FSAE Engine Dyno

A MAE 156B Capstone Senior Design Project

Sponsored By:

Triton Racing

UCSD's Formula SAE Team

Presented To:

University of California, San Diego

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Abstract

This Mechanical and Aerospace Engineering (MAE) 156B project, sponsored by Triton Racing, was based on the Formula Society of Automotive Engineers (FSAE) team's need to develop a power measuring device that can test small combustion engines. This device will allow the team to optimize engine performance, by increasing the team's competitiveness in the FSAE Lincoln international competition held annually. The project's mission was to develop an engine dynamometer (dyno) test bed to measure an engine's performance, primarily by monitoring the parameters of engine torque and rpm. The dyno provides a real-time visualization of collected data, and allows the engine to be tested under various conditions for differing durations. Critical to the project success were two additional features; reliability – to accurately and consistently test and diagnose new and used engines; and safety – since the dyno would be operated by FSAE students in close proximity. The requirements of the dyno included simple operation, capability of measuring the power of a 89.48 kiloWatt (kW) engine, measurability and recordability of revolutions per minute (rpm) and torque, and expandable capabilities of the test bed in the future. The dyno components consists of: a welded steel frame, a manufactured aluminum water brake, a driveline connecting the engine to the water brake unit, a closed water cooling system, and a data acquisition system. All components of the dyno were designed, manufactured, and assembled in parallel by members of the MAE 156B team. The team was also able to run a mock test to verify the data acquisition system and ability to write data to an editable text file.

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Chapter 1: Project Description

Project Background

Triton Racing is the University of California, San Diego's Formula SAE team. Formula SAE is a collegiate society with an engineering project focus. The project is to design, build, test, and compete a Formula style race car from scratch (Fig. 1). The multidisciplinary team includes not only many types of engineers, but also students in business and other non-engineering majors who have a passion for design projects and in particular, race cars. At the end of each school year, Triton Racing competes against 80+ teams in an international competition held in Lincoln, Nebraska. This competition incorporates project management, engineering, and business to give teams a holistic view into real-world projects and their requirements.

Figure 1: Triton Racing's Cars On Display
(From Left: TR-9, TR-11, TR-15, TR-14, TR-13)

The competition gets tougher each year, and teams like Triton Racing are pushed to innovate and improve their design and testing methods to achieve the best results. Currently, the team is seeking to improve the performance of various parts and systems by utilizing better instrumentation for testing. The team has focused on measuring engine performance, and the proposed solution is to create an engine dynamometer (dyno) that will test the engine and measure specific parameters like torque and hp versus rpm.

Dynos are currently used in a wide range of applications. One such application is emissions testing, where road-like conditions are simulated to load a car's engine, and the results of the test are compared to national and state-wide standards. An engine dyno allows a user to measure engine torque and rpm while also providing real-time data visualization. The engine's performance can be measured and recorded across a range of different conditions and durations. This ultimately allows the team to perform diagnoses on new or used engines to determine their reliability and performance, while also providing measurable feedback for iterative improvements for competition.

There are two main types of dynos that exist for the team's application: chassis and engine (Fig. 2). A chassis dyno measures the power delivered to the road through the wheels of the car. An engine dyno measures the power output of the engine that goes directly into the transmission. To satisfy their testing needs, Triton Racing has been able to use a chassis dyno at a nearby shop in San Diego, but the cost and availability of the dyno limit the team. In addition, the chassis dyno does not provide the most accurate performance of the engine. Therefore, the engine dyno solution created by this Capstone Senior Design team enables Triton Racing to best characterize their engine performance at a fraction of the cost, and with much more flexibility in their own schedule. Furthermore, the dyno created features an engine subframe specifically designed for the Yamaha FZ6R motorcycle engine the team currently uses. The subframe will also allow for adjustability — to test a new engine; only a new subframe will have to be made. Additionally, the dyno allows for increased capabilities of testing with future developments by incorporating more sensors.

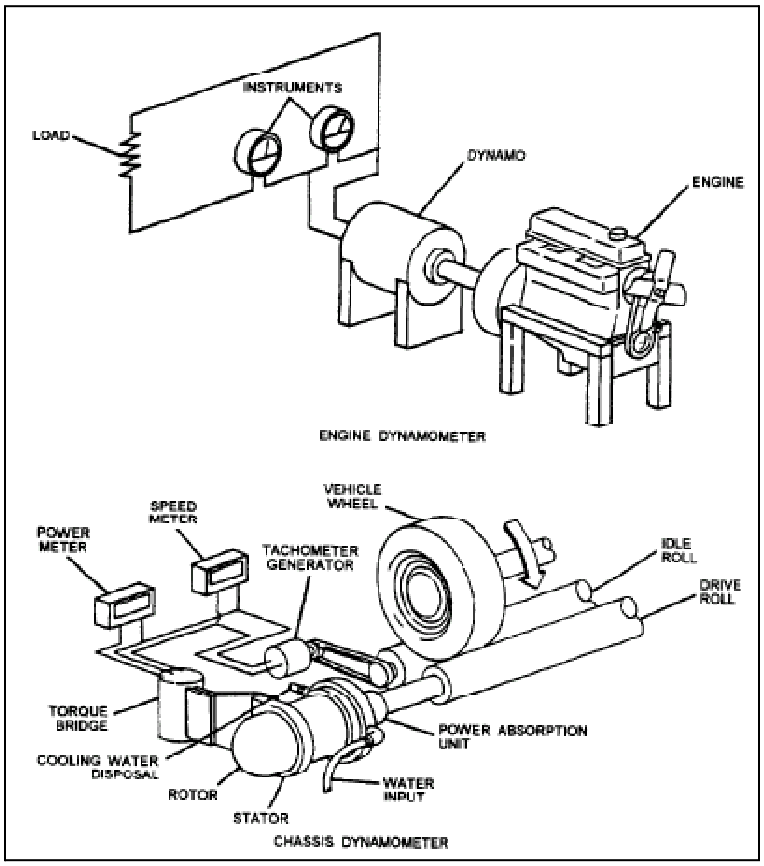

Figure 2: Engine (Top) and Chassis (Bottom) Dyno Diagram[1]

With this new dyno, Triton Racing will gain an upper hand in the building and testing of its race cars, enabling a possibility for better performance at competition. In addition, it also allows team members to learn and use a critical performance tool that is commonly used in the automotive industry and recreational projects.

Review of Existing Design Solutions

Engine dynamometers are made in a wide variety of horsepower capacities and range from a $1,000 to over a $100,000. A primary requirement of an engine dynamometer is to measure the power produced by an engine. In many of the different designs, the engine is loaded with a device that absorbs the engine power into a medium, such as water or air, using a pump or electrical load and fan.

The main categories of dynamometers are:

- Hydraulic
- Electric (AC, DC, eddy current)
- Water brake (toroidal and straight rotor)
- Mechanical brake
- Inertia-based

The simplest method to understand is the mechanical brake design, which typically includes some brake disks attached to a load cell for torque measurement, and an rpm sensor. Input engine power is converted to heat by the friction of the brake pads on the brake disk, and the torque and shaft rpm required to hold the engine at a constant speed are measured to calculate power as shown in equation (1.2.1). This design type has some drawbacks, however, including limited run time, inconsistent applied load from brake pads heating up and cooling down, and slow response time compared to other design solutions.[12]

$$Power = Torque * (Revolutions/Minute)/60 * 2 * \pi \qquad (1.2.1)$$

The least expensive type of dyno is an inertia based system. In this design, the engine spins a drum of a known inertia, usually through a relatively high gear reduction. The acceleration of the inertia drum is measured, which can then be correlated to a relative power value.[12] This system cannot test steady state performance and therefore is difficult to calibrate correctly. It should only be used for comparison purposes when altering some part on an engine after a baseline performance and tuning has been done.

Hydraulic dynos include a pump, valve, pressure gauge, and heat exchanger. This operation consists of an engine which spins a pump to pressurize hydraulic fluid as much as possible while the load is controlled by an outlet valve, where pressure is measured. Because of the nonlinearities of this loading approach and its limits on accuracy, these dynos are not typically used for engine tuning applications. In older systems, pressure is measured across the valve and correlated to horsepower through a table. The hydraulic fluid is assumed to be incompressible and valve volumetric flow rate is relatively constant. This approach is gaining traction in testing hybrid electric vehicles with complicated powertrain systems.

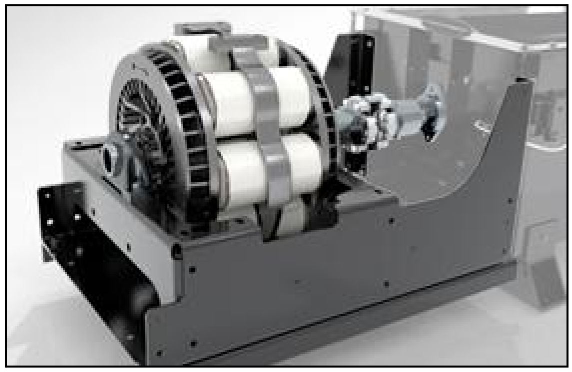

Figure 3: Dynojet 224x Eddy Current Dynamometer[9]

Eddy current dynamometers have the best engine control performance capabilities, and can achieve very smooth feedback control through an engine rpm range (Fig. 3). They operate by loading an engine with a rotor system that is slowed by eddy currents. Heat in the coils is dissipated into the surrounding air by a fan connected to the shaft. Electricity used by the eddy current coils is measured to give power as shown in equation (1.2.2). Unfortunately, due to their expensive cost, they are usually limited to high performance automotive applications, where reliable new testing systems can cost at minimum tens of thousands of dollars.[12]

$$Power = Current * Voltage \qquad (1.2.2)$$

AC and DC generator dynamometers also operate by using electricity. They are not as accurate or smooth as eddy current dynamometers but they can be made at a lower cost. They operate by loading an engine with a motor connected to a resistor load bank of variable capacity where the energy is dissipated as heat. Electricity generated by the generator is measured to give horsepower.

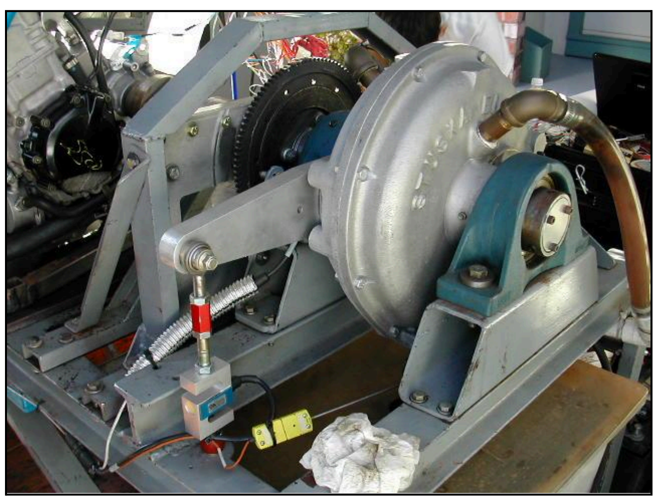

Figure 4: Stuska Dyno Used By Berkeley FSAE Team[8]

Water brakes, also known as hydraulic brakes; hydraulic retarder; or an inefficient pump; absorb engine power into water or another fluid that is pumped through the system. Across different water brake designs, the most common way to measure engine power is to continuously measure torque and input shaft rotational speed (Eqn 1.2.1).

An SAE paper on the Theory of Hydraulic Dynamometers states:

> The hydraulic dynamometer (Fig. 5) consists essentially of an impeller keyed to the main shaft and a stator mounted on trunnions. The rotor and the stator are each provided with an annular half-torus facing each other and fitted with radial blades. When the toroidal space is filled with water and the impeller is rotated by an external source of power, a toroidal flow is established outward in the impeller, which acts as a centrifugal pump, and inward in the stator, which acts as a stalled turbine. Due to the shock experienced by the water in passing from the rotating impeller to the stator, the torque impressed on the water by the impeller is transferred to the stator, on which it can be measured by any of several devices. [3]

Water brakes are also used in large automotive bus applications as a backup in case of brake failure, as they have a very large energy dissipation capacity relative to their weight.

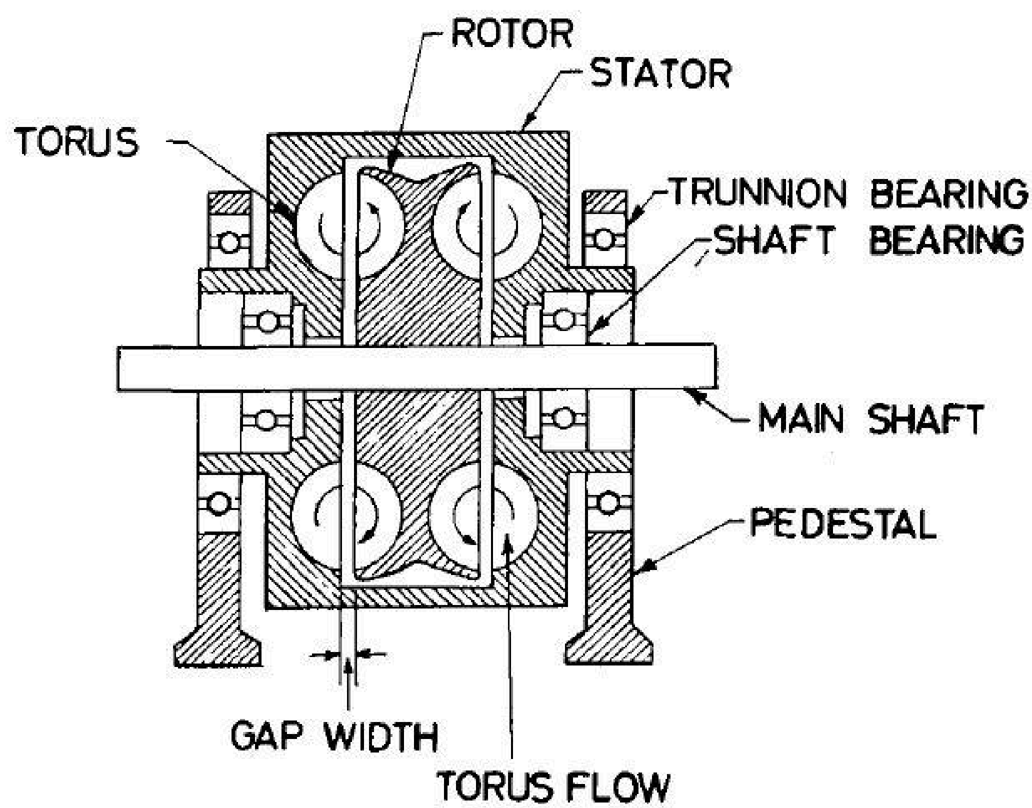

Figure 5: Diagram of a Water Brake[3]

Recently, automotive enthusiasts and other senior design teams have built their own dynamometer systems with varying degrees of success. This is likely due to the recent trend in electronics sensors and data acquisition becoming much less expensive as well as an increased access to online knowledge, resources, and manufacturing capabilities. This project aimed to bring the best out of a low budget dyno system, utilizing knowledge gained from previous published attempts at such a system.

Statement of Requirements

- The project requires the design and manufacture of an engine dynamometer test bed that will:
  - Measure maximum power output of 90 kW (120 hp)
  - Measure power while changing engine rotational speed
  - Measure power with repeatability of ±0.1 kW (0.134 hp)
  - Perform a duty cycle of 50% at one minute intervals
    - one minute of continuous operation, followed by one minute of no power transmission
  - Be safe to operate for a user with minimal experience
  - Be adjustable so that different engine models can be tested
  - Be compact and easy to transport

Deliverables

The following will be provided to the FSAE team at the end of the project:

- A functional water brake engine dynamometer test bed with:
  - A 30.48 cm (12 inch) diameter water brake with a torque arm
  - Water recirculation system including: a pump, a pump filter, and a 26.5 L reservoir
  - Cooling system composed of two radiators
  - Mobile frame enclosing all moving components
  - Sub-frame for ease of mounting different engines
  - Integrated sensors including: S-type load cell, rpm sensor, thermistors
  - Programmed data acquisition with LabVIEW and myRIO
  - Emergency stop button
  - Operator's manual
  - Report with design, analysis, test data, and overall performance review
  - Bill of Materials (BOM)
  - SolidWorks CAD files and drawings

Chapter 2: Description of Final Design Solution

Design Solution

The heart of this project's design solution is a custom water brake dynamometer and a myRIO data acquisition device (DAQ). This dynamometer was designed to test a motorcycle engine for use in the Formula SAE program. Other primary design considerations besides those listed in the functional requirements above are a relatively small budget and a high performance capability. In Figure 6 some of the main components are annotated. The engine under test is mounted onto a subframe that is connected to the rest of the system through U-bolts. Dual radiators are used to dissipate the power generated by the engine and absorbed by the water brake. Safety panels on the sides, front, and top protect the operator from any unpredictable accidents. In addition, the operator will stand behind the control station and monitor system parameters at all times the system is running. Operator controls include a inlet and outlet flow control valve, an emergency stop switch, a fan switch, throttle, clutch, and shifter controls.

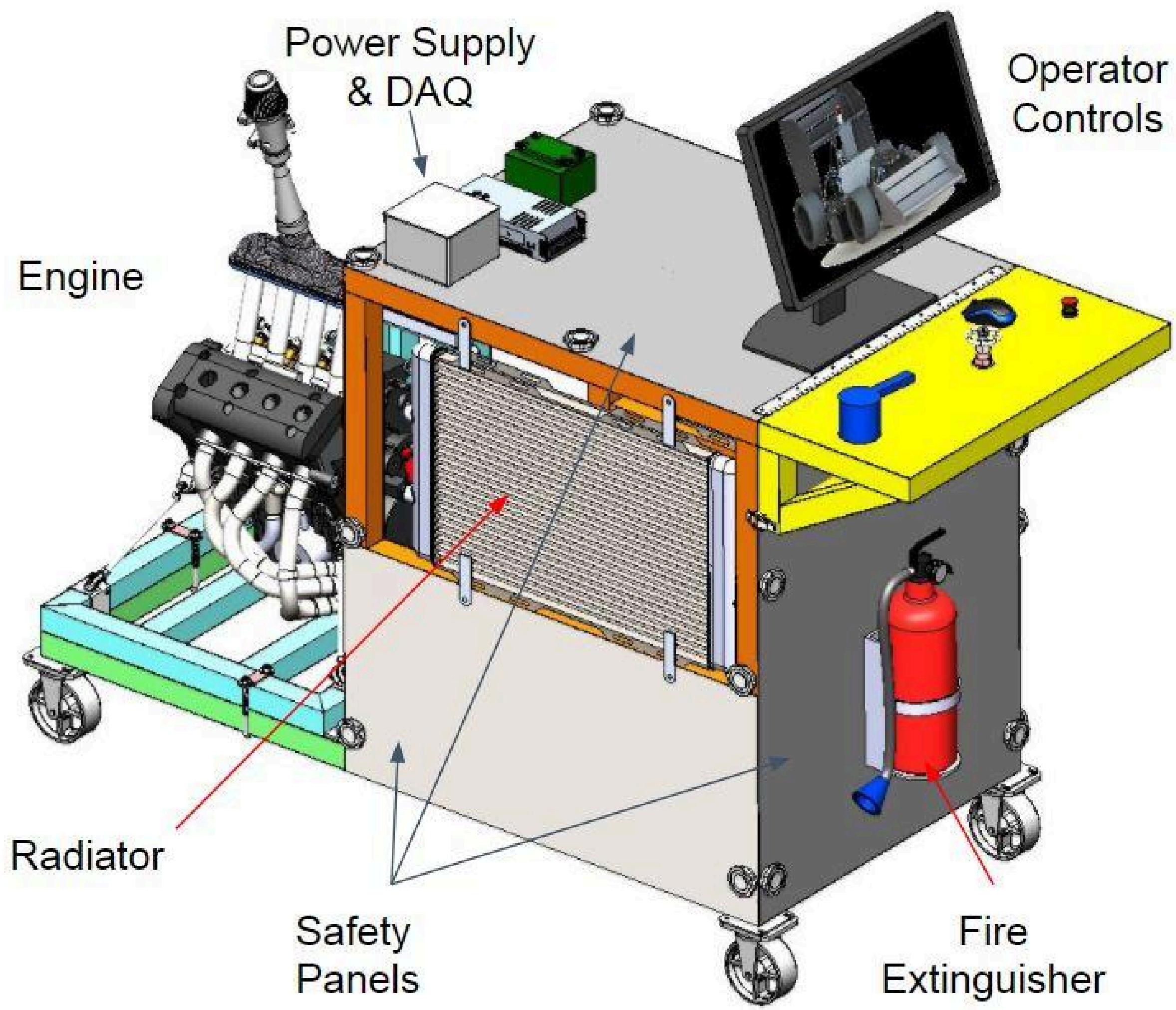

Figure 6: Isometric View of Final Design Solution

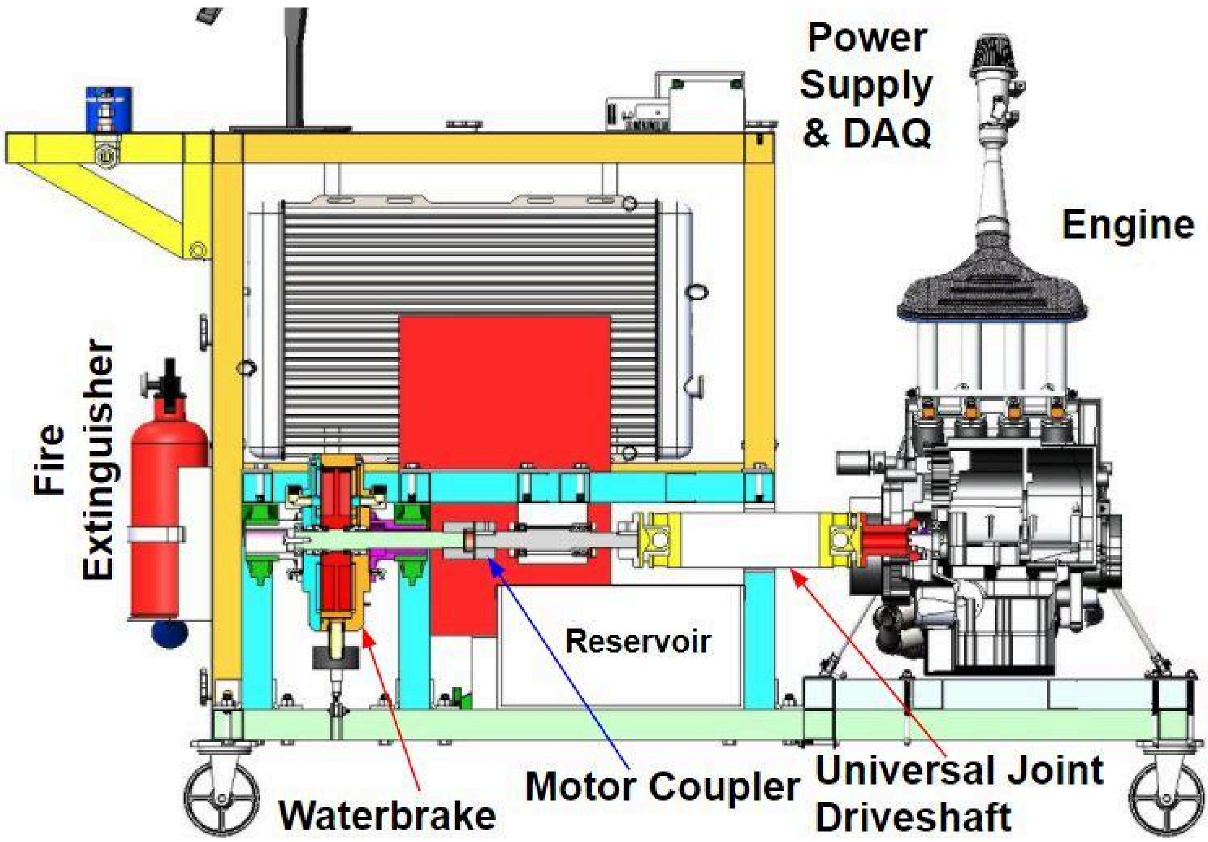

Figure 7: Section View of Final Design Solution

In order to measure engine power, measurements of shaft speed and torque produced by the water brake are needed (Eqn 1.2.1) To measure shaft speed, a Littelfuse Hall-Effect Sensor detects the rotation of a gear tooth wheel implemented on the output shaft of the engine. For continuous torque measurement, an AnyLoad NH-500 load cell is mounted at a moment arm, 0.1524m(6in) away from the rotational axis of the water break's output. The calibration mounting distance for the load cell is 0.3048m (12in). The load cell assembly is mounted on rod ends to ensure the load cell is purely in compression and includes a sandwich mount vibration isolator to filter out unwanted disturbances in the data.

The current design is operated manually by opening or closing the inlet valve to the absorber system, thereby restricting the flow of water to the rotor blades and reducing the load on the engine. With this method, the engine output speed can be varied between idle and redline for a given throttle position. If the inlet valve is in a completely open state, with the maximum water flowrate into the water brake, the engine will stall. Figure 8 shows the completed system.

Figure 8: Final Design

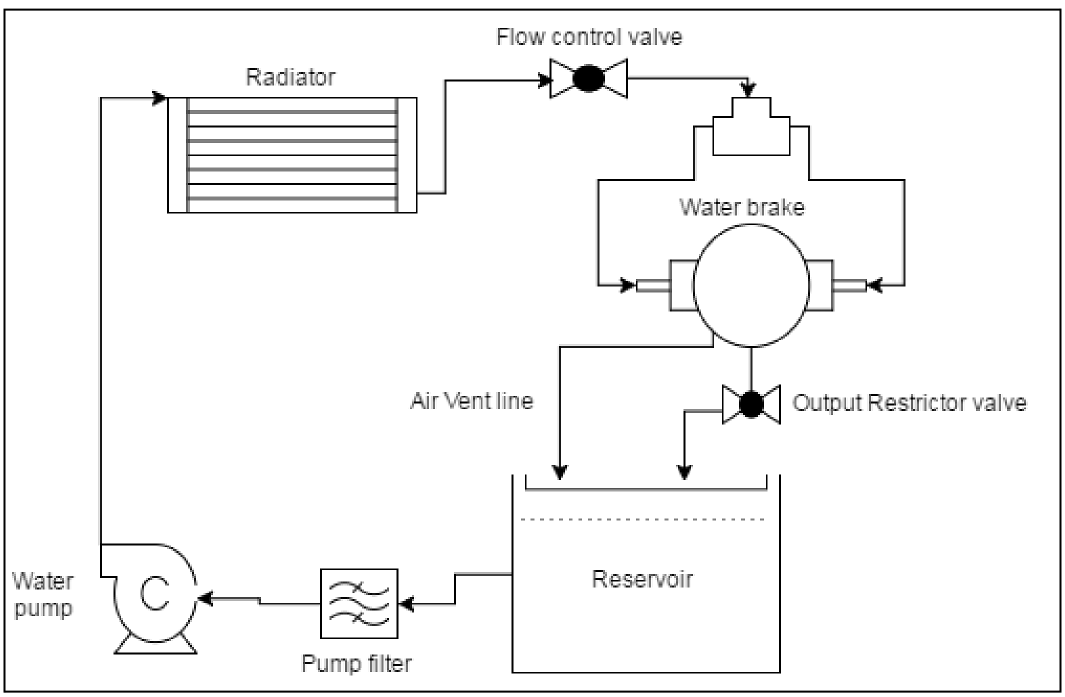

Figure 9: Water System Connection Diagram

The water system, as shown in Figure 9, includes several key components including: a pump filter, reservoir, circulator pump, radiator, two control valves, and flexible PVC tubing. The 26.5 L reservoir is actuated by a Taco 0011 circulator pump, which is primed by being located beneath the water level of the reservoir. From the pump, the water travels through a radiator and up to an inlet flow control valve actuated by the operator. From the inlet valve, the water flows into a T-fitting and into both inlets of the water brake for even loading. An air vent line connected to the reservoir at atmosphere prevents both high and low pressure from accumulating inside the water brake. An output restrictor valve allows the water brake to function in a specific power range for each engine used, by allowing for controlled unloading. Water brake output flow can be up to 55°C before testing must be stopped to let the system cool down.[13]

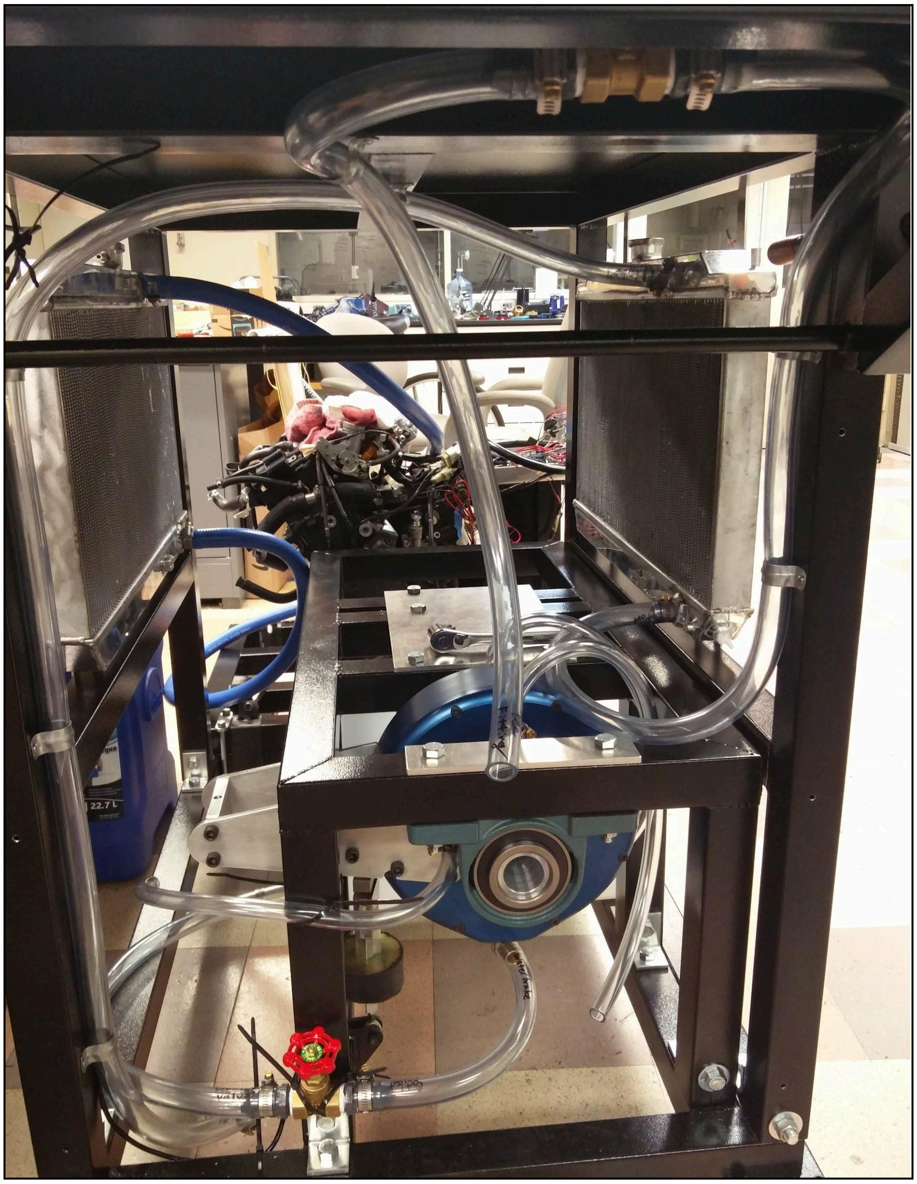

Figure 10: Water System Plumbing on Final Design

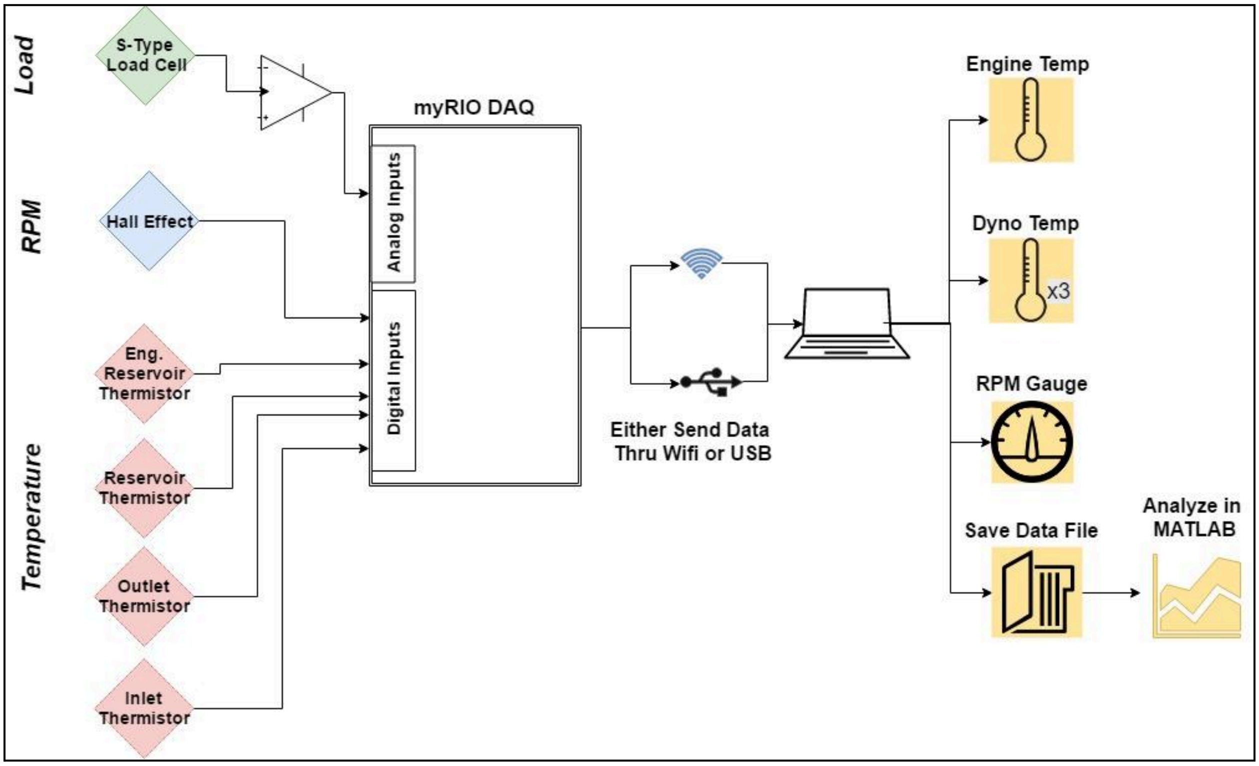

Figure 11: DAQ System Connection Diagram

The data acquisition connection circuit is shown in Figure 11 and includes a myRIO DAQ system which was donated to the project by National Instruments. Connected to the data acquisition system are four one-wire waterproof thermistor DS18B20 temperature sensors, a AnyLoad NH-500 load cell, and a Littelfuse Hall-Effect Sensor for shaft speed.

The fan power circuit includes a 12 V, 360 W power supply to power all four of the 80 W radiator fans. The pump is powered directly off of mains power voltage, as it includes its own controller circuitry.

The engine power circuit includes a Yamaha FZ6R battery to start the engine and provide power for the engine harness. An emergency stop switch is put in circuit to allow shutdown of the system at any time via shutting off power to the fuel injection system. All frame components are grounded to reduce stray electromagnetic interference (EMI). An extension cord is run from where testing is done outside to the nearest building or to an onsite generator.

Chapter 3: Design of Key Components

3.1 Frame

Functional Requirements

- Provide mounting for all system components
- Allow system mobility on a variety of outside surfaces
- Provide comfortable control panel usage for a range of operator heights
- React engine torque through the frame to the load cell

Five subframes make up the overall structure of the system (Fig. 12). The subframes are connected at specific locations with 6.35 mm (¼ in.) thick L-brackets. The engine subframe is connected with U-bolts to allow for easy removal and flexibility in the future. Overall, this modular frame design choice allows for easy access when performing maintenance or fixing a specific subcomponent of the design. This design also allows for adjustability should a subframe or driveline component need alignment with shims due to manufacturing uncertainties. The frame design is relatively lightweight (91 kg) compared to other dynamometers as it needs to be mobile. The frame was powder coated to a durable outdoor black finish as shown in Figure 13. The design is intended to provide a long lasting sturdy platform for testing many years into the future.

