 Load Cell Amplifier [25]

Therefore, the following circuit is used to record and amplify the force measurements from the S-type load cell.

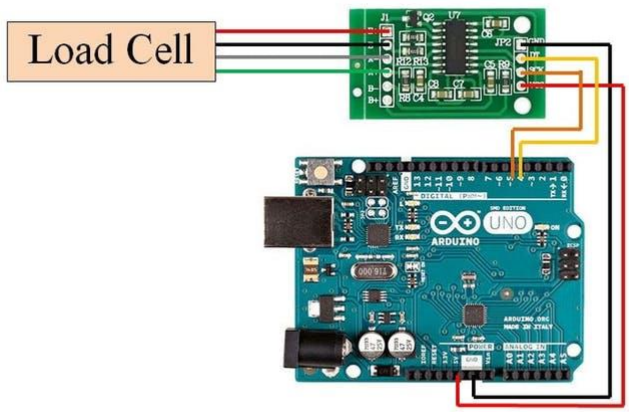

Figure 16: Load Cell Circuit [25]

With the load cell circuit made, the next step is to calibrate the load cell. This is done with a 25lb weight. By applying a known weight to the load cell, the HX711 library can extrapolate accurate weight readings from the load cell. Since the Wheatstone bridge has a linear relationship, only one known weight reading is required. The Arduino Uno will be powered by a laptop/computer, which is listening to the Serial Monitor on the Arduino. Another scope requirement is to have the data save to a file, to be used and referred to in the future.

To determine the RPM, a hall effect sensor will be used. The end goal is to have the dyno connect to the Schulich racing website. This is a future task for the Schulich Racing software team to complete. The code is already written in their tech stack but will need to massage the code to their specific application. To date, there is a small website written with Nodejs and React, which is what the Schulich Racing website uses. This way, they should be able to take the code and modify it slightly to fit their application. To see or download the code for the project, see the following GitHub repository: https://github.com/olinanderson/Small-Engine-Dynamometer.git. The code that is uploaded to the Arduino Uno is provided in the Appendix. For all other code, see the GitHub link provided above.

The following circuit is used to detect changes in magnetic fields (a magnet attached to the rotating driveshaft).

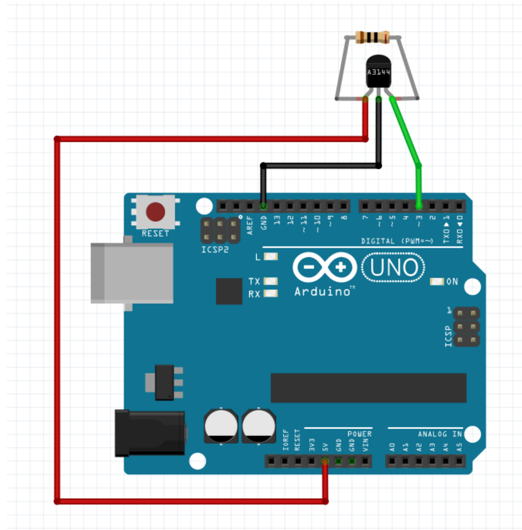

Figure 17: Hall Effect Circuit

The purpose of the pulldown resistor is to signal if there is moving current in the circuit (from the induced magnetic field in the hall effect sensor), then make the according calculations for RPM. The way it calculates RPM is by counting how many times it has passed the sensor and dividing by the time it took to do so. There are a few other parameters that adjust the accuracy or number of times that it needs to pass by to count as RPM, which is configured as global variables in the Arduino code, which is in the Appendix A. Figure 17 shows the entire circuit, which will be soldered together permanently during the manufacturing process.

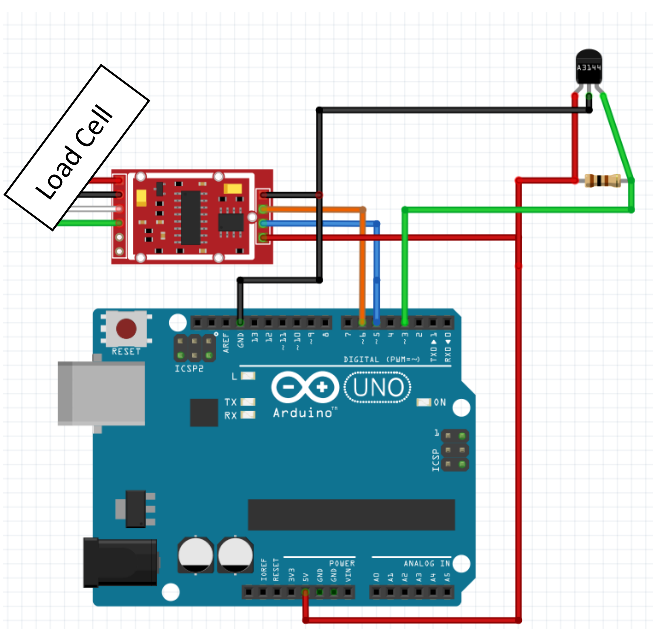

Figure 18: Overall Arduino Circuit

The hall effect sensor will be fastened to a 3D printed stand that is near the shaft. A magnet will be fastened to the shaft with either glue or tape. The circuit board and components will be housed inside a 3D printed containment box, which will be fastened to the frame. This is to prevent any wires coming lose from the soldering joints, and so that any additional accidents do not damage the circuit boards. Figure 18 below shows the 3D printed hall effect sensor stand and circuit board containment that are both fastened to the frame.

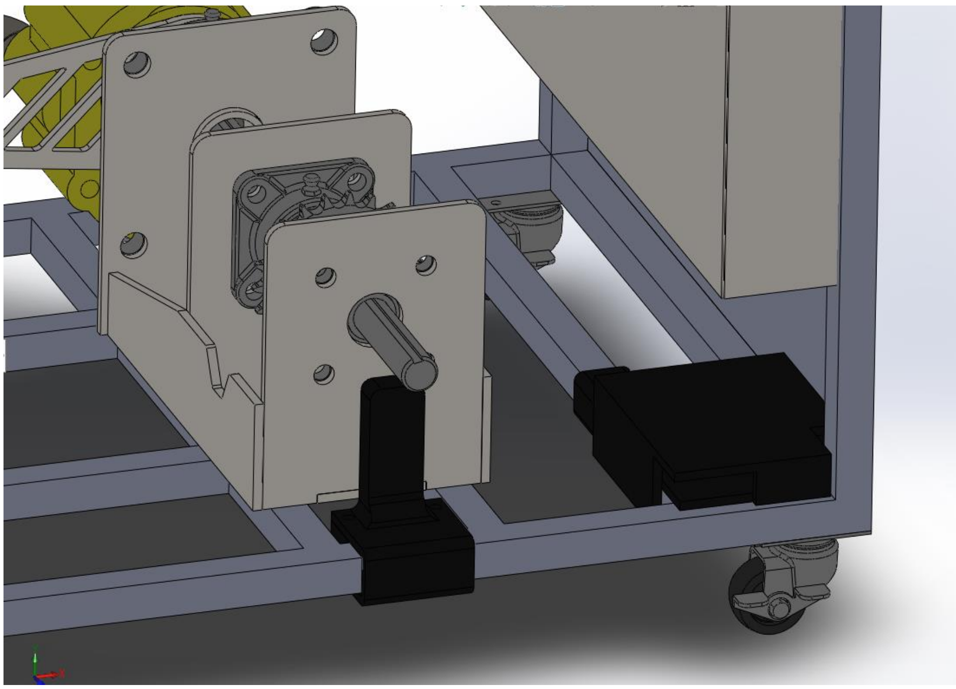

Figure 19: 3D Printed Parts (black)

3.3 Engineering analysis

3.3.1 Hydraulics

The first part of the analysis of the hydraulic system requires determining the minimum amount of pressure required in the hydraulic system. The minimum pressure needed is equal to the pressure lost in the hydraulic system. The hydraulic losses in the system can be defined as head loss. In this case, head loss is the sum of the major and minor losses in the system. The major losses in the system are due to frictional losses within the tubing itself. The minor losses in the system are due to frictional losses in the pipe fittings. Additionally, the control valve requires a minimum of 5psi in the system to be able to accurately control the flow. The total pressure drop in the system can be determined using the following equations:

$$\Delta P = 5psi + \sum h_{Lmajor} + \sum h_{Lminor} \quad (5)$$

Where the major and minor hydraulic losses are defined by the following, and the 5psi comes from the pressure required by the flow control valve [22].

$$h_{Lmajor} = f\frac{l}{D}\frac{V^2}{2g} \tag{6}$$

$$h_{Lminor} = K_L\frac{V^2}{2g} \tag{7}$$

Where f is the friction factor and is a function of Reynolds number and relative roughness of the tube. For fully developed laminar flow, the friction factor depends solely on the Reynold number. K_L is the loss coefficient and is dependent on the geometry of the component [22]. In the above equations, l is the pipe length, D is the pipe's inner diameter, V is the velocity of the fluid and g is the gravitational constant.

