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Baja Drive Line Test Development

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Abstract - This paper discusses the development and setup of a test facility to model the performance of the Briggs and Stratton Model 20 engine, used in the SAE Baja competition. The test platform is integrated with sensors to measure various engine parameters including horsepower, torque, RPM, an assortment of temperatures, intake air flow rate, fuel flow rate, cylinder pressure, and air/fuel ratio measurements. The system employs a DC motor as the dynamometer, a dynamometer controller to provide controlled load to the engine, sensors, and data acquisition hardware. All measured parameters were within range of expected results and theoretical estimates. A repeatability and reliability assessment of the system was not conducted due to time constraints and limited availability of the test facility; however all sensors were tested and validated. Certain physical design requirements included the ability to adapt the engine to the DC motor via a drive system, ability to tilt the engine, robustness, and safety. All design requirements were met. Limitations in the drive system observed during testing were partly resolved through the design and development phases.

INTRODUCTION

The SAE Baja competition mandates the use of the same engine by all participants, a Briggs and Stratton Model 20, henceforth referred to as the engine. As a competition requirement, the engine cannot be modified and must consist of all factory components. Participants are allowed to evaluate engine performance and replace components.

This project focused on the development and setup of a test facility for the RIT Baja team. Engine parameters of interest for the test system included: RPM, torque, horsepower, various temperatures, air/fuel ratio, intake air flow rate, fuel flow rate, and cylinder pressure. Ability to tilt the engine was incorporated in the design of the test stand to evaluate performance at varying angles. A Siemen's DC motor and dynamometer controller were used as the dynamometer, henceforth referred to as the dyno. This dyno and the attached mounting table, designed in MSD project P06112, were adapted for this project.

STAND DESIGN METHODOLOGY

Concept Selection

An engine test stand, adaptable to the dyno mounting table, was designed. Criteria for the test stand included: ease of mount and dismount, ability to tilt the engine, and a rigid platform able to withstand potential vibration.

Several stand concepts were evaluated using a Pugh's matrix. The design shown in Fig. 1, was chosen due to its rigidity and durability.

Figure 1. Engine stand

The stand is designed such that the engine can be tilted to +/-30˚. When mounted, the design of the rotational arc allows the stand's centerline to align with that of the engine shaft. This greatly simplifies the drive system by having a constant center distance from sprocket to sprocket.

The engine is bolted to the cradle of the stand which consists of a plate with rockers and mounting brackets. The rockers are integrated with a support arm to reduce the moment produced by shifting the center of gravity as the engine is rotated. This ensures stability, as well as, reduces stress in the bolted connections while at tilted positions. In addition, mounting brackets used as locking mechanisms for the cradle are bolted to the upright sides at the desired angular adjustment.

The stand was bolted to the dyno mounting table utilizing an existing hole pattern. Runners were attached to the stand that included this hole pattern in the form of slots. This provided a method of chain tensioning. These and all other parts of the stand were constructed of 1018 Cold Rolled Steel plate. Bolted connections were chosen as they allowed for simple modifications compared to welded connections. Furthermore, provisions for mounting an optical encoder, needed for pressure measurements, were made by attaching a bracket to the top of the stand.

Modeling and Analysis

The modeling and vibration analysis of the stand were carried out in SolidWorks. Finite Element Analysis (FEA) for the design was performed in ANSYS 9.0. Engine torque and weight at the different tilt angles were modeled for the loading scenarios on the stand as shown in Fig. 2. In ANSYS, the bottom of the stand was modeled as a *fixed* connection. Frictional and bolted connections were modeled as *no separation* and as *bonded,* respectively. Further analysis was carried out on the effect of torque on the stand runner. Simulations and hand-calculated values of the above were compared and found to be within range. Both produced acceptable factors of safety in excess of five.

Figure 2. FEA loading scenario of the stand

The engine has a maximum frequency of 75 Hz, corresponding to expected peak engine speed of 4500 RPM. Vibration analysis showed the lowest of the first five modes of the stand to have a value of 375 Hz. This implies that the stand will not reach a natural frequency during normal use.

Manufacturing

Parts of the stand were manufactured using a Computer Numerical Controlled (CNC) mill and programming was performed in MasterCam software. All parts were manufactured with 0.005 inch tolerance to the specified dimensions. Secondary operations were performed on the rockers and mounting brackets to drill and thread mounting holes.