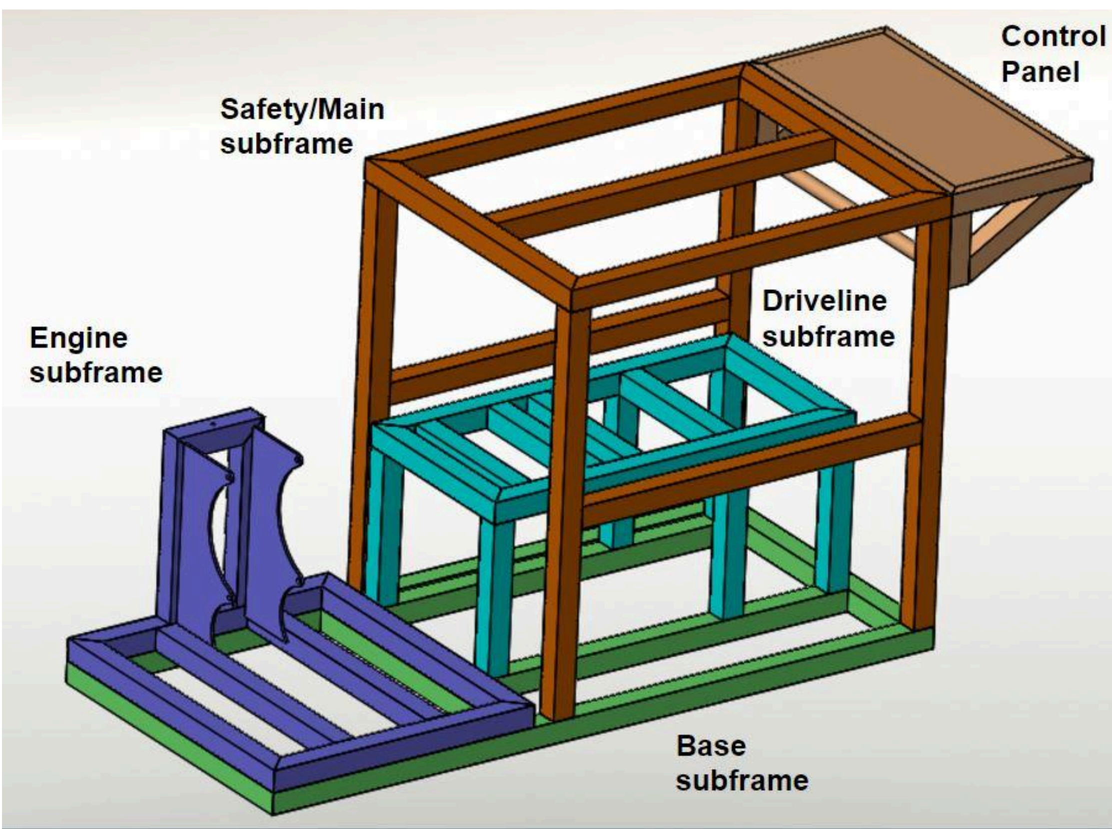

Figure 12: Subframe CAD Assembly

Figure 13: Subframe Final Assembly

Analysis on the performance of the frame design was done with Solidworks FEA using beam elements and applying loads at the joints. Two load cases were considered. First, the engine torque output (75 Nm) and consequential water brake reaction torque were applied to two of the four corner frame joints. Two fixed joint constraints were applied on the other two of the four frame corners to simulate a ground constraint and fully constrain the model. These boundary conditions allowed for simulation of the rigidity of the frame when exposed to this torsional load. The results, which are shown in Figure 14 reveal a maximum deflection of less than a millimeter which is acceptable in this design and will not impact measurement performance or system function.

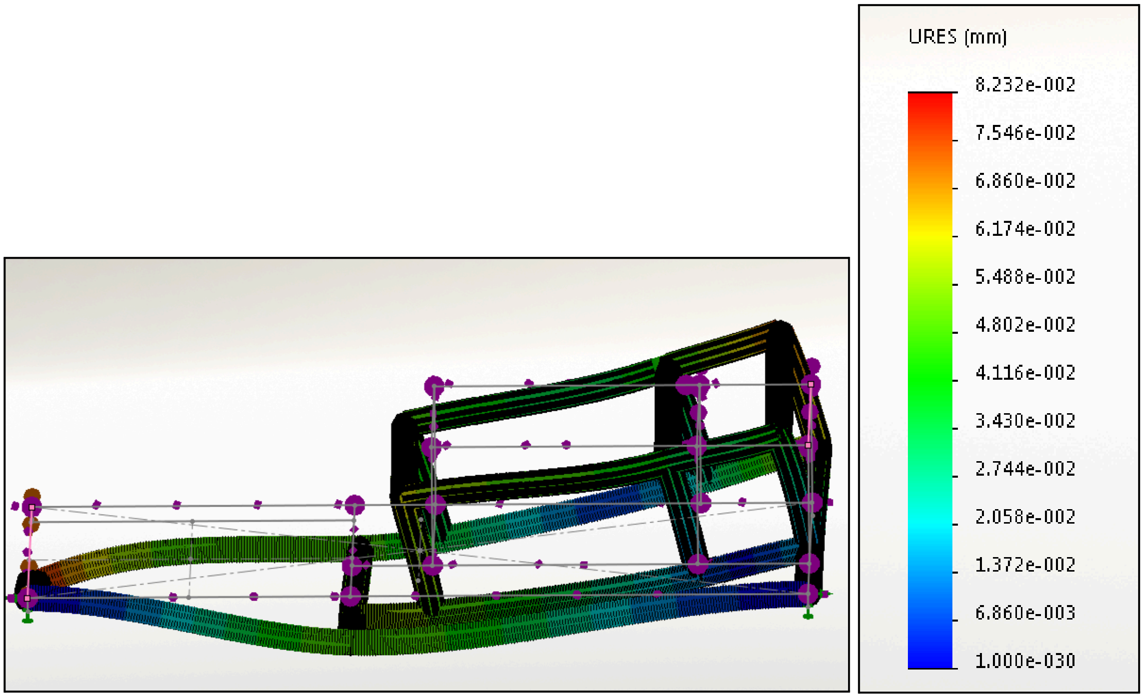

Figure 14: FEA of Frame Under Load Case 1 (Maximum Engine Torque)

The second load case considered the weight of the components on the base frame as well as additional weight from persons or other misuse conditions. Load was applied at a worst case scenario condition, in the middle of the frame. In a similar fashion to the first load case, all four of the outer corners of the frame were fixed to simulate a flat ground surface reacting the force. The cargo weight was estimated at around 1300 N (300 lbf) and the results are shown in Figure 15. These results show the frame has a factor of safety of 4.

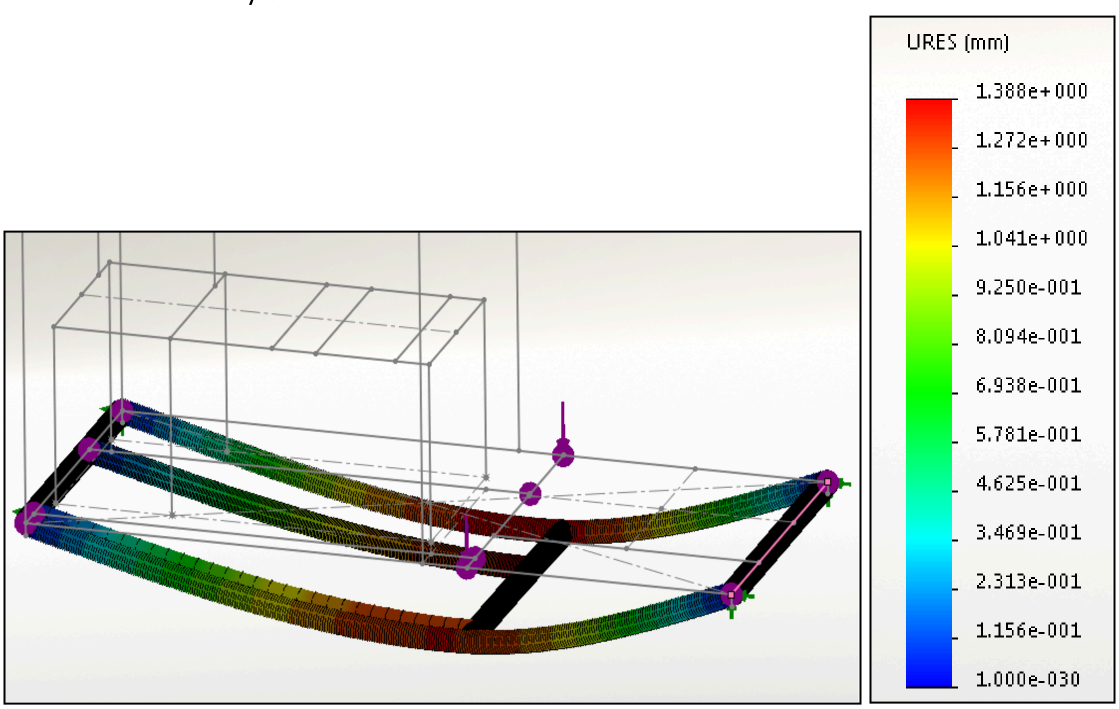

Figure 15: FEA of Frame Under Load Case 2 (Frame Cargo Weight)

3.2 Water Brake

Functional Requirements

- ➢ Ability to absorb and dissipate 90 kW (120 hp) of power continuously
- ➢ Rotate at speeds between 1500 and 6000 rpm

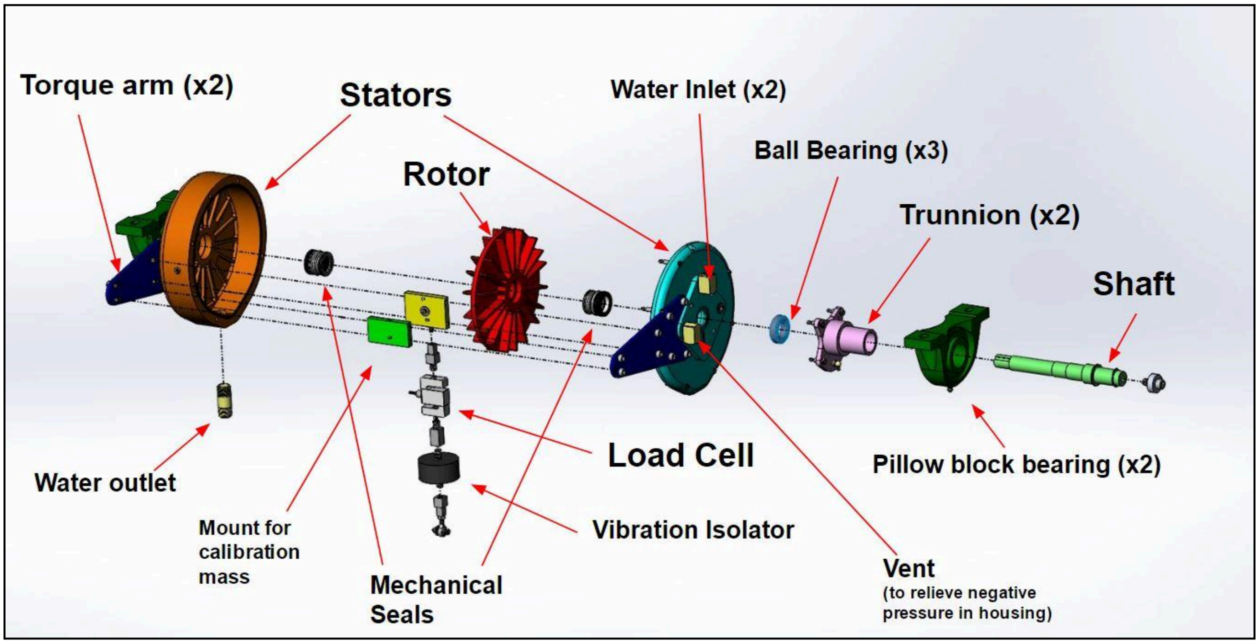

Figure 16: Exploded View of Water Brake

Since the water brake unit must absorb all of the power output by the engine tested, it is the most critical component of the dynamometer system. Designing the water brake was thus an iterative process which took several weeks to complete. Since this project must be completed within 12 weeks, designing a water brake from the ground up was a difficult task; however, water brakes have been in use since the late 19th century as devices for absorbing mechanical power and are well understood in their respective industry. To expedite the design process of the water brake unit, the design was loosely based on a currently existing water brake: the XS-111, which is manufactured and sold by Stuska Dynamometer. [13]

Torque capacity of a water brake is governed by the following equation:[3]

$$T = K * N^2 * D^5 \tag{3.2.1.1}$$

Where T is the torque capacity, K is the capacity constant, N is the rotational speed of the machine, and D is the diameter of the rotor. The capacity constant, related to the geometry of the rotor and stators, requires intense calculations and testing beyond the scope of this project. As such, known parameters from the XS-111 were used to determine the appropriate sizing of the water brake.

The water brake assembly consists of multiple important parts (Fig. 16), of which several were designed and manufactured as prototypes for the purpose of this project, including the rotor, stators, trunnions, and torque arms. A cross sectional view of the assembled water brake unit can be seen in Figure 17.

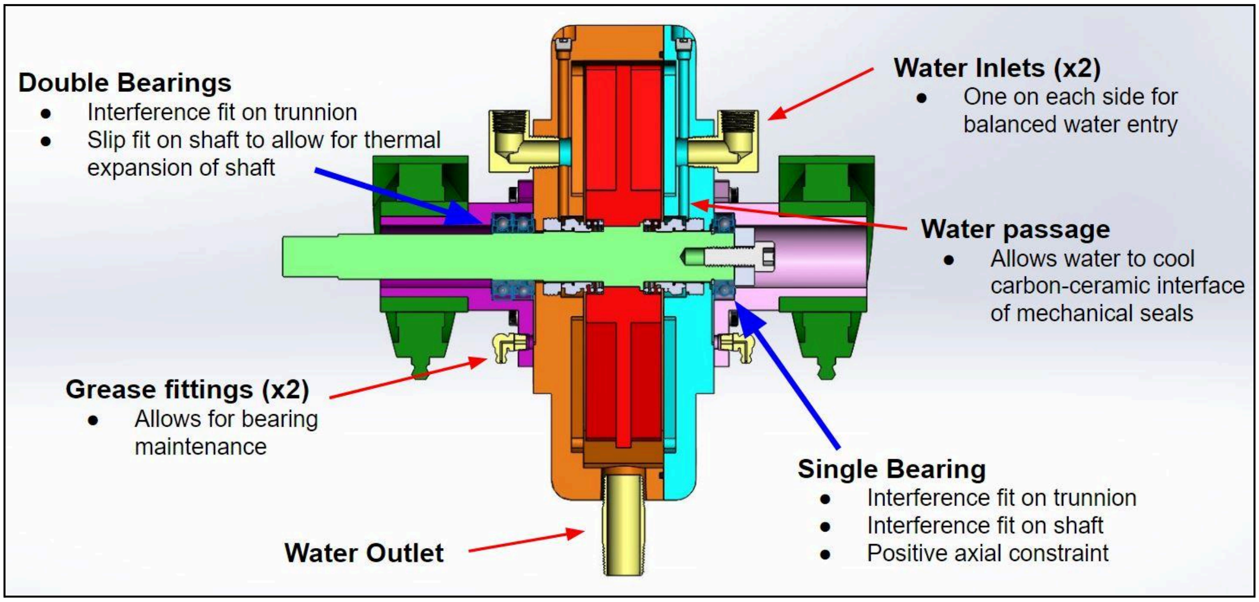

Figure 17: Section View of Water Brake

Rotor

The rotor in the XS-111 has a diameter of 30 cm and is rated for 300 kW. Power is directly proportional to torque (Equation 1.2.1) and the torque capacity is proportional to the fifth power of the diameter of the brake (Equation 3.2.1.1). With these realizations, the parameters of the XS-111 were used to approximate the minimum required diameter of the water brake rotor, which was calculated to be 20.5 cm. To account for uncertainty, the rotor was designed with a diameter of 24 cm, giving a theoretical power limitation of 200 kW. Furthermore, the rotor vane profile was designed to be similar to the XS-111, with minor exceptions to simplify the manufacturing process. To reduce system harmonics, the rotor blades are variably spaced such that they are not equidistant to each other (Fig. 18).

After completion of manufacturing, the rotor was coated with type 2 anodizing to prevent galvanic corrosion between the shaft and rotor. The assembly final shaft assembly was professionally balanced for high speed rotation.

The stainless steel water brake shaft is heat shrink press fitted into the rotor as an 80 micron interference fit. Typically in applications such as these, positive retention with a shaft key is preferred, however, this was deemed unnecessary to satisfy the requirements. The required interference between the water brake shaft and rotor was calculated using a program which considers material properties including yield strength, modulus of elasticity, poisson's ratio and friction coefficient of the mating materials, as well as interference and contact area. Results are shown in Table 1. With an 80 micron interference, the theoretical torque required for relative movement between the rotor and shaft is 1744.5 N-m. Since the maximum torque specified by the design requirements is 110 N-m, a factor of safety of 10.6 is achieved.

Table 1: Rotor-Shaft Interference Fit Holding Force at Different Temperatures

Temperature Condition	80 °C (MAX Condition)	20 °C (MIN condition)
Engine torque(ft*lbs)	55	55
Engine final drive gear ratio (2:1)	110	110
Engine torque reacted at press fit diameter (lbs)	1744.5	1744.5
Press fit calculator results (N)	30524	81444
Press fit calculator results (lbs)	6937.3	18510.0
Factor of Safety at MAX Temp condition	4.0	10.6

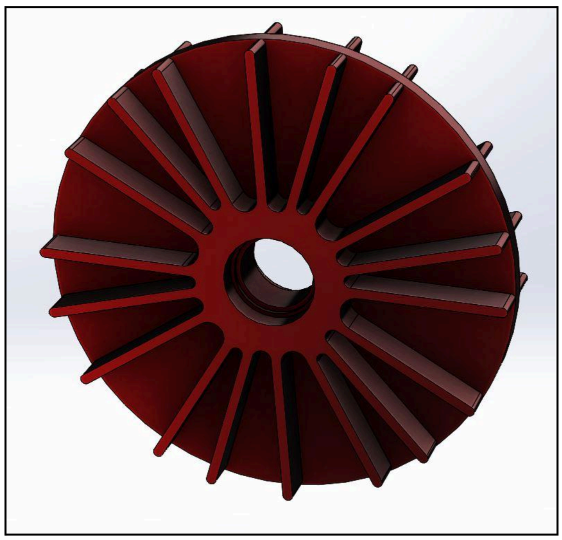

Figure 18: Rotor Design

Stators

The stators (Fig. 19 - 20) provide a chamber for the rotor to operate as well as machined features by which viscous and shear forces will accumulate and provide a measurable torque corresponding to the output of the engine being tested. The stator has vanes which cause torus flow as the water circulates. As the rotor rotates, water is sheared in between rotor and stator blades[3]. Furthermore, the stators will house dynamic pump seals to prevent leakage of the system. Each stator was designed with a water cooling pathway from the water inlet to the carbon-ceramic interface of the seals to prevent overheating. An outlet at the bottom of the stator allows water to escape the chamber at an adjustable rate via a gate valve.

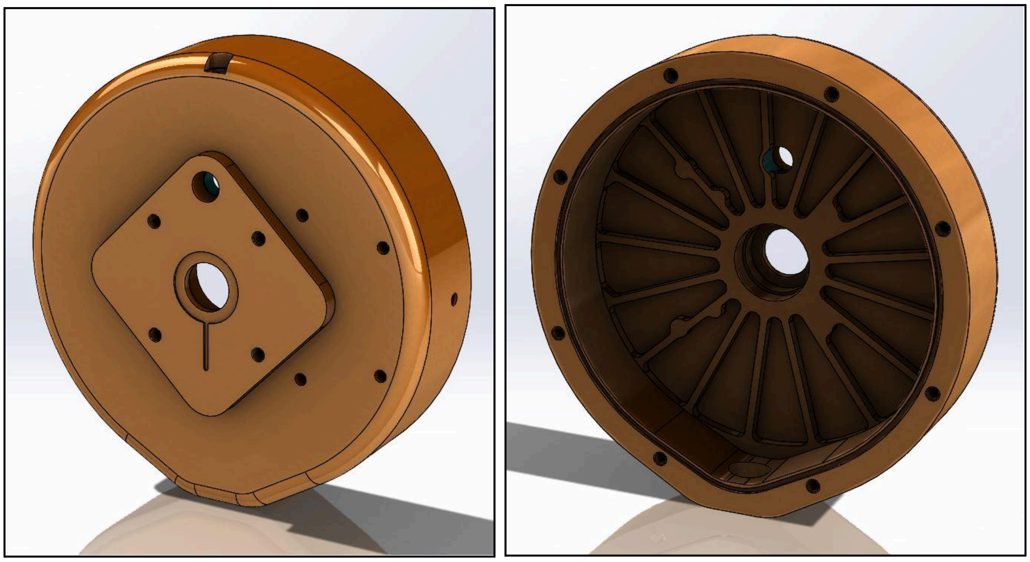

Figure 19: Driveline Side Stator Design

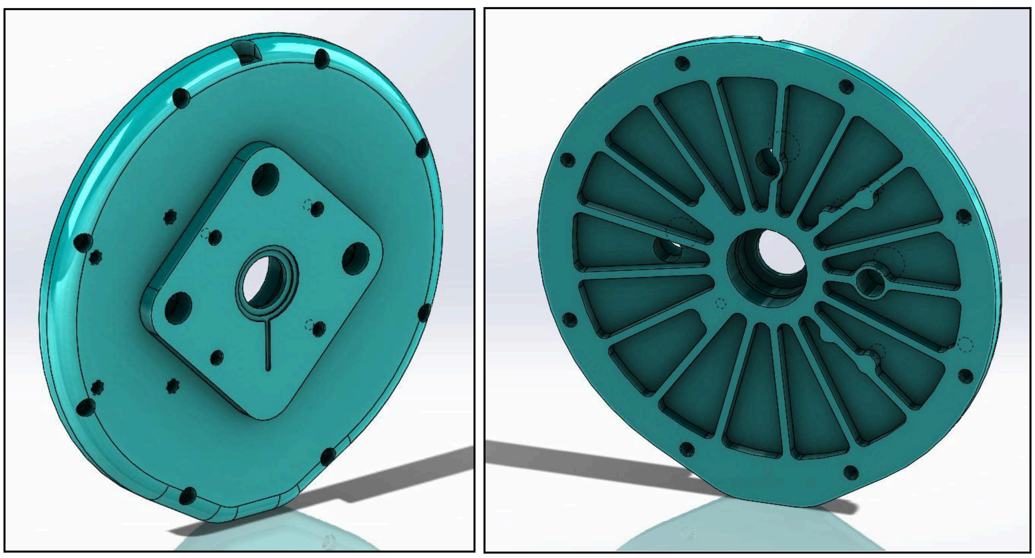

Figure 20: Outer Side Stator Design

Trunnions

The trunnions (Fig. 21) house the internal bearings as well as grease fittings for periodic maintenance of the bearings. The trunnions mount directly to the stator and provide a provision for the water brake to be mounted to large external pillow block bearings which reduce the system to a one degree of freedom (rotational) system. The water brake has two unique trunnions, one of which holds a single 6006 ball bearing with an interference fit on the outer race of the bearing,and the second holds two 6006 ball bearings, both with an interference fit on their outer race. The bearings are axially constrained due to the interference fit, as well as the sidewall of the trunnion and stator when assembled. Four threaded holes are included in the single bearing trunnion for ease of disassembly (See operator's manual in Appendix H for more details).

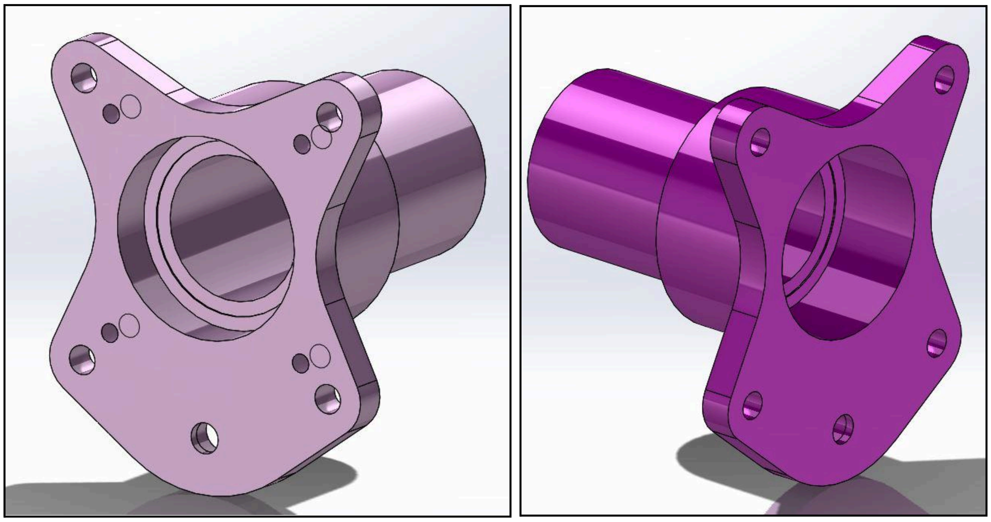

Figure 21: Trunnion Design
Axially Constrained Side (Left); Slip Fit Side (Right)

Torque Arm

The torque arm (Fig. 22) is mounted to the outside of each stator at its base and is tapered to a narrower section where it is attached to the load cell which is fixed to the frame via a rod end joint. The arm thus reacts to the torque produced by the water brake such that the system is fully constrained and purely in compression. The load cell is mounted 0.1524m (6 in.) away from the rotational axis of the water brake and the calibration mounting distance for the load cell is 0.3048m (12 in.).

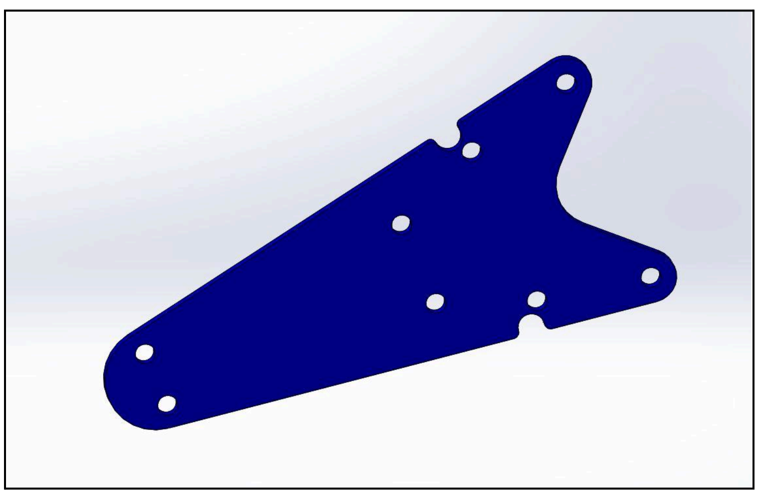

Figure 22: Torque Arm Design

3.3 Radiators

Functional Requirements

- ➢ Dissipate maximum engine power output 90 kW (120 hp) at steady state with no wind
- ➢ Operate with water at a maximum of 100^{o} C for engine cooling
- ➢ Operate with water at a maximum of 55^{o}C for the water brake side
- ➢ Minimize cost

Comparison of Designs Considered

The dual radiators in this dynamometer design are used to exchange heat energy from the engine coolant and the water brake system into the air. Critically, the radiators must be able to operate in a stationary condition which is unlike the configuration typically used in moving automobiles. The radiator fans alone must provide enough cross flow to generate sufficient heat transfer.

There are many different types of radiators that could be used in this application. To narrow the design scope, a thermodynamic analysis was done to estimate the radiator requirements. This analysis is presented in section 4.1.1. In summary, through the analysis, it was found that an automotive radiator would meet and exceed performance requirements.

Now that the size requirement was taken care of, the focus was shifted to optimizing the cost of this component. After investigating various aftermarket, OEM, and custom fabrication solutions, it became clear that the best price to performance ratio was found in an after market solution that was modified to fit the project's requirements.

Final Design Choice

With the results of the performed analysis, an inexpensive radiator which was comparable to the one used in the analysis was found on eBay and retrofitted to mount into the design and tubing size (Fig. 23). The radiator used in this design has a core size of 66.67cm x 32.385cm x 4.445cm and is an automotive type after-market radiator originally meant for an Acura Integra. Modifications made to the radiator are shown in Figure 24. The fans are mounted to the radiators with springs to prevent vibrations from transferring into the frame. Custom fan shrouds were also manufactured to fit the radiators.

The second radiator used in this design is for engine cooling during testing. It was determined by the sponsor that the same model radiator could be used for both cooling the water brake system as well as cooling the engine.

The dual fans used for each radiator have the following specifications:

- Maximum Fan Flowrate (m^3/s): .816
- Fan Diameter (cm): 30.6
- Thickness (cm): 6.35
- Number of Blades: 10 blades
- Maximum Fan (rpm): 2,250 rpm
- Amp Draw (A): 12.70

Figure 23: Aftermarket Automotive Radiator Used Final Design

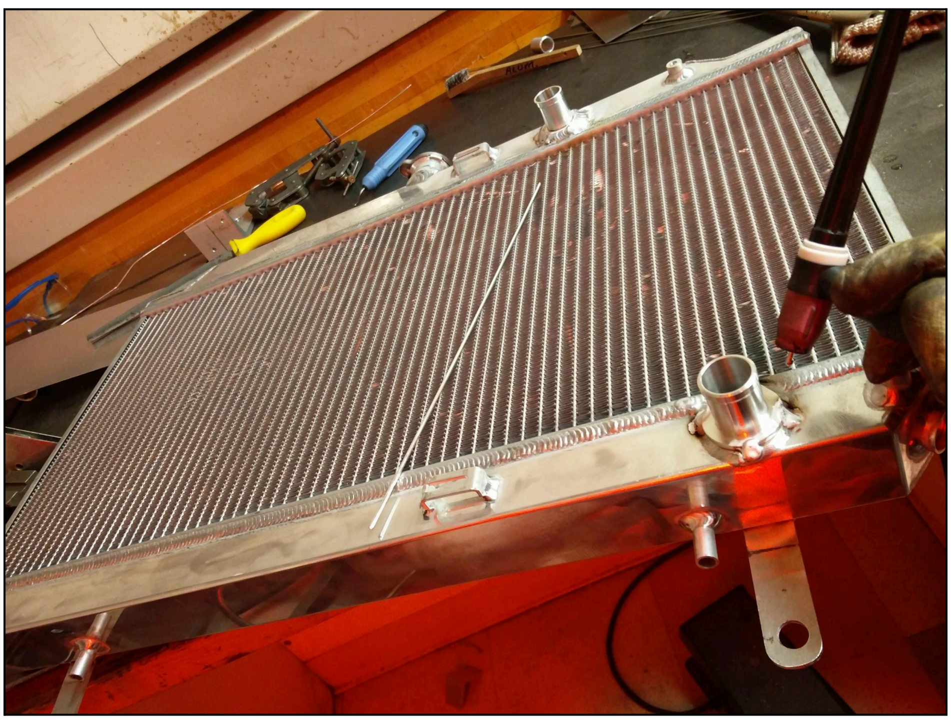

Figure 24: Mounting Brackets and Inlet/Outlet Modifications Made

3.4 Pump

Functional Requirements

- ➢ Pump water at a flow rate of 0.946 L/s (15 gpm) max at 3.32 meters of head
- ➢ Operate with water at a maximum of 50 ℃
- ➢ Be compact and fit on the frame

The pump is used to draw water from a reservoir, through a radiator for cooling, and to the water brake to create a load that is read with a torque arm and load cell. A water circulation pump was chosen because it best met the functional requirements of the system listed above. It is necessary that the water pump in the dynamometer system provides a high enough flow rate to load the water brake. The Stuska XS-111 Dynamometer operation manual states the maximum flow rate needed for their water brake system design is 0.946 L/s (15 gpm).[13] A pump flow analysis was done to determine the total head loss including frictional losses in the water system. Since this calculation was done without the actual fittings and tubings that will be used, the results will vary slightly to the true head loss. Through flow and head loss calculations, it was concluded that there would be a total head loss of 3.32 m (10.89 ft) as shown in Chapter 4.2.1. This information is useful to determine the specifications required for pump performance. Therefore, a proof of concept for the water system was performed to ensure that the pump functions as needed. The pump was able to empty the 26.5 L reservoir in ten seconds and was found to be sufficient as this is a flow rate of 2.5 L/s.

Pump Designs Considered

Table 2: Pros and Cons of Pump Types Considered

Model	Pros	Cons
Red Lion	6.309E-4 m^3/s (17 gpm) max, 2 year warranty	high price, not self-priming
Graymills Immersion	High operating temp, submersible and self-priming	Most expensive, made for coolant
TACO 0011	High operating temp, 32 feet max head	Not self-priming
Astro 210 Cl	Check valve, meets requirements	High price
Wayne	Self-priming, durable cast iron, 1.514E-3 m^3/s (24 gpm) max	High price, not self-priming
Arksen	Self -priming, lowest cost	9.65E-4 m^3/s (15.3 gpm) max

Final Design Choice

After careful consideration of multiple water pumps, the TACO 0011 (Fig. 25) pump was chosen. After reviewing the spec sheet and the pump curve, the team determined that this pump would work best for the system because at 3 m of head it produces 1.4 L/s, and this would allow for frictional losses through the fittings and piping system since the water brake requires a max of .946 L/s for loading. This pump is typically used in household hot water applications and has an operating temperature of 110 ℃ maximum, which was another deciding factor because the water will be at 50℃. The TACO 0011 was also the least expensive pump, and it was possible to cut cost by purchasing it on eBay. Low cost was a deciding factor for pump consideration, as the project budget was very tight.

Figure 25: TACO 0011 Pump[15]

3.5 Data Acquisition (DAQ)

Functional Requirements

- ➢ Simple to program and read data
  - ○ Any Triton Racing member should be able to operate the dyno after some basic instructions and safety guidelines are explained.
- ➢ Have expandable capabilities to incorporate more sensors in the future
- ➢ Sampling rate (critical for rpm and load cell) should be at least 500 Hz
  - ○ 500 Hz was the maximum signal frequency the rpm sensor was chosen to read (See “Rpm Sensor” in Section 3.6 “Sensors”)
- ➢ Withstand typical outdoor environmental conditions in San Diego as well as moderate engine vibrations
- ➢ Multi year service life with occasional heavy use for a couple weeks at a time

Comparison of Designs Considered

Three options were considered for use of a DAQ system: Arduino, Raspberry Pi, or a National Instrument’s DAQ. The DAQ was considered solely because of the availability of DAQs by the MAE department for testing purposes. Initially, the team had chosen a Raspberry Pi over the Arduino because of its general ease of usage for data visualization as well as having experience with programming it. However, after a lecture on LabVIEW and the DAQs owned by the MAE department by Greg Specht, the team saw a potential for using the National Instruments hardware. After exploring sponsorship opportunities from National Instruments and representative Ingo Foldvari, the team reasoned that the free DAQ as well as the resources available with programming in LabVIEW gave enough motivation to switch over. In addition, testing and programming with the DAQ could begin immediately since one was provided by the Electronics Lab of the MAE department rather than having to wait for a Raspberry Pi to be purchased and delivered. In addition, the Triton Racing team primarily consists of MAE students who are taught to use LabVIEW. This advantage would allow more members to be familiar with programming in LabVIEW and thereby minimize the learning curve.

Final Design Choice

Once the National Instruments brand of DAQ systems was decided upon, the choice of which DAQ to use was next. Between meeting the basic requirements of the six sensors to be implemented, the myDAQ and myRIO were the available options. While both met all the remaining requirements, after speaking to the National Instruments representative and the Triton Racing team, it became apparent that the myRIO (Fig. 26), would give the team the best chance to grow and incorporate more sensors and control as needed. In addition, the myRIO also has many more analog and digital I/O pins than what was required, as well as a separate field programmable gate array (FPGA) which could expand future dyno capabilities. While the myRIO’s cost was more than that of the myDAQ or significantly more so than that of the Raspberry Pi, the team was able to ensure a donation of an additional myRIO by National Instruments, which nullified the cost issue. Conclusively, the myRIO was the best choice , and the team could jump-start testing and programming within a week of choosing the DAQ system.