At idle the engine has a rotational speed of 1250-1350 rpm and a torque of 12.0lbft. Using equation (1) the minimum power produced by the engine is 2.85HP. Assuming that the engine is being tested in 4th gear, the pump shaft would be spinning at 290 rpm. Using equation (3) at this speed shows that the pump is producing 5.68 gpm or 0.368 L/s. Then by rearranging equation (4) to solve for pressure; the pressure produced in the system at minimum engine power is 688.1psi.

At idle, the liquid flows at 2.034 ft/s and has a Reynolds number of 494. From this Reynolds number it is concluded that the flow is laminar. The friction coefficient can be found using the following:

$$f = \frac{64}{Re}, [22] \tag{8}$$

Resulting in a friction factor of 0.129. Using equation (16), the pressure loss per foot of pipe is 0.04 psi/ft. Now using equation (14) and knowing that the final hydraulic design uses 15 feet of hydraulic line with an inner diameter of 0.75 in, it can be determined that the major losses in the system are 0.616 psi.

The next step in determining the minimum pressure is to determine the minor losses within the system. The final hydraulic design includes three 90-degree threaded elbows and 16 straight connections. Each 90-degree elbow has a loss coefficient of 1.5 and each straight connector has a loss coefficient of 0.08. This means that the system has a total loss coefficient of 5.8. Now using equation (15), and the liquid velocity at idle calculated above, the minor losses in the system are 0.371 psi.

The final source of pressure losses in the hydraulic system are due to elevation changes within the system. These kinds of pressure losses are easy to calculate the pressure drop only depends on the physical fluid properties and the elevation change in the system. In the hydraulic system, the pump has to push the fluid up a vertical distance of 3 ft. The pressure drop due to the elevation change can be calculated using the equation below:

$$\Delta P = \rho g h \tag{9}$$

Where, ρ is the density of the liquid, g is the gradational constant and h is the elevation change. Using the 3ft height drop and the fluid density of 54.25 lb/ft^3, the pressure drop due to the elevation change can be determined to be 1.13 psi.

From the calculations above, the minimum pressure that the pump needs to be able to produce is 7.12psi. To create the minimum pressure at the previously calculated minimum engine flow, the engine has to produce 0.09 horsepower. The Yamaha R6 engine produces 2.85 horsepower at idle which exceeds the minimum horsepower requirement of the system.

Another consideration for this system is the heat gained by the oil in the system. The largest factor for heat gain in the system is heat gained from the pump. Assuming the energy lost due to the pump's efficiency is all transferred into heat, the heat gained by the oil can be described by the following equation.

$$\Delta T = \frac{Power * (1 - efficiency)}{c_p Q \rho} \tag{10}$$

Where power is the power generated by the engine in kilowatts. Efficiency is the pump efficiency. C_p is the specific heat capacity of the oil in kj/kgK. Q is the volumetric flow rate in m^3/s and ρ is density of the oil. At the maximum power conditions, the temperature gained by the system at steady state is 2.56^{o}C. Since it takes time for the system to reach steady state, the rate at which the oil heats up can be mitigated by having a large oil reservoir. The reservoir helps keep the oil cool by allowing the mixing of hotter and colder oils.

3.3.2 Driveline

To determine if the pump can work with the engine, first we must calculate the maximum allowable torque for the pump. The maximum power input that the pump can withstand is 121.43HP, therefore when rearranging equation (1), the torque is given by:

$$Torque = \frac{Horsepower \ * \ 5252}{RPM} \tag{11}$$

Using this equation, it is found that the torque that the pump shaft can handle at the maximum power input is 211.83lbft. Knowing that the highest permissible torque is 211.83lbft, the power transmission within the engine needs to be analyzed. To analyze the transmission, a simple gear train was constructed using the R6 gear ratios. These are specified by the engine manufacturer within the engine service manual. [17]

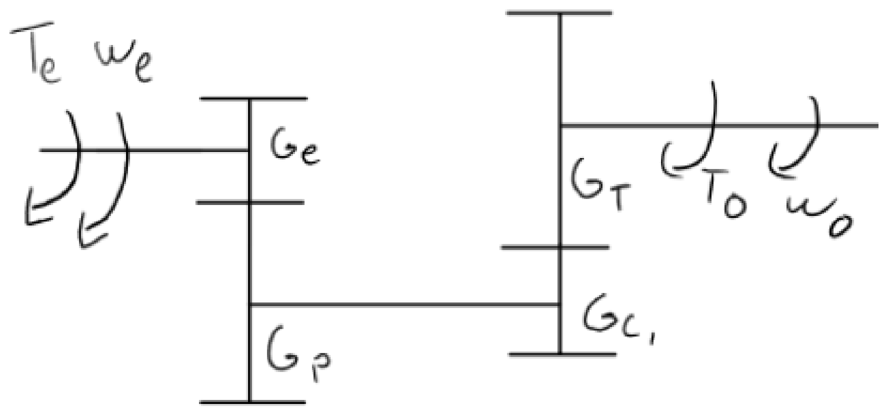

Figure 20: Simplified diagram of the engine's internal transmission

Figure 19 shows a gear train that is a simplified schematic of the engine's transmission. Where T_e is the torque produced by the engine at the crank shaft and ω_e is the rotational speed of the crank shaft. G_e and G_p are the two gears that make up the primary reduction of the engine. G_{c1} and G_T are the two gears that make up the transmission reduction. Finally, T_O is the resulting torque at the engine output sprocket and ω_O is the rotational speed of the engine output sprocket. Using this gear train, the following equations can be developed using gear ratios.

$$\frac{G_P}{G_e} = 2.07, [17] \quad (12)$$

$$T_O = T_e \frac{G_e}{G_P} \frac{G_{c1}}{G_T} \quad (13)$$

$$\omega_O = \omega_e \frac{G_p}{G_e} \frac{G_T}{G_{c1}} \quad (14)$$

By using these equations and the transmission reductions found in the R6 service manual, the engine output power and rotational speed for each of the transmission reductions can be found. This is when the engine is operating at its maximum horsepower.

Table 1: Engine output sprocket torque and speed for each transmission ratio

Gear	Transmission reduction ($\frac{G_T}{G_{c1}}$) [14]	Engine output speed (RPM)*	Engine Output Torque (lbft)**
1st	2.583	2241.08	255.4
2nd	2.000	2894.35	197.76
3rd	1.667	3472.53	164.8
4th	1.444	4008.80	142.78
5th	1.286	4501.33	127.16
6th	1.150	5033.66	113.7

* Assuming the maximum engine speed is 12000RPM
** Assuming the maximum engine torque is 47.7lbft [17]

Now using the values calculated in the table above, another simple gear train can be created. This is to determine the gear ratio between the engine output sprocket and the spur gear on the pumps drive shaft.

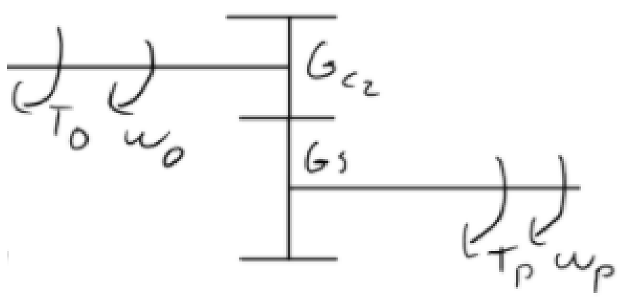

Figure 21: Gear train representing the gear ratio for the engine sprocket and the spur gear

Using the gear train in figure 20, the following equations can be derived to determine the required gear ratio to satisfy the pump requirements.

$$G_s = G_{c2}\frac{\omega_O}{\omega_P} = G_{c2}\frac{T_P}{T_O} \tag{15}$$

Where G_s is the number of teeth on the spur gear and G_{c2} is the number of teeth on the engine output sprocket. It is important to note that the engine output sprocket has 16 teeth on it. To find the number of teeth for the spur gear, a maximum pump speed needs to be determined. For this analysis, a safety factor of 5% was applied on the rotational speed. Therefore, the rotational speed of the pump was initially determined to be 2850 RPM. Then the number of teeth were rounded to the nearest tooth. The resulting rotational speed and resulting torque was calculated by manipulating equation (12). However, for gear 2 since the values from the engine were very close to the values desired for the pump, a 1:1 gear ratio would be used. The table below shows the number of teeth required for the spur gear as well as the resulting rotational speed and torque for the pump driveshaft at the maximum values presented in *Table 1*.