Safety and Durability

To meet safety requirements, several modifications were made to the initial stand design. The rocker was extended to minimize stress and provide support for the mounting bolts. The mounting bolts were also increased in size to 5/16 inch. The larger bolts allowed for more torque to be applied and thus more clamping force. Furthermore, all bolted connections were made using fine threaded bolts as the larger minor diameter provides slightly more strength [1].

Single-cylinder internal combustion engines, such as the test engine, are highly prone to vibration. To address this, the final stand design incorporated several features to dampen vibrations. The mounting brackets were designed with provisions to allow small pockets for rubber strips. Additionally, the plate of the stand was designed such that the corners could be cut down slightly and rubber bumpers installed, if necessary.

A handle was made to aid in titling the engine. The handle, which attaches to the engine, reduces the possibility of being burnt from hot surfaces when trying to adjust the stand.

Stand Results & Discussion:

Upon assembly of the stand, it was observed that some bolted connections did not align properly. To resolve this, the through holes were slightly oversized by 1/32 inch and the slots in the mounting brackets were re-machined to a larger width. In hindsight, it would have been better practice to design it with more tolerance. Additionally, dowel pins could have been used where location was critical. Overall, the stand met its design goals.

DRIVE SYSTEM METHODOLOGY

The primary function of the drive system is to connect the DC Motor output shaft with that of the engine. Several considerations were made in selecting a drive system. First, the drive ratio had to facilitate the operation of both the dyno and the engine within their dynamic ranges. Second, the drive system had to be adaptable to the titling ability of the engine. The latter was primarily handled during the stand design. Third, the final design should allow for quick installation and removal of the drive system. Safety and affordability were also paramount.

These parameters led the team to several concepts. The top three included modifying the existing chain drive architecture, upgrading to a new chain drive system, or designing a new timing belt system. While timing belt systems can outperform chain drive, the latter was chosen due to its lower cost. A preexisting 525 chain architecture was selected as it offers a fast lead time, low cost, and a simple bolt-on operation for quick setup.

Drive System Setup:

A 66 tooth aluminum sprocket with 5/8 inch pitch was custom ordered. The sprocket was designed to the dimensions and 5 bolt pattern of the existing dyno mounting hub. A 17 tooth steel sprocket was used on the engine output shaft resulting in a 3.88:1 drive ratio. According to the American Chain Association, the current setup allows for a 3.23 HP rating [2], which is below the anticipated horsepower of the engine. It is also stated that "substantial increases in rated speed loads can be utilized, as when a service life of less than 15,000 hours is satisfactory, or when full load operation is encountered only during a portion of the required service life"[2]. Applications of 525 chain and sprocket sizing on motorcycles were also investigated to validate the feasibility of the drive system. Additionally, chain length and sprocket center to center spacing was computed using an equation available in the Machinery's Handbook [1]. The stand base was slotted to allow for horizontal movement, forward and backward of 1/2 inch and 1 inch respectively, to assemble and tension the chain. The final design minimized power losses and eliminated the need for a tensioner system.

Safety

To reduce safety risks, a chain guard was designed to cover the drive system. This prevented open pinch points and enclosed any catastrophic failure of the drive system. The chain guard was manufactured with 14 gauge cold rolled sheet steel in compliance with OSHA standards. [3]

To combat the scenario of engine seizure or an extreme overload from the dyno, a "safety hub" was designed. The smaller engine sprocket is mounted to the safety hub using bolts designed to shear in an overload scenario. The sprocket has a delrin bushing that lacks a keyway. This allows the sprocket to stay mounted and continue to spin on the shaft when broken away from the hub. Preliminary calculations pointed towards using two grade 2, 1/4 inch bolts, necked down to approximately 3/32 inch diameter at the sprocket to hub interface to facilitate this disconnect. The bolts were designed to shear at approximately 34 lb-ft to prevent an overload condition. This value is about twice the expected output of the test engine, but below the operating range of all other system components.