Figure 26: myRIO Data Acquisition System[17]

3.6 Sensors

The base requirements of the project deemed that solely torque, rpm, and temperature would need to be measured by the test bed. Future improvements by Triton Racing would include the expansion of the sensor portfolio as the team saw fit.

Load Cell

Introduction

The primary measurement sensor that is required in the dyno is the load cell. The load cell is attached to a torque arm which is securely mounted onto the water brake. As the water is pumped through the system, it will exert a torque on the brake as the rotating shaft inside pushes the water to the stator walls. This torque is exhibited through the torque arm located at a set distance of 15.25 cm (6 in.) away from the center of the brake to create a small vertical force. To measure this force, an S-type load cell was mounted in line with the force through two tie rods that mount to the torque arm and the base of the frame (Fig. 27).

Functional Requirements

- ➢ Mass requirement of 150 kg for worst-case scenario (see calculation below)
- ➢ Capable for use in tension and compression setups
  - ○ Decision to purchase load cell was made before final frame design was completed

Determining the mass requirement (from above):

From previous engine dyno data that was provided by the Triton Racing team, the team knew that the torque exerted by a 64 kW (85 hp) engine would be 163 Nm (120 ft. lb.). To consider a 90 kW (120 hp) engine as stated in the functional requirements of the project, a simple ratio calculation was made since no true method of determining max torque can be used.

$$\frac{120\ hp}{85\ hp} = \frac{x\ Nm}{163\ Nm}$$

(3.6.1.1)

This ratio gave the torque requirement for the system to be 230 Nm (170 ft lb). Since the load transmitted through the torque arm to the load cell decreases as the length of the arm increases, a worst-case design for the torque arm was assumed to be 15.54 cm (0.5 ft.) since that is the smallest length the arm can be since it is mounted to the 31.08 cm (1 ft.) diameter water brake. As such, the capacity required by the load cell would be 1480 N (340 lbf). Assuming gravity constant as g= 9.81 m/s^2, the mass requirement is 150 kg.

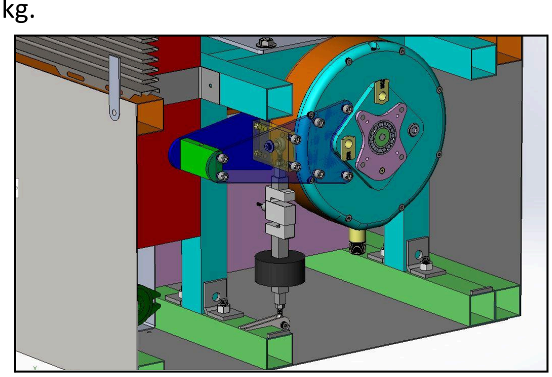

Figure 27: Section View of the Final Design Showing Load Cell Mounting Location

Comparisons of Designs Considered

An in-depth comparison of designs considered can be seen in the Appendix I. First, the type of load cell selected for the final design was chosen based on their different applications and use. As such, S-type load cells were chosen, since professional dyno manufacturers as well as many enthusiasts use S-type load cells in their dynos. Secondly, the cost of the load cells was another consideration since the team had a limited budget to work with. A less expensive option could be considered because the design of the load cell system allows for the possibility to replace the original load cell when Triton Racing's budget permits to improve the precision (and/or) accuracy measurement capabilities. While comparing the options, the team wanted to choose a load cell with a capacity well beyond what was required of the dyno, to prepare for any accidental issues like an object falling and overloading the cell.

Final Design Choice

Initially the DTec Load Cell (see Appendix "Individual Component Analysis") was decided upon, however, after looking into the supplier further, the team reasoned that getting the load cell from Australia would incur additional import costs as well as potential for significant delays in receiving the part. As such, a similar load cell was chosen from AnyLoad, which is based in the U.S. (Fig. 28). In addition the AnyLoad NH-500 load cell was rated for a capacity of 224 kg (500 lbs), which is well beyond what is required of the component. This load cell is also powered with 10 V, which is a standard supply voltage that can be supplied through either the DAQ or directly through a power supply. Another justification for using this load cell was the material of the load cell itself--it is made of nickel-plated steel which has excellent corrosion resistance. The nickel-plated steel load cell would be able to prevent any rust or water from damaging the component itself.

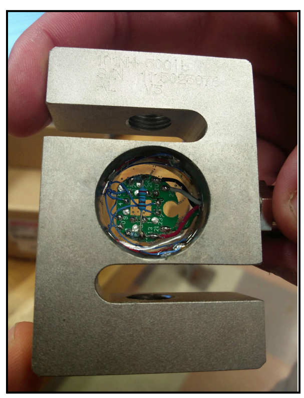

Figure 28: AnyLoad 101NH-500lb Load Cell Internals

Rpm Sensor

Introduction

In professional dynamometers, both the engine and dyno rpm is measured using rpm sensors. The dyno rpm is measured in order to first calculate the engine rpm, and then the engine horsepower. This is done by knowing the transmission ratio from the engine to the dyno which is often provided in the engine specifications. The rpm of the engine is additionally measured find the transmission ratio (if it is unknown) or to validate the known ratio. With the tight budget, the 156B team determined that the sole dyno rpm sensor would be enough, especially since the transmission ratio was known. In the future, Triton Racing would easily be able to implement a rpm sensor for the engine to eliminate some error with using provided transmission ratio.

Functional Requirements

- With a transmission gear ratio of 1.995 and a known max engine rpm of 15,000 the sensor would need to measure to 7500 rpm
    - With four teeth designed on the flywheel:
        - 7500 * (4 teeth) / 60 min = 500 Hz
- Compact design to fit tight spacing along the driveline
- Easily mountable, usable, and adjustable

Comparison of Designs Considered

There are three categories of rpm sensors: rotary (shaft encoders), optical (photoelectric), and proximity (magnetic)[10]

Rotary types sensors work great for high resolution systems since the pulse output is guaranteed to be symmetric[10] . In addition, they are incorporated into the driveline itself, reducing any external space requirements. However, they are not as easy to incorporate into a pre-designed system and are more expensive because of their higher resolution.

Optical type sensors are primarily used in low-resolution settings because they cannot measure as many pulses per rotation[10] . Photoelectric sensors work by sensing a reflection off a target placed on the rotating shaft. Photoelectric sensors are more moderately priced than encoders, but their drop in accuracy at higher rpms is a significant concern. In addition, since most of the driveline is metallic, reflection noise is a possibility since the sensor will likely read more than just the reflective target.

Proximity sensors are a common application in dynamometers. Their resolution is low to medium depending on the number of pulses. A pulse is generated by the magnetic field induced by the magnet located in the sensor as well as the magnetic trigger. Using these sensors, therefore requires a trigger wheel, which is commonly designed as teeth on a gear. The gear must be magnetic and its teeth must be designed to meet the specific requirements of the proximity sensor in order for a signal to be generated. These sensors incorporate a simple design and are relatively easy to mount. In addition, the cost of proximity sensors is low, which makes it ideal for the project’s application.

Final Design Choice

Out of the three sensor types, it was determined that the proximity sensor would be the best choice. Most of these types are sensors are rated for 10 kHz or more and are relatively inexpensive. In addition, basic mounting is possible and easy as long as proper spacing to the trigger wheel is accounted for. The only additional work created by using this sensor is designing and manufacturing a trigger wheel, which is not a difficult task. Using an encoder would have been ideal in the case that higher resolution was preferred, but since the max rpm is only 7500, this was not needed. Besides, incorporating the rotary encoder into the existing driveline would require a redesign of the entire dyno driveline.

Once narrowed down to just proximity sensors, the next deterministic quality was the ease of use. The team wanted to minimize the circuitry required to use the sensor since it would be more wires and components to keep track off. A good option that incorporated an analog-to-digital converter (ADC) was ultimately chosen. Figure 29 shows the Littelfuse Hall-Effect Sensor - 55505 chosen.

Figure 29: Littelfuse 55505 Hall-Effect Sensor[14]

Temperature Sensor

Introduction

For safety reasons, four temperature sensors were included into the system. Three would measure the water temperature at critical locations: inlet of the water brake, outlet of the water brake, and reservoir. Because the internal water temperature of the brake system should not exceed 50°C, active monitoring of the temperature sensors through the DAQ will notify the operator if the maximum temperature is exceeded, prompting the operator to safely turn off the dyno. The fourth sensor measures the coolant for the engine side at the location right before the radiator.

Functional Requirements

- ➢ Temperature rating of at least 60 °C
- ➢ Response time of less than 10 seconds
- ➢ Waterproof
- ➢ Usable with myRIO DAQ

Comparison of Designs Considered

Three common temperature sensors were considered for use in the system: thermocouples, resistance temperature detectors (RTDs), and thermistors.

Thermocouples are the most common in lab settings because of the ease of use. In addition, they are more accurate and have a very fast response time. However, thermocouples are more expensive in the long run because it requires an amplifier board. In terms of design, thermocouples also require design consideration of the cold junction. Placement of the cold junction is critical to performance of the thermocouple. As such, the dyno system design was determined not to be ideal for a thermocouple because a proper cold junction location could not be guaranteed.

RTDs as temperature sensors are currently the best option for quality and for a narrower temperature range than thermocouples. RTDs are also more reliable over a longer time period and therefore readings can be accurate and precise. Like thermocouples, RTDs require connection in a bridge configuration, as well as an amplifier. Because of their varied use in applications, waterproof RTDs are now common.

Thermistors are the least expensive temperature sensors and also the easiest to set up since calibration tables are provided by manufacturers. The resistor in a thermistor is much more temperature sensitive than standard resistors. In addition, thermistors are more accurate in a smaller temperature range, making it ideal for the dyno since the temperature range is about 20 °C (room temperature water) to 50°C (worst case temperature of water in the water brake). In addition, waterproof thermistors are easily found.

Final Design Choice

The team chose to go ahead with a thermistor because of the low-cost, applicability with the temperature range needed, and because one option that was found had already been used with a myRIO and LabVIEW. This Sparkfun waterproof temperature sensor (Fig. 30) incorporates a standard thermistor - DS18B20 - encased in a protective plastic covering shielding it from water.

The added benefit of using this sensor was that it was a 1-wire interface that could directly output temperature readings by communicating over a data line to the microprocessor. Communication with the four temperature sensors through the myRIO required a custom designed FPGA VI (see Appendix M).

Figure 30: Sparkfun Temperature Sensor - Waterproof (DS18B20)

Chapter 4: Prototype Performance

Theoretical Predictions

Thermodynamic Simulation

To predict cooling system performance, a thermodynamic simulation of the system design was done. This simulation also helped explore the design space and was helpful for choosing a radiator. To find power requirements, previous dynamometer data was studied. As shown below in Figure 31, maximum engine power occurs at 9000 rpm and results in 63 kW (85 hp). To give a factor of safety of 1.4, the system is designed to be rated for 90 kW (120 hp).

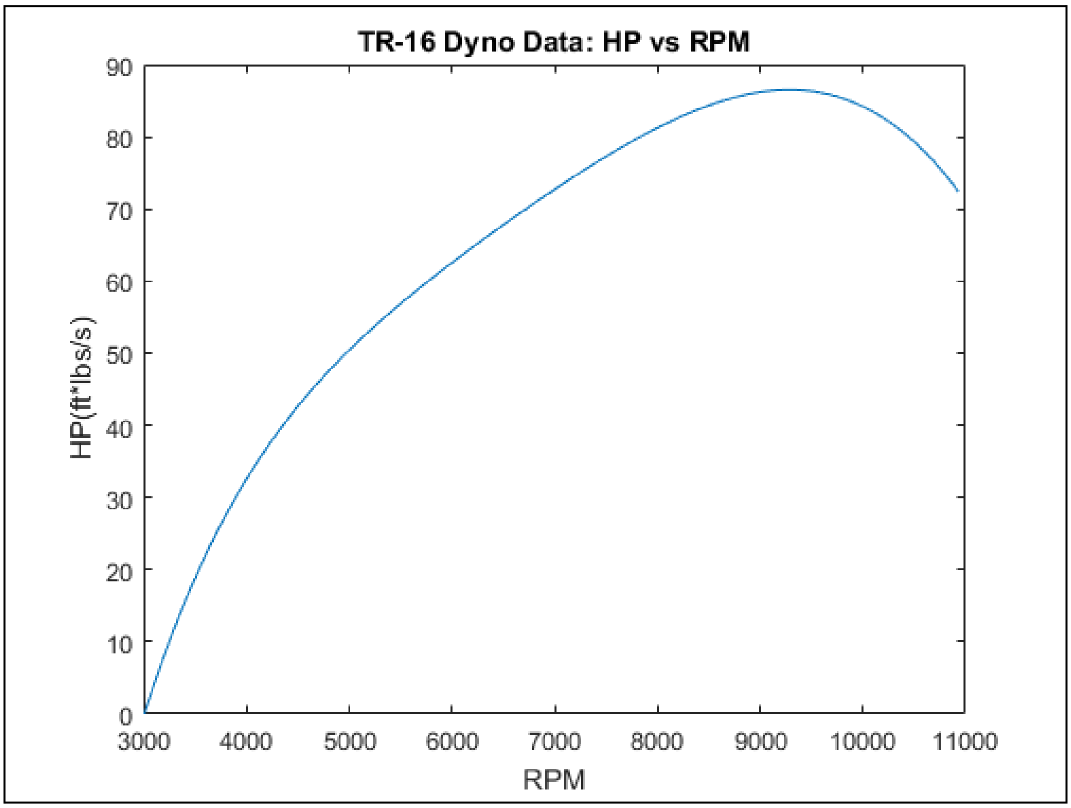

Figure 31: Triton Racing Team Engine Dyno Data

Assumptions used in the thermodynamic simulation include:

- Constant flow rates
  - constant inlet/outlet water flow rate(assume no effect from the water brake vent)
  - constant input kW
  - flow rate of 0.00063 m^3/s (10 GPM)
- Worst case ambient external temp
  - 30^{o} C
- Reservoir capacity
  - 37.85 liters

    - Instant reservoir mixing at every timestep
- Radiator efficiency
    - Effectiveness - Number of Transfer Units (NTU) method
    - pulled results data from study done on automotive radiators for accurate numbers
- Water brake efficiency
    - assume 100% energy transfer
- Pump energy
    - add as heat input (<1% of total)

To find an appropriate radiator size, a thermodynamic simulation of the system design was done to explore the design space. To find power requirements, previous dynamometer data was studied. As shown below in Figure 32, maximum engine power occurs at 9000 rpm and results in 63 kW (85 hp). To give a factor of safety of 1.4, the system is designed to be rated for 90 kW (120 hp).

The thermodynamic analysis solved the system in Figure # for every timestep and iterated the results until a steady state operating condition was reached.

Figure 32: Aftermarket Radiator Used in Analysis[4]

Table 1: Basic Radiator Dimensions

$L_{radiator}$	$H_{radiator}$	$W_{radiator}$	W_{tube}	H_{tube}	L_{fin}	W_{fin}	H_{fin}	N_{tube}	N_{fin}
inches	inches	inches	inches	inches	inches	inches	inches		
26.125	18	2.375	1	0.083	0.156	2.3	0.003	86	780

Table 2: Properties of Water at Average Temperature

ρ_{water}	$c_{p,water}$	Pr_{water}	k_{water}	μ_{water}
kg/m^3	J/kgK	unitless	W/mK	kg/sm
982.20	4185	5.26	0.59	7.61E-4

Table 3: Properties of Air at Average Temperature

ρ_{air}	$c_{p,air}$	Pr_{air}	k_{air}	ν_{air}
kg/m^3	J/kgK	unitless	W/mK	m^2/s
1.16	1007	.74	2.50E-2	2.20E-5

Figure 33: Aftermarket Radiator Specifications in Analysis[4]

Water tables[7] were used for enthalpy calculation. For the radiator component, data from an independent lab test[4] was used for an overall effectiveness-NTU parameter to simulate heat rejection performance. Energy balance was used to simulate the effect of the reservoir.

The energy balance equation for the water brake is shown below and simplifies down to heat energy being added to the system. This energy balance equation is to solve components in a thermodynamic system.[7]

$$\Delta E_k + \Delta E_P + \Delta U = \sum_{\text{input streams}} m_j\left(\hat{H}_j + \frac{u_j^2}{2} + gz_j\right) - \sum_{\text{output streams}} m_j\left(\hat{H}_j + \frac{u_j^2}{2} + gz_j\right) + Q - W_s \tag{4.1.1.1}$$

The radiator efficiency equation allows the calculation of the radiators heat rejection capability for solving the temperature at the inlet and outlet of the radiator. This equation is used to get an aggregate heat transfer efficiency for a compact cross flow radiator based on temperature.[6]

$$\varepsilon = 1 - \exp^{\left[\left(\frac{1}{C_r}\mathrm{NTU}^{0.22}\right)\left(\exp^{(-C_r \mathrm{NTU}^{0.78})} - 1\right)\right]} \tag{4.1.1.2}$$

The equation for the reservoir step is shown below through energy balance, and is solved for the outlet temperature.

$$T_{outlet} = \frac{m'_{res} T_{res_{old}} + m'_{in/out} T_{inlet}}{m'_{res} + m'_{in/out}} \tag{4.1.1.3}$$

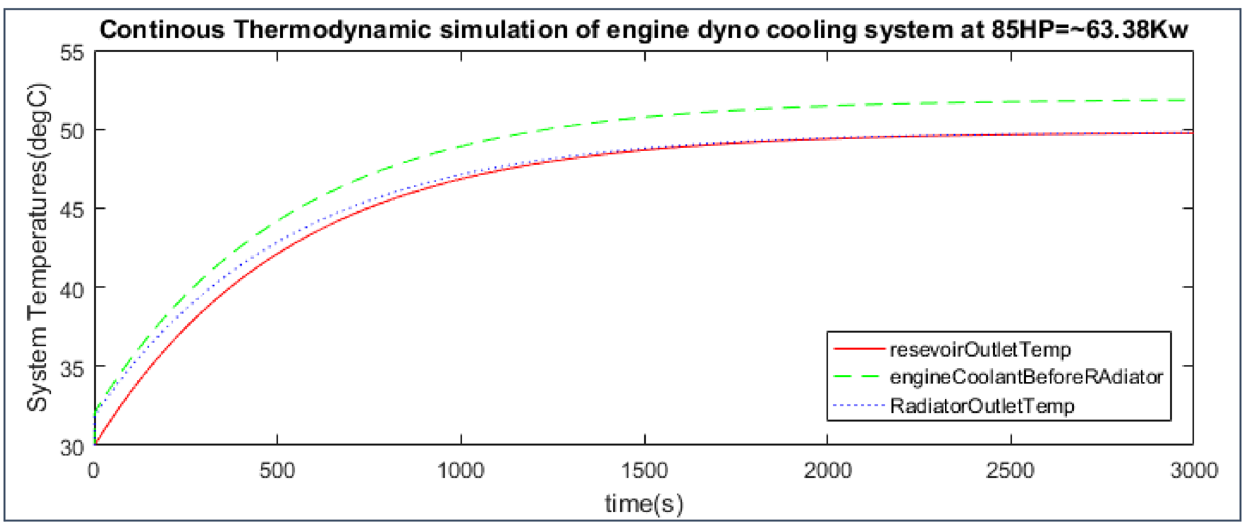

Figure 34: Thermodynamic Simulation Results of Cooling System Temp vs. Time

As shown in Figure 34, system temperature reaches an equilibrium about 50°C after over 40 minutes of a constant engine output power of 63.38 kW (85hp). This worst case analysis matches the design goal of less than 55°C.

Thus, the system is predicted to allow for continuous testing of the sponsor's current engine.

Water Flow Rate

In order to choose a pump with enough flow rate at a certain head, it was necessary to calculate the total head loss through the system. Head loss occurs when fluid flows inside a pipeline and the friction between the stationary pipe wall and the fluid causes the hydraulic energy of the fluid to be converted to thermal energy which is lost. This loss of energy in the fluid is called head loss.

The system in this prediction consisted of a reservoir, radiator, and pump. The head loss calculation was done before all fittings and plumbing were decided upon, therefore some assumptions were made for a theoretical prediction of the total head loss.

The following assumptions were made to calculate the total head loss:

- ➔ 25 feet of flexible tubing
- ➔ Max flow rate of 9.946×10^{-4} m^3/s (15 GPM according to the Stuska manual)
- ➔ Fittings include a non-return valve and a bellmouth outlet
- ➔ Inner diameter of pipe is .0195 m
- ➔ The static head is .6096 m (the height from the reservoir to the pump outlet)

The total head loss through the system is

$$H_{total} = H_{static} + H_{dynamic} \qquad (4.1.2.1)$$

The static head is the energy loss of the fluid due the height of the fluid column, as mentioned in the assumptions to be .6096 m.

The *Darcy Weisbach Equation* is used to calculate the dynamic head through a system with incompressible fluid due to friction

$$H_D = k\frac{v^2}{2g} \qquad (4.1.2.2)$$

Where υ is the velocity in the pipe (m/s), which was determined from the volumetric flow rate equation

$$Q = vA \qquad (4.1.2.3)$$

Therefore,

$$v = \frac{Q}{A} \qquad (4.1.2.4)$$

From the assumption that the max flow rate, *Q* needed is 9.946×10^{-4} m^3/s and *A* is the cross sectional area, $A=\pi r^2$ with a 0.0195m diameter pipe is $0.0011945m^2$. The velocity was calculated to be .79225 m/s.

Where k is the Darcy loss coefficient

$$k = k_{fittings} + k_{pipe} \qquad (4.1.2.5)$$

$$k_{pipe} = \frac{fL}{D} \tag{4.1.2.6}$$

$k_{fittings}$ are the loss coefficients due to fittings in the piping system. These include a non-return valve k=1, pipe entrance k=.05, and bell-mouth outlet k=0.2 for a total of 1.25.

k_{pipe} is the loss coefficient due to pipe friction where f is the friction factor, L is the length of the pipe, and D is the inner diameter of the pipe. The friction factor *f* was determined through finding the Reynolds number

$$R_e = \frac{vD}{\nu} \tag{4.1.2.7}$$

where υ is the velocity in the pipe (m/s), D is the inner diameter and ν is the kinematic viscosity for water which is $1\,x\,10^{-6}$(m^2/s). R_e was determined to be 15,092 which means that the flow is turbulent following the Reynolds number condition that $R_e > 4{,}000$ is turbulent flow.

For turbulent flow, the friction factor *f,* follows the Blasius equation

$$f = .316R_e^{-1/4} \tag{4.1.2.8}$$

Therefore, *f* was determined to be .0285.

Finally, k_{pipe} was calculated to be 11.4. And the total loss coefficient *k*, is 12.65.

Using these values the dynamic head H_D was calculated to be 0.4046m. Therefore, the total head, (Eqn. 4.1.2.1) is 1.014 m. This value was used to determine the pumping head needed to account for this head loss.

Load Cell Predictions

In preparation for testing the dyno system, the load cell was calibrated after water brake assembly onto the frame was complete. Refer to Appendix H "User Manual" for details on the calibration procedure of the load cell. Details on connecting the load cell to the myRIO can be found in the "Test Conditions" section below (Fig. 37). Due to time restrictions, the team was unable to use proper weights for this calibration procedure, and instead had to use the varying weights of the team members as replacements. The test was conducted twice to determine the repeatability of the load cell. However, due to the fact that team members had to be used in place of dead weights, perfect balance on the torque arm could not be achieved which likely resulted in different voltage readings.

Table 3: Data From Load Cell Calibration

Mass Used	True Load (kg)	Load at Load Cell (kg)	Test 1 Voltage (V)	Test 2 Voltage (V)	Difference in Voltage
Steel Cylinder	8.53	17.06	0.402	0.416	0.014
Person 1	61.69	123.38	2.601	2.588	0.013
Person 2	85.18	170.36	3.552	3.533	0.019
Person 3	92.17	184.34	3.78	3.75	0.03

From this data, a linear fit was imposed onto the two tests to determine the equation of best fit. The results in Figure 35 show that the two equations are fairly similar to each other, giving a good sense of repeatability in the measurement data.

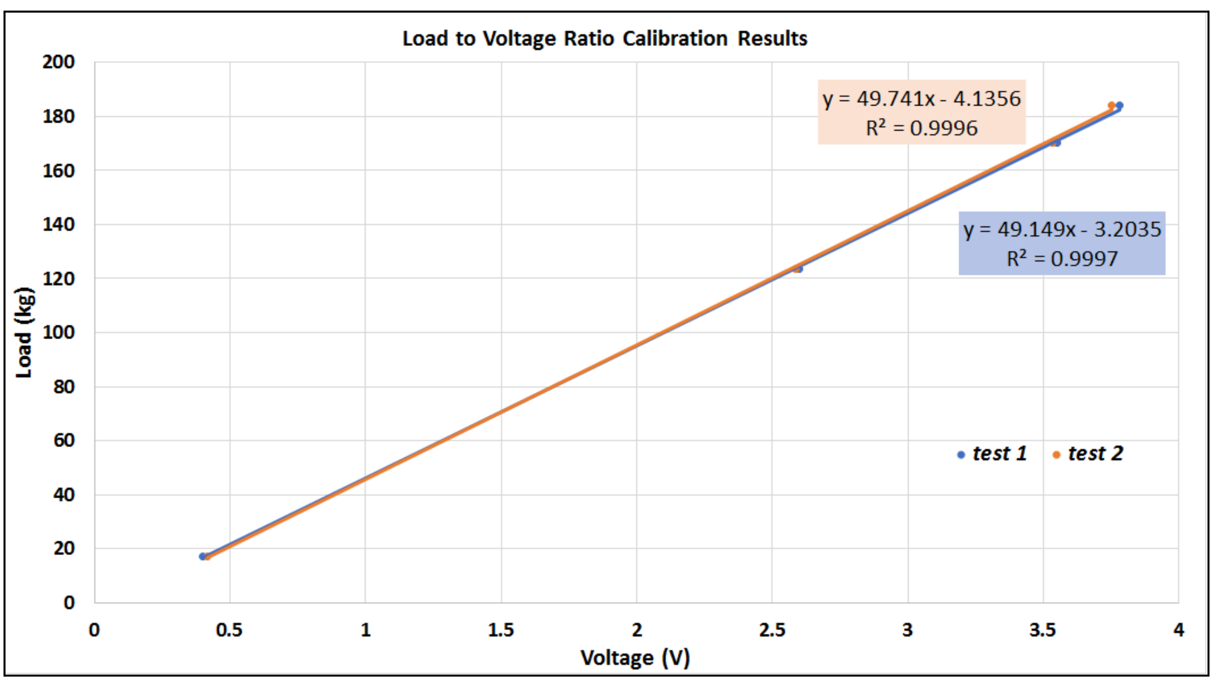

Figure 35: Plot of Linear Fits to Experimental Data From Calibration

In theory, by taking the data from Test 1 and using the determined linear equation, output voltages from the load cell can be determined by knowing the torque applied by the water brake to the torque arm. At max predicted power of the Yamaha FZ6R (using previous dyno data) the max torque exerted by the water brake onto the torque arm would be 230 Nm (see Chapter 3.6 “Load Cell”). This torque is transmitted at a length of 0.1524 m (6 in.) to the load cell which would feel 153 kg. Using the equation from above, it is assumed that the voltage reading of the load cell in the myRIO 3.178 V. Table 4 below summarizes the theoretical voltage values for a variety of loads applied that can be checked upon testing.

Table 4: Theoretical Load Cell Voltage Outputs For Given Loads

Load Cell Load (kg)	Theoretical Voltage Reading (V)
25	0.574
50	1.082
75	1.591
100	2.100
125	2.608
153	3.178

Test Conditions

Preliminary Testing

To ensure proper functionality of the dynamometer system, a series of preliminary tests were conducted, examining the durability and functionality of several of the key components.

Testing of the pump was done by performing a proof-of-concept of the reservoir and pump assembly by running water through the pump from a full 26.5 L reservoir to a drain, and measuring the time it took to fully drain the reservoir. Further testing of the entire water system was later conducted by connecting the reservoir, filter, pump, radiator, and water brake with flexible PVC tubing to ensure that no leaks occurred, and then measuring the time to drain the system.

The data acquisition system was initially tested as a proof-of-concept connecting substitute devices to a myRIO DAQ to hold the place of a sensor for the purpose of programming the data acquisition (Fig. 36). Potentiometers were used in place of thermistors to measure water temperature, as well as the load cell to measure water brake torque. A square wave signal generator was used in place of a hall effect sensor to measure rpm of the motor. After assembly of the dynamometer system, the corresponding sensors were connected to the DAQ and sample data was shown in the VI interface, confirming that the sensors were connected properly.

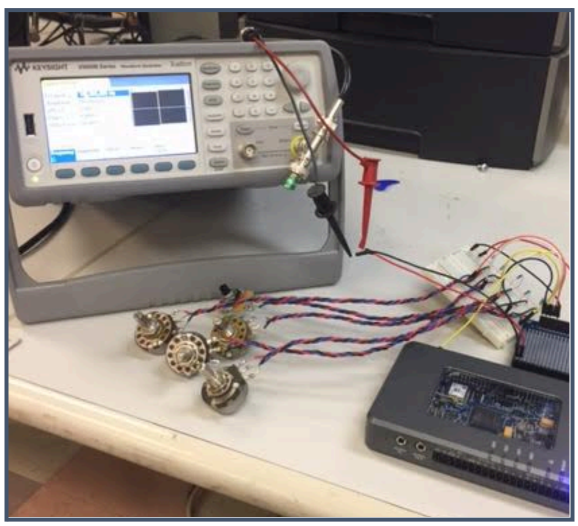

Figure 36: Proof of Concept Setup for myRIO DAQ

The next step of testing the DAQ required implementing the actual sensors that are used in the dyno. The four DS18B20 thermistors were connected to digital I/O pins on Connector C of the myRIO. The 55505 Hall-effect sensor was connected the digital I/O pin on Connector A. Finally, the AnyLoad load cell was connected through an instrumentation amplifier and an operational amplifier into an analog input pin on Connector A (Fig. 37).

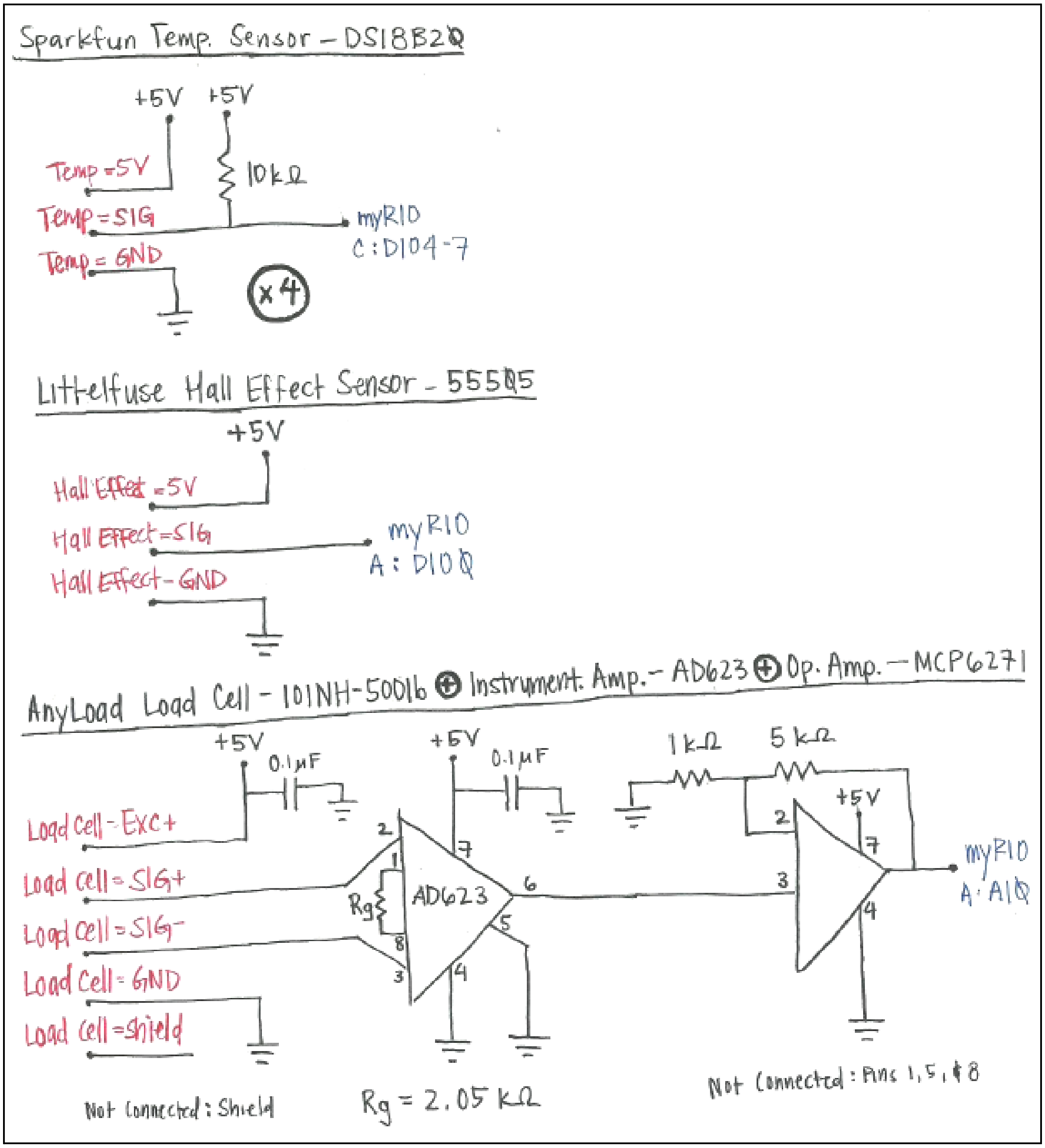

Figure 37: Sensor Connection Guide to myRIO

Communication with the digital and analog pins on the myRIO required programming a VI specifically for the FPGA Target. This was primarily done for the 1-wire temperature sensors, but had to incorporate the remaining sensor pins on Connector A as well in order for the custom FPGA VI to read all the pins connected to sensors. If Triton Racing were to add more sensors in the future, they would need to first program it into the FPGA VI, and incorporate it into the Real-time VI which is where the operator will see the necessary parameters displayed on the Front Panel. See Appendix J for details on the VIs incorporated. Thus the validation of the data transfer from all the sensors to the visual interface of the VI was observed.