Table 2: Secondary gear ratio and resulting driveshaft speed and torque

Gear	Number of teeth on the spur gear (G_s)	Resulting rotational speed of the driveshaft (rpm) (ω_p)	Resulting torque on the driveshaft (lbft) (Tp)
1st	12	2988.11	191.55
2nd	16	2894.35	197.76
3rd	20	2778.04	206.0
4th	22	2672.53	205.24
5th	25	2880.85	198.69
6th	28	2876.37	198.98

For tuning purposes typically professionals use a gear where the internal gear ratio is as close to 1:1 as possible. In the R6 engine this would mean using 6th gear, however the racing team mainly uses gears 1-4 during competition, and when using a spur gear with 28 teeth, the cost and size of the gear greatly increases. Having a larger gear would mean suspending the pump higher up which would have more catastrophic results should the pump ever fall. Running the car in 4th gear allows the pump to sit low to the frame base while maintaining a gear ratio below 1:1.5. The cost of a 22-tooth sprocket is around 22 USD less and around 2 in smaller in diameter. For these reasons it was decided that the R6 would be tested while in 4th gear.

3.3.3 Structural Components

Since this dyno will be outputting high loads through the driveshaft, all structural components attached to the force output must be designed to withstand the expected loads. The two components analyzed for this scenario are the moment arm and the frame base. The maximum load potentially experienced by these components during operation will be 47.7 lbf as specified by the hydraulic pumps operating parameters. To analyze the stresses experienced in these components, the built-in finite element analysis tool in SolidWorks was used. This was deemed adequate for our purposes as the loading is simple enough and the components are not expected to be near failure. The factor of safety plots after analysis are shown in figures below:

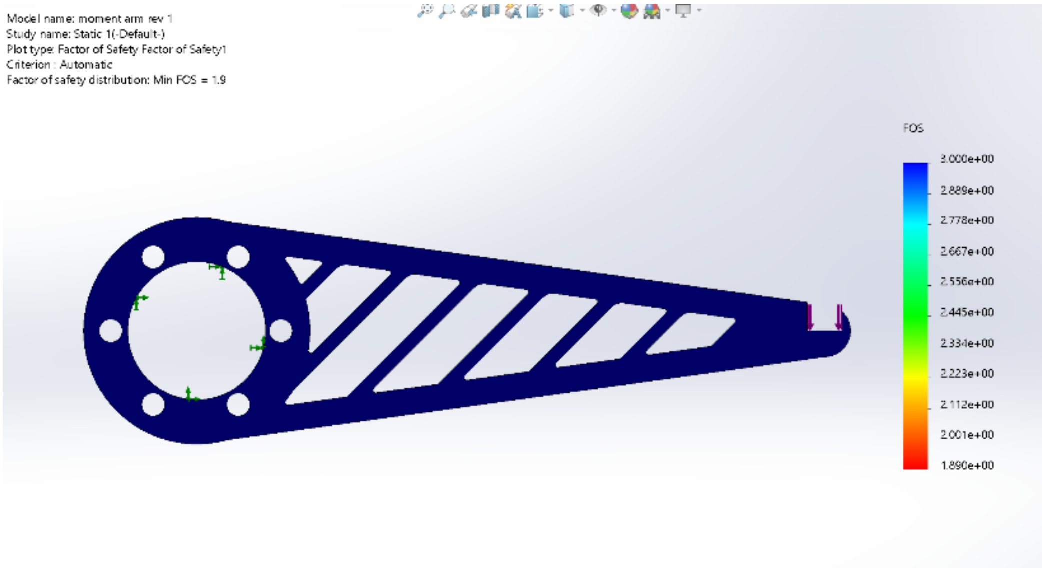

Figure 22: Factor of Safety Plots for the Moment Arm

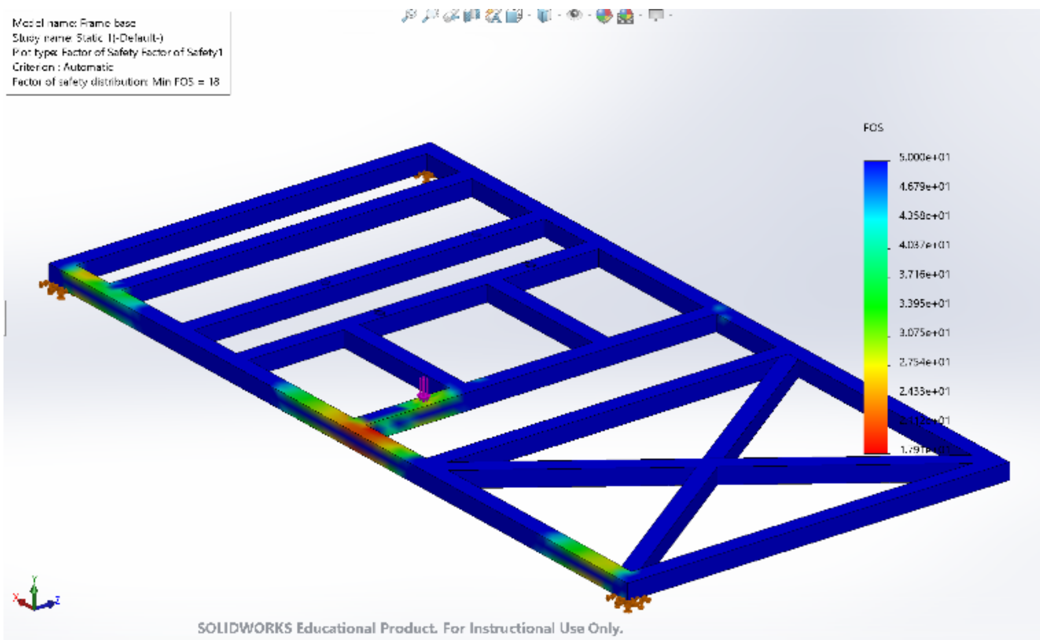

Figure 23: Factor of Safety Plots for the Frame Base

When subjected to the expected max load of the dyno setup, the minimum factor of safety is 1.8 and 18 for the moment arm and frame base, respectively. This ensures that our designed structural components can withstand the maximum operating parameters of the dynamometer.

4 Prototyping and Testing

4.1 Manufacturing Details

4.1.1 Driveline

A water jetting process was used for the driveline housing. A waterjet allows for precise and quick cutting of materials. 2D dxf patterns were created from the cad model and sent off to a machine shop to be cut out of 0.25” mild steel as designed. This assembly was then welded using a tig welding process and ER-70 filler rod by the group. All driveline components such as the driveshaft and bearings were ordered from McMaster-Carr and used to maintain concentricity when welding the assembly. By bolting on the bearings to their appropriate plates and running the driveshaft through them, the plates were constrained to be parallel. To ensure squareness, appropriate clamps were used to hold pieces together and checked using a machinist’s square. With the entire assembly tack welded together, a final squareness and concentricity check was performed and then the assembly was fully welded.

The spline to driveshaft coupler and bushing were both made on a manual lathe. Metal for the bushing was purchased from Metal Supermarket in Calgary. A 3” diameter piece of cold-rolled steel was used to create the bushing. A sizeable amount of material was removed from the piece purchased. The final machined part incorporates a 0.001” oversize on the bearing fit. This allows for a shrink-fit and subsequently a very tight fit when assembled into the bearing. After being welded concentrically to the moment arm both pieces were put into a freezer to shrink the material. With the bushing cold and dimensionally smaller now, it was pressed onto the 2” bearing, seating against the face of the bearing.

4.1.2 Frame

The frame was also tig welded with ER-70 filler rod. The 1” square tube mild steel bars were also purchased from Metal Supermarket. The frame was fully tacked with all bars in place and inspected. Squareness was measured and within 0.0625”, flatness was also measured using a level as a straight-edge. This ensured that all bars were co-planar, and none were standing proud, eliminating potential issues with later assembly. After all quality checks passed, the frame was fully welded. The top and rear sheet were later screwed on using #10 self-tapping sheet metal screws. This created the platform for the hydraulic flow control valve and pressure-relief valve to rest on.