For further validation of the safety hub, a test fixture was used to replicate shear on the bolts. The fixture replicated the area of the clamping force, layout, and respective widths of the hub and sprocket. The fixture was used in conjunction with MTS tensile compression equipment. Three different 1/4 inch, grade 2 reduced diameter bolts, as well as one non-reduced grade 2, and one non-reduced grade 5 bolt were tested. Each trial was repeated three times to validate results

A graph of load versus displacement for each of the five different diameters' three test average can be seen in Fig. 3.
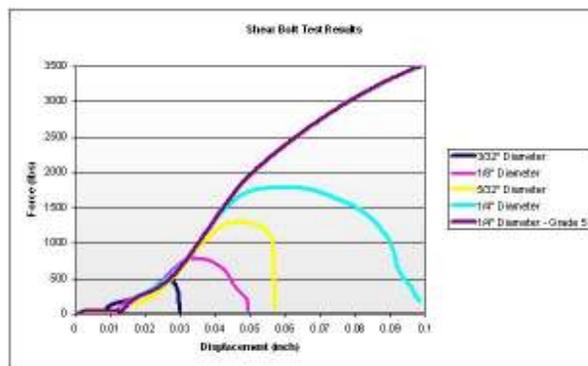

Figure 3. Results of shear bolt testing

Overall, the results showed good repeatability and trends were as expected. The experimental average peak force of the grade 2, reduced to 3/32 inch diameter, bolts was equivalent to 36 lb-ft of torque. Necking occurred, and the bolts snapped at a lower average value equivalent to 32 lb-ft of torque.

Manufacturing / Procurement of Parts:

As mentioned earlier, the dyno sprocket was custom ordered to be a direct bolt-on operation with the existing dyno hub. The smaller sprocket is an off the shelf McMaster-Carr piece sold only for number 50

chain. The sprocket width near the teeth was reduced to accommodate narrower 525 chain. This was accomplished by facing the teeth of the sprocket on the lath, holding it by the shear hub then chamfering the teeth to resemble the stock profile. The bore was increased to approximately 1.25 inches and a McMaster-Carr bushing, cut for length, was press fit into the center of the engine sprocket. The sprocket was also drilled with two 1/4 inch through holes to match the hub that holds it. The hub was purchased from McMaster-Carr with a keyway and bore matched to the engine output shaft. The hub attaches the sprocket to the engine using two shear bolts. A set screw was added to the hub's keyway using the mill. A second similar, but larger diameter hub with a set screw and a circular three bolt pattern, was used to attach the encoder wheel when cylinder pressure traces were run. When the encoder hub and wheel are not used, a 1.1 inch spacer is used between the sprocket hub and engine. Both setups use the same washer and engine shaft bolt to encapsulate the components of the system. All bolts used on the drive system employ thread locker and lock washers to reduce loosening effects of vibration.

Drive Results & Discussion:

The drive system serves its primary functions as expected. The discussed setup allows for the drive system to be modified later if desired. Loosening of hub bolts, observed during initial dyno runs, was resolved using thread locker and lock washers. Using the stand for chain tensioning worked well, but increased setup time. As a result, use of a stand-alone tensioner system is recommended. Infrequent chain slackening observed during initial dyno runs, required all mounting bolts in the stand to be loosened and re-tightened after tensioning, a laborious process. View windows were later put into the chain guard to facilitate inspection of chain tension without removal.

The original dyno hub was machined from aluminum. Silting was observed in this hub between dyno runs. The hub was thus replaced with a larger steel hub with the same mounting geometry.

The designed shear bolts fatigued frequently due to their low quality and the alternate loading. This alternate loading was an effect of the engine having only one cylinder, and thus only providing power once every other cycle, while the dyno provides constant power in opposition to this movement. Attempts to resolve this issue using grade 5 bolts, which were still safe for the system, resulted in failure of the safety hub. These fractures occurred at stress concentrations created by the keyway and clamping slots used for mounting the hub. Further inspection indicated that casting flaws may have also contributed to the failure. The fractured hub was replaced with a machined steel hub. The new hub did not implement the slotted clamping technique and had a thicker face to provide more stability and thread engagement for the shear bolts. This hub proved to be an improvement, however occasional loosening of the shear bolts from vibration still occurred. A solution using bolts with nylon inserts and shear pins instead of bolts has been proposed. The pins may perform better under the alternate loading of the engine, and the nylon inserts in the bolts may prove to be more effective than thread locker.

