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IMPACT OF E20 FUEL ON A HIGH-PERFORMANCE, TWO-STROKE ENGINE

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By

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Abstract

The purpose of this project was to explore the emissions, combustion, and performance effects of running a gasoline/ethanol fuel mixture of 20 percent by volume (E20) in a fuel-injected, two-stroke engine. The engine was operated at five engine speeds that corresponded with the EPA 5-mode emissions test for snowmobile engines. Single parameter sweeps were conducted along with a preliminary recalibration of the test engine at two E0 target values (lambda and mid-pipe temperature) using E20 fuel.

Baseline testing showed that running E20 fuel produced a leaner air/fuel mixture compared to E0, resulting in higher lambda values for all modes and higher mid-pipe temperatures in modes 1 and 2. The increase in lambda resulted in lower CO and THC emissions at all modes and an increase in formaldehyde and acetaldehyde emissions. An increase in CO_2 and NO emissions followed the trend of increasing mid-pipe temperature at modes 1 and 2.

Single parameter sweeps were performed by changing one engine parameter at a time and sweeping over a range of predetermined values. Engine parameters included injection time (duration), injection end angle, and ignition timing. Increasing the amount of fuel injected into the combustion chamber decreased lambda values, decreased mid-pipe temperatures, increased CO and THC emissions, and decreased CO_2, NO, formaldehyde, and acetaldehyde emissions. Advancing the ignition timing decreased mid-pipe temperatures which decreased CO_2, NO, formaldehyde, and acetaldehyde emissions. CO and THC emissions were increased with the advancement of ignition timing. Opposite trends could be seen with retarding ignition timing, except with NO emissions where retarding ignition timing also resulted in a reduction in NO emissions. Adjusting the injection end angle showed little effect on performance, but increases in CO_2, NO, formaldehyde and acetaldehyde emissions were seen at large advances of degrees.

Recalibration of injection parameters for E20 fuel to meet E0 baseline lambda values was performed by increasing the injection timing values in the ECU. This created a richer mixture at all modes when compared to the E20 baseline test, while some modes were still leaner than stoichiometric. Matching lambda values resulted in mid-pipe temperatures that were still higher than the E0 baseline test in modes 1 and 2. CO emissions were still lower in all modes except in mode 3 as well as THC emissions except for an increase of two percent in mode 1. CO_2 and NO emissions saw a decrease in mode 1 although both values were still higher than the baseline E0 test.

Meeting E0 mid-pipe temperatures with E20 fuel resulted in a higher lambda value at modes 1, 4 and 5. CO emissions followed these trends with higher values in modes 2 and 3 when compared to the E0 baseline test. CO_2 emissions were opposite CO emissions with increases at modes 1, 4 and 5. NO and THC emissions saw an increase at mode 1 and decreases in modes 2 through 4.

Chapter 1: Introduction

Testing various ethanol content fuels in snowmobile engines has taken place in previous years showing that there is little evidence supporting that the current ethanol concentrations found at gas stations are hazardous to current snowmobile engines [1]. This current research project looks at an ethanol concentration of 20 percent ethanol by volume. The octane rating was maintained at a value of 91-93 AKI, which can be found in premium gasoline available at gas stations.

One of the problems with adding ethanol to gasoline is that air/fuel ratio changes. The air/fuel ratio is the ratio between the mass of air and the mass of fuel being combusted. This is show in equation 1 below [2].

$$AFR = \frac{m_{air}}{m_{fuel}} \tag{1}$$

An air/fuel value that is unique to each fuel is the stoichiometric air/fuel ratio. This is where there is theoretically the correct amount of fuel to completely combust with a corresponding amount of air. For ethanol, the stoichiometric air/fuel ratio is smaller, meaning that there is less mass of air required to completely combust the same mass of fuel; E0 gasoline has an air/fuel ratio of 14.7 [2], and E20 has an air/fuel ratio of 13.4. There is a certain range where each engine can run above and below this stoichiometric air/fuel ratio. This can be defined as lambda in equation 2.

$$Lambda\ (\lambda) = \frac{AFR_{actual}}{AFR_{stoich}} \tag{2}$$

E20 is a leaner mixture than E0 due to the additional element of oxygen in ethanol which results in higher combustion temperatures [1]. These higher temperatures can be hard on both marine engines and snowmobile engines since they can spend a lot of time operating at high temperatures; for instance, running at wide open throttle under heavy load. Most of these engines are either carbureted or have no/minimal electronic means of compensating for this fuel change.