With the frame created, it was prepared for painting. This is to ensure the mild steel will not corrode when left outside or in presence of moisture. The first step in preparation was grinding and sanding down the frame bars. This removed the mill scale and any oxidation already present on the bars. The paint does not adhere properly or cleanly without proper bar preparation. Once the bars were sanded, a coat of paint primer was sprayed on. This creates a base layer that helps the coat of paint stick and adhere properly. Following this, three coats of paint were sprayed on using a pneumatic paint sprayer. The frame was left to dry before attaching all the remaining subsystems.

4.1.3 Hydraulics

The main assembly of the hydraulic system was having the hose connectors crimped to the hose. However, for the system operating pressure being up to 3000 psi, it would have been impossible to crimp the fittings on the hose by hand. Once the frame was assembled and the pump installed, the lengths of the hydraulic hose were measured with additional slack and were sent off to Greenline Hoses in Calgary, who cut the hoses to the specified lengths and then placed the crimp fittings onto the end and used a hydraulic crimping tool to crimp the connector fittings to the ends of the hose. The hoses were ready to be picked up 2 days after ordering. This short lead time is part of the reason that Green Line was selected to provide the hydraulic hoses for this project.

Once the hoses were picked up, the hydraulic system was assembled. The first step in the assembly was attaching the SAE to 1" NPT adapters to the pump, pressure relief valve and control valve. Then the hose connecting the control valve to the pressure relief valve was installed. The next step was to connect the pump outlet to the pressure relief valve. This step caused problems because the pump was fixed stationary to the frame and as a result one person had to hold the control valve and pressure relief valve and allow them to rotate while tightening the hose to the control valve. The next step was to install the floor flange to the bottom of the bucket. The floor flange seals the bucket and is threaded so that the hose can connect to it. To do this, first a seal made of rubber gasket was cut out and oil resistant gasket maker was applied to both sides and affixed to the floor flange. The floor flange is attached to the bucket using four ¼ x 1" bolts. These bolts were covered in gasket maker before being tightened to the floor flange to seal the threads. Once the bucket was sealed, a 3" nipple was connected to the floor flange and a union was connected to the bottom of the nipple. Then the hose that connects to the pump inlet was connected to the union. Finally, the pressure relief valve and the control valve were connected to the lid of the bucket by cutting a hole in the bucket and attaching them to a 90-degree elbow to the underside of the lid. The elbows were then pointed to the sides of the bucket to direct flow to swirl around the side of the bucket instead of directed straight down into the bucket. This helps stop the oil from splashing around inside the bucket and from applying a downward force in the bucket due to the fluid flow. On all the threaded connections both anti-seize and oil resistant Teflon tape were applied. The anti-seize keeps the joint lubricated so that it is possible to take the hydraulic system apart in the future. The Teflon tape helps to protect the system from leaking at the threaded joints. Once all the hydraulic lines were installed the control valve and pressure relief valve were bolted to the frame.

4.1.4 Electronics

After the circuits were designed and tested using a breadboard and non-permanent wires, the circuit was ready to be permanently soldered to a breadboard. For this task, a solderable breadboard was used as well as 22-gauge wire to connect 5V and GND to each positive and negative terminal on the breadboard. Throughout the process of soldering, a multimeter was used to verify continuity of the circuit. Each new soldering joint was checked for continuity/short circuit to verify that the circuit worked as intended. Once the circuit was soldered, it was ready to solder the load cell wires to the HX711 board. Once this is done, the circuit is permanently fastened to the dynamometer. To fasten the A3144 hall effect sensor to the stand, electrical tape was used as it provides a slight barrier of protection against any abrasive hits/accidents. Zip ties were used to fasten loose wires to the frame of the dynamometer. Once the circuit was built and fastened to the dynamometer, the Arduino was plugged into a laptop. The first step was to calibrate the load cell, so the force readings were accurate. To do this, a known weight of 25lbs was placed on the load cell, while running the calibration script. The calibration script is used to find the value that is used to scale the load cell to read accurate force readings. Once the calibration value was found, the code including RPM and load cell arithmetic was compiled and verified, it was pushed to the Arduino. The Arduino Serial Monitor was used to verify that the values were reading correctly. To verify the weight readings, one of the students stood on the load cell and compared the weight reading to what they read on a scale, and it was within +- 2 lbs. To verify the readings of the RPM values, it was connected to a drill at max speed, and it was reading within +- 15 RPM, as a construction drill is rated at 2000 RPM at max speed.

Once the data was being accurately output on the serial monitor, the team made a custom app to record the data and manipulate into a real-time graph. To do this, there were two servers made. The first server was running in Nodejs, which listened for the COM port on a specific baud rate. The second server was a website application which ran with React, which was on its own server as well. The data which was coming through the COM port was relayed to a WebSocket which then streamed the real time data to the front end React application. The React application then graphs the livestream using a framework called Recharts. Another part of the backend server that the team wanted was an ability to save the data to a json file. To do this, there is a global array of objects, which contains torque, rpm, and a timestamp. With every new change from the COM port, the new value is added to the array and then the array overwrites the previous array in the file. This means that while new data is streaming, the file will always record the data stream and not miss any points. Note however that the file name would need to be changed to not be overwritten with each new run.

The final manufactured prototype is shown below:

Figure 24: Full Manufactured Prototype

4.2 Testing Procedure and Details

After the prototype was completely manufactured, a testing procedure was determined, and a testing period was set for a day with favorable weather to be able to test the prototype outdoors. However, due to Covid-19 restrictions, the Yamaha R6 engine was unable to be used for testing. Instead, tests were performed on a Honda CRF 450r dirt bike. This dirt bike produces less power than the R6 engine, so the upper limits of the system were unable to be tested. The CRF 450r has a maximum power of around 50HP, while the Yamaha R6 has a maximum power of around 90HP. We were able to test the dirt bike up to an engine rpm of 1500, which produced a power of approximately 1.0 horsepower. Initially, this power value was thought to be much lower than what was expected but these kinds of dirt bikes produce minimal power until the engine speed reaches around 2000 rpm as seen in figure 25 below.

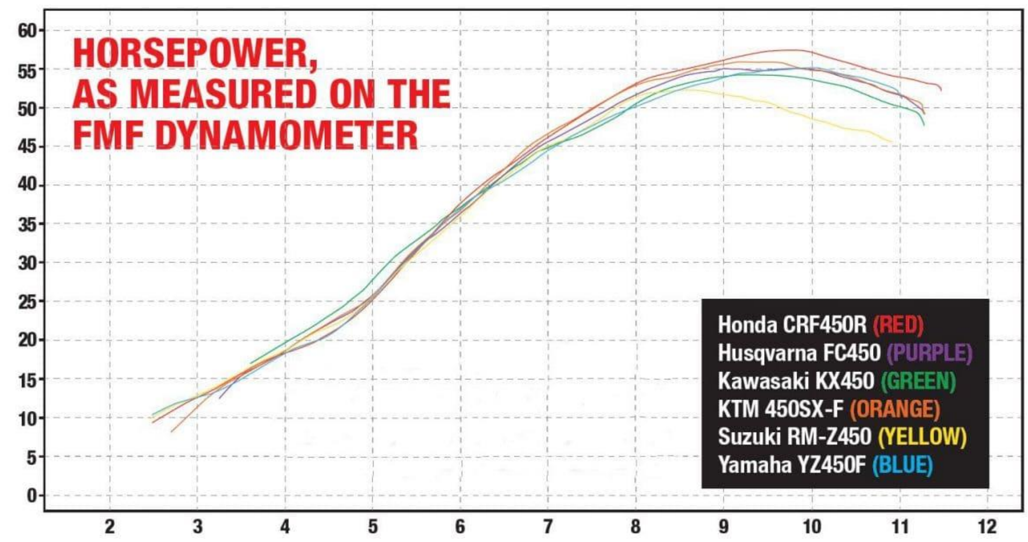

Figure 25: Various MX Dyno Results [24]

While researching to see what the experimental power of the Honda dirt bike is, it was apparent that most tests do not go below 2000rpm. Since the lower portion of the graph is relatively linear, when one extrapolates to around 1000rpm, it equates to 1.0hp. This is consistent with the experimental results recorded from the testing setup. Figure 26 below shows real test data that was recorded from the small engine dynamometer. The left y-axis contains torque in ft-lbs, and the right y-axis contains power in HP. The x-axis is revolutions per minute.