CONTROL SYSTEM METHODOLOGY

The sensor requirements for this project were determined through meetings with members of RIT SAE Baja. Engine parameters monitored and logged during testing (ordered by priority) include:

- Engine Speed
- Engine Output Torque
- Engine Horsepower
- Oil Temperature
- Cooling Fin Temperature
- Cylinder Pressure
- Air/Fuel Ratio
- Carburetor Bowl Temperature
- Intake Air Temperature
- Fuel Flow Rate
- Exhaust Gas Temperature
- Intake Air Flow Rate

In addition to the engine parameters specified, closed loop control of the DC motor by the DyneSystems DL4-334 dynamometer controller required measurements of:

- DC motor Speed
- DC motor Torque

The data acquisition system hardware includes products from National Instruments, Innovate Technology, and TFX Engine Technology, made available by the Mechanical Engineering Department. The PCI-6034E Data Acquisition (DAQ) card from National Instruments was used. The device features a 200 kHz sampling rate, 16 single-ended / 8 differential analog inputs, and 8 Digital I/O's (DIO). It is connected to a SCXI-1001 chassis, which houses and interfaces with the other input modules.

Air/Fuel ratio (AFR) measurements are made using an LM-2 kit from Innovate Technology. The kit includes a wideband Bosch lambda sensor and LM-2 digital air/fuel ratio meter. The LM-2 features two fully programmable linear analog outputs which were coupled to the DAQ system. The AFR and lambda

value, were determined through the linear relationships in equations (1) and (2):

$$AFR = 3.008V_{out} + 7.2933 \qquad (1)$$
$$\lambda = 0.2V_{out} + 0.5 \qquad (2)$$

where AFR is the air/fuel ratio, typically between 7.35 and 22.39, and V_{out} is the analog voltage output. Lambda is the ratio of actual AFR to stoichiometry.

Temperature measurements are acquired using sheathed K-type thermocouples from Omega Engineering. This type was chosen because it provided a range of -328°F to 2228°F which covers all expected temperatures. All thermocouples have an ungrounded junction which offers higher electrical isolation at the cost of response time. The thermocouples are connected to the DAQ card via TNC-2095 Terminal Block and SCXI-1102 Thermocouple Input Module. In addition, this hardware handles all signal conditioning.

A piezoelectric pressure sensor, Type 6125C from KISTLER, was used for cylinder pressure measurements. The sensor has a measuring range of 0-4.351 ksi and outputs varying capacitance with respect to pressure. The output from the sensor is connected to a dual mode charge amplifier type 5010B, also from KISTLER, that converts the measured charge to user-scalable voltage. The output of the amplifier was connected to the DBS 5 data logger from TFX Engine Technology. The KISTLER system was used largely because of its availability; the department already owned all the necessary hardware and software for its use. All pressure measurements were analyzed using TFX combustion pressure analysis software. The setup was integrated with an optical crankshaft encoder to provide RPM and crank position to the software. An encoder wheel was constructed for this setup.

A Model 112B10 spark plug cylinder pressure transducer and Model 422E03 in-line charge converter from PCB Piezotronics were acquired. This second sensor was requested by the Baja team since it was much simpler to implement on different engines. Due to lead time, the sensor was never implemented, but could be wired in the same fashion as the KISTLER sensor.

To calculate the power output of the engine, both torque and speed of the DC motor were measured. An Interface SM-500 load cell, mounted on the dyno mounting table, was used to measure output torque of the engine. The load cell has a 500 lb load capacity, which correlates to 665 lb-ft of torque at the DC motor. The load cell was connected to the Dyn-Loc IV dynamometer controller, which provides excitation voltage and signal conditioning. A hall-effect type sensor, which also interfaces with the dynamometer controller, was used to measure the speed of the DC motor. The Dyn-Loc IV dynamometer controller employed both torque and speed measurements for closed-loop control of the DC motor. The analog output from both sensors was coupled to the DAQ system to provide a real-time graphical display of horsepower, torque, and motor speed.