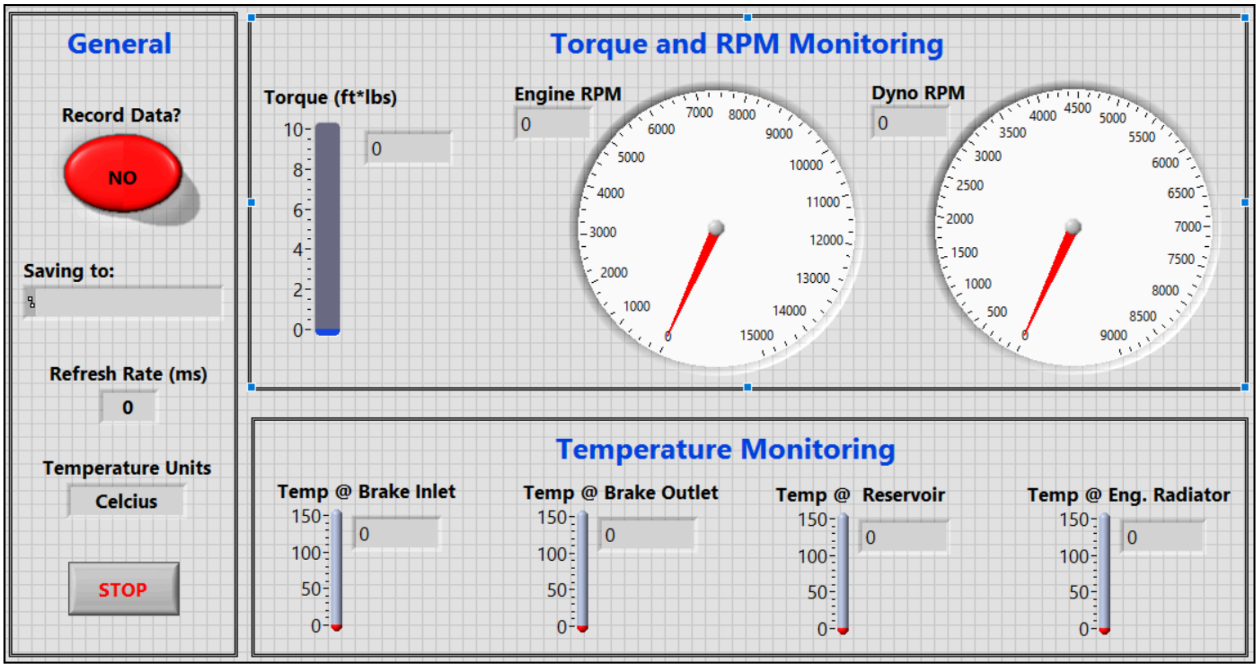

Figure 38: LabVIEW Interface for Dyno Operator

Final Test: Mock Setup

Since manufacturing of the dyno driveline was not completed by the end of the quarter, the 156B team chose to set up a preliminary test that proved the functionality of the DAQ, sensors, and the water system. This test eliminated the requirement for the driveline to be completed because there was not enough time to machine the sprocket adapter that was needed to connect the driveline to the engine. In this test, a waveform generator was used in the place of the Hall-effect sensor to simulate the changing rpm. The load cell was tested by applying a weight to the torque arm and compressing the load cell. Water was run through the system and into the water brake to test that there were no leaks and that the three thermistors in the water system (Inlet, Outlet, and Reservoir) all gave the same temperature reading. In addition, a top-level monitoring VI- “Dyno Data Monitoring”- was created so that the operator could simultaneously monitor and save data collected from a test cycle. The final visual interface for the dyno operator can be seen in Figure 38. When first running the program, the operator would be prompted by the VI to identify a file location to store data in the form of an Excel file type. Once accepted, the VI would begin to monitor the parameters while also storing the data into the Excel file. This VI utilizes a variety of network variables that are called out from the “Dyno (RT)” VI which primarily deals with data processing from analog and digital inputs to required variables like frequency, dyno rpm, and load.

Results

The mock test performed as expected and a portion of the resulting Excel file is below in Table 5. The adjustments in rpm and torque can be seen over the samples taken at one second intervals. In addition, the temperatures at the inlet, outlet, and reservoir all maintained about 21°C. The engine radiator temperature sensor was not connected which resulted in the irregular (-0.0625) value. With this test, the overall functionality of the DAQ and sensors was proven to ensure that the Triton Racing team would have an accurate DAQ system once the FZ6R Yamaha engine is connected to the dyno and tested. Also, the simple data file outputted from LabVIEW would be easy for Triton Racing to work with in MATLAB to create the necessary power curves using the recorded data.

Table 5: Dyno Mock Test Sample of Data File

Time	Dyno Rpm	Engine Rpm	Torque	Inlet Temp	Outlet Temp	Reservoir Temp	Engine Radiator Temp
6/15/2017 01:06:12.516	299.9871	586.4748	3.90096	21.75	21.3125	21.5	-0.0625
6/15/2017 01:06:17.200	299.9402	586.3832	4.00932	21.75	21.3125	21.5	-0.0625
6/15/2017 01:06:18.200	300.0067	586.5131	3.90096	21.75	21.3125	21.5	-0.0625
6/15/2017 01:06:19.201	300.0007	586.5014	4.00932	21.75	21.3125	21.5	-0.0625
6/15/2017 01:06:20.201	300.0302	586.5591	4.00932	21.75	21.3125	21.5	-0.0625
6/15/2017 01:06:21.201	299.9723	586.4459	4.00932	21.75	21.3125	21.5	-0.0625
6/15/2017 01:06:22.202	299.9472	586.3968	4.00932	21.75	21.3125	21.5	-0.0625
6/15/2017 01:06:23.203	300.033	586.5645	3.90096	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:24.203	299.959	586.4199	3.90096	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:25.204	300.0253	586.5495	3.7926	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:26.204	299.9639	586.4295	3.95514	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:27.204	299.9947	586.4896	3.84678	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:28.205	300.0125	586.5245	3.90096	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:29.206	299.9803	586.4615	19.99242	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:30.206	299.9711	586.4435	32.34546	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:31.207	299.9787	586.4584	35.00028	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:32.207	300.002	586.504	36.3006	21.75	21.3125	21.5	-0.0625
6/15/2017 01:06:33.207	300.0615	586.6203	35.70462	21.75	21.3125	21.5	-0.0625
6/15/2017 01:06:34.208	299.9986	586.4973	34.45848	21.75	21.3125	21.5	-0.0625
6/15/2017 01:06:35.208	396.0579	774.2933	33.97086	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:36.209	851.1776	1664.052	4.00932	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:37.209	1066.723	2085.443	27.74016	21.75	21.375	21.5	-0.0625
6/15/2017 01:06:38.209	1197.326	2340.771	40.25574	21.75	21.3125	21.5	-0.0625

Chapter 5: Design Recommendations and Conclusions

Design Recommendations for the Future

The team foremost recommends Triton Racing to incorporate electronic control of the inlet valve of the water brake and throttle for the engine. This will greatly reduce the skill level necessary to operate the dynamometer successfully and will also provide better and more repeatable testing. Electronic control also allows the system to measure engine performance under simulated racing conditions. This can be achieved by recording throttle and fuel data while racing at a racetrack and then extrapolating the water brake inlet valve positions needed to achieve those conditions.

Another recommendation from the team is to expand the sensor portfolio allowing the operator to monitor many more engine performance parameters than are currently measured in this design. Additionally, this expansion advances Triton Racing's ability to tune and modify their engines to lead to better performance overall at competition. Possible sensors include engine rpm (inductive clamp on spark plugs), O^2 measurement in the exhaust pipe for better air to fuel ratio tuning, temperature sensors at various places throughout the engine cooling system, multiple intake pressure sensors fuel sensor, airflow sensor, throttle position sensor (from throttle control lever) and others.

Comparison of data taken from a dyno run on a professional $50k system is recommended for validating the accuracy and quality of this system's measurements. Also, compensation factors for air temperature, pressure, humidity and other variables affecting engine performance can be found in SAE engine dyno testing papers or in the Stuska Dynamometer operator's manual. They are necessary to achieve good results when dynoing an engine.

Safety Considerations

A primary concern of the project was and will continue to be safety. The drive shaft of the dyno has a significant inertia and will be rotating at 6,000 rpm which results in a significant amount of rotational kinetic energy. Any failure point along the driveshaft may result in damage to the system. The operator, when standing behind the control panel, is protected from projectiles by plates of perforated sheet metal. However, the operator is not protected from hot water or electricity discharges. In the event of drive shaft failure, whipping is prevented by a 6.35 mm (1/4 inch) thick steel collar according the National Hot Rod Association (NHRA) safety requirements for drive shafts. More information about this standard can be found below in the "Applicable Standards" section. In addition, steel mesh surrounds the frame, which encompasses the drivetrain and dyno to provide another level of safety. Certain safety precautions will be taken to ensure that the operator and anyone else stays out of the plane of rotation of the driveline system; the operator (and anyone else) shall stand behind the dyno out of the rotation plane of the driveline to prevent any exposure to impact, should the system fail.

In addition, the dyno's water circuit will operate at a maximum temperature of 42°C as concluded by the thermodynamic analysis. Energy.gov[2] states that the recommended temperature for hot water in a home is 48°C , since 60°C can cause scalding. Any direct contact with hot water in the dyno due to a failure in the water circulation system will cause a minor (safe), temperature shock if skin-contact is made. The safety precaution to mitigate this will be to ensure that the operator's arms and legs are fully covered by clothing.

More safety guidelines can be found in Appendix H (User Manual).

Applicable Standards

There are many standards to adhere to when testing combustion engines due to their high energy operating characteristics. The following are standards which apply to this project.

- SAE J1349 Revision JUN90. This standard provides conversion factors to compensate for varying air temperature and density during dynamometer operation, allowing for dyno testing to take place in a wide range of climate conditions, reducing testing constraints.

- OSHA 1910.106. Provides guidelines for safely containing flammable liquids without risk of combustion, proper labelling for notification of hazardous substances.

- OSHA 1910.212. Provides standards for safety in regards to machine operation, general safety requirements and good practices.

- OSHA 1910.219: Mechanical power-transmission apparatus. This standard provides details on power transmission devices and is relevant to sections of this project including the bearing shaft, driveshaft, and torsional coupler.

- NHRA general regulation number 2:4. This standard provides guidelines for safely containing a high rpm driveshaft in the event of a catastrophic failure by means of a safety loop designed to prevent the driveshaft from causing harm.

- ANSI/ASME B1.20.1. This standard defines the dimensions of national pipe thread (NPT) fittings used for plumbing the water system of the dynamometer.

In addition to these standards, common bolt fasteners and heim joints were used throughout the project which are designed according to SAE standards. Further, electronics were wired according to the National Electric Code (NEC) guidelines.

Impact on Society

An engine dynamometer can provide numerous advantages to UC San Diego's Formula SAE team Testing of internal combustion engines and better education of engineering fundamentals conveyed through this project will help future engineers in the racing team receive a better engineering education. From a different perspective, however, this project sustains the growing negative environmental impact of internal combustion engines in a period of time where the world needs to move to alternative energy sources.

Environmental

An important environmental impact is fuel consumption. It is projected that running the engine dyno for a couple of hours will consume on average 19 liters (5 gallons) of fuel. This of course contributes to pollution and overall air quality. Noise pollution is another consideration, although it is likely that this project will be used in a more remote location, such as a parking lot. The noise level due to operating the engine will be in the range of 60 to 70 dB, which will require ear protection for the operator or anyone near the dyno.

Fabrication of this dyno requires steel and aluminum as well as some more minerals for the electronic equipment. This project constitutes only a single dynamometer build and therefore not many resources compared to the entire economy. The average lifecycle for a well-built dyno is five to twenty years, assuming there are no accidents. Therefore, the impact of requiring replacement parts and components will not take a large toll on the environment's natural resources.

Also, the dyno has a water recirculation system, instead of a total loss system that will help to reduce the amount of water use. This was done for environmental considerations to help water conservation.

Global

Taking the environmental impacts and expanding them to a larger scope to cause a global impact is only likely if dynos are more easily fabricated. If this project proves successful, other FSAE teams will potentially chose to build their own engine dynos. However, the likelihood is still small because the resources required for the project as well as the maintenance does not entice many teams. Only a few have attempted to create a dyno of their own. It is important to consider that in theory, more dynos will cause more environmental impacts due to the potential for unregulated greenhouse gas emissions and wasted water.

Economic

The economic impact will only affect FSAE teams, and nevertheless it is a small impact for most teams. While the price of building a dyno would be about $2,300 (not including custom fabrication of components), on average, teams have budgets of $30,000 or more. An engine dyno project requiring 10% of the budget would likely cause a double-take, but not a disastrous problem. Teams can properly budget out a dyno across a couple years if they are worried about the price tag.

Current Infrastructure

The one physical effect on infrastructure caused by this project would be the required 1.67 m^2 (18 ft^2) of floor space to store the engine dyno. The mobility of the designed dyno allows the team to use the dyno in a variety of places, including ones without a space constraint (like a parking lot). The only requirement for the project is that the university be able to support the safe storage of the dyno with adequate project space for the Triton Racing team.

Another possibility for affecting current infrastructure would be the ability for engineers to get experience collecting experimental data and practice optimization of systems by using this dyno for testing their engines. This would be a huge benefit for Triton Racing which does not have the budget to pay for a dyno run at a shop, or the free time to organize such an event. Overall, using this dyno will allow Triton Racing to increase their performance at competition in both the design presentations and actual racing events.

Lessons Learned

The first lesson the team learned was that the design process takes a lot of time. For example, the water brake, even with the Stuska manual for the water brake design, it still took a significant amount of time to go through design iterations and ensure that all components were designed correctly according to the drawings. The second lesson was that using more resources to better understand the scope of the project would have been helpful so the team wouldn't underestimate the amount of work required for the project. There are plenty of reports and documentation by other universities who have also taken on similar projects. Using more resources would also help narrow down the project from the beginning, rather than towards the end.

A main lesson learned was to not underestimate the time it takes to manufacture and assemble. The team initially allocated more time for designing and less time for manufacturing because it was understood that members of the Triton Racing team would lend more of their time to the manufacturing process. This was a mistake because there were many parts to manufacture and the team did not receive as much help as was initially perceived and this added significant time to manufacturing and cut into testing time.

Another big issue is the limited budget the team was facing. Keeping to the budget required cutting corners in some places of design and manufacturing. This inadvertently led to risks including potential part failures, rushed decisions, and performance gaps. A larger budget of about $2,500 would be advised for a similar project that requires full manufacturing of the system. More money would be required to purchase a water brake from a manufacturer, as was the original plan. To accommodate this, a large budget of around $5,000 would be recommended. In addition, the team has learned that more time is needed for this project. Fourteen weeks simply was not enough when considering the full scale of design and manufacturing required, as well as the amount of testing and calibrating time needed. If a larger budget would have allowed, buying a water brake would have been helpful and would have reduced the time it took to design and manufacture and would have made it possible to allocate more than one week for testing.

Another lesson is that strictly following the Gantt chart is crucial to completing the project in time. Ordering all parts early is also essential to allow for shipping errors and long lead times that are inevitable when purchasing parts online, especially from overseas suppliers.

This project also allowed the team to learn more practical and hands-on engineering skills. Manufacture and assembly of components is essential for a Mechanical engineering student to learn in order to gain an upper hand in the competitive job market.

Overall, the lessons learned include time, project, and budget management.

Conclusions

The water brake engine dynamometer test bed will be of great use to the Triton Racing team. Having their own dyno will allow for ease of testing that will enhance the performance of the race car in future competitions. This MAE 156B project taught the team valuable lessons in project management, budgeting, time management, as well as practical engineering skills.

Acknowledgments

Foremost, the team would like to thank both MAE 156B professors: Dr. Jerry Tustaniwskyj and Dr. Jack Silberman, for their support and guidance throughout the quarter and particularly in regards to this project. In addition, the team would also like to thank Pedro Franco, Ian Richardson, Steve Roberts, Greg Specht, and Thomas Chalfant who were all part of the MAE 156B instructional team.

In terms of outside sponsors and advisors, the team would like to thank project and FSAE advisor Rob Shanahan for his continuous dedication and help to the project. A special thanks goes out to Bill Schlossnagel at Schlossnagel Racing for providing the ability to visit and learn about his own dynamometer for which the team used as a basis of design. In addition, the large generosity of Action PowderCoating and Anocote enabled the team to have a powder coated frame and anodized water brake in time for assembly. Also the team would like to acknowledge Ingo Foldvari and National Instruments for the generous donation of the myRIO DAQ unit, as well as for help and support with using it.

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Appendices

A: Project Management

A1: Positions, Tasks, and Responsibilities

- MAE 156B Team
    - Overall design, manufacturing, and assembly: Luke Bockman
    - Water brake design and manufacture: Justin Moreno
    - DAQ, sensors selection, preliminary testing, project management: Chinaar Desai
    - Website, water system design and assembly: Cassandra

- Sponsor Team:
    - Triton Racing Faculty Advisor: Rob Shanahan
    - Triton Racing Technical Lead: Daniel Morris
    - Triton Racing Powertrain Lead: Brad Anderson

A2: Risk Reduction Effort

- Water Brake vs. Torque Converter
    - During the initial design process the team researched the type of dynamometers that existed to see what type would be chosen for the FSAE team. After some deliberation, the team narrowed the choice down to a torque converter or a water brake dyno. While researching the two types, the team also looked into the feasibility of buying or manufacturing each type. From this, the team was able to reduce risk by noting the issues that could result by choosing one type over the other. In the end, the water brake was chosen for the project based on this risk reduction effort.

- Thermodynamic Analysis
    - A basic thermodynamic simulation was created in MATLAB to determine the temperature that the water system would reach when kinetic energy from the engine was converted in thermal energy in the water through the water brake (See Chapter 4; Theoretical Predictions; Thermodynamic Simulation). First, the team looked at existing dynamometer data to determine the power output of the same engine that was going to be tested (63 kW). Then by combining assumptions (ie. steady state and constant flow rate), radiator specifications, design requirements, and water tables into the energy balance equation and effectiveness-NTU method a simulation was determined at each time step. The simulation showed a maximum temperature of 50^{o}C and allowed the team to choose components for the design therefore reducing the risk of failure for the dyno system.

- Visit to Schlossnagel Racing
    - Since the team did not have any experience working with dynamometers before, the project sponsor was able to arrange a meeting with Bill Schlossnagel, at Schlossnagel

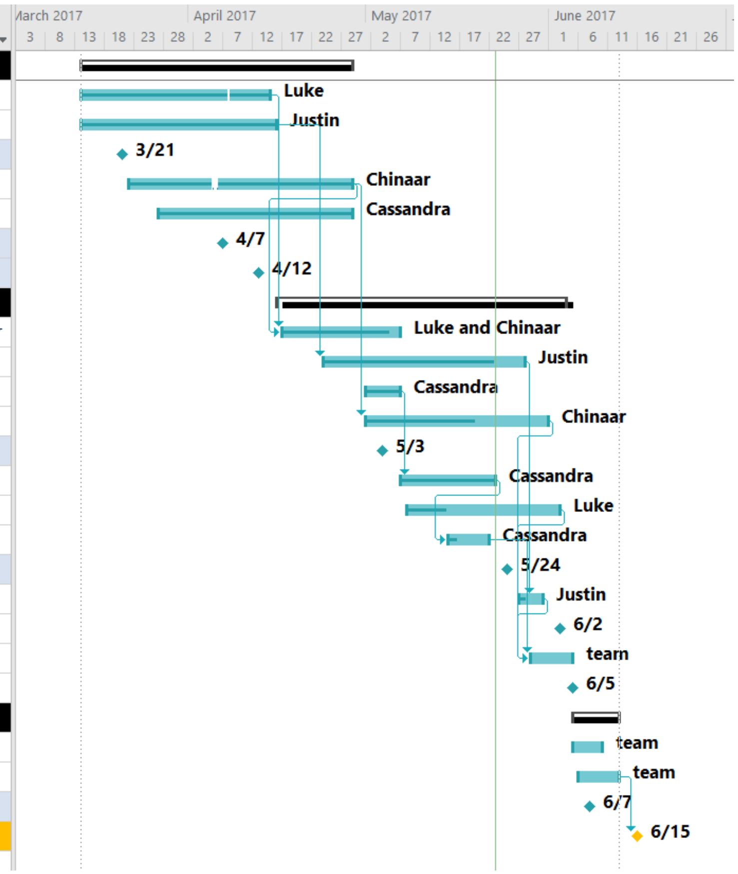
Racing, who had a custom designed water brake engine dyno. By speaking with Mr. Schlossnagel and the project sponsor, the team was able to take note of design features, safety considerations, as well as experience Schlossnagel had with operating his own dyno. The greatest benefit came from Schlossnagel's Stuska manual which was utilized for water brake design. The Stuska manual was for the water brake absorber unit that was used in Schlossnagel's dyno which eventually aided in the design of the team's own water brake unit.

A3: Intermediate Deadlines

- Final Water Brake Design: 4/15
- Finalized Design of Entire System: 4/28
- Pump Demonstration: 5/22
- Water Brake Manufacture: 5/27
- Working Sensors and VI: 5/31
- Completed Fabrication: 6/2 (late and incomplete)
- Working Machine: 6/5 (late and incomplete)
- Testing Complete: 6/12 (incomplete, but revised plan for mock test)

A4: Gantt Chart

	% Complete	Task Name	Duration	Start	Finish	Resource Names
1	**99%**	**Design**	**45.38 days**	**Tue 3/14/17**	**Fri 4/28/17**	**team**
2	100%	Frame and Overall	30.75 days	Tue 3/14/17	Fri 4/14/17	Luke
3	100%	Water Brake	32.38 days	Tue 3/14/17	Sat 4/15/17	Justin
4	*100%*	*Risk Reduction*	*0 days*	*Tue 3/21/17*	*Tue 3/21/17*	
5	100%	Torque Arm and Other Sensors	36.75 days	Wed 3/22/17	Fri 4/28/17	Chinaar
6	100%	Water System	33 days	Sun 3/26/17	Fri 4/28/17	Cassandra
7	*100%*	*Individual Comp. Analysis*	*0 days*	*Fri 4/7/17*	*Fri 4/7/17*	
8	*100%*	*Design Proposal*	*0 days*	*Wed 4/12/17*	*Wed 4/12/17*	
9	**64%**	**Build**	**48.38 days**	**Sun 4/16/17**	**Sat 6/3/17**	**team**
10	90%	Frame and Subframe	20 days	Sun 4/16/17	Sat 5/6/17	Luke and Chinaar
11	85%	Water Brake Manufacture	33.38 days	Mon 4/24/17	Sat 5/27/17	Justin
12	100%	Water System Purchase	5.63 days	Mon 5/1/17	Sat 5/6/17	Cassandra
13	60%	Working Sensors and VI	30.75 days	Sun 4/30/17	Wed 5/31/17	Chinaar
14	*100%*	*Proof of Concept*	*0 days*	*Wed 5/3/17*	*Wed 5/3/17*	
15	100%	Pump Demo	15.75 days	Sat 5/6/17	Mon 5/22/17	Cassandra
16	25%	Driveline Manufacture	25.38 days	Mon 5/8/17	Fri 6/2/17	Luke
17	15%	Demo of working system and validation	7 days	Mon 5/15/17	Mon 5/22/17	Cassandra
18	*100%*	*Preliminary Design Review*	*0 days*	*Wed 5/24/17*	*Wed 5/24/17*	
19	25%	Water Brake Assembly	3.38 days	Sat 5/27/17	Tue 5/30/17	Justin
20	0%	Completed Fabrication	0 days	Fri 6/2/17	Fri 6/2/17	team
21	0%	Assembly of Full System	6.38 days	Mon 5/29/17	Sun 6/4/17	team
22	0%	Working Machine	0 days	Mon 6/5/17	Mon 6/5/17	team
23	**0%**	**Testing**	**7.38 days**	**Mon 6/5/17**	**Mon 6/12/17**	**team**
24	0%	Basic testing and data collection	4.38 days	Mon 6/5/17	Fri 6/9/17	team
25	0%	Sensor calibration and real testing	6.38 days	Tue 6/6/17	Mon 6/12/17	team
26	*0%*	*Final Design Review*	*0 days*	*Wed 6/7/17*	*Wed 6/7/17*	
27	0%	Poster Presentation	0 days	Thu 6/15/17	Thu 6/15/17	team



B: Drawings/Layouts

Reference folder “FinalCAD&Drawings” attached on website to find all 42 drawings in PDF format.

C: Bill of Materials (BOM)

BOM for FSAE Engine Dynamometer				
Subassembly	Manufacturer Vendor	Part Number	Description	Qty
Test Cell Frame	Custom	N/A	Control subframe	1
	Custom	N/A	Safety subframe	1
	Custom	N/A	Driveline subframe	1
	Custom	N/A	Engine subframe	1
	Custom	N/A	Base subframe	1
	Custom	N/A	Tie rod mounts	2
	Custom	N/A	Motor tie rod	2
	Custom	N/A	Top panel	1
	Custom	N/A	Upper side panel - perforated	2
	Custom	N/A	Lower side panel - perforated	2
	Custom	N/A	Front panel - perforated	1
	Custom	N/A	Parts tray	1
	Custom	N/A	Pressure distribution plate - 2 hole	2
	Custom	N/A	Pressure distribution plate - 6 hole	1
	Supplied by Sponsor	N/A	#10-32 aluminum bolts for mounting radiators	8
	Supplied by Sponsor	N/A	M6x1 Vibration isolator screws	4
	Supplied by Sponsor	N/A	sheet metal screws	30
	Home Depot	ZAB201EG-10	Steel L brackets	17
	McMaster-Carr	1608A44	Piano hinge	1
	McMaster-Carr	306T45	2" Square U-bolts	4
	McMaster-Carr	91185A921	1/4-20 hand screws (pack of 10)	24
	Home Depot	802254	1/2"-13 nut	25
	Mcmaster-Carr	91236A724	1/2"-13 x 3" bolt	24
	Mcmaster-Carr	91236A726	1/2"-13 x 3 1/2" bolt	4
	Marshalls Hardware	G002097	1/2"-13 x 4" bolt	4
	Home Depot	Model # 807240	1/2 washers (box of 50)	

	Grizzly Industrial	G8177	5" Swivel casters with brake	4
	Supplied by Sponsor	N/A	Washers	50+
Water Brake	Custom	N/A	Rotor	1
	Custom	N/A	Stator - driveline side	1
	Custom	N/A	Stator - outer side	1
	Custom	N/A	Trunnion - driveline side	1
	Custom	N/A	Trunnion - outer side	1
	Custom	N/A	Thrust washer	1
	Custom	N/A	Torque arm	2
	Custom	N/A	Water brake shaft	1
	McMaster-Carr	98491A136	Shaft key	1
	Custom	N/A	1/2-20 <-> 5/16-24 adapter	1
	Custom	N/A	1/2-20 <-> 1/2-13 adapter	1
	Custom	N/A	1/2-13 <-> 5/16-24 adapter	1
	Marshalls Hardware	G003086	1/2-20 threaded rod	2
	Custom	N/A	Load cell rod end mount plate	2
	Pac Seal	9281K352	Mechanical seal	2
	VXB	KIT18473	2" Pillow Block bearing	2
	Nachi	6006ZZE	6006 ball bearing	3
	Marshalls Hardware	B221031	.3125 x .75 shoulder screw	8
	Supplied by Sponsor	N/A	.3125 x 1.25 shoulder screw	9
	Supplied by Sponsor	N/A	.375 x 1 shoulder screw	1
	Supplied by Sponsor	N/A	5/6 rod end	2
	Supplied by Sponsor	N/A	1.25" external retaining ring	1
	Marshalls Hardware	B011249	.375-16 x .625 SHCS	16
	Mcmaster-Carr	1095K27	Grease fitting	2
	Mcmaster-Carr	9213K62	Load cell vibration isolator	1
	Mcmaster-Carr	160003	1/8 NPT plug fitting	2
	Home Depot	UC138LFA	3/4 in. Brass PEX Barb x 1/2 in. Male Adapt.	1
	Mcmaster-Carr	9151K19	90 Degree Elbow Adapter, 1/2 NPTF Female x Male	2
	Mcmaster-Carr	160004	1/2 NPT plug	1
	Marshalls Hardware	32-314	3/4" NPT 90 degree brass hose barb	1
Driveline	Custom	N/A	Bearing block	1
	Custom	N/A	Bearing cover	2

	Custom	N/A	Bearing block shaft	1
	Custom	N/A	Shaft key	2
	Custom	N/A	Driveshaft coupler	1
	Nachi	6006ZZE	Nachi Bearing Shielded C3 Japan 30x55x13	3
	Lovejoy	68514462016	Torsional coupler hub	1
	Lovejoy	68514462070	Torsional couple isolator (98 shore A)	1
	Mcmaster-Carr	90967A297	30mm external retaining ring	1
	Supplied by Sponsor	N/A	Universal joint driveshaft	1
Water Circuit (Dyno)	US Plastic Depot	14257	Water reservoir	1
	Custom	N/A	Sheet metal water reservoir lid	1
	TACO	Taco 0011	Water pump	1
	KO Speed	RA-AI90MT-2+RAF-12+FMK-X2	Radiator and fans	2
	Marshalls Hardware	L-19 1305	3/4" NPT PVC hose barb	8
	Marshalls Hardware	100-404NL	3/4" NPT gate valve	2
	eBay	N/A	3/4-Inch barbed PVC bulkhead fitting	3
	McMaster-Carr	5415K14	Worm-Drive Clamps for Firm Hose and Tube	20
	McMaster-Carr	44615K425	threaded pipe	1
	Supply House	LF777SI	3/4" NPT Water filter brass	1
	Lowes	Item # 130931	Apollo 1-in Tee Barb Fitting	1
	Home Depot	Internet #202270619	1 in. x 3/4 in. Brass PEX Barb Reducer Coupling	2
	Custom	N/A	1.25" to 3/4" NPT hose bung	2
	McMaster-Carr	5231K451	PVC Tubing 1/2" ID, 11/16" OD (10ft)	1
	McMaster-Carr	5231K385	PVC Tubing 3/4In ID, 1in OD (25ft)	1
	Supplied by Sponsor	N/A	2 hole pipe strap	8
	Home Depot	UC138LFA	3/4 in. Brass PEX Barb x 1/2 in. Male Adapt.	3
Water Circuit (Engine)	FlexFab	FlexFab 5526-087	7/8" silicone heater hose	1
	Custom	N/A	1.25" to 7/8" hose bung	2

Electronics & Data Acquisition	National Instruments	myRIO-1900	DAQ for system	1
	AnyLoad	101NH-500lb	Load Cell (500 lb capacity)	1
	Mouser	584-AD623ANZZ	Instrumentation amp (AD623) for load cell circuit	1
	Microchip	MCP6271	Op amp for load cell circuit	1
	N/A	0.1 microFarad	Capacitor for low pass filter	2
	N/A	2.05 kOhm	Gain resistor for instrumentation amp	1
	N/A	1 kOhm	Gain resistor for op amp	1
	N/A	5 kOhm	Gain resistor for op amp	1
	Sparkfun	SEN-11050	DS18B20 - waterproof thermistor	4
	N/A	10 kOhm	Pull-up resistor for temp sensors	4
	Mouser	653-A22E-M-01	Omron A22E-M-01 E-stop switch	1
	Digikey	55505-00-02-A-ND	Littelfuse 55505 RPM Sensor	1
	SUPERNIGHT	N/A	360W DC 12V 30A Power Supply	1
	Caltric	CZ1743BA125CZ	AGM Battery - 12V 10 AH CCA 160	1

D: List of Suppliers and Information

Amazon

Type: Online
Contact: N/A
Parts: Water Pump Fittings

AnyLoad

Type: Online
Contact: Edmund Young
edmund.young@anyload.com
Customer Service Manager
973-866-7931
Parts: Load Cell

DigiKey

Type: Online
Contact: N/A
Parts: Rpm Sensor

eBay

Type: Online
Contact: N/A
Parts: Pump, Hose, Power Supply, Radiators, Battery, Bulkhead

Grizzly

Type: Online
Contact: N/A
Parts: Casters

Home Depot

Type: In-Store
4255 Genesee Ave, San Diego, CA 92117
Contact: N/A
Parts: L-Brackets, Nuts, Washers, Barbs

Industrial Metal Supply

Type: In-Store
7550 Ronson Rd, San Diego, CA 92111
Contact: N/A
Parts: Metal Stock, Tubes, and Mesh

Marshalls

Type: In-Store
8423 Production Ave, San Diego, CA
Contact: N/A
Parts: Barbs, Valve

McMaster Carr

Type: Online
Contact: N/A
Parts: U-bolts, Hinge, PVC Tubing, Pipe Fittings, Clamps, Bolts, Grease Fittings, Seals, Isolator, Fuel Tank, Key Stock

Motion Industries

Type: Online
Contact: N/A
Parts: Torsional Coupler Hub and Spider

Mouser

Type: Online
Contact: N/A
Parts: E-Stop and Instrumentation Amp

Sparkfun

Type: Online
Contact: N/A
Parts: Waterproof Thermistors

Supply House

Type: Online
Contact: N/A
Parts: Water Filter

US Plastic Depot

Type: Online
Contact: N/A
Parts: Water Reservoir

VXB

Type: Online
Contact: N/A
Parts: Bearings and Pillow Blocks

E: Specification Sheets for Key Purchased Parts

Alphabetized

Analog Devices AD623 Instrumentation Amplifier (Through Mouser)

ANALOG DEVICES

Single and Dual-Supply, Rail-to-Rail, Low Cost Instrumentation Amplifier

Data Sheet **AD623**

FEATURES

- Easy to use
- Rail-to-rail output swing
- Input voltage range extends 150 mV below ground (single supply)
- Low power, 550 μA maximum supply current
- Gain set with one external resistor
  - Gain range: 1 to 1000
- High accuracy dc performance
  - 0.10% gain accuracy (G = 1)
  - 0.35% gain accuracy (G > 1)
- Noise: 35 nV/√Hz RTI noise at 1 kHz
- Excellent dynamic specifications
  - 800 kHz bandwidth (G = 1)
  - 20 μs settling time to 0.01% (G = 10)

APPLICATIONS

- Low power medical instrumentation
- Transducer interfaces
- Thermocouple amplifiers
- Industrial process controls
- Difference amplifiers
- Low power data acquisition

GENERAL DESCRIPTION

The AD623 is an integrated, single- or dual-supply instrumentation amplifier that delivers rail-to-rail output swing using supply voltages from 3 V to 12 V. The AD623 offers superior user flexibility by allowing single gain set resistor programming and by conforming to the 8-lead industry standard pinout configuration. With no external resistor, the AD623 is configured for unity gain (G = 1), and with an external resistor, the AD623 can be programmed for gains of up to 1000.

The superior accuracy of the AD623 is the result of increasing ac common-mode rejection ratio (CMRR) coincident with increasing gain; line noise harmonics are rejected due to constant CMRR up to 200 Hz. The AD623 has a wide input common-mode range and amplifies signals with common-mode voltages as low as 150 mV below ground. The AD623 maintains superior performance with dual and single polarity power supplies.

Table 1. Low Power Upgrades for the AD623

Part No.	Total V_S (V dc)	Typical I_Q (μA)
AD8235	5.5	30
AD8236	5.5	33
AD8237	5.5	33
AD8226	36	350
AD8227	36	325
AD8420	36	85
AD8422	36	300
AD8426	36	325 (per channel)

FUNCTIONAL BLOCK DIAGRAM

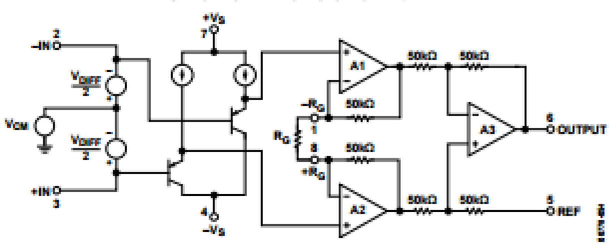


Figure 1.