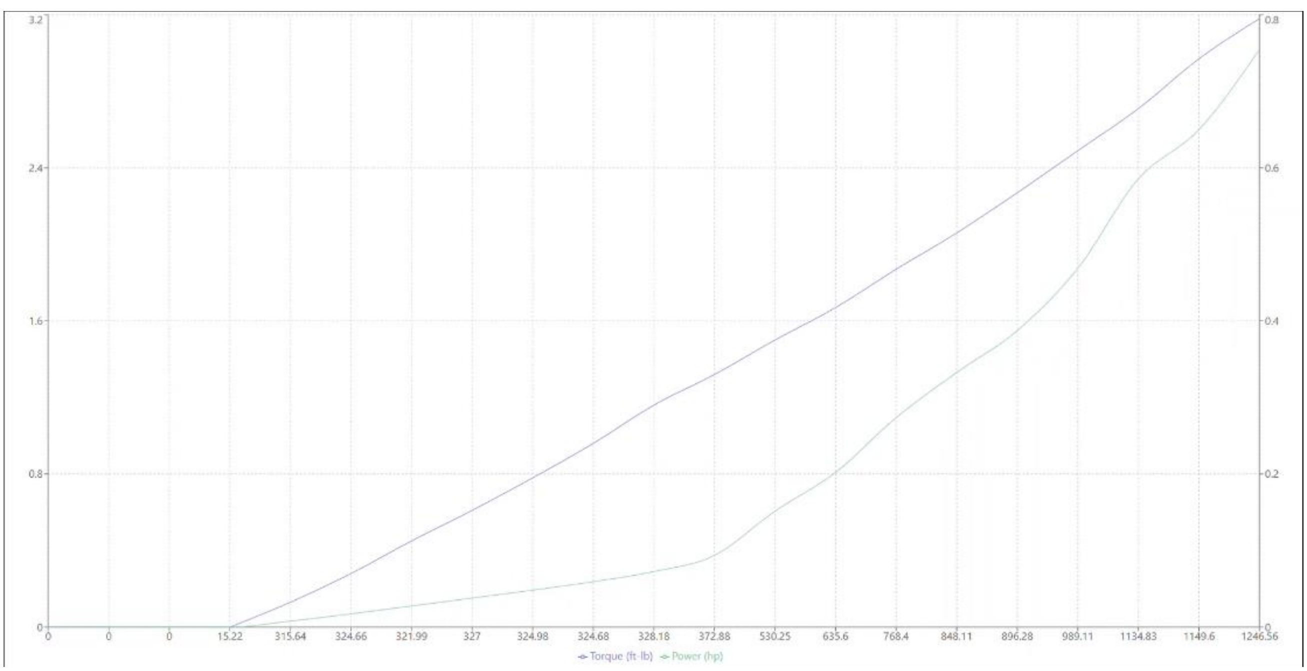

Figure 26: Experimental Test Results

Due to the improvised test setup which contained a wooden jig attached to the rear tire of a CRF450r, the test was limited to 1400rpm due to safety concerns. Throughout the testing, the group did multiple runs to around 1400rpm, which each resulted in relatively similar data output from the load cell. This means that the testing setup is consistent and repeatable, which was another large deliverable that the team wanted to achieve. With the repeatability of the testing setup, the Schulich Racing team should be able to tune engine parameters and determine if it is increasing or decreasing the power of the engine.

4.3 Descriptions of test scenarios / experiments

In order to test the manufactured prototype, a few scenarios were created and executed to attempt to test all aspects of the dyno. The first test involved a sweep in throttle and rpm without any hydraulic load. As previously specified, the testing was limited to around 1500 rpm from a safety standpoint. Therefore, the actual range tested varied from 300 rpm to 1500 rpm. The hydraulic flow control valve was fully backed off to remove any restriction in the flow. The CRF 450r was then put into first gear and the clutch released to allow for rotation at engine idle. After ensuring that both torque and rpm were being recorded, the test began. By slowly increasing the amount of throttle applied, the bike began to spin faster. As the rpm increased, as expected, the force on the load cell also increased. The throttle was then released, the system allowed to settle back down, and the test was repeated.

The second test scenario involved varying hydraulic flow and examining the resulting system reaction. This test was also attempted in two forms, one keeping the bike at idle and another with a higher engine rpm. The first test had the dirt bike running at idle. Once the force and rpm values reached steady state, the hydraulic control valve then slowly restricted. As the needle valve closed in the valve, the system started to register an increase in load. This behavior was expected and verified with the test. Closing the hydraulic flow control valve increase backpressure on the pump and thus requires more load from the engine to spin the hydraulic pump. The second test of this kind replicated the same procedure at a higher rpm. The hydraulic valve was fully opened, and the bike was released to idle. Following this, a moderate throttle increase was performed, and the bike was held around 1000 rpm. With the bike held at 1000 rpm the flow control valve was once again closed slowly. The results showed an increase in load as the control valve reached a fully restricted position.

From these test scenarios, it was verified that the engine dyno works. The system successfully takes engine input and can drive the hydraulic pump as intended. It was also demonstrated that the flow restriction from the hydraulic control valve successfully works as intended. The increase in load as the valve was restricted demonstrates the creation of backpressure in the system. Another important discovery was the repeatability and consistency of the system. When placed under the same testing procedure, the results matched up and were consistent. This ensures that the results received from the dyno can be used to justify engine tuning choices and serve as empirical evidence.

5 Project Management

5.1 Roles and responsibilities of team members

There are common responsibilities that are shared by each team member. Each team member is responsible for producing high quality of work. Also, each team member is responsible for meeting deadlines that were set at the start of the project. If any issues arise, they must inform the team of delays to the project schedule. Additionally, each team member is responsible for contributing to the presentations and reports that are capstone course requirements. The following paragraphs will outline each team members individual responsibilities, however everyone is responsible for helping with all parts of the project.

Tyler's role on the team is to design the hydraulic and drivetrain systems. He is responsible for the selection of all parts and equipment needed for these systems. The parts selected for these sections must be able to work in the system and fall within budget. Additionally, he is responsible to perform the required calculations to verify the system. Tyler is also responsible for the assembly of the hydraulic system. Additionally, Tyler is responsible for ensuring that the assembly of the hydraulic system works as intended and is free of leaks. Also, he is responsible for updating the schedule spreadsheet for all tasks that fall under his scope of work.

Conrad's role on the team involves overall team leadership, pump analysis, manufacturing strategies. The leadership role involved calling and scheduling meetings. Taking notes on overall project progress and developing a plan of action for the group to work on. Conrad took part in the pump selection process, analyzing various styles and models and reporting the info back to the team. Justifying and performing the final confirmation on the Hydreco pump with teammate Tyler. Another responsibility is the design of the driveline assembly. Conrad is also responsible for all welding required throughout the project.

Josh's responsibility on the team is to design the frame of the dyno, create and maintain a budget and to communicate the budget status to the sponsor. He is responsible for a frame that meets all requirements of the sponsor, as well as any requirements created by the designs of the other sub-systems. Additionally, he must ensure that the frame can withstand any forces and vibrations so that anyone can operate the system safely. Furthermore, the budget created for this project must be kept updated by Josh so that the team can source and purchase parts accordingly. Finally, Josh is responsible for sourcing raw materials and several of the parts required to manufacture the prototype.

Olin's responsibility on the team is to determine the data acquisition process and design circuits that will achieve the scope of this project. He is responsible for the Arduino code and communication with the Software/Electrical team members on Schulich Racing. He must ensure that the code is accurate and delivers meaningful information. Another responsibility will be to train Schulich Racing team members on how the data acquisition process works in case there needs to be any troubleshooting. He will need to construct the circuits and test them as well. A side responsibility of his will also be designing and 3D printing stands and containments for all data acquisition components.

5.2 Project schedule and milestones

The scheduling system used for this project is an excel spreadsheet that splits up the project into 6 subcategories which are: conceptual design, frame verification, hydraulic system verification, mechanical system verification, data collection system and fabrication. Each of these 6 subsystems are split into individual tasks. Each task has an associated progress, priority, start and end dates. During team meetings, time is spent updating the spread sheet and adjusting completion dates should a delay in the project schedule occur. Also, the spread sheet is periodically printed into a PDF document to use in the progress reports. Figure 27 below shows the printed spread sheet that was updated on April 4th, 2021.