Most current production snowmobile engines operate with an open-loop engine management system which does not compensate for changes in ethanol content. This results in higher exhaust system temperatures and different emissions constituents to be produced. This current research project utilized a new production two-stroke engine that is fuel injected, allowing the ECU to precisely control fuel delivery and a temperature sensor in the exhaust that protects the engine if exhaust temperatures get too high allowing compensation.

The testing format for this experiment was conducted as a 5-mode test. This is the standard for the industry and is used to certify the emissions levels of snowmobile engines. Table 1 below shows what engine speed and torque values are associated with each mode. 5-mode testing points vary for each engine because each mode is based on the maximum engine speed at peak torque and the maximum torque value. Weight factors are also applied to each mode for emissions where the sum of all modal emissions must not exceed a maximum value.

Table 1. 5-mode testing points for the Arctic Cat 600 DSI engine

Mode	Speed Percent (%)	Engine Speed (RPM)	Torque Percent (%)	Torque (ft-lbs)	Emissions Weight Factors
1	100	8250	100	67	0.12
2	85	7015	51	33	0.27
3	75	6190	33	22	0.25
4	65	5360	19	13	0.31
5	Idle	1650	0	0	0.05

With permission from the engine manufacturer, access was granted to the engine management system resulting in the ability to change three main engine parameters. These parameters included injection time (duration), injection end angle, and ignition timing. After baseline E0 and E20 5-mode tests were acquired, engine parameter sweeps were completed to see how changing these engine parameters affected emissions, combustion stability, exhaust system temperatures, and overall engine performance.

After analyzing the sweep results, the goal was to reduce exhaust system temperatures utilizing two E0 targets. These targets included matching the E0 baseline lambda values and mid-pipe temperatures with E20 fuel at each mode.

Chapter 2: Goal and Objectives

The goal of this experiment was to run a gasoline/ethanol fuel mixture of 20 percent ethanol by volume in a modern fuel-injected two-stroke engine while recording emissions, combustion, and performance data.

The first objective of this experiment was to construct a dyno cell that utilized an eddy current dyno to run an Arctic Cat 600 DSI two-stroke engine. Emissions and combustion data was required for testing along with monitoring temperatures and pressures of the engine.

The second objective was to baseline test the setup with E0 and E20 fuel to get a starting point for the recalibration process. Following the baseline tests, single engine parameter sweeps were conducted to study the change in lambda, exhaust temperature, and emissions.

The final objective was to adjust engine parameters to match lambda and mid-pipe temperatures of E0 fuel when operating on E20 fuel, to see if these values would produce similar emissions and exhaust temperatures. Note that operating at the same lambda values with two fuels of different ethanol contents will match the ratio of the actual air/fuel ratio over the stoichiometric air/fuel ratio. However, this does not match the mass flow rates of the different fuels which results in different performance and emissions values.

Similar lambda values will produce different exhaust temperatures creating different emissions results. This hypothesis is similar for matching mid-pipe temperatures if the lambda values are different at each matching mid-pipe temperature.

Chapter 3: Literature Review

The researchers at National Renewable Energy Laboratory (NREL) conducted tests to follow up previous research done on ethanol concentrations in fuels. A strict protocol was used in this testing to show how a closed loop engine control system can mitigate the impact of a higher ethanol content fuel [3]. There were three fuel blends (E0, E15, and E20) that were splash blended to meet a percent volume of ethanol per total volume of fuel. Each fuel blend group had vehicles assigned to it; four cars from the year 2009 (when the test started) and two cars from the year 2000 for a total of 18 vehicles. These cars were chosen based on the manufacturer, model year, number of vehicles sold and registered, and whether or not the vehicle had calibration changes done to control higher ethanol content fuels at wide open throttle (WOT). For the E0 and E15 fuel, new vehicles were purchased and broken in to about 4,000 miles before starting the testing. The E20 2009 vehicles and all year 2000 vehicles were bought used after passing an initial inspection which deemed the vehicle acceptable to use for this test. For the model 2009 vehicles, baseline tests were recorded and then the vehicle ran a standard road cycle on a dynamometer to put miles on the car before stopping and taking emissions data. Emissions data was taken at 60,000, 90,000, and 120,000 miles. All vehicles from the year 2000 were bought used after an inspection and then baseline tested. There were two more emissions tests done at 25,000 and 50,000 miles after the baseline test with the standard road cycle being used to accumulate the miles. At each test interval, a cold start FTP75 3 bag dynamometer test was conducted for each vehicle. This test is a city driving test that varies engine speed to simulate city driving over a time for about a half hour.