Rev. E Document Feedback

One Technology Way, P.O. Box 9106, Norwood, MA 02062-9106, U.S.A.
Tel: 781.329.4700
Technical Support www.analog.com

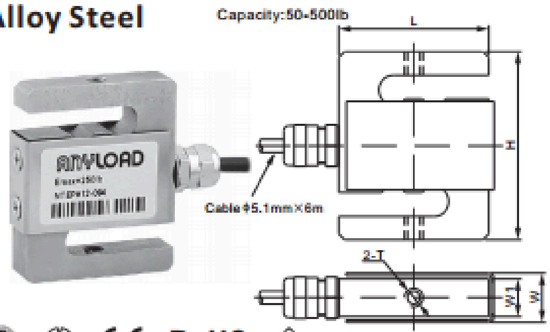
Full data sheet can be found at: http://www.mouser.com/ds/2/609/AD623-877194.pdf

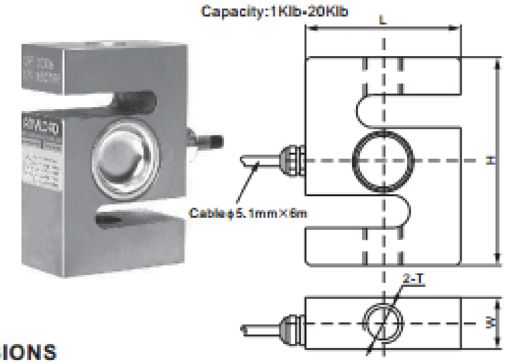
AnyLoad 101NH-500 Load Cell

101NH

ANYLOAD

Alloy Steel





C E RoHS

S-BEAMS

DIMENSIONS

RATED CAPACITY	H	L	W	W1	T
lb/inches					
50–250	2.50	2.00	0.65	0.50	1/4–28 UNF–2B
500	3.00	2.00	0.89	0.75	1/2–20 UNF–2B
1,000–3,000	3.00	2.00	1.11	-	1/2–20 UNF–2B
5,000; 10,000	4.00	3.00	1.00	-	3/4–16 UNF–2B
15,000	5.50	4.00	1.25	-	1–14 UNS–2B
20,000	7.00	4.92	2.00	-	$1\frac{1}{4}$–12 UNF–2B
kg/mm (conversion of above dimensions)					
22.68–113.4	63.5	50.8	16.6	12.7	1/4–28 UNF–2B
226.8	76.2	50.8	22.6	19.1	1/2–20 UNF–2B
453.6–1,360.8	76.2	50.8	28.2	-	1/2–20 UNF–2B
2,268; 4,535.9	101.6	76.2	25.4	-	3/4–16 UNF–2B
6,803.9	139.7	101.6	31.8	-	1–14 UNS–2B
9,071.8	177.8	125.0	50.8	-	$1\frac{1}{4}$–12 UNF–2B

SPECIFICATIONS

Full Scale Output	3.0 mV/V ± 1%	Recommended Excitation	10V (15V Maximum)
Zero Balance	±0.02 mV/V	Insulation Resistance	>2 [50V DC] GΩ
Non-linearity	< ±0.017%	Compensated Temperature Range	-10°C to 50°C / 14°F to 122°F
Repeatability	< ±0.017%	Safe Overload	150% of full scale
Hysteresis Error	< ±0.017%	Breaking Overload	300% of full scale
Creep in 30 min.	< ±0.023%	Seal Type / IP Rating	Environmentally Sealed / IP67
Input Resistance	350 Ω ± 7	Cable Color Code	Exc+ Red, Exc- Black, Sig+ Green, Sig- White, Shield Bare
Output Resistance	350 Ω ± 3		
Element Material	Alloy Steel, Nickel Plated		

Rated Capacity	Part Number	Shipping Weight (kg) Approx.	INTERCHANGEABLE PRODUCTS						
			Coti-Global		Rice Lake		Vishay Celtron	Vishay Revere	Vishay Sensortronics
50lb	101NH-50lb	0.6	CGSB-50lb		RL20000-50lb		STC-50lb	363-50lb	60001-50lb
100lb	101NH-100lb	0.6	CGSB-100lb		RL20000-100lb		STC-100lb	363-100lb	60001-100lb
200lb	101NH-200lb	0.6	CGSB-200lb		RL20000-200lb		STC-200lb	363-200lb	60001-200lb
250lb	101NH-250lb	0.6		CGSB-1-250lb	RL20000-250lb			363-250lb	
500lb (H:3.0")	101NH-500lb	0.7	CGSB-500lb	CGSB-1-500lb (H:2.5")	RL20000-500lb (H:2.5")	RL20001-500lb	STC-500lb	363-500lb (H:2.5")	60001-500lb
1Klb (H:3.0")	101NH-1Klb	1.0	CGSB-1K	CGSB-1-1K	RL20000-1Klb (H:2.5")	RL20001-1Klb	STC-1K	363-1K (H:2.5")	60001-1K
1.5Klb (H:3.0")	101NH-1.5Klb	1.0	CGSB-1.5K	CGSB-1-1.5K	RL20000-1.5Klb (H:2.5")	RL20001-1.5Klb	STC-1.5K	363-1.5K (H:2.5")	60001-1.5K
2Klb (H:3.0")	101NH-2Klb	1.0	CGSB-2K	CGSB-1-2K	RL20000-2Klb (H:2.5")	RL20001-2Klb	STC-2K	363-2K (H:2.5")	60001-2K
2.5Klb (H:3.0")	101NH-2.5Klb	1.0	CGSB-2.5K	CGSB-1-2.5K	RL20000-2.5Klb (H:2.5")	RL20001-2.5Klb	STC-2.5K	363-2.5K (H:2.5")	60001-2.5K
3Klb (H:3.0")	101NH-3Klb	1.0	CGSB-3K	CGSB-1-3K (H:4.0")	RL20000-3Klb (H:4.0")	RL20001-3Klb	STC-3K	363-3K (H:4.0")	60001-3K
5Klb (H:4.0")	101NH-5Klb	1.5	CGSB-5K (H:4.25")	CGSB-1-5K	RL20000-5Klb	RL20001-5Klb (H:4.25")	STC-5K (H:4.25")	363-5K (H:4.0")	60001-5K (H:4.25")
10Klb (H:4.0")	101NH-10Klb	1.5	CGSB-10K (H:4.75")	CGSB-1-10K	RL20000-10Klb	RL20001-10Klb (H:4.75")	STC-10K (H:4.75")	363-10K (H:4.0")	60001-10K (H:4.75")
15Klb	101NH-15Klb	3.0	CGSB-15K						60001-15K
20Klb	101NH-20Klb	6.8	CGSB-20K		RL20000-20Klb		STC-20K		60001-20K

100lb-500lb: NTEP 1:4 500 Class III, Single Cell; 1Klb-20Klb: NTEP 1:5 500 Class III, Single Cell

200kg-1.5t: OIML MAA C3, Y=17 000; 2t-10t: OIML MAA C4.5, Y=14 000

www.anyload.com

Littelfuse 55505 Hall Effect Speed Sensor (Through DigiKey)

Littelfuse Expertise Applied | Answers Delivered

Hall Effect Sensors
Flange Mount Geartooth > 55505

55505 Hall Effect Flange Mount Geartooth Speed Sensor

RoHS

Description

The 55505 is a robust flange mounting hall effect sensor primarily for geartooth speed sensing. Protected against severe and automotive environments. It is capable of switching up to 27Vdc and 20mA and available as three-wire (voltage output) version. It's case design encorporates brass mounting bush. It is ideally suited to wheel, transmission and camshaft speed sensing, geartooth sensor, rotary encoders, industrial, commercial, and automotive applications.

Features

- Ferrous target speed sensing
- Internal circuit protection
- Protection against severe and automotive environments
- Self adjusting magnetic range
- Automotive grade circuit protection
- High speed operation
- Short circuit and reverse voltage protection
- On board 10-bit A/D converter

Benefits

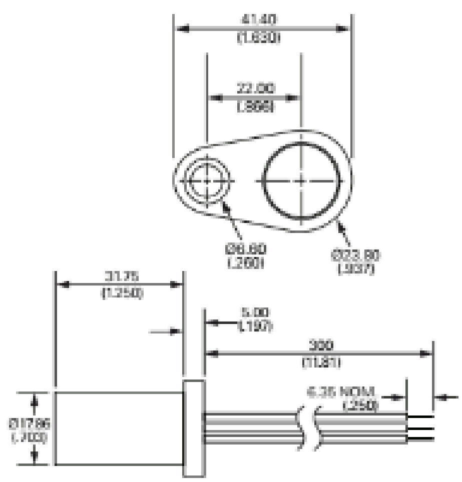
- No chopper delay
- Immune to target run out
- Rotational alignment to target not critical

Applications

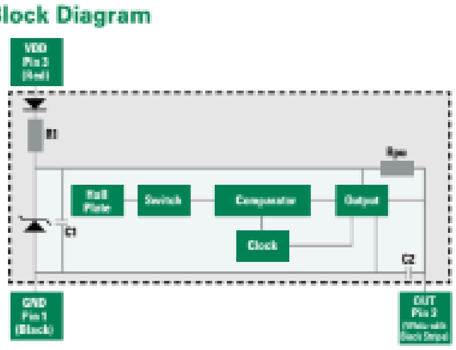
- Wheel, transmission and camshaft speed sensing
- Geartooth sensor
- Rotary encoders
- Industrial
- Commercial
- Automotive

Dimensions

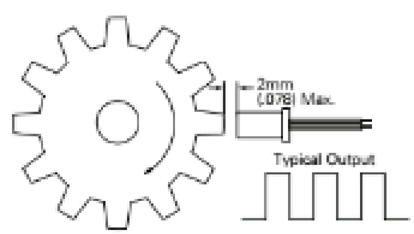
Dimensions in mm (inch)



Block Diagram



Application Example (Geartooth Sensor)



 Revised: 10/15/15
Specifications are subject to change without notice.
littelfuse.com

Lovejoy 68514462016 Jaw Coupling Hub (Through Motion Industries)

MOTION® INDUSTRIES

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Product Categories Mechanical Power Transmission > Shaft Couplings, Collars & U-Joints > Elastomeric Flexible Couplings > Jaw Shaft Couplings > Jaw Coupling Hubs > 68514462016

Lovejoy 68514462016

Jaw Coupling Hub - CJ28/38, Finished with Keyway, Curved Jaw, 2.5600 In OD, 3.54 (Aluminum) (A Style) In, 3.54 (Aluminum) (B Style) In, 3.54 (Powder Metal) In, 5.51 (Aluminum) (BX Style) In, 5.51 (Powder Metal) In Length, Aluminum, Powdered Metal

Mi Item #: 02067483

0 Reviews - Be the first to review this item.

Please Note: Photo may NOT represent actual item, please refer to title and product specifications for all details.

How can we make our Product Content better?

$40.32 each

IN STOCK

QTY Add to Cart

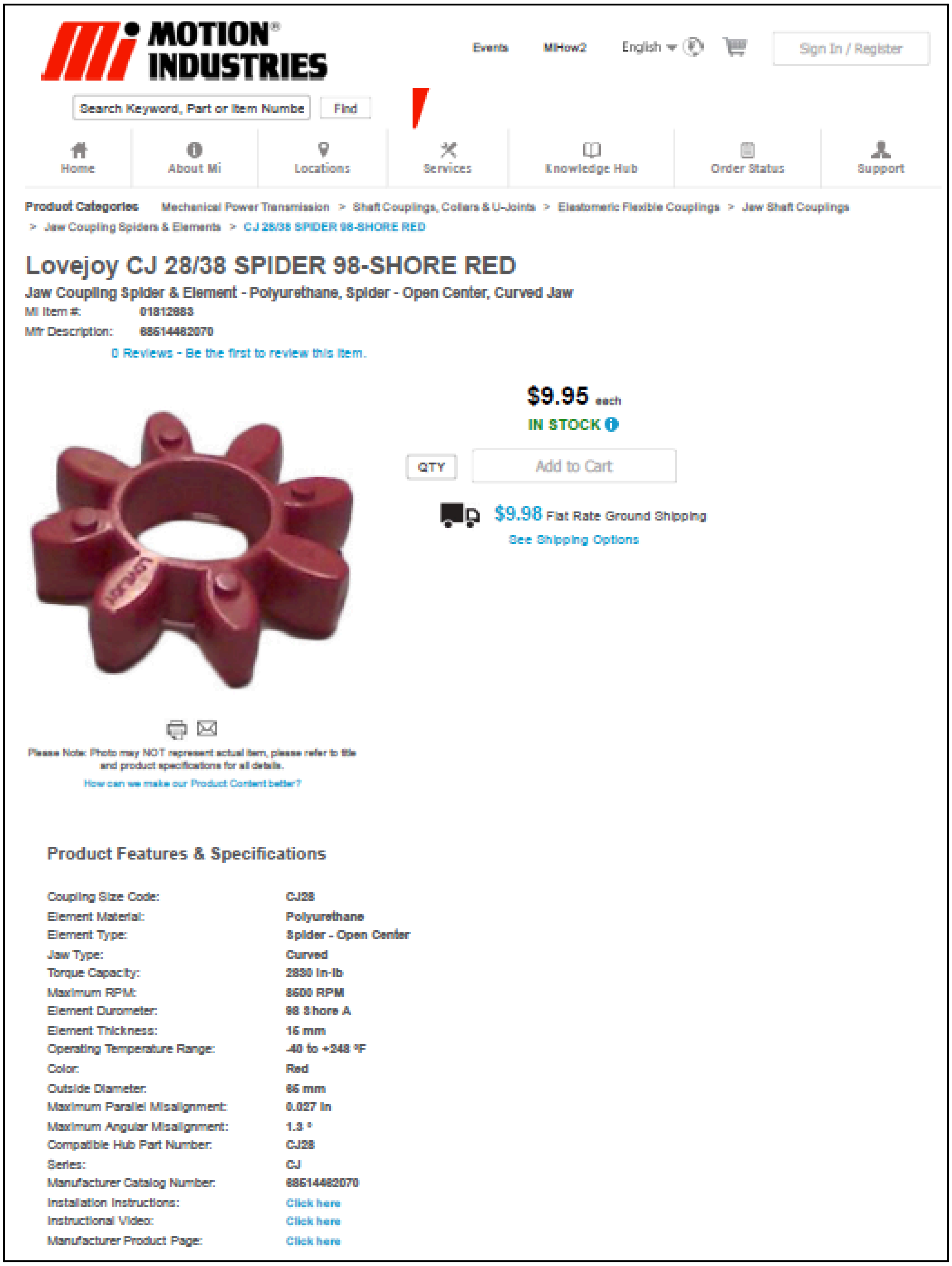
$9.98 Flat Rate Ground Shipping

See Shipping Options

Product Features & Specifications

Coupling Size Code:	CJ28/38
Bore Type:	Finished with Keyway
Bore Diameter:	1.000 In
Jaw Type:	Curved
Hub Material:	Aluminum, Powdered Metal
Outside Diameter:	2.5600 In
Length Through Bore:	1.38 (Aluminum) (A Style) In, 1.38 (Aluminum) (B Style) In, 1.38 (Powder Metal) In, 2.58 (Aluminum) (BX Style) In, 2.58 (Powder Metal) In
Coupling Overall Length:	3.54 (Aluminum) (A Style) In, 3.54 (Aluminum) (B Style) In, 3.54 (Powder Metal) In, 5.51 (Aluminum) (BX Style) In, 5.51 (Powder Metal) In
Hub End Diameter:	1.88 (Aluminum) (A Style) In, 1.88 (Powder Metal) In
Torque Capacity:	1680 (Urethane Spider-92 Shore A-Yellow) In-lb, 2830 (Urethane Spider-98/95 Shore A-Red) In-lb, 3540 (Urethane Spider 64 Shore D-Green) In-lb, 814 (Urethane Spider-80 Shore A Sizes 14-38-Blue) In-lb
Maximum RPM:	8500 RPM
Screw Clamp Type:	Set Screw
Series:	CJ
Manufacturer Product Page:	Click here
Instructional Video:	Click here
Manufacturer Catalog:	Click here

Lovejoy CJ 28/38 SPIDER 98-SHORE RED (Through Motion Industries)



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Product Categories Mechanical Power Transmission > Shaft Couplings, Collars & U-Joints > Elastomeric Flexible Couplings > Jaw Shaft Couplings > Jaw Coupling Spiders & Elements > CJ 28/38 SPIDER 98-SHORE RED

Lovejoy CJ 28/38 SPIDER 98-SHORE RED

Jaw Coupling Spider & Element - Polyurethane, Spider - Open Center, Curved Jaw

MI Item #: 01812883

Mfr Description: 68514482070

0 Reviews - Be the first to review this item.

$9.95 each

IN STOCK

QTY Add to Cart

$9.98 Flat Rate Ground Shipping

See Shipping Options

Please Note: Photo may NOT represent actual item, please refer to title and product specifications for all details.

How can we make our Product Content better?

Product Features & Specifications

Coupling Size Code:	CJ28
Element Material:	Polyurethane
Element Type:	Spider - Open Center
Jaw Type:	Curved
Torque Capacity:	2830 in-lb
Maximum RPM:	8500 RPM
Element Durometer:	98 Shore A
Element Thickness:	15 mm
Operating Temperature Range:	-40 to +248 °F
Color:	Red
Outside Diameter:	65 mm
Maximum Parallel Misalignment:	0.027 in
Maximum Angular Misalignment:	1.3 °
Compatible Hub Part Number:	CJ28
Series:	CJ
Manufacturer Catalog Number:	68514482070
Installation Instructions:	Click here
Instructional Video:	Click here
Manufacturer Product Page:	Click here

Microchip MCP6271 (Through Steve Roberts)

MICROCHIP MCP6271/1R/2/3/4/5

170 µA, 2 MHz Rail-to-Rail Op Amp

Features

- Gain Bandwidth Product: 2 MHz (typical)
- Supply Current: I_Q = 170 µA (typical)
- Supply Voltage: 2.0V to 6.0V
- Rail-to-Rail Input/Output
- Extended Temperature Range: –40°C to +125°C
- Available in Single, Dual and Quad Packages
- Parts with Chip Select ($\overline{CS}$)
  - Single (**MCP6273**)
  - Dual (**MCP6275**)

Applications

- Automotive
- Portable Equipment
- Photodiode Amplifier
- Analog Filters
- Notebooks and PDAs
- Battery Powered Systems

Available Tools

- SPICE Macro Models
- FilterLab® Software
- Mindi™ Circuit Designer & Simulator
- MAPS (Microchip Advanced Part Selector)
- Analog Demonstration and Evaluation Boards
- Application Notes

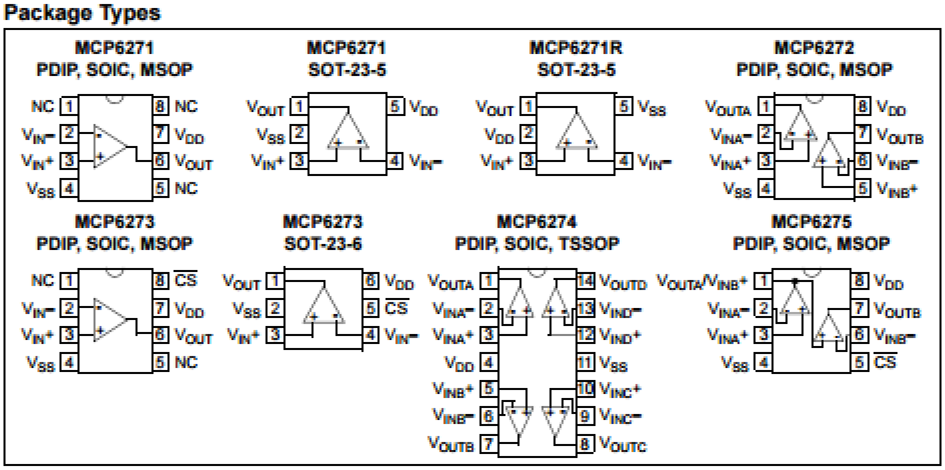
Description

The Microchip Technology Inc. MCP6271/1R/2/3/4/5 family of operational amplifiers (op amps) provide wide bandwidth for the current. This family has a 2 MHz Gain Bandwidth Product (GBWP) and a 65° Phase Margin. This family also operates from a single supply voltage as low as 2.0V, while drawing 170 µA (typical) quiescent current. The MCP6271/1R/2/3/4/5 supports rail-to-rail input and output swing, with a common mode input voltage range of V_{DD} + 300 mV to V_{SS} – 300 mV. This family of op amps is designed with Microchip's advanced CMOS process.

The MCP6275 has a Chip Select input ($\overline{CS}$) for dual op amps in an 8-pin package and is manufactured by cascading two op amps (the output of op amp A connected to the non-inverting input of op amp B). The $\overline{CS}$ input puts the device in low power mode.

The MCP6271/1R/2/3/4/5 family operates over the Extended Temperature Range of –40°C to +125°C, with a power supply range of 2.0V to 6.0V.

Package Types



 DS21810F-page 1

Nachi 6006ZZE Shielded Bearings (Through VXB)

6006ZZE Shielded Nachi Bearing
One Bearing
Made in Japan

6006ZZE Nachi Bearing, the 6006ZZE inner diameter is 30mm, the 6006ZZE outer diameter is 55mm, 6006ZZE width is 13mm.

- Item: 6006ZZE Nachi Ball Bearing
- Type: Nachi Deep groove Ball Bearing
- Closures: Double shielded with Metal shields Z/ZZ/2Z
- Radial Clearance: C3
- Dimensions: 30mm x 55mm x 13mm/Metric
- ID (inner diameter)/Bore: 30mm
- OD (outer diameter): 55mm
- Width/Height/thickness: 13mm
- Size: 30 x 55 x 13 mm
- Quantity: One Bearing
- Dynamic load rating Cr: 13,200 N
- Static load rating Cor: 8,300 N
- Limiting Speed:
  - Grease Lubrication: 13,000 RPM
- Equal: 6006 ZZ / 6006 2Z
- Made in Japan

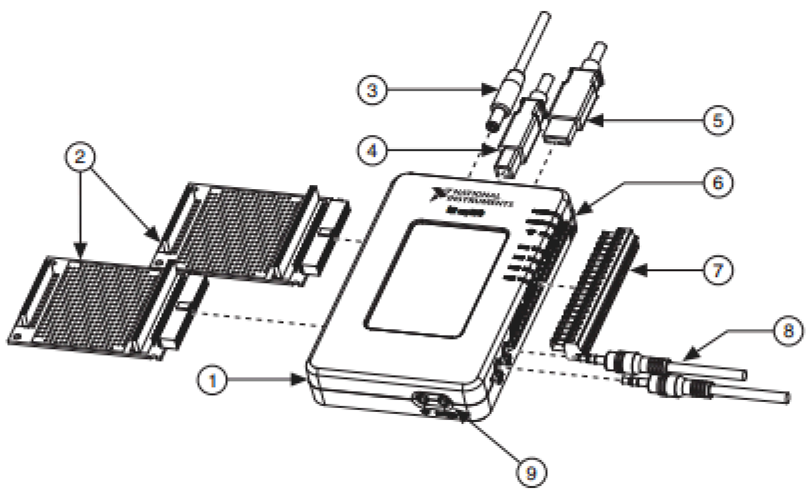
National Instruments myRIO-1900

USER GUIDE AND SPECIFICATIONS

NI myRIO-1900

The National Instruments myRIO-1900 is a portable reconfigurable I/O (RIO) device that students can use to design control, robotics, and mechatronics systems. This document contains pinouts, connectivity information, dimensions, mounting instructions, and specifications for the NI myRIO-1900.

Figure 1. NI myRIO-1900



1	NI myRIO-1900	6	LEDs
2	myRIO Expansion Port (MXP) Breakouts (One Included in Kit)	7	Mini System Port (MSP) Screw-Terminal Connector
3	Power Input Cable	8	Audio In/Out Cables (One Included in Kit)
4	USB Device Cable	9	Button0
5	USB Host Cable (Not Included in Kit)		

NATIONAL INSTRUMENTS

Full user guide and specifications can be found at: http://www.ni.com/pdf/manuals/376047c.pdf

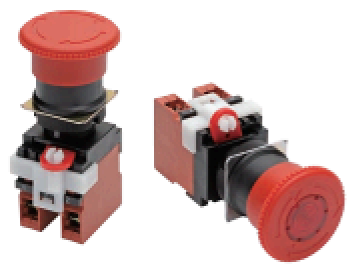
Omron A22E Emergency Stop Switch (Through Mouser)

Emergency Stop Switch (22-dia./25-dia.)

A22E

CSM_A22E_DS_E_18_2

Install in 22-dia. or 25-dia. Panel Cutout (When Using a Ring)



For the most recent information on models that have been certified for safety standards, refer to your OMRON website.

- Direct opening mechanism to open the circuit when the contact welds ⊖.
- Safety lock mechanism prevents operating errors.
- Easy mounting and removal of Switch Blocks using a lever.
- Mount three Switch Units in series to improve wiring efficiency (with non-lighted Switch Units, three Units can be mounted for multiple contacts).
- Finger protection mechanism on Switch Unit provided as a standard feature.
- Use 25-dia. ring to install in 25-dia. panel cutouts.
- Install using either round, or forked crimp terminals.
- Oil-resistant to IP65 (non-lighted models)/IP65 (lighted models)
- A lock plate is provided as a standard feature to ensure that the control box and switch are not easily separated.

⚠ Be sure to read the *"Safety Precautions"* on page 17.

Model Number Structure

Model Number Legend (Completely Assembled)

........Shipped as a set which includes the Operation Unit, Lamp (lighted models only), and Switch.

A22E [L] - [M] - [24A] - [01] - [ ] - [ ]
(1) (2) (3) (4) (5) (6)

1. Lighted/Non-lighted

Code	Description
None	Non-lighted
L	Lighted *

* Lighted Emergency Stop Switches are available only for the medium (M). push-lock turn-reset models.

2. Head Size

Code	Size	Description
MP	Medium 40 dia.	Push-pull
S	Small 30 dia.	Push-lock turn-reset
M	Medium 40 dia.	
L	Large 60 dia.	

3. Light Source

Without Voltage Reduction Unit

Code	Description	Operating Voltage
None	Non-lighted	---
6A	LED	6 VAC/VDC
12A		12 VAC/VDC
24A		24 VAC/VDC

With Voltage Reduction Unit

Code	Description	Operating Voltage
T1	LED	100 VAC
T2		200 VAC

Equipped with 24-VAC/DC LED.

4. Contacts

Code	Description
01	1NC
11	1NO + 1NC
02	2NC (1NC + 1NC)
12	1NO + 2NC (1NC + 1NC)
03	3NC (1NC + 1NC + 1NC)

5. Configuration

Code	Configuration
None	Switch only
B	Switch with Integrated Control Box

6. Configuration

Code	Configuration
None	Neither "EMO" nor "EMS" printed, arrows engraved in red.
EMO	"EMO" and arrows printed in white.
EMO-RD	"EMO" printed in white, arrows engraved in red.
EMS	"EMS" and arrows printed in white.
EMS-RD	"EMS" printed in white, arrows engraved in red.

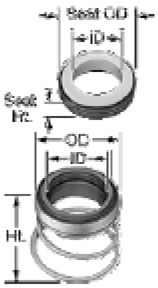
Pac-Seal Type 21 Oil-Resistant Pump Shaft Seal (Through McMaster-Carr)

McMASTER-CARR.

Oil-Resistant Pump Shaft Seal

Flexible, Buna-N Diaphragm, 1-1/4" ID, Trade No. 185

In stock
$18.64 Each
9281K352



For Motion Type	Rotary
Material	304 Stainless Steel
ID	1 1/4"
OD	1.875"
Height	1.062"
Seat	
OD	1.875"
Height	0.437"
Material	
Diaphragm	Buna-N Rubber
U-Cup	Buna-N Rubber
Seat	Ceramic
Washer	Carbon
Diaphragm Hardness	Durometer 65A (Medium)
Maximum Pressure	150 psi
Maximum Speed	5,000 fpm
Temperature Range	-40° to 212° F
Trade No.	185
Manufacturer/Brand Equivalent	John Crane Type 21 Pac-Seal Type 21 Sealol Type 43 CE Sealol Type 43 CU Short
Features	Extended Spring
For Use With	Ammonia Animal Oils Boric Acid Butyl Alcohol Calcium Hydroxide Citric Acid Diluted Salt Solutions Ethylene Glycol Glycerin Grease Methanol Mineral Oils Motor Oils Petroleum Fluids Sodium Bicarbonate Transmission Fluids Vegetable Oils
RoHS	Not Compliant

An extended spring on these seals accommodates variations in seal cavity length and pump misalignment. All have a stainless steel outer shell that locks the diaphragm and washer together, which makes them easy to install and eliminates the need for bonding adhesives. They are commonly used in circulation and dispensing pumps.

Seals that come with a seat are furnished as two pieces. The seats have a U-cup seal that is the same material as the diaphragm.

304 stainless steel seals are comparable to Pac-Seal Type 21, John Crane Type 21, and Sealol Type 43 CE and 43 CU short.

Sparkfun/Dallas Semiconductor Waterproof Temperature Sensors (DS18B20)

PRELIMINARY

DALLAS SEMICONDUCTOR

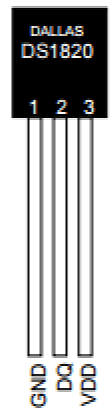
DS18B20 Programmable Resolution 1-Wire® Digital Thermometer

www.dalsemi.com

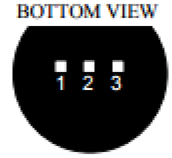
FEATURES

- Unique 1-Wire interface requires only one port pin for communication
- Multidrop capability simplifies distributed temperature sensing applications
- Requires no external components
- Can be powered from data line. Power supply range is 3.0V to 5.5V
- Zero standby power required
- Measures temperatures from -55°C to +125°C. Fahrenheit equivalent is -67°F to +257°F
- ±0.5°C accuracy from -10°C to +85°C
- Thermometer resolution is programmable from 9 to 12 bits
- Converts 12-bit temperature to digital word in 750 ms (max.)
- User-definable, nonvolatile temperature alarm settings
- Alarm search command identifies and addresses devices whose temperature is outside of programmed limits (temperature alarm condition)
- Applications include thermostatic controls, industrial systems, consumer products, thermometers, or any thermally sensitive system

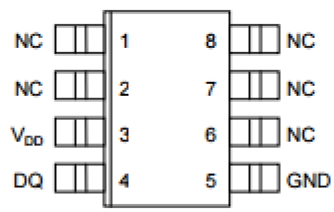
PIN ASSIGNMENT



BOTTOM VIEW



DS18B20 To-92 Package



DS18B20Z
8-Pin SOIC (150 mil)

PIN DESCRIPTION

GND - Ground
DQ - Data In/Out
V_{DD} - Power Supply Voltage
NC - No Connect

DESCRIPTION

The DS18B20 Digital Thermometer provides 9 to 12-bit (configurable) temperature readings which indicate the temperature of the device.

Information is sent to/from the DS18B20 over a 1-Wire interface, so that only one wire (and ground) needs to be connected from a central microprocessor to a DS18B20. Power for reading, writing, and performing temperature conversions can be derived from the data line itself with no need for an external power source.

Because each DS18B20 contains a unique silicon serial number, multiple DS18B20s can exist on the same 1-Wire bus. This allows for placing temperature sensors in many different places. Applications where this feature is useful include HVAC environmental controls, sensing temperatures inside buildings, equipment or machinery, and process monitoring and control.

1 of 27 050400

Full data sheet and specifications can be found at:
https://cdn.sparkfun.com/datasheets/Sensors/Temp/DS18B20.pdf

TACO 0011 Pump Spec Sheet

Submittal Data Information
Model 0011 Cartridge Circulator

Submittal Data # 101-034
Supersedes: 06/10/13

Effective: 01/12/15

Features

- Exclusive ACB Anti-condensate baffle with ambient air flow-Protects motor windings against condensate buildup
- High velocity performance — compact design
- Quiet, Efficient operation
- Direct drive-Low power consumption
- Unique replaceable cartridge design — field serviceable
- Self lubricating
- No mechanical seal
- Unmatched reliability-maintenance free
- Universal flange to flange dimensions
- Cast Iron or Stainless Steel construction

Materials of Construction

Casing (Volute):........Cast Iron or St. Steel
Stator Housing:........Aluminum
Cartridge:..................Stainless Steel
Impeller: ....Non-Metallic
Shaft:........................Ceramic
Bearings:..................Carbon
O-Ring & Gaskets:.....EPDM

Model Nomenclature

F – Cast Iron, Flanged
SF – Stainless Steel, Flanged

Performance Data

Max. Flow: 31 GPM
Max. Head: 31 Feet
Min. Fluid Temperature: 40°F (4°C)
Max. Fluid Temperature: 230°F (110°C)
Max. Working Pressure: 150 psi
Connection Sizes:
3/4", 1", 1-1/4", 1-1/2" Flanged

Certifications & Listings

c UL US LISTED FOR INDOOR USE ONLY

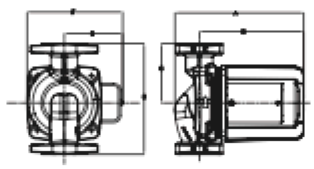
NSF Low-Lead Compliant

Application

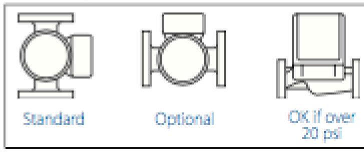
The Taco 0011 is specifically designed for high head / medium flow applications in large residential or light commercial closed loop hydronic heating and chilled water cooling systems. Ideal for high pressure drop boilers, fan coil units, heat exchangers, large radiant heating and heat recovery/geothermal systems. The Stainless Steel 0011 can be used on open loop systems. The unique replaceable cartridge contains all of the moving parts and allows for easy service instead of replacing the entire circulator. Universal flange to flange dimensions and orientation allows the 0011 to easily replace other models. Compact, low power consumption design makes it ideal for high-efficiency jobs.

Pump Dimensions & Weights

Models	Casing	A		B		C		D		F		G		Ship Wt.	
		in.	mm	in.	mm	in.	mm	in.	mm	in.	mm	in.	mm	lbs.	Kg
0011-F4	Cast Iron	7-1/2	191	6-1/8	156	3-1/2	89	3-3/8	86	5-5/8	143	6-1/2	165	12.0	5.5
0011-SF4	St. Steel	7-1/2	191	6-1/8	156	3-1/2	89	3-3/8	86	5-5/8	143	6-1/2	165	11.0	5.0



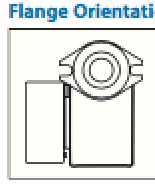
Mounting Positions



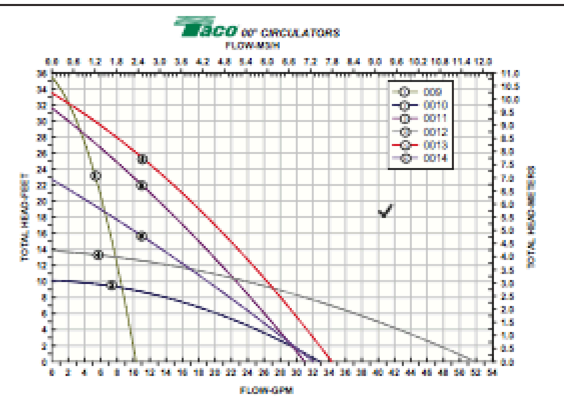
Electrical Data

Model	Volts	Hz	Ph	Amps	RPM	HP
All Models	115	60	1	1.76	3250	1/8
Motor Type	Permanent Split Capacitor Impedance Protected					
Motor Options	220/50/1, 220/60/1, 230/60/1, 100/110/50/60/1					

Flange Orientation



Performance Field - 60Hz



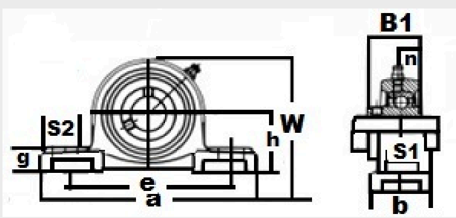
UCP211-32 Pillow Block Bearing (Through VXB)

UCP211-32 Pillow Block Bearing
Set Screw Type High-Center Height
2" Inch Inner Diameter

UCP211-32 Pillow Block Bearing, UCP211-32 is Set Screw Type High-Center Height, Inner Diameter is 2" Inch.

- Item: UCP211-32 Bearing
- Type: Pillow Block Mounted Bearing
- Bolt Size: 5/8" inch
- Basic Load Dynamic (Cr): 43,400 N
- Basic Load Static (Cor): 29,400 N
- Bearing No: UC211-32
- Housing No: P211
- Quantity: One Bearing

Dimensions (Inchs)									
h	a	e	b	S1	S2	g	W	B1	n
2-1/2"	8-5/8"	6-23/32"	2-3/8"	7/8"	25/32	3/4"	4-29/32"	2-3/16"	7/8"



2" Inch Bearing UCP211-32 + Pillow Block Cast Housing Mounted Bearings

Kit18473

Watts LF777SI Wye-Pattern, Lead Free Filter (Through Supply House)

LEAD FREE*

Series LF777SI, LFS777SI

Wye-Pattern, Lead Free Cast Strainers
NSF Certified to NSF 61 Annex G

Sizes: ⅜" – 3" (10-75mm)

LF777SI

Series LF777SI, LFS777SI Wye-Pattern, Lead Free* cast strainers are designed to protect plumbing system components from dirt, rust and other damaging debris. The Series LF777SI and LFS777SI feature Lead Free* construction to comply with Lead Free* installation requirements.

Features

- NSF 61 Annex G certified
- Lead Free* cast copper silicon alloy body and cap
- Wye-pattern
- Tapped retainer cap
- Closure plug
- Special flared screen opening on upstream end to provide unrestricted flow through the strainer

Models

LF777SI – ⅜" – 3" (10 – 80mm) threaded connections
LFS777SI –½" – 2" (15 – 50mm) solder connections†

Specifications

A wye-pattern, Lead Free* cast strainer to be installed as indicated on the plans. The strainer must have a tapped retainer cap and closure plug. Strainer shall be rated to 400psi (27.6 bar) WOG; 125psi (8.6 bar) WSP for sizes ⅜"-2" (10-50mm) and 300psi (20.7 bar) @ 210ºF (99ºC); 125psi (8.6 bar) WSP @ 353ºF (178ºC) for sizes 2½"-3" (65-80mm). The strainer shall be constructed using Lead Free* cast copper silicon alloy. Lead Free* strainers shall comply with state codes and standards, where applicable, requiring reduced lead content. Strainer shall be a Watts Series LF777SI (threaded ends) or LFS777SI (solder ends).