Conceptual Design

Name	Progress	Priority	Timeline - Start	Timeline - End
Pump Selection	100%	Critical	2020-10-01	2020-10-20
Frame Concept	100%	High	2020-10-01	2020-11-30
Hydraulic System Concept	100%	Medium	2020-10-20	2020-11-30
Data Collection System	100%	Medium	2020-10-20	2020-11-30
			2020-10-01	**2020-11-30**

Mechanical System Verification

Name	Progress	Priority	Timeline - Start	Timeline - End
Component List	100%	Low	2020-11-08	2020-11-15
Force Analysis	100%	Medium	2020-12-01	2021-02-14
Sprocket Choice	100%	High	2021-01-01	2021-02-14
Driveshaft Setup	100%	High	2021-01-01	2021-02-14
			2020-11-08	**2021-02-14**

Frame Verification

Name	Progress	Priority	Timeline - Start	Timeline - End
Frame Analysis	100%	Medium	2020-12-01	2021-02-14
Structural Analysis	100%	Medium	2020-12-01	2021-02-14
CAD Drawings	100%	Low	2021-02-01	2021-02-14
Bill of Materials	100%	Low	2021-02-01	2021-02-14
			2020-12-01	**2021-02-14**

Data Collection System

Name	Progress	Priority	Timeline - Start	Timeline - End
Component List	100%	Low	2020-11-08	2020-11-15
Wiring Diagram	100%	Medium	2021-02-01	2021-02-14
Programing	100%	Medium	2021-01-01	2021-02-14
			2020-11-08	**2021-02-14**

Hydraulic System Verification

Name	Progress	Priority	Timeline - Start	Timeline - End
Pump CAD Modelling	100%	Medium	2020-11-10	2020-11-14
Component List	100%	High	2020-11-08	2020-11-15
Hydraulic Fitting List	100%	Low	2020-11-08	2020-11-15
System Pressure Analysis	100%	Medium	2020-12-14	2021-02-14
Oil Temperature Analysis	100%	Medium	2020-12-14	2020-12-26
System CAD Drawings	100%	High	2021-01-01	2021-01-31
			2020-11-08	**2021-02-14**

Fabrication

Name	Progress	Priority	Timeline - Start	Timeline - End
Frame Welding	100%	High	2021-02-15	2021-02-27
Plumbing Hydraulic System	100%	High	2021-02-27	2021-03-03
Reservoir Welding	100%	High	2021-02-27	2021-03-06
Sensors + Wiring	100%	High	2021-02-27	2021-03-06
Final Drive System	100%	High	2021-02-27	2021-03-17
			2021-02-15	**2021-03-17**

Figure 27: Project Schedule Printout

5.3 Cost of the project's developments

At the beginning of this project, multiple meetings with the sponsor occurred to create a reasonable budget for prototyping. Several budgets were made for different dyno setups which were presented to the sponsor for their final selection. After reviewing the different options from a technical standpoint, the design team decided that the most practical setups for the project would be the water-based rotor break or an oil-based hydraulic brake.

A worst-case scenario budget was created for both of the final choices. Each budget consisted of major subsystem costs and any potential miscellaneous items needed for a working prototype. The cost of these items was then reasonably inflated to account for any potential design changes, shipping costs, taxes and tariffs. In a final meeting with the sponsor, the early high-level estimates were presented to the leadership team. Advantages and disadvantages of different setups were discussed along with the estimations and a final design direction was chosen. This ended up being the hydraulic brae style dynamometer and a preliminary budget of $2,800.00 was set for the prototype.

Item group	Item	Item use	Sub-Items	Estimate ($ CAD)	Current Costs ($ CAD)	Difference ($ CAD)
Dyno Frame	Steel frame	Frame to mount engine and pump, must withstand torque and vibrations.	Steel tubing	$300.00	$333.10	-$33.10
			Steel plate	$200.00	$0.00	$200.00
	Miscellaneous Items		Casters, connections, bolts, etc	$150.00	$39.32	$110.68
Oil System	Pump	Supplies load to engine		$450.00	$491.02	-$41.02
	Fittings	Oil system connections		$150.00	$634.06	-$484.06
	Reservoir	Large oil storage		$150.00	$53.00	$97.00
	Flow control valve	Regulates pressure		$300.00	$174.17	$125.83
	Pressure release valve	Safety measure to avoid over pressuring system		$200.00	$156.00	$44.00
	Miscellaneous Items		Hoses, Valves, etc	$150.00	$47.48	$102.52
Cooling system	Radiator	Cooling the oil system due to high temperatures		$150.00	$0.00	$150.00
	Fans			$50.00	$0.00	$50.00
Drivetrain	Driveshaft			$100.00	$204.53	-$104.53
	Waterjet parts			$0.00	$242.04	-$242.04
	Miscellaneous Items		Gears, chains, bearings, etc	$100.00	$420.90	-$320.90
Data	Load Cell	Measuring torque on pump		$150.00	$63.39	$86.61
	Volumetric flow metre			$100.00	$0.00	$100.00
	Miscellaneous Items		Sensors, wiring, controls, etc	$100.00	$72.78	$27.22
Misc tax/Shipping				$0.00	$55.93	-$55.93
			Total:	$2,800.00	$2,987.72	-$187.72

Figure 28: Early High-Level Cost Estimates and Final Prototype Costs

The final cost of the prototype was $2,987.72 as shown in figure 28, the overall cost was overbudget by $187.72. Although the sponsor had set the initial limit of $2,800.00, they approved the extra spending to ensure that the prototype worked properly and safely.

The frame of the prototype came in under budget, as not much had changed from early concepts with the frame design. The oil system was one of the large contributors to the spending on the prototype. Due to changes in the design, the cost of fittings exceeded the original budget by a large margin. These design changes allowed other components to come well under the budget, but the cost still exceeded original estimates by $155.73. The cooling system originally required was deemed unnecessary due to design changes in the oil system, the money was reallocated to the other sub-systems. The driveshaft was the most extreme case of overspending on the prototype due to several reasons. The main reason was due to the fact that the waterjet components were originally going to be manufactured for a reduced cost from a sponsor of the Schulich Racing team. This fell through and the design team was forced to find an emergency source for waterjet cuts, which drastically increased costs. The other reason was that larger bearings were needed to support the driveshaft than originally thought, again driving the cost up. The final cost of the drivetrain system was $867.67, which ended up being $667.67 over-budget. Finally, the data subsystem came in $213.83 under-budget. This allowed the design team to divert these funds to the drivetrain and oil systems.

5.4 Use of resources and contact hours

The original plan of using the University of Calgary Makerspace was altered. Due to local Alberta covid lockdown restrictions, the university closed the makerspace. The closure of the engineering machine shop also cancelled the possibility of water jetting and machining on location and alternative resources were sought after.

An old coworker along with a friend of Conrad both offered help and time with welding. Scheduling time to meet and use the equipment was done with respect to all covid restrictions in place at the time of use. For the frame welding, an Everlast PowerTIG machine was used. This welder allowed for the smooth delivery of high frequency dc power. Perfectly controllable and able to generate enough power to completely weld the frame. The driveline was welded together using a Lincoln Electric Square Wave 200. Once again a capable machine producing proper high frequency dc power to weld the 0.25" mild steel plates together.

To machine the bushing and spline-to-shaft coupler, an old coworker was contacted by Conrad. This coworker owns a Colchester Chipmaster Lathe. This lathe, although smaller, was able to handle the size of work needed. As the biggest part to be machined started off as a 3" diameter piece of cold-rolled steel. The Colchester lathe along with a roughing cutter, boring bar and finishing cutter were all that was required to machine the pieces.

To 3D print any of the containments for the electronics, Olin's 3D printer was used. To solder any of the breadboard together, there had already been everything needed, which includes soldering wire, a soldering iron, wire, etc. Therefore, the resources used for the construction of the electronics section were already at the student's disposal. To construct the testing apparatus, the student also had scrap wood and screws from previous projects that would help to contribute to building the fastener which attached to the rear dirt bike tire.

6 Conclusions and Recommendations

6.1 Project summary

Overall, this capstone project was a success. All deliverables were met, and a functioning dynamometer was designed and manufactured. The process began with outlining the sponsor requirements. Which were a compact engine dynamometer that is able to test up to 107 horsepower and 12,000 engine rpm. All the while remaining close to the initially agreed upon budget of $2800. The total final price for the project was $2,987.72.

The final design chosen was a hydraulic brake dyno. This design was the most cost-effective design for this horsepower requirement. The design also allows for easier manufacturing and assembly, which has proven itself important with regards to the Covid-19 pandemic. The hydraulic brake setup powers a hydraulic pump using the engine being tested. A restriction in the flow creates back pressure on the pump and in turn loads the engine. This increases the force required to circulate oil in the system, which is measured from the pump casing as it rotates.

To house the entire system a mild steel frame was designed from 1" square tubing and sat on top of castor wheels. This allows all subsequent systems direct fastening to the frame through bolts and nuts while keeping the entire project mobile. The dimensions for the frame are 2'x4'x3.5' allowing for a compact footprint that can be easily stored. The frame was tig welded and later painted to prevent any future corrosion. The hydraulics of the project involve a Hydreco 9WL hydraulic pump with a displacement of 4.53 cubic inches. At max operating conditions, the pump creates 58.88 gpm of flow.