For mass air flow rate measurements, a series 641RM Air Velocity Transmitter from Dwyer, under loan from the Mechanical Engineering Department, was used. The device measured air velocity within a range of 250 FPM to 15,000 FPM and provided an analog output to the DAQ system. The sensor probe was placed in an engine intake air snorkel with known area to determine volumetric flow rate. The linear analog output can be converted to velocity by equation (3),

$$v = v_{max} \frac{(V_{out} - 1)}{4} \qquad (3)$$

where v is the air velocity in FPM, v_{max} is the user-selected velocity range, and V_{out} is the analog voltage output. The mass flow rate was computed from equation (4) below:

$$\dot{m} = \rho A v \qquad (4)$$

where $\dot{m}$ is the mass flow rate in kg/s, ρ is the air density in kg/m^3, and v is measured air velocity in m/s[4]. Assuming dry air the intake air density is given by equation (5),

$$\rho = 360.77819 T^{-1.0036} \qquad (5)$$

where T is intake air temperature in Kelvin and ρ is air density in kg/m^3[5].

A stepper motor and DPY50611 motor controller from Anaheim Automation, mounted on the dyno frame and coupled to a lever arm with a cable attached to the engine throttle, was used to control the throttle position. The stepper motor was connected to the controller that interfaces with the PC via USB. The throttle can also be actuated manually by a lever arm.

Manufacturing

During the design phase, a new mounting system and MS connector were manufactured to connect a small-sized SM-150 load cell to the dyno mounting table and Dyn-Loc IV Controller respectively. The mounting system consisted of female rod-ends that connected to the load cell via threaded plugs. The plugs were manufactured with 1/2 inch thread and 1/4 inch thread at opposite ends. As a result of intermittent torque spikes that occurred when controlling the DC motor,

the SM-150 load cell that operated at a lower 150 lb capacity was not used in the final design.

Modifications were also made to the design during sensor integration. An aluminum tube mounted on the engine intake inlet was installed to direct and accurately measure intake air flow rate. The velocity transmitter was placed inside the tube through a rubber grommet. The sensor was supported using an aluminum angle bracket.

Thermocouples were generally mounted with tape. This allowed easy placement of the sensors as the team saw fit. Epoxy was also purchased for the Baja team to use in the future when a more defined location was decided. For measuring the temperature of the oil, a brass plug was manufactured that would fit in the oil fill hole. A purchased swage-loc compression fitting was then put into this plug to hold the thermocouple probe in place. A similar fitting, made of steel, was welded to the top of the muffler in order to mount the exhaust thermocouple probe. This fitting was placed such that the probe's tip would be as close as possible to the exit air stream from the head.

To mount the oxygen sensor, a through hole was drilled in the side of the muffler and a threaded bung was welded in place. The bung was provided with the LM-2 kit.

Cylinder pressure measurements were acquired using two sensors, the KISTLER and PCB system. The latter used a spark plug modified by PCB. The KISTLER system was mounted in the cylinder head. The sensor was placed at the intake side of the head such that the sensor protruded near the edge of the flywheel cover. The head of the engine was held in a mill and drilled using a provided step drill. A detailed procedure of this process was provided to RIT Baja.

The optical encoder was mounted using an aluminum bracket attached to the engine stand. The trigger wheel for the encoder was machined on a rotary table such that there were eight teeth. The teeth were polished to be as reflective as possible and the notches were painted flat black. This was the protocol laid out in the TFX setup documentation. The wheel was mounted using a hub purchased from McMaster-Carr. Slots were milled into the encoder wheel so it could be rotated to the proper orientation to locate top dead center of the engine.

LabVIEW Graphical User Interface

Data acquisition and engine throttle control was implemented through National Instrument's LabVIEW program. The DC motor was controlled using Dyne Systems Dyn-Loc IV dynamometer controller. For safety, all control and data acquisition modules were located in a monitoring room away from the testing room, giving the user remote access during testing.

As shown in Fig. 4, the LabVIEW Graphical User Interface (GUI) featured multiple tabs, each for different tasks. Post-analysis to generate plots was handled using MATLAB script. Prior to each run, the user can specify test configurations using the "Settings" tab or use default values. During each run, the GUI displays real-time engine speed, torque, horsepower, temperatures, fuel-flow rate, and AFR under the "Acquisition" tab. Data logs of each parameter are created in a time-stamped CSV file located in a dated-folder. Real-time plots of engine speed and other debug parameters, such as output voltages of the velocity transmitter and LM-2 data logger, are available under the "Analysis" tab. The user can also enter comments in the "Observation" tab during runs. At the end of each run, comments were processed and logged in time-stamped text files. Post-analysis of acquired data in the CSV file were performed using MATLAB script that plots the acquired parameters. The script gives the user the ability to plot one run, compare two runs from the same engine, or compare two different engines.