Materials

Body:	Lead Free* cast copper silicon alloy
Retainer Cap:	Lead Free* cast copper silicon alloy
Plug	Lead Free* brass
Gasket:	NBR
Standard Screen:	#20 mesh, 304 stainless steel

Pressure – Temperature

Maximum Working Pressure:

⅜"–2" (10-50mm)

400psi (27.6 bar) WOG @ 210ºF (99ºC)
125psi (8.6 bar) WSP @ 353ºF (178ºC)

2½"–3" (65-80mm)

300psi (20.7 bar) WOG @ 210ºF (99ºC)
125psi (8.6 bar) WSP @ 353ºF (178ºC)

***The wetted surface of this product contacted by consumable water contains less than 0.25% of lead by weight.**

Full data sheet can be found at:
http://s3.supplyhouse.com/product_files/Watts%20-%200379112%20-%20Submittal%20Sheet.pdf

F: Designs Considered

As a project involving many custom components, each component went through a design, manufacturing, and assembly phase. Particularly, the water brake, driveline and frame went through multiple iterations of design before being manufactured. In this section some of the milestone design iterations of these subsystems will be discussed.

The driveline was first designed with a gear ratio of 2, to lower rotational speed of the dynamometer. After considering safety implications, and the possibility of a chain failure, the frame and driveline were redesigned to accommodate a drive shaft driveline.

The initial design concept is shown in Figure A. Design started with the intention of driving the dynamometer with a chain for cost and adjustability reasons. Due to the high CG, poor mobility, and lack of safety in this early design, a more thought out design was created next.

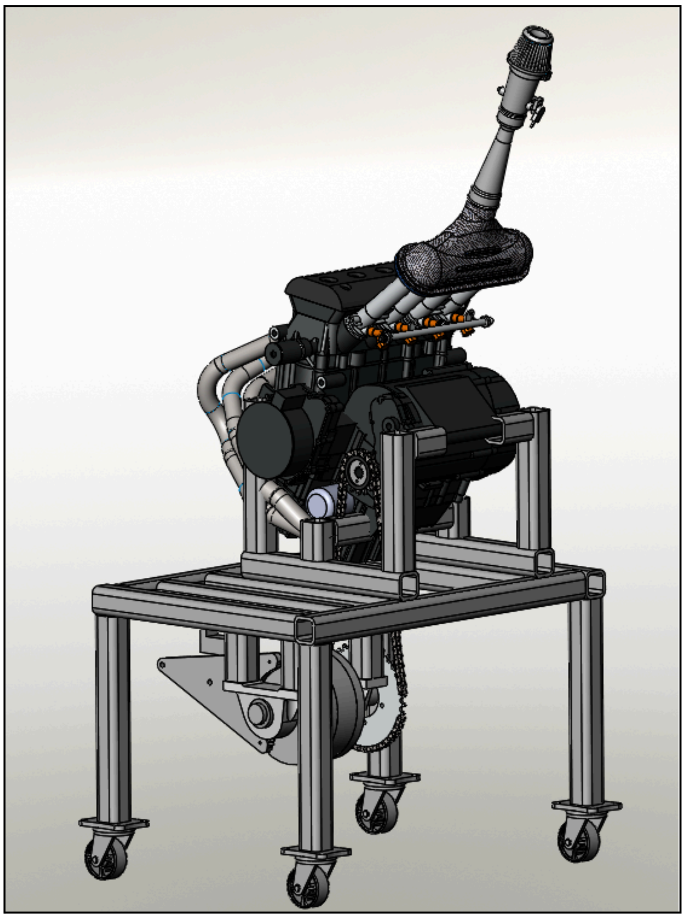


Figure A: Initial Design Concept of Frame

The second milestone design was still in the early stages and exploring the solution space of where the major components would be located and how they would operate together. It is shown in Figure B. A separate safety frame with panels was added to the design. The system CG was lowered to the ground for better stability. Also, a motor coupler was added to the driveline to help decrease vibration from the chain. A flaw of this design is that pillow block bearings were used to spin at high RPMs and the safety frame was not a good solution to the operator safety problem.

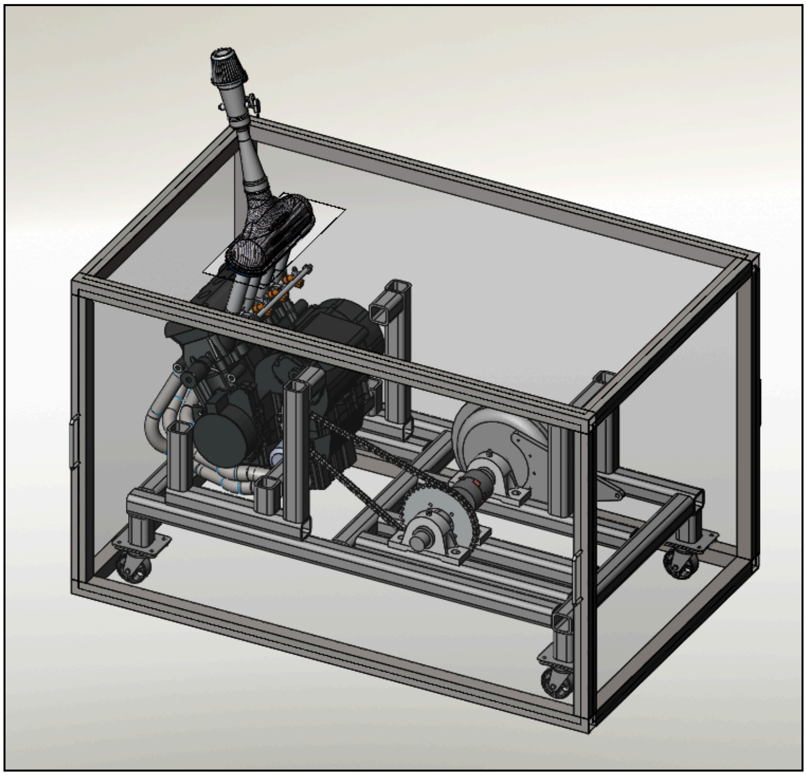


Figure B: Second Design Concept of Frame

The third major design milestone included many of the components in the final design. It is shown in Figure C. A reservoir for a closed water system was added and to be made from welding sheet metal to the frame. In retrospect, welding a sheet metal aluminum reservoir would have been a more cost effective and customizable solution than buying the expensive polypropylene reservoir that was included in the final design. Also consider that many glues and epoxies don't bond to polypropylene so the final design used ratchet straps to hold it in place. At this point in the design phase, the engine subframe needed a redesign and other early stage design considerations were being figured out and implemented.

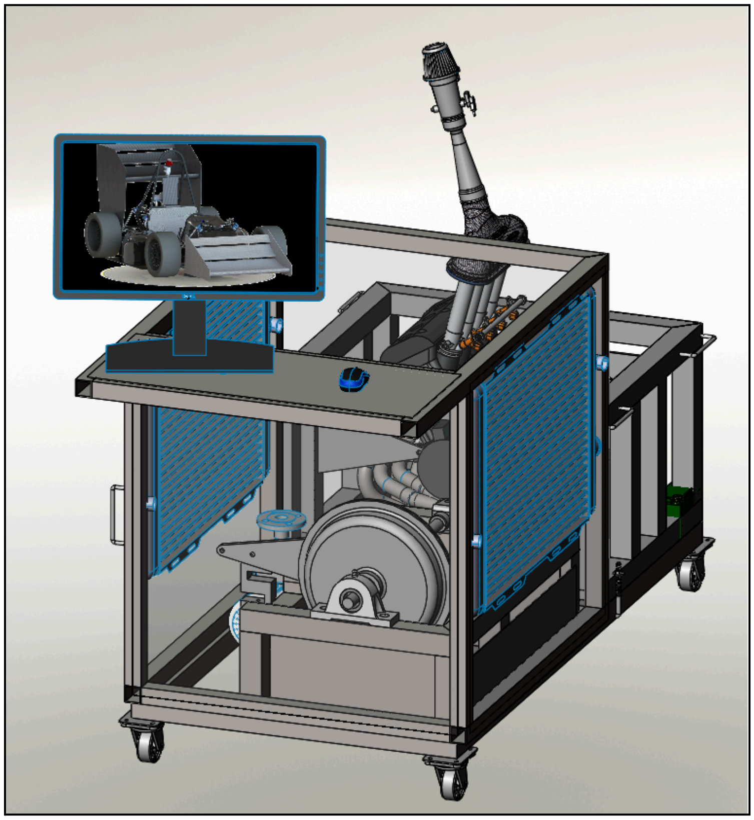


Figure C: Third major Design Concept of Frame

The water brake was initially designed with a tight tolerance machined feature to concentrically locate the trunnions to the stators. Problems with this design included increased manufacturing difficulty, and no provision for concentrically locating the stators with each other. Additionally, this design included stators which were identical to each other, however, since the outlet was at the bottom center of the stators (not shown in Figure D), the design left no simple method for creating a watertight seal. Lastly, this design included provision for a single bearing on each stator side, with interference fits on both inner and outer race of the bearing, neglecting the consequences of thermal expansion of the shaft as the system heats up.

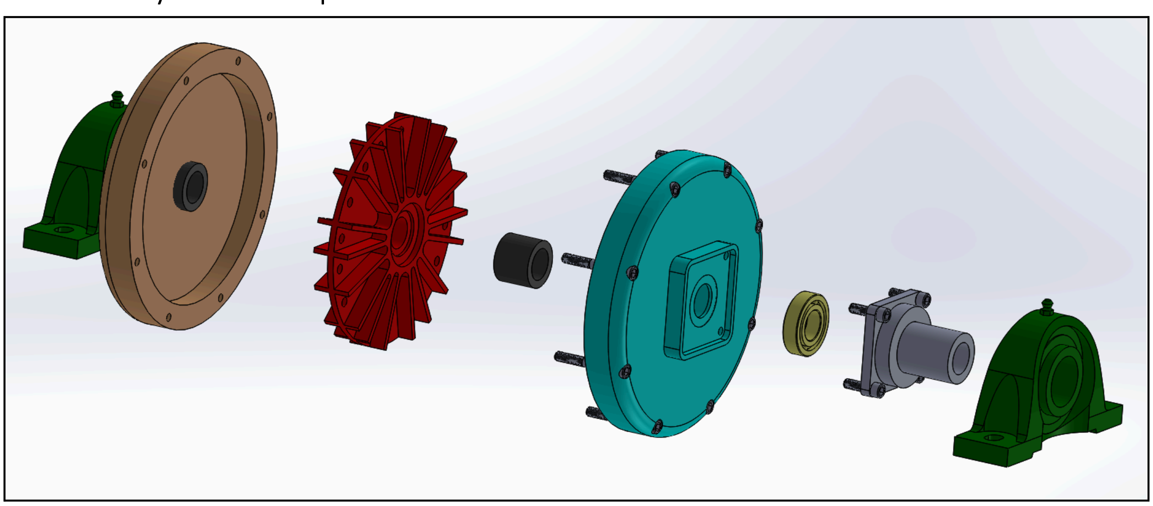


Figure D: Initial Design of the Water Brake

The second design iteration (Fig. E) included multiple changes the addition of precision ground shoulder screws to accurately and concentrically locate the stators to each other. This design change meant that the machined components can be manufactured with a larger tolerance, since they were no longer relied on for location. Further, addressing the issues with the initial design, the second iteration incorporates three bearings, of which two have clearance fits on their internal diameter to allow for thermal expansion of the shaft. Also, the stators are unique from each other, one of which is flat, and the other with a deep pocket to house the rotor, and provide enough space for a water outlet. Since the water outlet is now threaded into only one stator, an o-ring can easily be added to create a watertight seal. Finally, the design included a water passage to allow for water to cool the mechanical seals. The key difference between the second and final design iteration of the water brake is the stators, which were initially designed and manufactured with a flat wall. After discussing viability with a Stuska engineer, the team realized that viscous forces and change in momentum alone would not suffice for creating enough torque to meet the design requirements. Thus, pockets with equidistant blades were machined into the stators to induce toroidal flow and water shearing[3].

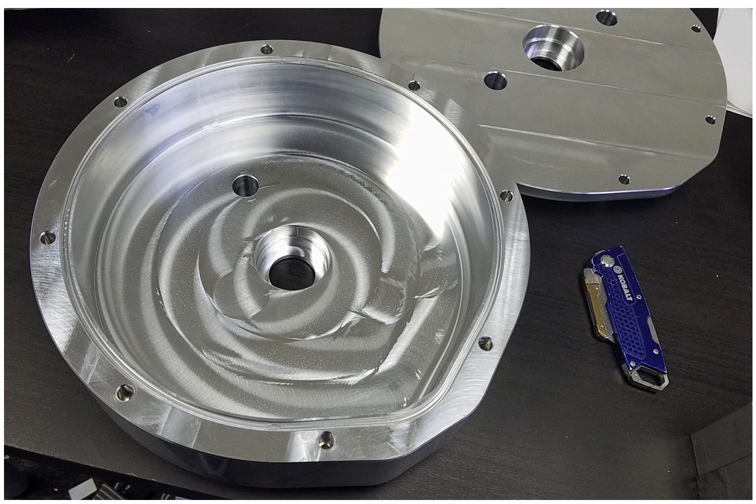


Figure E: Second Iteration of Water Brake Stators

G: Budget

FSAE Engine Dyno Budget	
Item	**Cost**
Frame	
Stock Steel Tubes	$224
Stock Steel Plates and Mesh	$134
Radiators and Fans	$254
Accesories (Casters, Hinge, Ubolts)	$121
Frame Subtotal	***$733***
Pump System	
Pump	$208
Reservoir	$127
Fittings and Barbs	$114
Filter	$21
Gate Valve	$21
Hosing	$90
Accesories (L-brackets, Nuts, Washers, Clamps)	$58
Pump System Subtotal	***$638***
Water Brake	
Round Aluminum Stock (stators and rotors)	$180
Bearings	$86
Pillow Block Bearings	$108
Grease Fittings	$10
Shaft Seal	$37
Water Brake Subtotal	***$422***
Data Acquisition	
Load Cell	$113
Instrumentation Amp for Load Cell	$12
Temperature Sensors	$40
Hall-Effect RPM Sensor	$22
Vibration Isolator	$11
Data Aquisition Subtotal	***$197***
Engine	
Fuel Tank	$27
Engine Battery	$32
Bolts for Mounting	$18
Engine Subtotal	***$76***
Drivetrain	
Round Steel Stock (Coupler to Srocket and Threaded Washer)	$11
Square Steel Stock (Trunnions and Bearing Blocks)	$74
Key Stock	$1
Torsional Coupler	$91
Drivetrain Subtotal	***$177***
Other	
Emergency Stop Switch	$45
Power Supply (Fans and Electronics)	$21
Other Subtotal	***$66***
Project Total	**$2,310**
Budget Allocation	*$2,200*
Overbudget	*$110*

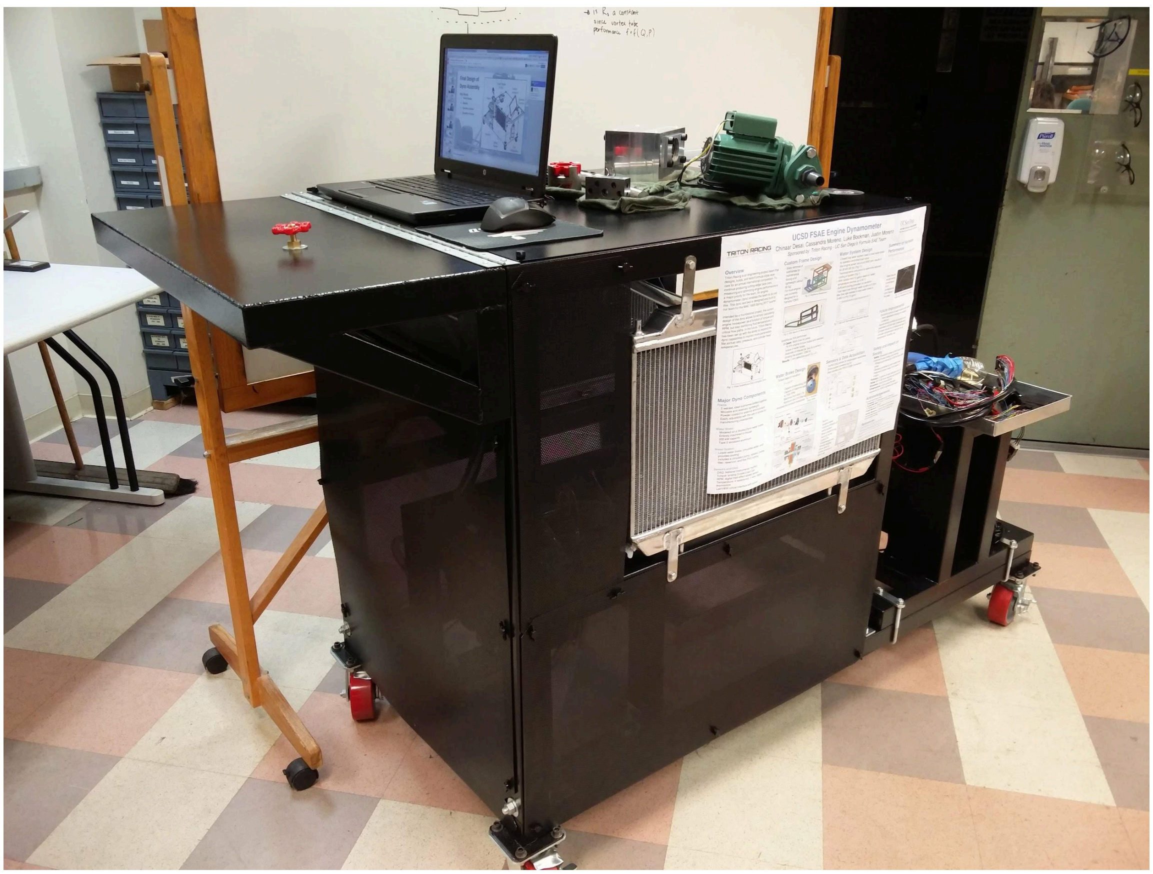
H: Users/Maintenance/Safety/Operation Manual

TRITON RACING
UNIVERSITY OF CALIFORNIA, SAN DIEGO

UC San Diego
JACOBS SCHOOL OF ENGINEERING
Mechanical and Aerospace Engineering

Engine Dynamometer

Operator's Manual



Last Updated: 6/9/2017

UCSD Formula SAE Engine Dynamometer

Table of Contents

Assembly

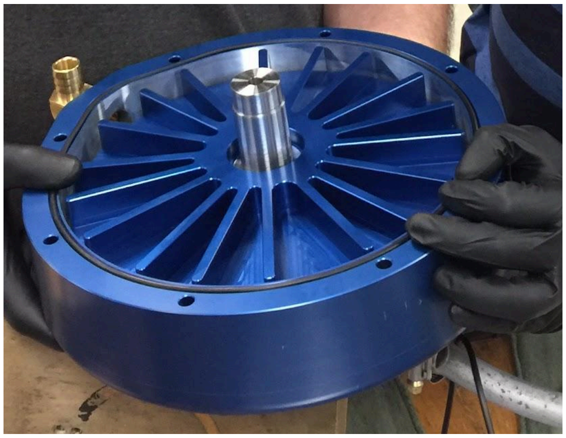
Water Brake

2 people recommended

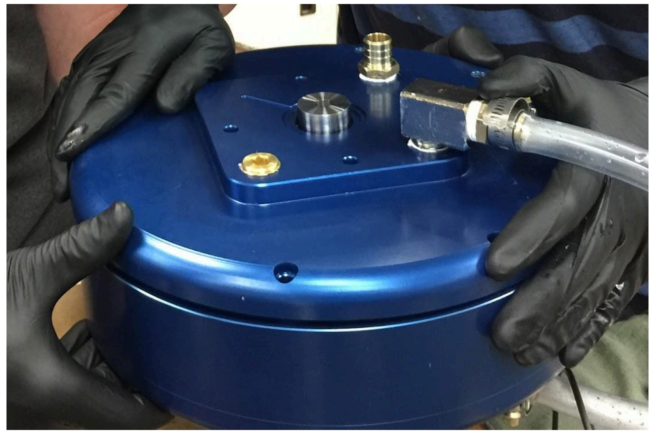
1. Make sure all assembly components are clean and free of dust, dirt, and grease.
2. Using press tool, press (x2) 6006ZZE bearings into driveline side trunnion.
3. Using (x4) 5/16” x .75” shoulder screws, fasten driveline side trunnion to the thick driveline side stator.
4. Clean (with warm soapy water) (x2) seal type 21 (1 1/4” shaft) mechanical seals
5. Apply a small amount of soap to the rubber outside and install the seal seat into driveline side stator.
6. Clean and install (x1) mechanical seal seat into thin outer side stator in the same manner as the driveline side stator.
7. Apply a small amount of soap to the rubber bellows of (x1) mechanical seal and Install onto rotor/shaft assembly on the keyed side of shaft.
8. Carefully install driveline side stator with mechanical seal onto the shaft/rotor assembly. Avoid touching the polished sealing faces to anything so that the seals do not get damaged.

Step 8.

9. Carefully flip the assembly over.
10. Repeat step 7 on the opposing side of the rotor/shaft.
11. Install .139” diameter o-ring into o-ring groove on driveline side stator.



Step 11.



Step 12.

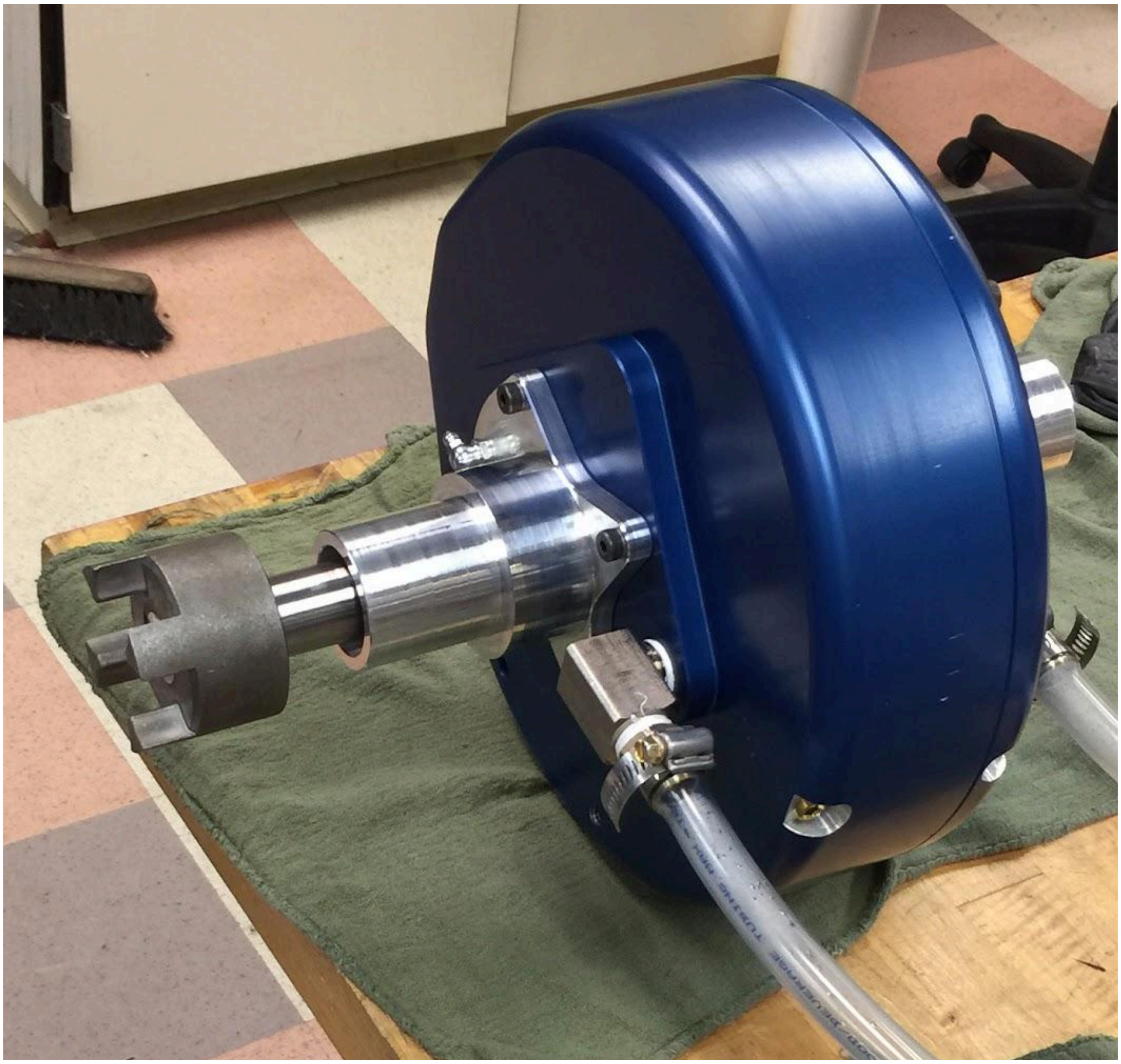
12. Align the stators and carefully install outer side stator onto rotor shaft.
13. Fasten stators to each other using (x8) 5/16” x 1.25” shoulder screws. Use german torque spec “gutentite”.

Step 13.

Step 17.

14. Install (x1) external retaining ring onto shaft.
15. Using press tool, press (x1) 6006 bearing onto shaft, until it is flush with the shaft end.
16. Install (x1) bearing retaining washer using (x1) 3/8” shoulder screw.
17. Press assembly onto outer side trunnion such that the force is only transmitted through the outer race.

18. Fasten outer side trunnion to outer side stator using (x4) 5/16 x .75” shoulder screws.
19. Install (x1) key and (x1) driveshaft coupler onto the keyed side of the shaft with a press if necessary. (do NOT use a hammer to install the coupler as it could damage the bearings)



Completed water brake assembly.

Disassembly

Water Brake

Due to high possibility of damaging mechanical seals and bearings, disassembly is not recommended unless absolutely necessary.

If disassembly is required, 2 people are recommended.

Important note: if bearings are removed, from the trunnions, they must be discarded, since they bearing races were over loaded in the disassembly process. Replace with new 6006-2ZZ bearings.

Procedure 1: Access inside of stators, rotor, and mechanical seals:

1. Remove torsional coupler and key from shaft.
2. Remove (x8) 5/16” x .125” shoulder screws from stators.
3. Hold the assembly with the driveline side up, and pull the driveline side stator straight up. The bearings are a very close slip fit, so it should slide out easily as long as lifting force is applied parallel to shaft.

Procedure 2: Access bearings on outer side trunnion:

1. Remove (x4) 5/16” x .75” shoulder screws from outer side trunnion.
2. Using (x4) ¼-20 screws with at least 1” of thread length, thread a screw into each threaded hole of the trunnion until they contact the stator surface.
3. Slowly tighten each screw ¼ turn at a time until the trunnion is separated from the stator.
4. If the bearing remains on the shaft, follow the Procedure 1 to separate the stators from each other, and proceed to step 5. If the bearing remains inside the stator, skip to 6.
5. Remove the bearing by pressing the shaft through the stator such that the stator forces the bearing off of the shaft. Skip step 6.
6. Using an anvil of appropriate size, press bearing out of trunnion.

Procedure 3: Access bearings on driveline side trunnions

1. Remove (x4) 5/16” x .75” shoulder screws from driveline side trunnion.
2. Using an anvil of appropriate size, press bearings out of trunnion.

Maintenance

- Grease the grease fittings in the water brake trunnions with Mobile Mobilith SHC™ 100 or Lupriplate P/N L0189-098 every 50 hours of operation
- Grease the pillow blocks with Mobile Mobilith SHC™ 100 or Lupriplate P/N L0189-098
- Calibrate load cell
- Calibrate temperature sensors
- Calibrate shaft speed sensor
- Verify that drive shaft joints have grease in them
- Inspect drive shaft sliding spline seal to make sure its free from dirt and dust

General Operation

1. Fill up the reservoir with water and other additives to prevent mold growth
2. Full up engine radiator making sure vent hose is above radiator
3. Turn on master switch
4. Connect to the MyRIO
5. Turn on and fans
6. Close water brake valve almost all the way(make sure to leave at least a small amount of flow going into the water brake because **the water brake seals require water whenever they are rotating**
7. Disengage e-stop switch to turn the engine battery on.
8. Turn on engine in neutral
9. Slowly shift up to 6th gear
10. Start testing by using throttle lever and water brake valve
11. **Update this section after dyno has been proven in its first test**
12. After testing is completed, and press e-stop switch
13. Disconnect from the MyRIO
14. Turn off master switch
15. Empty engine coolant reservoir
16. Empty water brake system reservoir
17. Attempt to get a majority of the water out of the system using compressed air or other methods

Notes:

- Do NOT operate the system dry. The water brake Seals require water when they are rotating. Water brake can be upgraded to always have a small amount of water flowing into the seal cavity via the a water tap before the inlet valve to the plug fittings on the top of the stators.
- Master switch directly controls radiator fans, pump, and MyRIO DAQ systems
- If water is stuck in the water brake vent line, inlet valve response will become sluggish. This line should have a downward slope to ensure water doesn't get trapped in it. If steam is coming out of this line, then not enough water is entering the water brake and testing should be stopped because cavitation is likely present.

- The outlet valve should be set during initial testing and not changed unless a different engine is run. Mark the valve orientation with a paint pen after calibration to keep a consistent outlet diameter. Balancing the inlet available pressure and the outlet valve should allow the engine to change rpm at about the same rate as the valve change(in both positive and negative directions) with equal changes in the valve.

Safety Information

Operation	Hazards	Controls
• Moving cart around on casters	• Cart tripping over • Cart moving away • Crashing cart	• Lock casters when not moving • Do not move cart on steep inclines • Use 2 people to move cart at all times
• Hooking up engine to dyno	• If alignment is not made so the u-joint has equal angles on each side, intense driveshaft vibrations will occur	• Carefully align engine subframe and measure alignment • If large vibrations are present during operation, shut down all systems from the operator panel and investigate the cause.
• Continuous Dyno Operation	• Loud noises • If universal joint on driveshaft fails at high rpm, it will have a lot of kinetic energy	• Wear hearing protection when operating the dyno • Operate the dyno outside • Properly install driveshaft failure collar according to NHRA guidelines • Do NOT stand in the plane of rotation of any rotating components while operating the dyno. • Check to make sure all safety shields are installed prior to operating the dyno
• Operating water cooling system	• Warm water at 40 °C (120 °F) will be moving through the water brake side system • Hot water 90 °C (120 °F) will be moving through the engine side system	• If a leak occurs during operation, shut down all system components and wait for the water to cool down before any work is done to fix the system. • Unlike on a car, the water brake side radiator is not pressurized, however, the engine side radiator is pressurized so wait for the system to cool down before removing the radiator cap. • Don't operate the dyno without double checking the coolant pump and fans are working

		correctly and no airflow is blocking the radiators. • Monitor the system temperature and shut the system down if any temperature are hotter than expected on either the engine side or water brake side.
• General operation	• 120V main supply extension cord is high voltage • Catastrophic engine failure • Gasoline canister catches on fire (will replace with a fuel cell in the future)	• Use a grounded mains plug for the extension cord • Ensure frame grounding connections are making good contact • When operating the dyno, always stand behind the operator bench to ensure that any projectiles from any system components won't hit you. • Don't operate the dyno in a way that will cause catastrophic engine failure. • Step away from the dyno, use a chemical fire extinguisher(yellow type) to put out the flame.

SAE Correction Factors

Refer to SAE J607 standard for correction factors

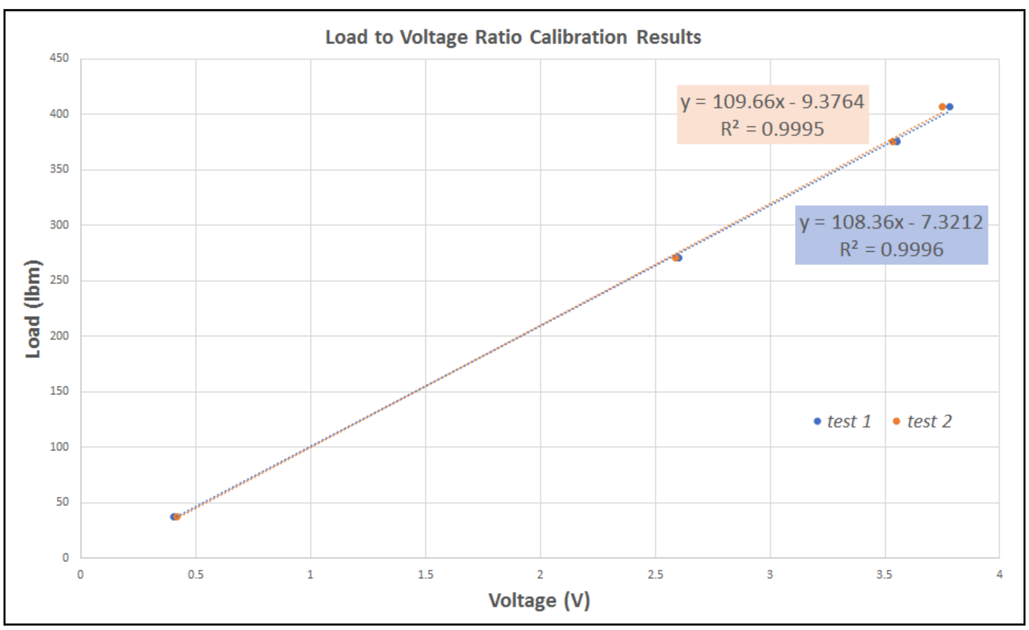
Calibration

Load Cell

The recommended calibration testing cycle for the team is each time the dyno is taken out for use. This is primarily due to the uncertainty of storage conditions that the dyno will be in. It is probable that the load cell will be accidentally loaded/overloaded through mishandling when in the project room. As such, it is best to ensure that the load cell is re-calibrated each time the dyno is used after a period of storage. In addition, in the possibility that the load cell is removed by disassembly of the water brake, re-calibration is mandatory.

With no load, the load cell outputs an offset voltage of 50-70 mV with a 5V supply. It is important to verify that the same offset voltage is read between placing calibration weights on the system to ensure the mechanical system has not changed during loading.

The observed linear relationship between voltage and the load is governed by $mx + b = y$. Where x is the voltage and y is the load. To determine m and b, known weights must be placed on the torque arm and the respective output voltage (out of the operational and instrumentation amplifier) must be recorded. Determining a linear fit from the data through MATLAB or Excel will give the corresponding values for the m and b parameters. Two tests are recommended to be done with the same weights to determine repeatability and ensure precision of the load cell. The example plot below was created as a result of preliminary calibration. The m and b values from here are not accurate due to imperfect testing methods, but should be used as a reference.



Example Plot of Linear Relationship Between Data Points

It is important to consider that the calibration weights will be placed at a distance farther than the location of the load cell. Currently, the design features the load cell attached to the torque arm at a distance of 15.24 cm (6 in.) away from center of the water brake. The calibration weight mount on the torque arm is located 30.48 cm (12 in.) away from the center of the water brake. Therefore, the load induced by a calibration weight at it's currently designed location is twice as much on the load cell.

In normal operation, the system was designed for a max load of 154 kg (340 lbm) to be exerted on the load cell by the water brake, though the capability of the load cell is up to 227 kg (500 lbm). As such, choosing a variety of calibration weights that fall in this range is ideal. Note that because of the 50-70 mV offset, a reading of 227 kg will exceed the load cell's voltage output capabilities, but it will not damage the load cell.

Improvements:

For calibration purposes, a longer torque arm would prove helpful since the current calibration weight location is in a tight space within the dyno frame. Other dyno manufacturers have created a separate, longer torque arm that attaches onto the existing torque arm to be used solely in calibration purposes.