To allow for fine adjustment on flow restriction, a Parker F2020s hydraulic flow control valve was fitted to the system. As a safety consideration, a pressure relief valve was installed and is set to open if the pressure reaches 2500psi. The driveline consists of 0.25” waterjet cut mild steel pieces. These were tig welded together to create the driveline housing. Bearings, a keyed driveshaft, key and a sprocket were all attached to the housing to make up the driveline. A 2” bearing and custom-made bushing allows for free rotation of the pump casing in order to properly transfer the load generated. The electronics feature an S-type load cell and hall effect sensor to measure rpm. The load cell is attached to the moment arm on the pump in order to measure the torque created. Through wiring the setup to an Arduino, the data is able to be read and saved in real time. Which can be later graphed to view horsepower and torque curves against rpm.

Tests were performed using a Honda CRF 450 dirt bike and the values that were recorded during the test matched to dyno values found online using the same bike. Most importantly, multiple tests were performed following the same procedures and the results recorded were similar between runs. This demonstrates that the created dyno is repeatable, a crucial detail. This signifies that the project sponsor will be able to use this project to verify and claim engine tuning justifications with proper empirical evidence to support it. Knowing that the dyno is repeatable, tuning choices made in between runs can be directly analyzed by running the same test again.

6.2 Suggestions for design improvements

During the course of this project, there were aspects that could have been done differently to make the project better. One recommendation is to make a permanent oil tank with welded-on bungs for the hydraulic system. The original hydraulic system included a tank, however due to Covid-19 restriction difficulties and constraints on time and budget during the manufacturing of this project, the tank was unable to be built. The addition of the tank would be beneficial because it would remove the risk of the bucket’s lid coming off during operation, and it would allow for a smaller profile in the back of the of the dynamometer. Another recommendation is to make an enclosure to house the chain. This enclosure would be for if the chain ever were to break or slip off either sprocket. The chain would then hit the walls of this enclosure instead of being sent in a random direction. This enclosure could be made of any kind of metal or wood and would just serve to contain the chain in the event of catastrophic failure. Another recommendation for this project would be to add a way to tension the chain. When the prototype was tested, the chain was able to heat up and expand, which then caused it to interfere with one of the plates on the driveline assembly. A chain tensioner would allow quick adjustments to the chain tension to ensure that the chain is properly tensioned at all times.

6.3 Lessons learned

Over the course of the past year, the group learned many lessons. Hurdles in the project along with the global pandemic caused many setbacks and made the entire project more difficult. The Covid-19 pandemic created massive unpredictability and volatility in every aspect of the project. From increasing material pricing, shipping lead times, stock sellout and local lockdown restrictions.

The first lesson learned was to budget additional time for all aspects of fabrication, shipping and services. Shipping lead times increased and had the potential to set back any delivery for materials and parts. Many of the original deadlines and timeframes for manufacturing followed the idea that the group would be able to use the makerspace at the University. With the closure of this space and the need to seek out exterior resources, a few deadlines ran longer than anticipated.

Another lesson learned was to diversify the group's service portfolio. The closure of the university due to lockdown measure led to the loss of our previously accessible workshop and the engineering machine shop. In response we had to outsource previously secured services such as water jetting and machining. The original plan was to waterjet and machine all the custom pieces with the engineering machine shop. As one member from the group previously interned at the machine shop and was allowed to continue using the equipment they previously used on internship. With this huge privilege gone, all equipment and help needed to be sought elsewhere. Luckily, the same group member had enough connections from their internship to borrow equipment and get manual machining done. However, this ties back into the previous point of budgeting more time. When receiving favors from colleagues for the use of a welder for example, the original schedule now has to change to match the colleagues. With a diverse portfolio of contacts, the chances of successfully getting work done increases.

During the assembly of the hydraulic system a few lessons were learned. The first is to use swivel fittings. These fittings would have made the assembly of the system much easier. The fittings that were used are effective, however they make the assembly harder because the threaded ends are unable to spin independently of the hose. This means that to tighten the hose the other end of the hose also has to be able to rotate, otherwise the hose will twist and apply a force that will loosen the joint. The second lesson learned with the hydraulic system was that accounting for a bend radius is imperative for the success of the system and a 4" bend radius needs to be converted into an arclength to determine the amount of hose needed to complete a bend. This lesson was learned when measuring the hydraulic hoses prior to ordering the supplier gave a set of guidelines for measuring how much hydraulic hose would be needed to complete a bend. For example, to make a 90-degree bend in the hydraulic hose, an additional 6.3" would be required.

References

[1] T. Lish, "What is a Dynamometer and How Does it Work?", *Setra.com*, 2020. [Online]. Available: https://www.setra.com/blog/test-and-measurement-dynamometer. [Accessed: 25- Nov- 2020].

[2] "FAT- Engine Dyno", *Focusappliedtechnologies.com*, 2020. [Online]. Available: http://www.focusappliedtechnologies.com/aed.html. [Accessed: 26- Nov- 2020].

[3] "Chassis dynamometer system | Testing systems | MEIDENSHA CORPORATION", *Meidensha.com*, 2020. [Online]. Available: https://www.meidensha.com/products/automobile/prod_01/prod_01_02/prod_01_02_03/index.html. [Accessed: 26- Nov- 2020].

[4] "Engine Dynos – Engine Dynamometers | Taylor Dyno", *Taylordyno.com*, 2020. [Online]. Available: https://www.taylordyno.com/products/engine-dynamometers/. [Accessed: 25- Nov- 2020].

[5] "Eddy Current Engine Dynos & Dyno Systems | Taylor Dyno", *Taylordyno.com*, 2020. [Online]. Available: https://www.taylordyno.com/products/engine-dynamometers/eddy-current/. [Accessed: 25- Nov- 2020].

[6] "Water Brake Engine Dynos – Hydraulic Dyno | Taylor Dyno", *Taylordyno.com*, 2020. [Online]. Available: https://www.taylordyno.com/products/engine-dynamometers/water-brake/. [Accessed: 25- Nov- 2020].

[7] D. Gitano, "Dynamometer Basics", *Skyshorz.com*, 2020. [Online]. Available: http://skyshorz.com/university/resources/dynamo_basics.pdf. [Accessed: 02- Dec- 2020].

[8] “Explain working of hydraulic brake dynamometer with sketch. | Mechanical Engg Diploma Topic wise Notes and Solutions", Mechdiploma.com, 2020. [Online]. Available: http://mechdiploma.com/explain-working-hydraulic-brake-dynamometer-sketch. [Accessed: 03- Dec- 2020].

[9] "4.53 cu in Hydreco 9WL-190503 Hydraulic Pump | Gear Pumps | Hydraulic Pumps | Hydraulics | www.surpluscenter.com", *Surpluscenter.com*, 2020. [Online]. Available: https://www.surpluscenter.com/Hydraulics/Hydraulic-Pumps/Gear-Pumps/4-53-cu-in-Hydreco-9WL-190503-Hydraulic-Pump-9-10044.axd. [Accessed: 03- Dec- 2020].

[10] “In-Line Mounted Flow, Pressure Control, Check and Ball Valves", *Fluidpowersolutions.ca*, 2020. [Online]. Available: http://www.fluidpowersolutions.ca/pdf/Parker%20Valve%20Industrial%20Colorflow%20In-Line%20Flow,%20Pressure%20Control,%20Check%20and%20Ball%20Valves.pdf. [Accessed: 03- Dec- 2020].

[11] "In-line Flow Control Valve – F Series", *Parker.com*, 2020. [Online]. Available: https://ph.parker.com/us/en/in-line-flow-control-valve-f-series. [Accessed: 03- Dec- 2020].

[12] "Hydraulic Systems and Fluid Selection", *Machinerylubrication.com*, 2020. [Online]. Available: https://www.machinerylubrication.com/Read/277/hydraulic-systems-fluid. [Accessed: 05- Dec- 2020].

[13] "HEAT TRANSFER OILS - Phillips 66 Lubricants", *Phillips 66 Lubricants*, 2020. [Online]. Available: https://phillips66lubricants.com/product/heat-transfer-oils/. [Accessed: 06- Dec- 2020].

[14] "TROJAN AW-SERIES ANTI-WEAR HYDRAULIC OILS", Industrialfluidsystems.com, 2020. [Online]. Available: https://industrialfluidsystems.com/Linked%20data%20pages/0404501-H&W-Trojan%20AW%20Series.pdf. [Accessed: 06- Dec- 2020].