A Simple Moving Average (SMA) could be applied to the data during post-analysis with MATLAB to smooth out short-term fluctuations and highlight long-term trends. The period of the SMA is simply the ratio of the sampling rate to the number of samples collected by the data acquisition hardware.
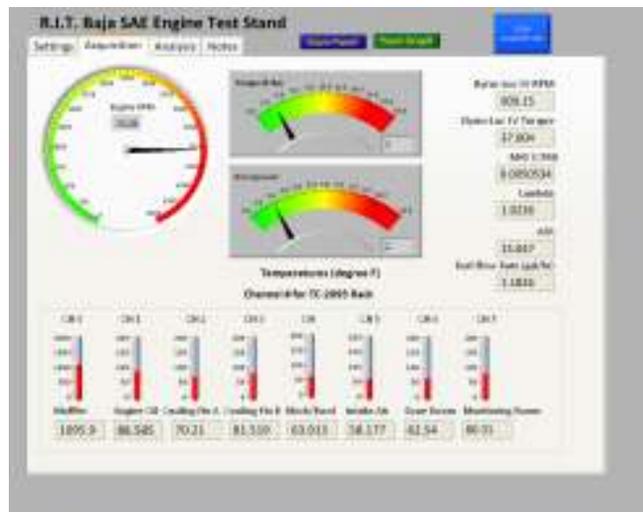

Figure 4. LabVIEW GUI showing "Acquisition" tab.

Results and Discussion of Data Acquisition

The SM-150 load cell was not used due to torque spikes that occurred on start up of the dyno. This load cell was replaced with the SM-500 load cell for its higher operating range.

For multiple runs, muffler temperature maxed out at 300°F due to the configurations of TC-2095

rack. Changing the DAQ configurations for expected thermocouple voltage output from 100 mV to 2500 mV resolved this issue. A test with a cigarette lighter was used to verify that higher temperatures up to 1300°F could be measured.

During the installation of the TFX pressure analysis software, driver conflict issues were experienced on the computer in the operating room. As a result, the DBS 5 data logger could not correctly acquire data from the sensors. To resolve this, the software was installed on a different PC. In addition, it was discovered that the PC must meet a display resolution requirement of 1028×768 to correctly display the software.

Another problem encountered was invalid AFR readings from the LabVIEW program in comparison to the values displayed by the LM-2 digital meter. This resulted from the LM-2 digital meter requiring connection to USB cable in addition to the power cable to output valid voltages.

OVERALL RESULTS AND DISCUSSION

Results & Discussion:

Plots of horsepower and torque, muffler temperature, and pressure generated from sample runs are shown in Figs 5-7.
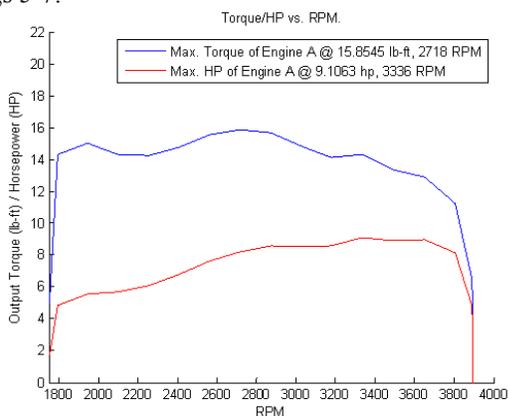

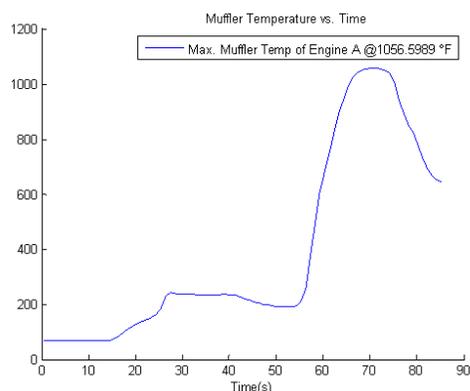Figure 5. Horsepower and Torque curve, SMA applied