Temperature Sensors

Note that in this application, the temperature sensors do not need to be accurate to more than ± 5 °C because they are used to monitor system status and not for precise calculation. To verify calibration of temperature sensors, first record ambient temperature data and make sure it is within 25 °C ± 10 °C or close to the expected outside ambient temperature. to make sure the sensor is connected and is outputting values. Next, get a glass of ice water and place the temperature sensor inside of it (if epoxied into a fitting, place fitting in too). Verify that the temperature probe is reading is 0 °C ± 1 °C. If not, take note of the reading in ice water and adjust the manufacturer given calibration table to match the sensor's current reading at different values. Use an independent sensor such as a multimeter connected thermocouple or RTD in steady state conditions to verify accuracy.

I: Individual Component Analysis

Torsional Damper

Justin Moreno

April 7, 2017

MAE 156B

Component analysis

Since our project involves measuring the torque output of an internal combustion engine, the vibrations produced by the engine must be controlled to reduce the amount of noise measured by the load cell. A critical component to handle these vibrations is a torsional damper, which would be designed to reduce the vibrations transmitted to the measuring device.

Dampers essentially function as a low pass filter, so the cut off frequency must be less than the lowest vibrational frequency of the motor. The motor has a rotational speed of 3000 RPM at idle, which is equivalent to a 50Hz excitation frequency. As such, the damper must have a cutoff frequency lower than 50 Hz.

There are many types of off the shelf torsional dampers available for purchase, so at first, narrowing down the large range of options was somewhat daunting, however, with some research, the selection was narrowed down to three options: Jaw Type, Shear type, and torsional (donut) type.

The jaw type coupling (Fig. 1) uses an elastomeric spider with metal protruding jaws that compress the elastomer when under load. This type of coupling has been around since 1927, and is one of the most widely used couplings available, as well as low in cost. While they are not designed to damp vibrations, they can be used as such if sized properly.

The shear type coupling (Fig. 2) was invented in the 1950's, a few decades after the jaw type coupling, as elastomer technologies started to become more advanced. Shear type couplings, as their name suggests, transmit torque with the elastomer only under shear load, providing excellent damping characteristics, but have a lower size to capacity ratio.

Finally, the torsional coupling (Fig. 3) were designed to be used with diesel engines to mitigate vibrations along the output shaft, as such, it is commonly regarded as the best type of coupling for torsional service. It consists of a donut shaped elastomeric component with metal inserts that are fastened to rigid hubs with bolts. The metal inserts transfer toque through friction to the hubs, then through the elastomer to the opposing hub.

After narrowing the selection to these types of couplings, a more detailed look at the important parameters of the selection was taken to help decide on the type to be used. Table 1 shows a detailed comparison on the three types examined. After weighing these options, the torsional coupling seems to make the most sense.

After the type of coupling is decided, the critical stiffness of the coupling must be determined, so that it will properly filter the high frequency vibrations. This can be done by simplifying the system as a two-mass system and using the following equation

$$C_{crit} = (\frac{2\pi in}{60})^2 * \frac{J_d J_l}{J_d + J_l}$$

Where

- C_{crit} is the critical stiffness (lb-in/rad)
- i is the number of excitations per revolution, in this case, the number of cylinders in the engine, 4.
- J_d is the mass moment of inertia of the driving side of the coupler, assumed to be 170 lb-in-s^2 (Cossalter).
- J_l is the mass moment of inertia of the loading side of the coupler, assumed to be 30 lb-in-s^2, based on CAD model data of the water brake rotor.
- n is the rotational speed at idle, 3000 RPM.

In the case of this project, the critical stiffness can then be calculated as 40*10^6 in-lb/rad. The stiffness of the damper should then be chosen as at least half of the critical stiffness to avoid resonance issues. By looking through the Lovejoy catalog for torsional dampers, it is realized that nearly their entire selection satisfies this requirement, so the damper that best fits our system can easily be determined to according to our water brake shaft sizing.

Appendix

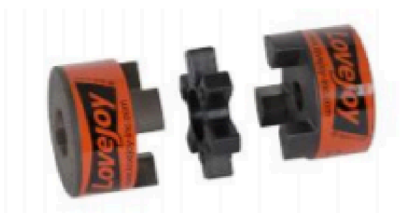


Figure 1. Jaw type coupling

Figure 2. Shear type coupling

Figure 3. Torsional type coupling

Coupling type			
Parameter	Jaw	Shear	Torsional
size to capacity ratio	good	fair	fair
max. misalignment	1.3 deg	2 deg	3 deg
speed capacity	good	fair	fair
damping capacity	good	excellent	excellent
cost	$	$$	$$

Figure 4. Comparison of couplings

References

- Cossalter, V. (2006). Motorcycle Dynamics (2nd ed.). Retrieved April 7, 2017.

- The Coupling Handbook. (2000). Retrieved April 7, 2017, from http://www.lovejoy-inc.com/thecouplinghandbook_part1.aspx

- Garrett (unknown last name). (2017, March 14). [Telephone interview]. Engineer at Power Test Dynamometers

Water Pump

Cassandra Moreno

MAE 156B

7 April, 2017

Individual Component Analysis

The pump system of the water brake dynamometer is an important component to analyze. The functional requirement of the pump is to pump water from a tank into the water brake by the use of a load control valve to provide the load to the water brake used to measure torque and RPM of the engine. It is necessary to purchase the correct pump to make sure that the adequate pressure and flow rate are possible to be supplied to the water brake.

The primary objective is to be able to test an engine with 85 HP and up to 120 HP. Therefore, the pump must be able to pressurize water to meet the respective load requirements. According to Stuska, a water brake manufacturer, the water brake must be supplied with a flow rate of 10 GPM (gallons/min) per 100 HP with a pressure of 45-60 psi. Also after consulting with our sponsor, we decided it was best to only have one pump and save money by making the water system more compact. With these considerations in mind, I drew a schematic of the dynamometer to see the various components that the pump would affect. As shown in Figure 1, the water will flow from a tank, through a radiator and pump and then to the water brake controlled by a load valve. This means that the pump would need to account for the pressure drop across the radiator. Next, I did fluid mechanics calculations to determine the head need from the pump to be able to load the water brake. The pump we need requires a flow rate of 15 GPM with a total head of 10.17 feet, with a 1" ID plastic flex hose.

I began my search for pumps, and by research from DYNOmite dynamometer sources I determined that I would need a centrifugal pump. I looked online at Water Pumps Direct and found a Red Lion portable transfer pump with capacity of 17 GPM for $155.00 with free shipping. This pump has a total suction head of 25 ft and total head lift of 147 feet. While I need only 7 feet of suction head and 3 feet of head lift for a pump to produce a flow rate of 15 GPM. The second pump I found was a Wayne self-priming cast iron water transfer pump from Northern Tool and Equipment with a maximum of 24 GPM and a total head of 100 feet. This pump is more expensive than the Red Lion, however, it is self-priming, which is a great feature to add less work to the operator of the dynamometer. The third pump I found was from Amazon, Arksen Stainless Booster Pump with 15.4 GPM and 25 foot suction head and 150 foot head lift for $79.99. This pump is the least expensive pump that meets the requirements, and it is self-priming.

In conclusion, the Arksen pump would be the best option because it is the cheapest option and meets the requirements, and we have a very tight budget. However, it is necessary to consider the option of needing more GPM of water depending on the water brake capacity, and we would therefore need the Wayne or Red Lion pump.

Figure 1: Dynamometer closed water system

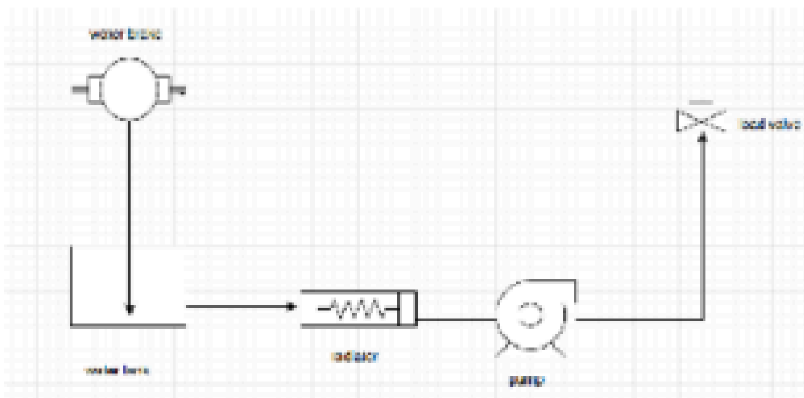


Figure 2: Types of Water Pumps

<u>Red Lion</u> <u>Wayne</u> <u>Arksen</u>

Pump comparison table:

Pump	Cost	Pros	Cons
Red Lion	$155.00	17 GPM, 2 year warranty	high price, not self-priming
Wayne	$169.99	self priming, durable cast iron, 24 GPM	most expensive- we have a very tight budget
Arksen	$79.99	self-priming, cheapest	15.3 GPM max, might not have enough GPM

References:

Red Lion pump: Water Pumps direct. waterpumpsdirect.com

Wayne pump: Northern Tool and Equipment. http://www.northerntool.com/shop/tools/product_7736_7736

Arksen pump: Amazon.

https://www.amazon.com/ARKSEN-Stainless-Shallow-Booster-Irrigation/dp/B00TQCSUZK/ref=sr_1_7?s=hi&ie=UTF8&qid=1491621857&sr=1-7&keywords=water+pump+self+priming

Keywords: Centrifugal pump, water transfer pump, water pump self-priming

For pump physics and calculations:

http://www.pumpfundamentals.com/download/calculate_press_anywhe.pdf

http://www.raeng.org.uk/publications/other/17-pumping-water

http://www.dynomitedynamometer.com/dyno-tech-talk/water_tanks.htm

http://www.stuskadyno.com/faq/

Load Cell

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MAE 156B | Section C00
April 7, 2017

Individual Component Analysis - Load Cell

Introduction

A water brake engine dynamometer (dyno) encompasses many sensors that are used to actively monitor the system and measure performance. These primarily include temperature, pressure, and torque sensors, and a tachometer and whatever else the operator of the dyno wants to monitor. The torque sensor is focused on in this report because the pressure sensor application is to still to be determined by the sponsor, the temperature sensors will be dependent on the closed water system parameters (also to be determined) and the tachometer, which will measure the RPM of the engine, is also dependent on a unresolved decision to either use direct or chain drive. The team knows the most about the parameters required for the torque sensor hence why it is discussed here. The torque sensor is a combination of a load cell and a known torque arm that is attached to the water brake as seen in Figure 1 in the Appendix. As the engine sweeps through its RPM range, a torque profile can be obtained by measuring the force exerted on the load cell by the water brake to keep the engine at its speed. Using the torque values, horsepower can be determined ($Hp = \frac{Torque * RPM}{5252}$) which is the critical parameter used for tuning an engine.

Functional Requirements of the Component

The first and most basic requirement of the load cell is that it needed to be work in compressive loading. The only sponsor-based requirements for the sensor comes from the overall objective of the dyno to measure to an accuracy of +/- 0.5 Hp and repeatability of +/- 0.1 Hp. In addition, previous dyno data used as a reference states that the max Hp of the current engine was 85, however this was with a restrictor in play. Therefore, the sponsors also want to consider the possibility of testing without a restrictor or with different engines so the maximum Hp requirement given to the team is 120. Using the torque and Hp curves from the dyno data (Figure 2 in Appendix) the max torque for 85 Hp was 60 ft*lbs. A ratio was taken of the increase in max Hp to calculate the increased max torque at the suggestion of the sponsor. 120/85 gives 1.411, which then multiplied by 60 ft*lbs gives 85 ft*lbs which is the max requirement. Additionally, by looking at similar engine dynos created by other universities and companies, the torque arm (though not designed) was assumed to be 1 ft. This gives a load cell requirement of 85 lbf or 378 N which gives 39 kg. Rounding 39 up to 50 kg, a more common design value, the final requirement was set. Some secondary requirements for the component would be a small design (since space requirements are tight for the enclosed system), simple mounting design to the frame and torque arm, and the load cell amplifier to work with the DAQ device (Arduino or Raspberry Pi).

Component Options

While strain gauges are the common load cell type used in industry, research was completed to see if other types (like hydraulic and pneumatic) could be applicable to the project. Using UCSD Library's Resource to the ENGnetBASE, load cell types were discussed and summarized in Figure 3 (in Appendix) taken from Chapter 7 in "Measurements and Safety". From the chapter, it was concluded that strain gauge type load cells seemed to be the best option for the application, and the validation came from the fact that existing dynos also use strain gauge load cells for similar accuracy and repeatability requirements.

Using the same resource from above, over 29 companies were identified as reliable suppliers of strain gauge transducers and some alternate suppliers were also considered. The options for the project application were either s-type or button-type strain gauge load cells. While researching other student projects[4][5] and talking specifically to Nick Kennard, another Mechanical Engineer who built his own water brake dyno, it was understood that an expensive load cell is not required especially since the are often built robustly. In addition, with a very tight budget, money needed to be reallocated to other parts of the system in order to ensure that the major mechanical components of the dyno would be built first. It will be easier to replace electronic components rather than mechanical parts and that is why the team decided that quality would be sacrificed on the electronic portions of the dyno.

Of the multiple options available, the decision was narrowed down to 3: SparkFun's TAS606, DTec's LCell-200, and Honeywell's 127-LF. The TAS606 is a button-type, while the other 2 are S-type. Base qualities these all had were that they could have a max capacity of at least 50 kg, could work with an Arduino or Raspberry Pi, and were compact in design. Honeywell was the recommended load-cell brand by the "Weight Detectors Load Cell" chapter, while DTec is one used specifically in dyno applications. SparkFun is the electronic distributor for DIY projects and is decently reliable as concluded by the team. The individual spec sheets for the 3 options can be found in the end of the Appendix. In addition, all 3 products would need an amplifier that would make the voltage readible and so they're recommended amplifier sub-component was considered as well. More research will need to be completed to determine the universality of some amplifiers for these products to minimize cost.

Summary Table and Results

Table 1 showing the pros and cons of the 3 products can be found in the Appendix. Several parameters were added to the initial requirements based on the load cell spec. sheets. In the end, the highest priority is given to cost and feasibility parameters such as load capacity, mounting, and microcontroller compatibility. Because electronics will come secondary in priority to the mechanical structure and water system itself, cost plays a bigger role than initially assumed. Mounting is an issue as well, but past dynos have used both button and s-type load cells with success and creating a working mount with either type will not be difficult. In addition, a secondary priority is to ensure that the dyno can eventually become an automatically controlled system, so starting the DAQ process now is also important and components need to work with the software the team decides to use.

Conclusion

As a result of this individual component analysis, the team will most likely use the Sparkfun TS606 due to the low cost, working capability, and room for error (if the product doesn't work well, the cost will not have set the team back significantly). There is also a wide resource available online from other users who have used the product which will help the team out when assembly and testing begins. However, the decision to buy a load cell is not an immediate one as it comes closer towards mid-quarter when the system will be in the manufacturing phase. And because load cells are not uncommonly used, the team will continue to ask around until mid-quarter for significant discounts or donations from companies and research labs for a s-type or button load cell just as other university dyno projects have successfully done in the past.

References

Report Resources

[1] "How to Calibrate the Load Cell on the Dyno". *Sport Devices.* Web. 5 Apr 2017 <http://www.sportdevices.com/dyno/load_cell.htm>

[2] "Weight Detectors Load Cell". *Instrument and Automation Engineers' Handbook Vol. 1: Measurement and Safety*. Sep 2016 , 1574 -1605. CRCnetBASE. Web. 3 Apr. 2017. <http://www.crcnetbase.com/doi/pdfplus/10.1201/9781315370330-135>

[3] Torque and Hp Curve Data for RPM Sweep for 2 FZ6R Engines. Provided by Triton Racing (Sponsor) and DynoShop.

[4] Duprey, Benjamin et al. "Redesign of Small Engine Dynamometer". *Rochester Institute of Technology.* May, 2004.

[5] Sanchez, Samuel. "Design, Reconstruction, and Evaluation of a Dynamometer for Quarter Scale". *California Polytechnic State University, San Luis Obispo.* 2009.

Spec Sheet Links

- Sparkfun TAS606: http://www.htc-sensor.com/products/151.html
- DTec LCELL-200: http://dtec.net.au/Downloads/Load%20Cell%20Data%20Sheet%20200kg.pdf
- Honeywell 127 LF: https://measurementsensors.honeywell.com/ProductDocuments/Load/Model_127_Datasheet.pdf

Contacts

1. Nick Kennard: Former mechanical engineering student who also created a Water Brake Dyno (Found on Youtube: https://www.youtube.com/watch?v=hmdZSjyUd8A)
2. Bill McCall: Application Engineering Specialist at Honeywell Sensing and Productivity Solutions. (Bill.McCall@honeywell.com | (614) 850-7826)
3. Futek: www.futek.com (Spoke to sales rep.; Only 5% rebate given for universities)
4. Sparkfun Electronics: www.sparkfun.com

Databases Searched

- Thomas Register: For load cell companies, then specifically those in California
    - Force Sensors, Load Cell, S-type, Button, California, 50 kg
- CRCnetBASE/ENGnetBASE: For load cell research
    - Keywords: Load Cell, S-type, Button, Comparison

Appendix

Figures

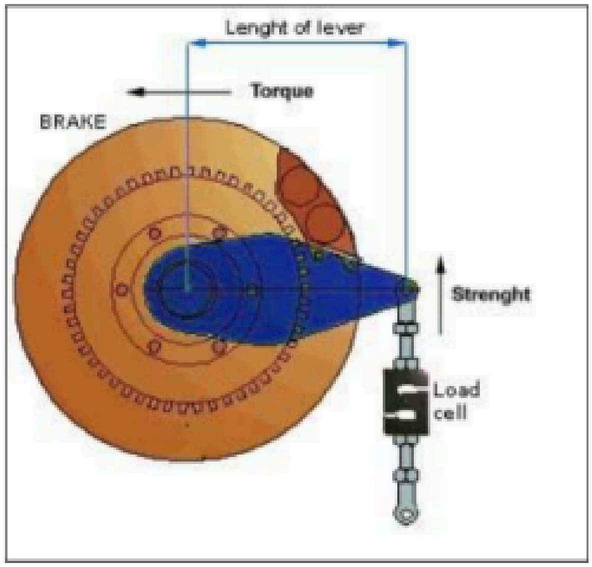


Figure 1: Diagram of Load Cell Application in Water Brake to Measure Torque[1]

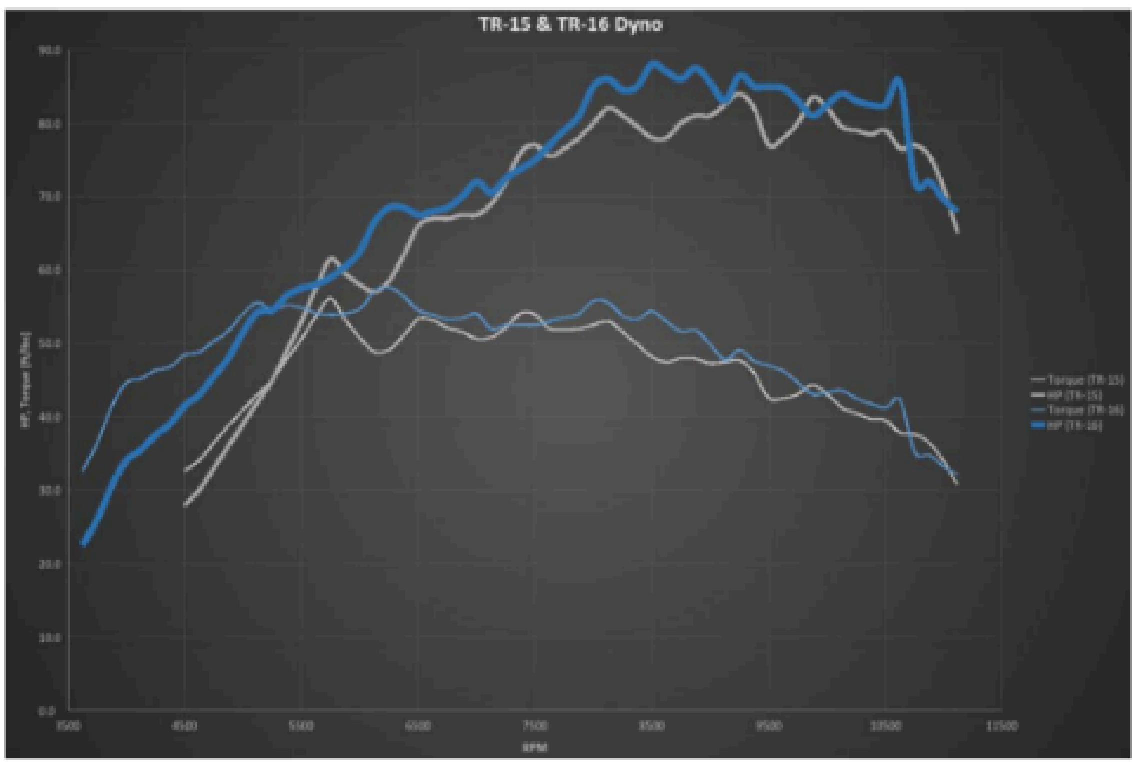


Figure 2:Dyno Data for 2 FZ6R Engines Tested At the DynoShop[3]

TABLE 7.19a
Load Cell Performance Comparison

Type of Load Cell	Weight Range*	Inaccuracy (% FS)	Applications	Advantages	Disadvantages
Mechanical cells					
Hydraulic	Up to 10,000,000 lb	0.25%	Tanks, bins, and hoppers; hazardous areas	Takes high impacts; insensitive to temperature	Expensive, complex
Pneumatic	Wide	High	Food industry; hazardous areas	Intrinsically safe; contains no fluids	Slow response; requires clean, dry air
Strain gauge cells					
Bending beam	10–5,000 lb	0.03%	Tanks; platform scales	Low cost, simple construction	Strain gauges are exposed, require protection
Shear beam	10–5,000 lb	0.03%	Tanks, platform scales, off-center loads	High side load rejection, better sealing and protection	
Canister	1000–500,000 lb	0.05%	Truck, tank, track, hopper scales	Handles load movements	No horizontal load protection
Ring and pancake	5–500,000 lb		Tanks, bins, scales	All stainless steel	No load movement allowed
Button and washer	0–50,000 lb, 0–200 lb typ.	1%	Small scales	Small, inexpensive	Loads must be centered; no load movement permitted
Other types					
Helical	0–40,000 lb	0.2%	Platform, forklift, wheel load, automotive seat weight	Handles off-axis loads, overloads, shocks	
Fiber-optic		0.1%	Electrical transmission cables; stud or bolt mounts	Immune to RFI/EMI and high temps; intrinsically safe	
Piezoresistive		0.03%		Extremely sensitive; high signal output level	High cost; nonlinear output

Figure 3: Load Cell Types in A Comparison Chart from "Weight Detectors Load Cell"[2]

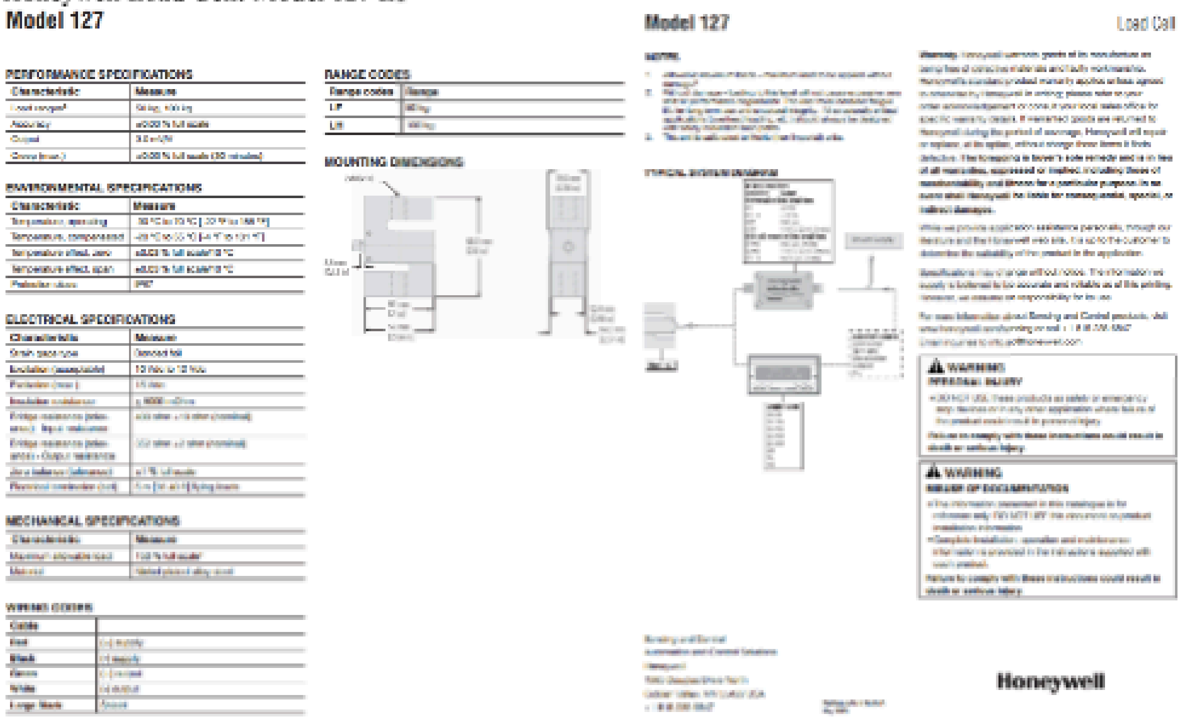
Table

Table 1: Pro/Con Comparison Chart of Load Cell Options

	Sparkfun TS606	DTec LCELL-200	Honeywell 127 LF
Pros	load capacity matches requirement; can work with 5V; works in compression; exceeds temp. range; low cost ($59); cheap amplifier known to work with it ($10)	larger than load capacity required; can work with 5V; works in compression; mount with 2 M12s; exceeds temp. range; works with dyno application; alloy steel material; mid price value ($98) + brand amplifier ($79)	load capacity matches; works in compression; Accuracy is better than required; exceeds temp. range; mouting ease with 2 M6 holes; 5 day lead time; nickel plated alloy steel means it's durable; reliable company
Cons	may need 10 V; accuracy not given; only can be mounted from the bottom with 3 M3 screws; Lead Time is 2 weeks; Stainless Steel Material	high load capacity (200kg) can cause accuracy and repeatability issues; Class C3 accuracy is Australian standard that can't be matched; total cost gives room for 1 quantity; unknown lead time (from Australia)	needs at least 10 V to work; price for part is very expensive ($296) and brand compatible amplifier too ($374); High cost gives room for 1 qty only and cuts into budget elsewhere

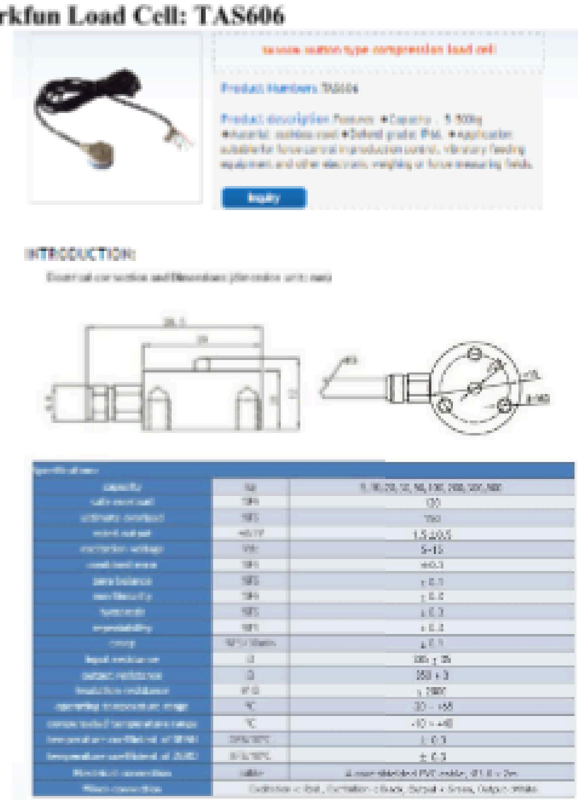
Spec Sheets

Honeywell Load Cell: Model 127 LF



DTec Load Cell: LCELL-200

Sparkfun Load Cell: TAS606



Frame

Luke Bockman

April 7, 2017

Component Report

Introduction

The sponsor, FSAE needs a water brake dynamometer for engine testing and tuning purposes. This particular engine under test is an Yamaha - FZ6R, 4 cylinder motorcycle engine with a mix of stock and non stock components. Torque and Horsepower produced by this model engine can be seen in Figure 6 in the appendix. Key points in Figure 6 are the max Torque produced during the testing of this engine as that moment and its reaction moment at the water brake will be used as representative loadings for the engine. The frame of this testing system is an integral part and options for it will be compared in this short writeup.

Functional Requirements of the component

Frame requirements for this application are high stiffness, reasonable mobility, and upgradability in the future. Frames for commercial dynamometers are commonly made from welded steel and are built for stiffness, upgradability, and longevity rather than mobility. They are almost always painted or powder coated to achieve a good durable finish.

Component Options

The frame in consideration will be built out of either welded steel or welded aluminum and possibly powder coated depending on the project sponsor's preferences and connections with local paint shops. Wheel or caster selection is the next important consideration in this design. Durable polyurethane casters are preferred as the likely terrain will be outdoor asphalt, concrete and must be able to handle small rocks and other debris.[5] Some industrial casters were compared at a local CNC machine shop, Mekilect LLC.[6] The driveline method used is another important component in this frame. It affects packaging, water brake loading and safety.

Conclusion

Steel as a material choice for the frame makes obvious sense when considering the functional requirements for this frame. Choosing the U-joints as the driveline method over the chain and sprocket system is also a good choice as it makes the design easier and increases safety. The Grizzly G8177 - 5" Heavy-Duty Swivel Caster w/ Brake are selling for an incredibly cheap price and seem to fit our application perfectly. They outperform the MCMaster Carr brand Creston Caster considerably and also at a lower price although they raise the systems overall CG and have thinner polyurethane material to wear on.

Appendix

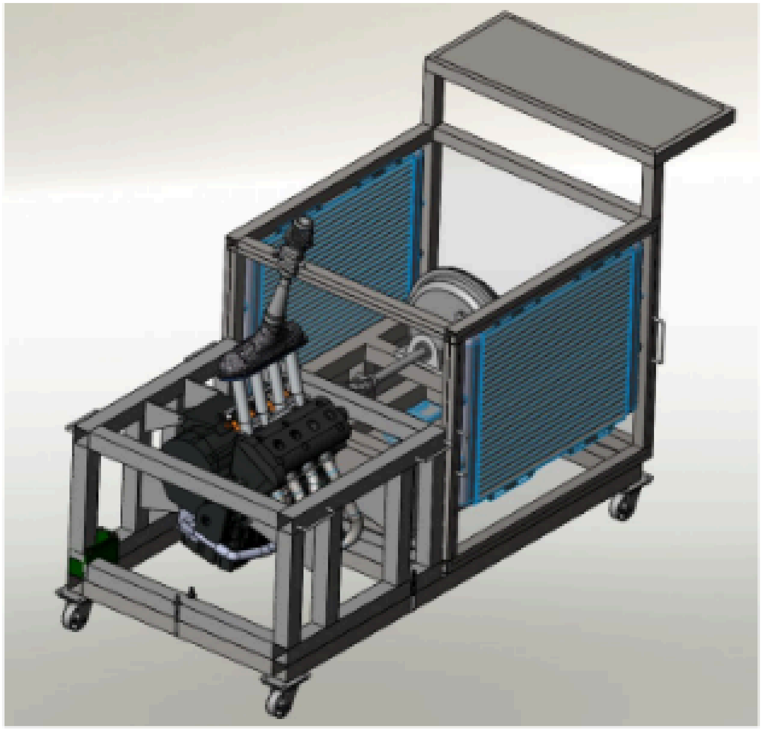


Figure 1:Iso View of Frame model showing dynamometer frame (model is not finished)

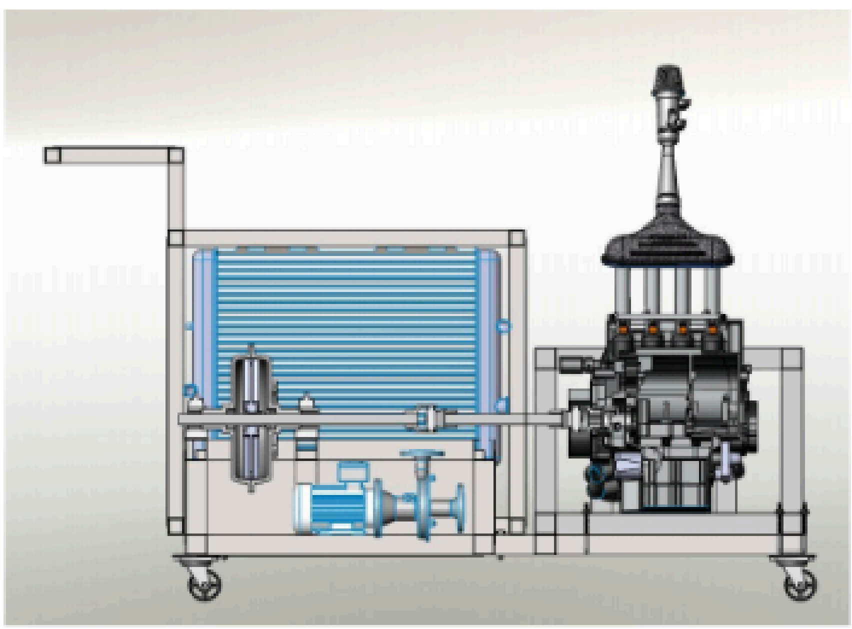


Figure 2: Section View of Frame model showing dynamometer frame (model is not finished)

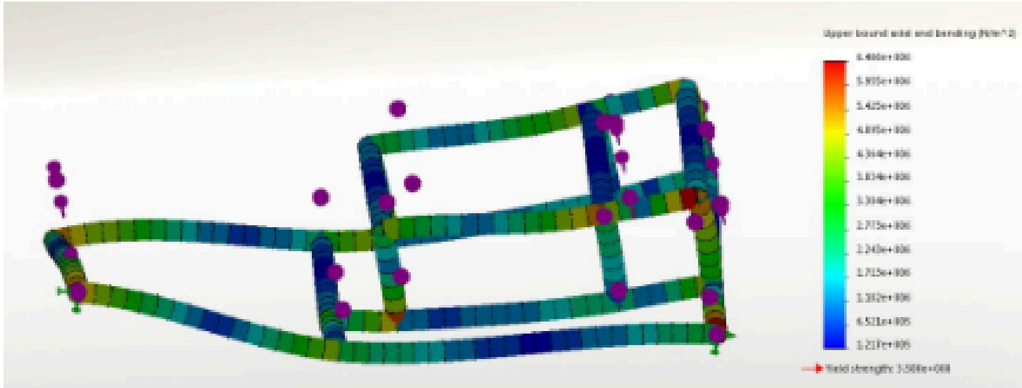


Figure 3: Frame Linear static analysis using 3d beam elements in solidworks solver

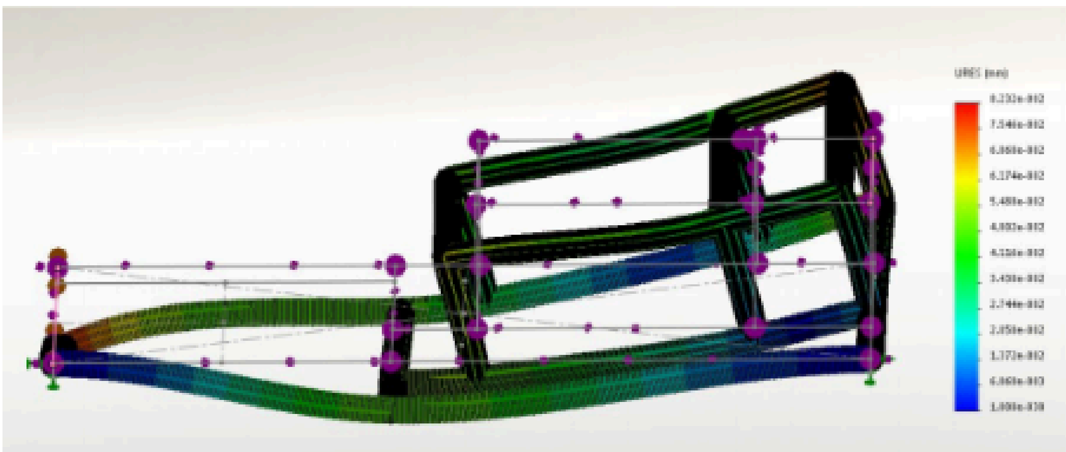


Figure 4: Same analysis as above but showing the original frame geometry and run with many more elements to verify the results have converged.