[15]"Certified Hydraulic Oil AW 32 Material Safety Data Sheet", 2015. [Online]. Available: https://webviewer.wercsmart.com/canadiantire_sds/client/GetDocument?id=50265bb2-35https://webviewer.wercsmart.com/canadiantire_sds/client/GetDocument?id=50265bb2-35c0-4843-9ecf-c857ac83062b&pid=91004540c0-4843-9ecf-c857ac83062b&pid=91004540. [Accessed: 06- Dec- 2020].

[16] "Splined Flexible Shaft Couplings", Mcmaster.com, 2020. [Online]. Available: https://www.mcmaster.com/shaft-couplings/for-shaft-type~spline/. [Accessed: 04- Dec- 2020

[17] Yamaha 2016 Service Manual YZFR6G, 1st ed. Yamaha, 2016.

[18] “Portable S-Type Beam High-Precision Load Cell Scale Sensor”, 2020. [Online]. Available: https://www.amazon.ca/Portable-High-Precision-Pressure-Tension-Weighing/dp/B077YHNNCP/ref=sr_1_1_sspa?dchild=1&keywords=load+cell&qid=1607285648&sr=8-1-spons&psc=1&spLa=ZW5jcnlwdGVkUXVhbGlmaWVyPUFLUkw2T0dKV05CSDgmZW5jcnlwdGVkSWQ9QTA2OTU0MzYyQlVTSEE5N0RMQTkxJmVuY3J5cHRlZEFkSWQ9QTAzMjY0MzMyM0JNS1pPUDFNRDdOJndpZGdldE5hbWU9c3BfYXRmJmFjdGlvbj1jbGlja1JlZGlyZWN0JmRvTm90TG9nQ2xpY2s9dHJ1ZQ==. [Accessed: 06-Dec-2020]

[19] “Load Cell Amplifier”, 2020. [Online]. Available: https://www.amazon.ca/Amplifier-Breakout-Converter-Raspberry-Microcontroller/dp/B07MTYT95R/ref=sr_1_1_sspa?dchild=1&keywords=load+cell+amplifier&qid=1607284182&sr=8-1-spons&psc=1&spLa=ZW5jcnlwdGVkUXVhbGlmaWVyPUFCVUgxVVc0M0RFMUImZW5jcnlwdGVkSWQ9QTA5NDUxOTgyVDhJSzJIMVZYWDFYJmVuY3J5cHRlZEFkSWQ9QTA5MTM1OTZJMjZXNTQzQ0tLNkEmd2lkZ2V0TmFtZT1zcF9hdGYmYWN0aW9uPWNsaWNrUmVkaXJlY3QmZG9Ob3RMb2dDbGljaz10cnVl. [Accessed: 06-Dec-2020]

[20] “Digital Force Gauge & Weight Scale w/ Loadcell & Arduino”, 2020. [Online]. Available: https://create.arduino.cc/projecthub/electropeak/digital-force-gauge-weight-scale-w-loadcell-arduino-7a7fd5. [Accessed: 06-Dec-2020]

[21] “Hall Effect Sensor Wiring Diagram | Wiring Resources”, 2020. [Online]. Available: https://circuitdigest.com/sites/default/files/circuitdiagram_mic/circuit-diagram-interfacing-hall-effect-sensor-with-arduino.png. [Accessed: 06-Dec-2020]

[22] "2010 Yamaha YZF-R6 on the Dyno", Cycle World, 2010. [Online]. Available: https://www.cycleworld.com/sport-rider/2010-yamaha-yzf-r6-on-dyno/. [Accessed: 07- Dec-2020].

[23] B. Munson, T. Okiishi, W. Huebsch and A. Rothmayer, *Fundamentals of Fluid Mechanics*, 7th ed. Hoboken, N.J: Wiley, 2013, pp. 426-440.

[24] “450 Shootout Winner and Dyno Results: The Wrap”, Dirt Bike Magazine, 2019. [Online]. Available https://dirtbikemagazine.com/450-shootout-winner-and-dyno-results-the-wrap/. [Accessed: 05- Apr- 2021].

[25] “Digital Force Gauge & Weight Scale w/ Loadcell & Arduino”, Project Hub, 2019, [Online]. Available https://create.arduino.cc/projecthub/electropeak/digital-force-gauge-weight-scale-w-loadcell-arduino-7a7fd5. [Accessed: 14-Feb-2021]

[26] “HX711 Arduino Library”, GitHub, 2017. [Online]. Available https://github.com/olkal/HX711_ADC [Accessed: 14-Feb-2021]

Appendix A

Main Arduino Code [26]

```
/*
   -------------------------------------------------------------------------------------
   HX711_ADC
   Arduino library for HX711 24-Bit Analog-to-Digital Converter for Weight Scales
   Olav Kallhovd sept 2017
   -------------------------------------------------------------------------------------
*/

/*
   Settling time (number of samples) and data filtering can be adjusted in the config.h file
   For calibration and storing the calibration value in eeprom, see example file "Calibration.ino"

   The update() function checks for new data and starts the next conversion. In order to acheive maximum effective
   sample rate, update() should be called at least as often as the HX711 sample rate; >10Hz@10SPS, >80Hz@80SPS.
   If you have other time consuming code running (i.e. a graphical LCD), consider calling update() from an interrupt routine,
   see example file "Read_1x_load_cell_interrupt_driven.ino".

   This is an example sketch on how to use this library
*/

#include <HX711_ADC.h>

//pins:
const int HX711_dout = 6; //mcu > HX711 dout pin
const int HX711_sck = 5;  //mcu > HX711 sck pin
const int hall_pin = 3;

//HX711 constructor:
HX711_ADC LoadCell(HX711_dout, HX711_sck);

// Global Variables
unsigned long t = 0;
// set number of hall trips for RPM reading (higher improves accuracy, lower improves refresh rate)
float hall_thresh = 20.0;

void setup()
{
  Serial.begin(115200);
  delay(10);

  Serial.println();
  Serial.println("Starting...");

  // RPM
  // make the hall pin an input:
```

```
  pinMode(hall_pin, INPUT);

  // FORCE
  LoadCell.begin();
  float calibrationValue;      // calibration value (see example file "Calibration.ino")
  calibrationValue = -6364.12; // uncomment this if you want to set the calibration value in the sketch

  unsigned long stabilizingtime = 2000; // preciscion right after power-up can be improved by adding a few seconds of stabilizing time
  boolean _tare = true;                 //set this to false if you don't want tare to be performed in the next step
  LoadCell.start(stabilizingtime, _tare);
  if (LoadCell.getTareTimeoutFlag())
  {
    Serial.println("Timeout, check MCU>HX711 wiring and pin designations");
    while (1)
      ;
  }
  else
  {
    LoadCell.setCalFactor(calibrationValue); // set calibration value (float)
    Serial.println("Startup is complete");
  }
}

void loop()
{
  //// RPM
  //// ===============================
  // preallocate values for tach
  float hall_count = 1.0;
  float start = micros();
  bool on_state = false;
  // counting number of times the hall sensor is tripped
  // but without double counting during the same trip
  while (true) {
    if (digitalRead(hall_pin) == 0) {
      if (on_state == false) {
        on_state = true;
        hall_count += 1.0;
      }
    } else {
      on_state = false;
    }

    if (hall_count >= hall_thresh) {
      break;
    }
  }
```

```
// print information about Time and RPM
float end_time = micros();
float time_passed = ((end_time - start) / 1000000.0);
// Serial.print("Time Passed: ");
// Serial.print(time_passed);
// Serial.println("s");
float rpm_val = (hall_count / time_passed) * 60.0;
// Serial.print(rpm_val);
// Serial.println(" RPM");
delay(1);        // delay in between reads for stability

String rpmString = rpm_val + String(" rpm, ");
String timeString = String("Time Passed: ") + time_passed + String(" s");

//// FORCE
//// ================================
static boolean newDataReady = 0;
const int serialPrintInterval = 1000; //increase value to slow down serial print activity

// check for new data/start next conversion:
if (LoadCell.update())
  newDataReady = true;

// get smoothed value from the dataset:
if (newDataReady)
{
  if (millis() > t + serialPrintInterval)
  {
    float i = LoadCell.getData();

    String forceString = i + String(" lb-ft");
    Serial.print(rpmString);
    Serial.print(forceString);
    Serial.println(timeString);
    newDataReady = 0;
    t = millis();
  }
}

// receive command from serial terminal, send 't' to initiate tare operation:
if (Serial.available() > 0)
{
  char inByte = Serial.read();
  if (inByte == 't')
    LoadCell.tareNoDelay();
}

// check if last tare operation is complete:
```

```
  if (LoadCell.getTareStatus() == true)
  {
    Serial.println("Tare complete");
  }
}
```