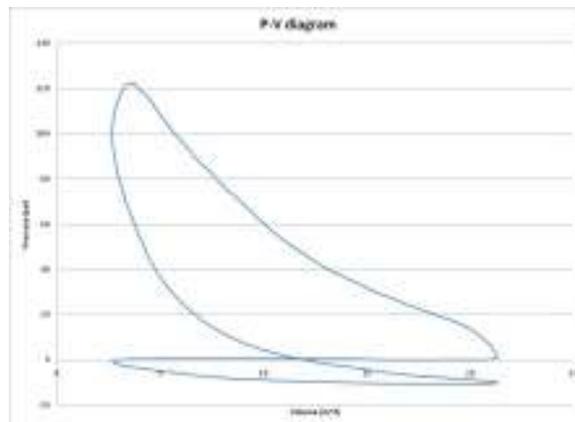Figure 6. Muffler temperature over time, SMA applied

Figure 7. P-V diagram

Due to time constraints and limited availability of the test facility, a rigorous repeatability and reliability assessment was not conducted, however post-analysis of collected data indicated the results were within range of expected values. Temperature values agreed with measurements taken with an infrared thermometer. Horsepower and torque measurements were found to be approximately 9.1 hp and 15.85 lb-ft respectively, near the rating of 10 hp and 14.5 lb-ft of torque, showing relative consistency. Intake air flow rates, fuel flow rates, and air/fuel ratios accurately matched their theoretical values.

A supplementary document highlighting step-by-step operating procedures was created. Safety risks and mitigation actions were also included in this document.

CONCLUSIONS AND RECOMMENDATIONS

The project was successfully completed in the provided time frame and within the allocated budget. With the exception of gauge reliability and repeatability tests, all other major design requirements were addressed.

A better failsafe mechanism, other than the shear bolts, would allow for faster setup time. The use of only two shear bolts may have also allowed slight movement of the small sprocket, eventually vibrating the bolts loose and breaking them. It has been suggested to the Baja team to use shear pins and bolts with nylon inserts to resolve loosening problems.

In addition, a new tensioning system that does not require frequent loosening and tightening of mounting bolts on the stand could be implemented. This may also reduce tension-related differences in acquired data.

The dyno controller has adjustable linear-acceleration (LAC) and feedback rate. Unfortunately, the feedback rate is not easily reconfigurable. Easy reconfiguration of the feedback rate would reduce

torque spikes and improve the resolution of the dyno. This feature is available in newer controller models; therefore replacement of this unit is recommended.

Furthermore, several issues related to the test facility should be addressed. The exhaust system is inadequate for an engine test facility and therefore needs serious and immediate attention. The fan lacks suction power and the exhaust hose is highly brittle. Currently, there are no air quality sensors to alert users of hazardous conditions.

The fire suppression system should also be upgraded. The only available fire suppressor is a handheld fire extinguisher. Additionally, there is no way to limit air flow into the room during a fire. There are also ventilation holes in the floor that would allow the fire to spread quickly to further parts of the building if not suppressed.

ACKNOWLEDGMENTS

REFERENCES

[1] J. M. Amiss, F. D. Jones, and H. H. Ryffel, *Machinery's Handbook* & *Guide to Machinery's Handbook*, 28th ed., Industrial Press, 2008, pp. 2466-67.

[2] The American Chain Association, "Improved Power Ratings for Single Strand Roller Chain," 2003 [Online]. Available: http://americanchainassn.org/acahpr122001R1.pdf. [Accessed: Oct. 12, 2009].

[3]OSHA, "Occupational Safety and Health Standards: Machinery and Machine Guarding", *Occupational Safety and Health Administration,*1910.219(f). [Online]. Available: http://www.osha.gov/pls/oshaweb/owadisp.show_document?p_table=STANDARDS&p_id=9847. [Accessed: Oct. 20, 2009].

[4] M. J. Moran and H. N. Shapiro, *Fundamentals of Engineering Thermodynamics,* 3rd ed., New York: J. Wiley, NY, 2008, pp. 149.

[5] "Density of Air vs. Temperature," [Online]. Available: http://users.wpi.edu/~ierardi/PDF/air_density_plot.PDF. [Accessed: Feb. 4, 2010].