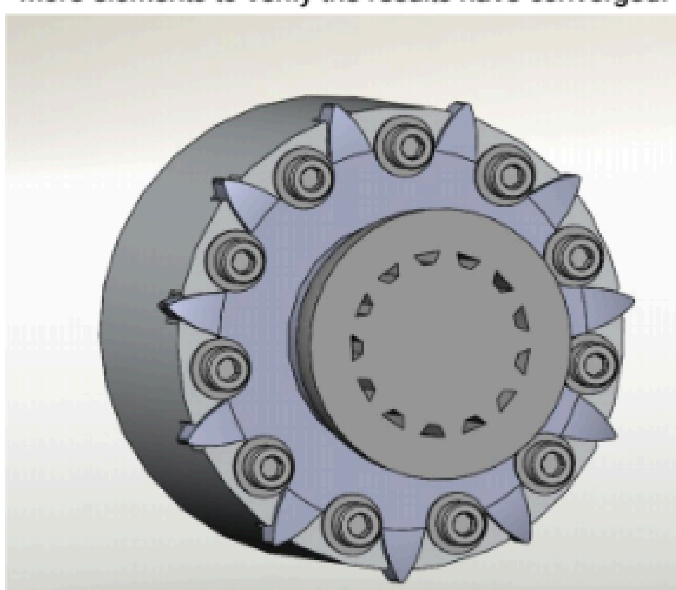


Figure 5: Coupling to the sprocket for the u-joint driveline method

Figure 6: FSAE Triton Racing previous dyno data (2015 and 2016 cars)

Pro/con tables

Material Option	Steel	Aluminum
Stiffness	E = 210 GPA [2]	E = 70 GPA (before welding)[3]
Equipment required	Can use a single phase supply for the welder	May need to use 3 phase supply line if welding 1/4" thick material
Welding skill required	Steel is relatively easy to weld[1]	Can be trick(but I have more experience with welding aluminum)
Weight of final frame	7850 kg/m^3 [2]	2712 kg/m^3 [3]
Cost	$1.5/lb [4]	$2/lb [4]
Effect on vibration	More mass = less vibration amplitude	Less mass = more vibration amplitude
Result	**winner**	**looser**

Driveline Method	U-joint w/ machined adapter	Chain and sprocket drive
Shock loading	Much less shock loading because of the reduced play in the U-joints	Shock loading whenever the chain becomes tensioned
cost	$100 + $25 + machining time and balancing time	$150 for chain and sprocket setup
packaging	Straight to Engine output sprocket	Slightly worse packaging
Safety	If driveline is broken, the axle will be kept	If chain is broken, it will be shot out with a whole lot of kinetic energy
Ease of use	Misposition of the subframe will be problematic. Fabrication needs tighter tolerances	Much easier and quicker to install. No power tools or sprocket change is necessary
Result	**winner**	**looser**

Wheel Type	Grizzly G8177 - 5" Heavy-Duty Swivel Caster w/ Brake	Creston Caster Swivel/Face Brake, 3" x 1-1/4" Urethane Wheel, 300 lb Capacity
Link to product website	http://www.grizzly.com/products/5-Heavy-Duty-Swivel-Caster-w-Brake/G8177	https://www.mcmaster.com/#9949t17/=173nmon
Load Capacity	600lbs ea	300lbs ea
cost	17.50	25.74
Mount Height	6.3"	4.312"
Wheel Material	PolyUrethane(thin)	Abrasion-Resistant Red Polyurethane on Polypropylene Wheels—Hard Tread (90A Durometer)
Brake	Yes, Face Brake	Yes, Face Brake
Result	**winner**	**looser**

References

[1] Collier, Jody "Tig Welding Aluminum" Accessed April 17th, 2017; http://www.weldingtipsandtricks.com/tig-welding-aluminum.html

[2] "AISI 1020 Steel, cold rolled" Accessed April 17th, 2017; http://www.matweb.com/search/datasheet.aspx?matguid=10b74ebc27344380ab16b1b69f1cffbb&ckck=1

[3] "Aluminum 6061-T6; 6061-T651" Accessed April 17th, 2017; http://www.matweb.com/search/DataSheet.aspx?MatGUID=b8d536e0b9b54bd7b69e4124d8f1d20a

[4] Industrial Metal Supply https://www.industrialmetalsupply.com

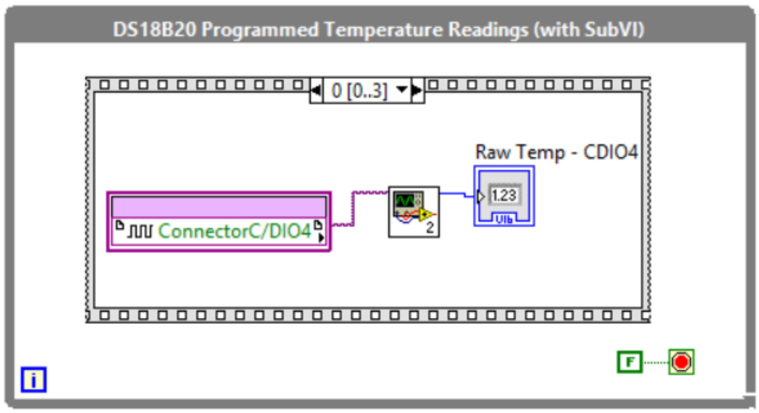
[5] Douglas Equiptment, "What to Consider When Purchasing Casters for Outdoors", http://www.douglasequipment.com/casters-1/what-to-consider-when-purchasing-casters-for-outdoors/

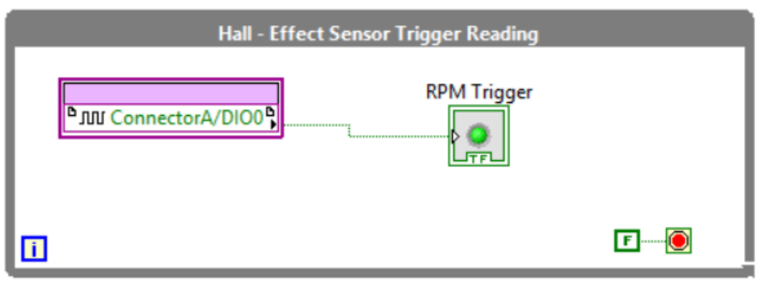
[6] Garret from power test

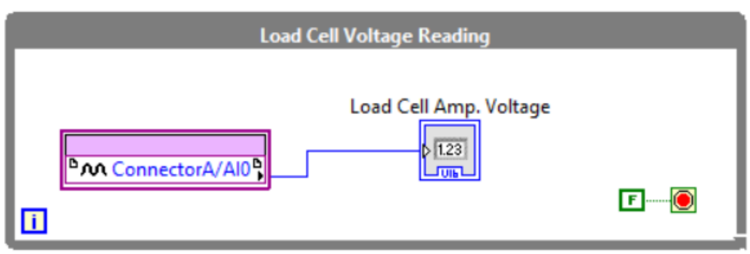
[7] Matt from Mekilect

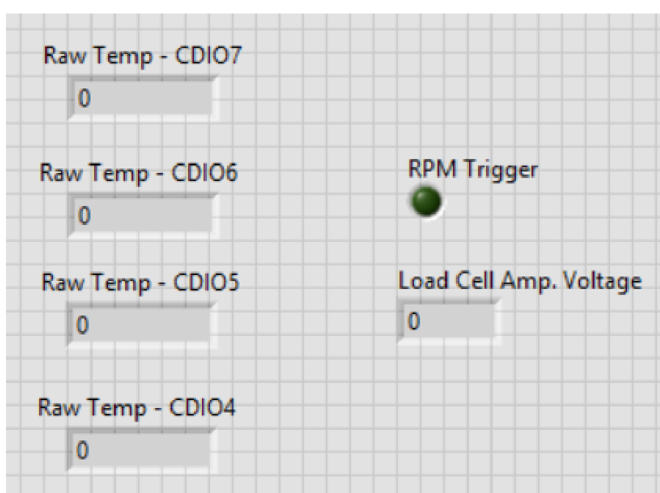
J: LabVIEW VI Diagrams

FPGA VI (Target VI) : “FPGA Multi Sensor.VI”

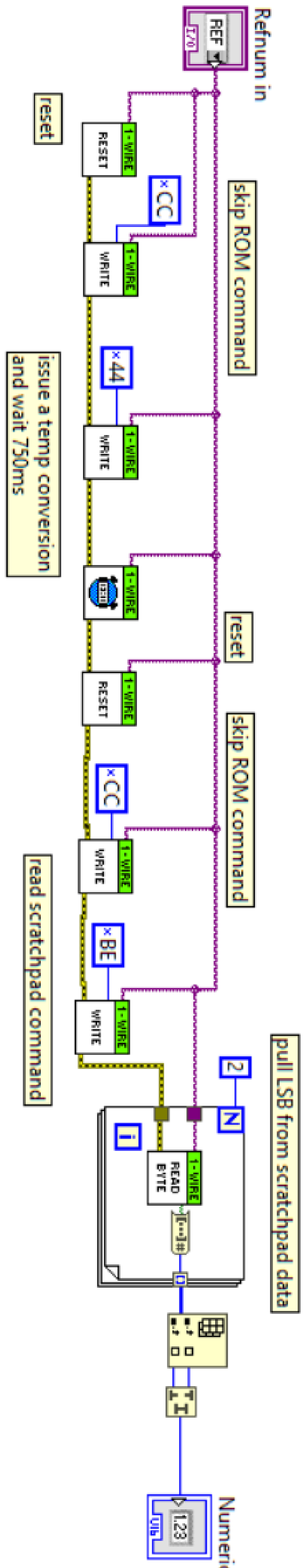






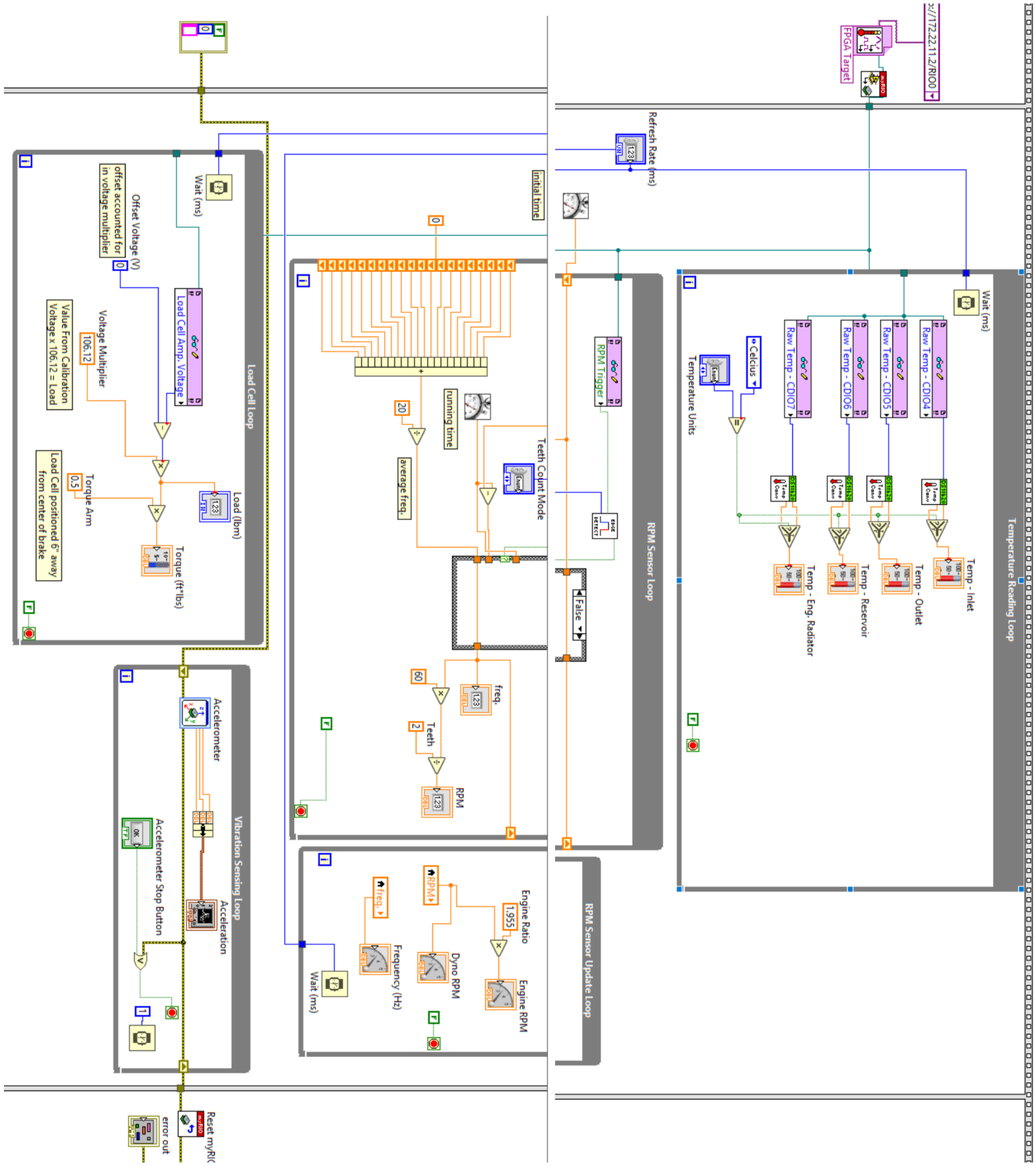


FPGA Temperature Reading (Target SubVI): “FGPA DB1820.VI”

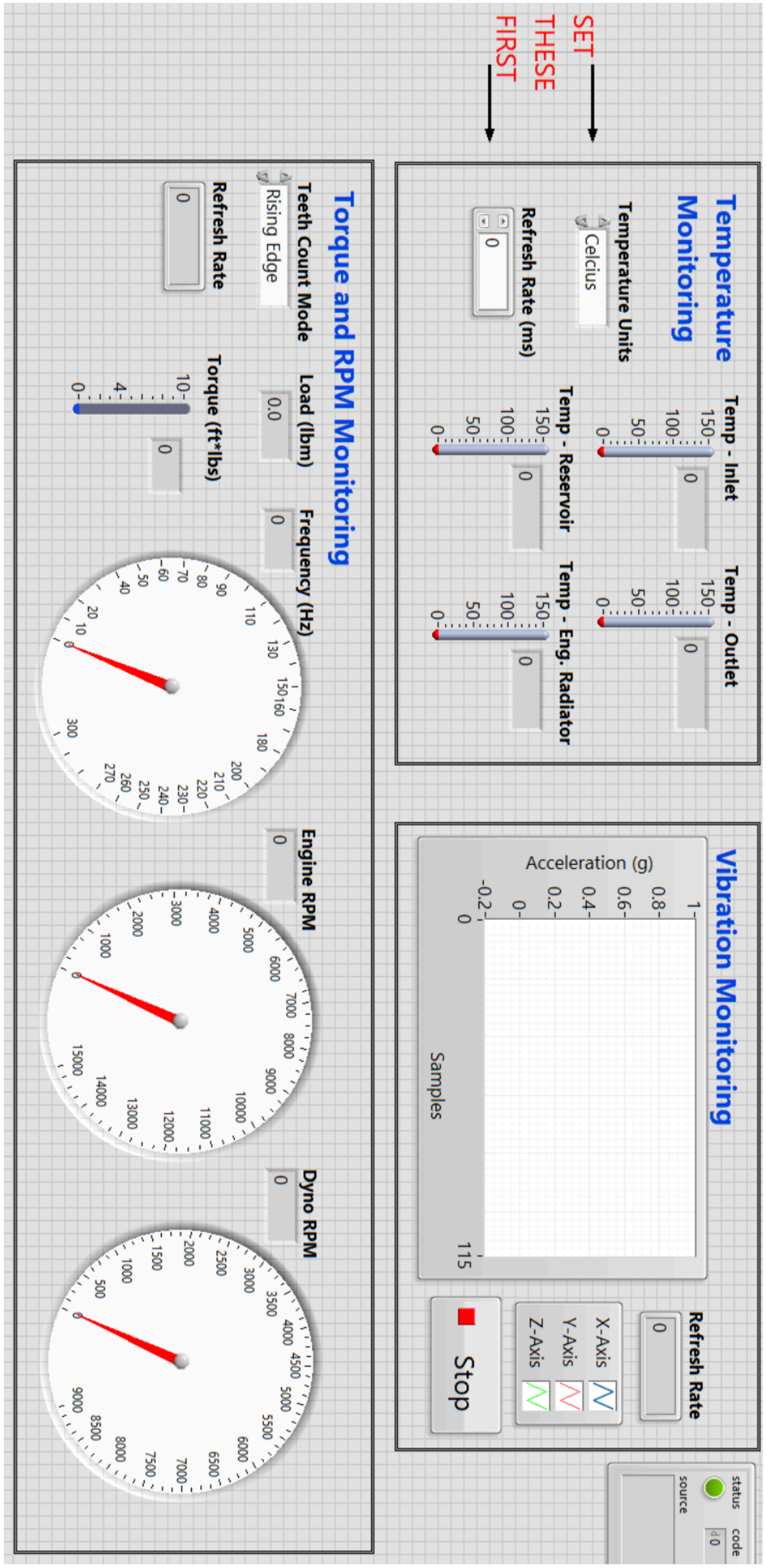


Dyno VI (Real-Time): “Dyno (RT).VI”

Block Diagram

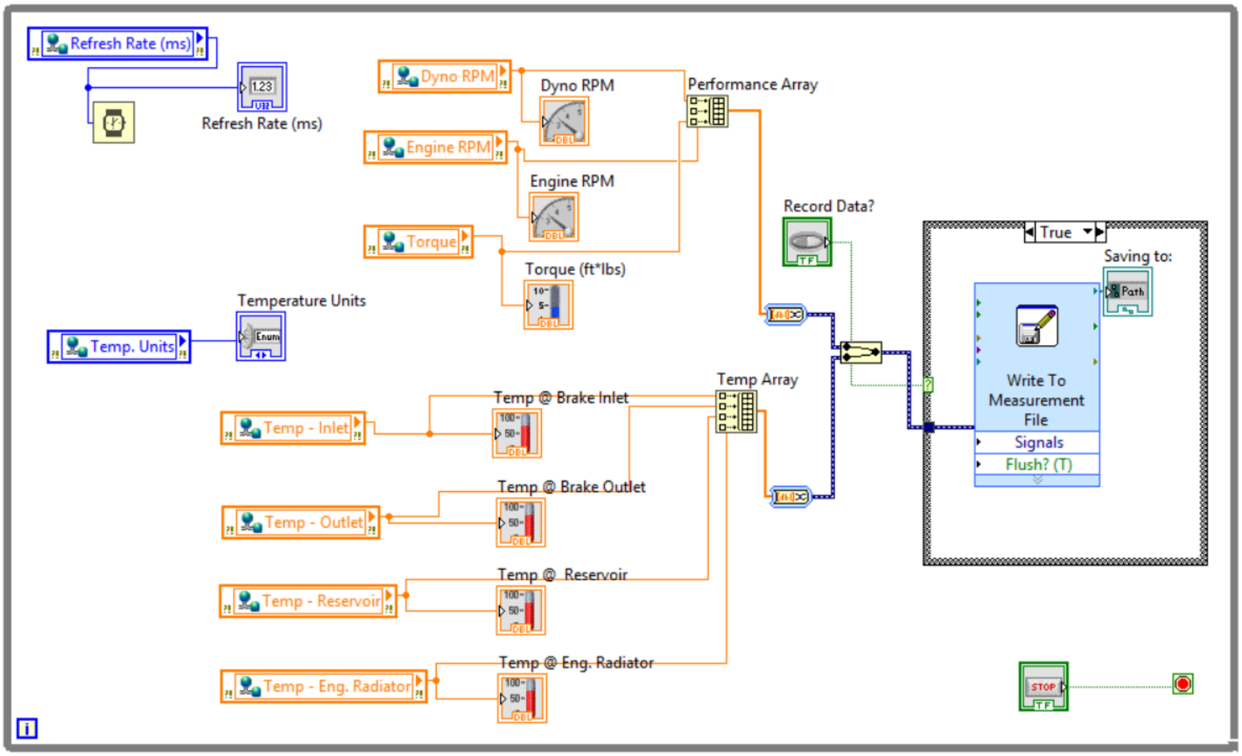


Front Panel


SET THESE FIRST
Temperature Monitoring
Temperature Units
Celcius
Refresh Rate (ms)
Temp - Inlet
Temp - Outlet
Temp - Reservoir
Temp - Eng. Radiator
Vibration Monitoring
Acceleration (g)
Samples
Refresh Rate
X-Axis
Y-Axis
Z-Axis
Stop
status
code
source
Torque and RPM Monitoring
Teeth Count Mode
Rising Edge
Refresh Rate
Load (lbm)
Torque (ft*lbs)
Frequency (Hz)
Engine RPM
Dyno RPM

Dyno Data Collection VI (Computer/Network VI)

Front Panel is Figure 38. Block Diagram:



For viewing all LabVIEW files, reference the “myRIO Project“ zip folder located on the team website.

K: Meeting Minutes (ATTN: Dr. Silberman)

Example 1: Sponsor Meeting - Saturday Week 6, 5/13/17

Department of Mechanical and Aerospace Engineering (MAE)
MAE156B

FSAE Engine Dyno – Sponsor Meeting Minutes

Spring Quarter, Saturday Week 6 – 5/13/17

Contact Information
Name: Chinaar Desai
e-mail: chinaar.desai@gmail.com

Agenda used on 05/13/2017

- Review agenda for the day.
  - Item 1: Current Progress
  - Item 2. Engine Drop-In
  - Item 3. Battery Purchase
  - Item 4. Additional questions/concerns

Contact Information

Contact	Dep.	Title	Phone	Email
Chinaar Desai	MAE	156B Member	818-470-0942	chinaar.desai@gmail.com
Cassandra Moreno	MAE	156B Member	562-619-8341	clmoreno1717@gmail.com
Justin Moreno	MAE	156B Member	951-595-5142	jlmoreno@ucsd.edu
Luke Bockman	MAE	156B Member	760-712-6665	lpbbockman@gmail.com

Rob Shanahan	Triton Racing	Project Sponsor	858-349-9431	r_shanahan@msn.com
Daniel Morris	Triton Racing	Triton Racing Liaison	916-207-7809	dkmorris@ucsd.edu
Dr. Jerry Tustaniwskyj	MAE	Professor		jtustaniwskyj@ucsd.edu
Dr. Jack Silberman	MAE	Professor		jacks@eng.ucsd.edu
Pedro Franco	MAE	TA		pfrancon@ucsd.edu

Future Meeting Information

Date	Time	Location
05/20/17	1:00-2:00	EBU2 Basement B35

Required Attendees	Dep.	W5	W6	W7
Chinaar Desai	MAE	P	P	
Cassandra Moreno	MAE	P	P	
Justin Moreno	MAE	P	P	
Luke Bockman	MAE	P	P	
Rob Shanahan	Triton Racing	P	P	
Daniel Morris	Triton Racing	P	P	

P = Present, E = Excused, A = Absent

Meeting Minutes

Review agenda for the day.
Take note of any change.

Item 1. Current Progress

- A lot of manufacturing of components under way
- Luke: Lead on frame components
- Justin: Lead on water brake components
- Chinaar and Cassandra supporting Luke

AR Get Mfg Help from Triton Racing – Daniel Morris to ask Triton Racing to help team out with manufacturing smaller items, 5/19/17

Item 2. Engine Drop-In

- Daniel filled in
- About ready to go
- Stock exhaust on its way
- Stock sensors in wiring harness should be good to go

AR Get Engine Dropped In Today – Chinaar to check in with Brad A. (Triton Racing) to get the engine dropped in by today, 5/13

Item 3. Battery Purchase

- On Purchasing List
- Look at Odyssey battery or Battery Mart for a free one
- Battery Needs: 12V, 10 AH, CCA 160; for motorcycles; lead acid

AR Look for Battery Donation – Chinaar to ask for sponsorship from Odyssey and Battery Mart for a free battery, otherwise we buy the battery ourselves

Item 4. Additional questions/concerns

- Driveshaft: We got one from Rob (sponsor) but it'll take time to setup. Luke to work on this.
- Powder coating: **Daniel (AR)** Talk to Action Powdercoating to get estimate for turn-around time for the frame to get coated
- Anodizing water brake: Get system anodized early. Drop off on Monday as well. **AR Daniel**
- Check with the Professors about running the dyno in the project room
- Fuel tank: consider getting sponsors for fuel cells (WOW solution); Worry about the fire hazard since it's a major safety concern **AR LUKE**

Agenda for next meeting

- Review agenda for the day
- Item 1: progress update
- Item 2:

Action Required List (AR)

Assigned	Date Due	Owner	AR/Status
W6. Sat.	W7.Fri	Daniel	**AR Get Mfg Help from Triton Racing / OPEN**
W6.Sat.	W6.Sat	Chinaar	**AR Get Engine Dropped In Today / OPEN**
W6. Sat.	W8.Wed.	Chinaar	**AR Look for Battery Donation / OPEN**
W6. Sat.	W7.Mon	Daniel	**AR Check for Anodizing and Powdercoating Sponsorship Opp. / OPEN**
W6.Sat.	W7.Sat	Luke	**AR Check for Fuel Cell Sponsorship / OPEN**

Completed AR's

Assigned	Date Due	Owner	AR/Status
W5.Sat.	W6.Sat.	Luke	**AR – Get Bearings (for free) / CLOSED;** we had to buy them instead
W5.Sat	W6 Mon.	Chinaar	**AR – RPM Sensor Finalized / CLOSED;** Hall Effect Sensor purchase

Example 2: Sponsor Meeting - Saturday Week 7, 5/20/17

Department of Mechanical and Aerospace Engineering (MAE)
MAE156B

FSAE Engine Dyno – Sponsor Meeting Minutes

Spring Quarter, Saturday Week 7 – 5/20/17

Contact Information

Name: Chinaar Desai
e-mail: chinaar.desai@gmail.com

Agenda used on 05/20/2017

- Review agenda for the day.
  - Item 1: Current Progress
  - Item 2. LabVIEW and Sensors
  - Item 3. What Needs to Be Made
  - Item 4. Additional questions/concerns

Contact Information

Contact	**Dep.**	**Title**	**Phone**	**Email**
Chinaar Desai	MAE	156B Member	818-470-0942	chinaar.desai@gmail.com
Cassandra Moreno	MAE	156B Member	562-619-8341	clmoreno1717@gmail.com
Justin Moreno	MAE	156B Member	951-595-5142	jlmoreno@ucsd.edu
Luke Bockman	MAE	156B Member	760-712-6665	lpbbockman@gmail.com

Rob Shanahan	Triton Racing	Project Sponsor	858-349-9431	r_shanahan@msn.com
Daniel Morris	Triton Racing	Triton Racing Liaison	916-207-7809	dkmorris@ucsd.edu
Dr. Jerry Tustaniwskyj	MAE	Professor		jtustaniwskyj@ucsd.edu
Dr. Jack Silberman	MAE	Professor		jacks@eng.ucsd.edu
Pedro Franco	MAE	TA		pfrancon@ucsd.edu

Future Meeting Information

Date	Time	Location
05/27/17	1:00-2:00	EBU2 Basement B35

Required Attendees	Dep.	W5	W6	W7
Chinaar Desai	MAE	P	P	P
Cassandra Moreno	MAE	P	P	P
Justin Moreno	MAE	P	P	P
Luke Bockman	MAE	P	P	P
Rob Shanahan	Triton Racing	P	P	E
Daniel Morris	Triton Racing	P	P	P

P = Present, E = Excused, A = Absent

Meeting Minutes

Review agenda for the day.
Take note of any change.

Item 1. Current Progress

- Anodized parts to be done by Tuesday
- Powder Coating complete by next Friday
- Redesigned fittings for radiators so the parts are ready to be manufactured
- Waiting to get bearings still

AR Call Anodizing – Daniel Morris to ask check in with Anocote on Tuesday, 5/23
AR Call Powder Coating - Chinaar to call on Wednesday to check Friday pickup, 5/24
AR Call VXB - Justin to call on Monday to check on delivery status, 5/22

Item 2. LabVIEW and Sensors

- RPM still needs to be fixed due to large amount of noise
- Temperature sensors need to be programmed on FPGA; Chinaar awaiting to hear back from NI representative about getting FPGA software
- Tabulated data file, if not directly to MATLAB
- Want rpm and torque graph to calculate power
- Max power and max rpm indicator ideal

AR LabVIEW FPGA – Chinaar to continue communication with NI rep (Ingo)

Item 3. Manufacturing List

- A bunch of items: trunnions, flywheel, stators (pending anodizing), rotor (pending anodizing), shaft, electronics holder (3d print?), pump mount, reservoir cover plate, sheet metal brackets, bearing blocks and caps, load cell arms, u-joint coupler, rod ends for engine...
- Luke needs to update manufacturing list on Drive
- Keep asking for help from Triton Racing

AR Update Manufacturing List- Luke Bockman to update Drive file and then assign manufacturing items to each member, and outsource other smaller parts to the team, 5/21

Item 4. Additional questions/concerns

- Check in on fuel cell sponsorship

Agenda for next meeting

- Review agenda for the day
- Item 1: progress update
- Item 2:

Action Required List (AR)

Assigned	Date Due	Owner	AR/Status
W6. Sat.	W8.Fri	Daniel	**AR Get Mfg Help from Triton Racing / OPEN AND EXTENDED**
W6. Sat.	W8.Wed.	Chinaar	**AR Look for Battery Donation / OPEN**
W6.Sat.	W8.Sat	Luke	**AR Check for Fuel Cell Sponsorship / OPEN AND EXTENDED**
W7.Sat	W8.Tues.	Daniel	**AR Call Anodizing / OPEN**
W7.Sat	W8.Wed	Chinaar	**AR Call Powder Coating / OPEN**
W7.Sat	W8.Mon	Justin	**AR Call VXB / OPEN**
W7.Sat	W8.Sat	Chinaar	**AR LabVIEW FPGA / OPEN**
W7.Sat	W7.Sun	Luke	**AR Update Manufacturing List / OPEN**

Completed AR's

Assigned	Date Due	Owner	AR/Status
W5.Sat.	W6.Sat.	Luke	**AR – Get Bearings (for free) / CLOSED;** we had to buy them instead
W5.Sat	W6 Mon.	Chinaar	**AR – RPM Sensor Finalized / CLOSED;** Hall Effect Sensor purchase
W6.Sat.	W6.Sat	Chinaar	**AR Get Engine Dropped In Today / CLOSED**
W6. Sat.	W7.Mon	Daniel	**AR Check for Anodizing and Powdercoating Sponsorship Opp. / CLOSED**

Example 3: Professors Meeting - Monday Week 3, 4/17/17

Department of Mechanical and Aerospace Engineering (MAE)
MAE156B

FSAE Engine Dyno – Professors Meeting Minutes

Spring Quarter, Monday Week 3 - 4/17/17

Contact Information

Name: Chinaar Desai
e-mail: chinaar.desai@gmail.com

Agenda used on 4/17/17

- Review agenda for the day.
  - Item 1: Current Progress
  - Item 2. Scope and Assignments
  - Item 3. LabVIEW
  - Item 4. Additional questions/concerns

Contact Information

Contact	Dep.	Title	Phone	Email
Chinaar Desai	MAE	156B Member	818-470-0942	chinaar.desai@gmail.com
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Justin Moreno	MAE	156B Member	951-595-5142	jlmoreno@ucsd.edu
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Daniel Morris	Triton Racing	Triton Racing Liaison	916-207-7809	dkmorris@ucsd.edu
Dr. Jerry Tustaniwskyj	MAE	Professor		jtustaniwskyj@ucsd.edu
Dr. Jack Silberman	MAE	Professor		jacks@eng.ucsd.edu
Pedro Franco	MAE	TA		pfrancon@ucsd.edu

Future Meeting Information

Date	Time	Location
05/20/17	1:00-2:00	EBU2 Basement B35

Required Attendees	Dep.	W5	W6	W7
Chinaar Desai	MAE	P	P	P
Cassandra Moreno	MAE	P	P	P
Justin Moreno	MAE	P	P	P
Luke Bockman	MAE	P	P	P
Rob Shanahan	Triton Racing	P	P	E
Daniel Morris	Triton Racing	P	P	P

P = Present, E = Excused, A = Absent

Meeting Minutes

Review agenda for the day.

Take note of any change.

Item 1. Current Progress

- Completed Frame FEA, only 4mm deflection
- Working on design of system
- Organizing purchase list

Item 2. Scope and Assignments

- Reduce scope (suggested by profs): define exactly what the team will do
  - Don't need to make a clean cut for the scope
- Divide group members into machining and instrumentation and pump system to breakdown work load
  - Luke and Justin: leads on manufacturing
  - Cassandra: pump and water system
  - Chinaar: instrumentation and DAQ

AR Reduce and Redefine Scope – Entire team to reconsider scope, 4/19

Item 3. LabVIEW

- Use hall-effect sensor for RPM on coupler
- Load cell will need amplifier; consult with greg on existing 170 instrumentation
- Use LabVIEW and NI DAQ to get data (work with Greg to get a myRIO or something)
  - DAQ can control voltage going into sensors too
  - Maybe get 2 DAQs (one for the Triton Racing team to keep)

AR Talk to Greg about LabVIEW - Chinaar to contact Greg about possibilities for DAQ, 4/21

Item 4. Additional questions/concerns

- Review gantt chart, and reflect changes in scope
- Conduct some safety analysis: radiator leak, splash protection, etc.
- Consider a dramatic failure case: the water brake doesn't work
  - We'll need to machine it wider if necessary

Agenda for next meeting

- Review agenda for the day
- Item 1: progress update
- Item 2:

Action Required List (AR)

Assigned	Date Due	Owner	AR/Status
W3.Mon	W3.Wed	Team	**AR Reduce and Redefine Scope / OPEN**
W3.Mon	W3.Fri	Chinaar	**AR Talk to Greg about LabVIEW / OPEN**

Completed AR's

Assigned	Date Due	Owner	AR/Status

L. Errata

Cover Page	Updated photo of final design
p.2	Updated Table of Contents to reflect accurate page numbers.
p.9	Adjusted alignment of Figure 2 caption.
p.12	Changed figure number from X to 5.
p.23	Modified first paragraph of rotor section to add equation reference, and elaborate on calculations.
p.24	Added figure reference for Figures 19 and 20.
p.25	Added figure reference for Figure 21.
p.26	Added figure reference for Figures 22.
p.27	Modified first sentence in Final Design Choice section. Also fixed unit typo for core size.
p.29	Changed wording of second-to-last paragraph to eliminate “cheap” and replace with “lowest cost”.
p.31	Introduction paragraph was removed as it was deemed not necessary and a duplication of information in the report. Also made small changes to functional requirements section.
p.32	Added citation to Figure 26: myRIO Data Acquisition System.
p.36	Changed wording of second-to-last paragraph to eliminate “cheapest” and replace with “inexpensive”.
p.37	Changed wording of last paragraph to eliminate “cheapest” and replace with “least expensive”.
p.39	Clarified statement of input vs output assumptions in the thermodynamic analysis.
p.44	Updated wording on Table 3 to better describe what was measured in the process. Replaced “weight” with “load”.
p.44	Replaced Figure 35 with an updated figure with text enlarged for readability.
p.47	Replaced Figure 37 with a more readable document scan.
p.47	Replaced Figure 38 with updated user interface after minor changes to

	LabVIEW VI
p.48	Added more details to “Final Test: Mock Setup” to better describe the test that was conducted on 6/14
p.49	Entered content for “Results” section to reflect the Mock Test results. Added Table 5 which reflects results of test.
p.57	Changed reference of Bill Schlossnagel from “Bill” to “Schlossnagel”. Also changed first person speech to third person speech.
p.60	Updated description for Appendix B: Drawings/Layout to reflect separate folder to access all 42 drawings
p.119 & 120	Added in Front Panel figure for Dyno (RT) VI, and Block Diagram for Dyno Data Collection VI.