After all of the testing was completed, it was concluded that running higher ethanol content fuels did not affect engine durability based on no reduction in compression test values. There was contradicting results where four of the six vehicles tested saw higher exhaust emissions with the E0 fuel than the other two alternative fuels, raising concern for the condition of the catalysts in the exhaust system. As the test continued, NO_x emissions saw an increase for all but one of the vehicles and little to no change was seen in hydrocarbon and CO emissions. Fuel economy decreased for both E15 and E20 fuels. It was discovered this value was proportional to the energy density of the fuel, where ethanol fuels have a lower energy density than gasoline. In conclusion, the vehicles tested on E15 and E20 fuels did not see higher exhaust emissions for all vehicles tested.

Research conducted at Michigan Technological University had the most relevance to the present research topic. Evaluating the Impact of E15 on Snowmobile Engine Durability and Vehicle Drivability [1] pertains to evaluating emissions using an E15 ethanol/fuel mixture in four different snowmobile engines. No engine tuning was done for E15 tests so their data can be compared with the current research baseline E20 data to find similarities. All tests except lab emissions used E0 gasoline mixed with E98 to create an E15 mixture, which was 15% by volume. For the lab emissions, E98 fuel was mixed with Indolene creating an E15 mixture resulting in a higher ethanol content fuel than the E0 fuel. The minimum octane rating was met for all of the manufactures, 87 octane for two engines and 91 octane for the other two engines.

A wide range of snowmobile engines were used including two and four strokes applications, different forms of aspiration such as carburetors, port fuel injection and direct injection, as well as fan and liquid cooling. It was not stated what combination of parameters were for each engine but it could be assumed that one combination was a fan cooled, carbureted two stroke from an older snowmobile, while the liquid cooled port and direct injected variants applied to more modern either two-stroke or four-stroke engines.

Engine emissions were taken for a pair of each of the four engines ran in the snowmobiles tested. One group was only run on E0, while the other group was run only on E15, but all engines went through the same brake-in procedure. After the brake-in was complete, 5-mode emissions were taken for all of the engines to get base line data before going into the durability testing. The durability testing consisted of a 6-mode test (where mode 6 was idle) ran on a cycle which was equal to the engine running for a total of 5,000 miles. Emissions was taken halfway through and at the end of the durability testing.

For some engines, a reduction in BSFC could be seen if there was an increase in power due to the higher octane rating of E15 in mode 1. The trend of higher exhaust gas temperature (EGT) was observed for all engines on E15, because ethanol fuel contains oxygen resulting in a leaner mixture, but still fuel rich when compared to the stoichiometric air/fuel ratio. All engines running on E15 saw a reduction in carbon monoxide emissions and HC emissions when compared to E0 emissions. The unregulated emissions constituent formaldehyde increased while 1, 3 butadiene emissions were generally reduced. Finally, there was a case where one engine experienced extremely high EGT's resulting in damage to the exhaust system rendering it unusable for future tests. This result of running leaner mixtures from higher ethanol content fuel results in higher exhaust temperatures.

BRP investigated the effects of butanol and ethanol fuel mixtures in a 3-cylinder, direct-injected two-stroke marine engine [4]. There was a total of six fuel blends tested. They included 100% Indolene (base fuel), 10% n-butanol, 15% n-butanol, 20% n-butanol, 10% ethanol, and 15% ethanol. All five of the fuels were mixed with Indolene and were blended on a percent volume basis. Combustion and 5-gas emissions analysis was done on the engine which was tested on a dynamometer. Two, 5-mode tests and two, wide open throttle tests were run with all six fuels.

Initial n-butanol results showed lower HC and NO_x combined emissions when comparing to 100% Indolene for all modes minus mode 4, where there were misfire problems. There were also reductions in CO emissions with increasing concentrations of butanol in the fuel mixture. A minimal increase in CO_2 was seen with increasing amounts of n-butanol. An increase in EGT's was seen for mode 1, but decreases in EGT's was noted for modes 2-4, which was assumed to be affected by the changing air/fuel ratio of the fuel mixture. There was no loss in power with increasing percentages of n-butanol in the fuel.

Ethanol/Indolene fuel mixtures had higher values of lambda compared to the baseline fuel, resulting in lower CO emissions. There was a slight increase in combined HC and NOx with the ethanol mixture when compared to the n-butanol mixture and a much larger increase in CO_2 emissions with ethanol, especially at mode 1 and increased concentration of ethanol in the fuel. Combustion analysis of both alternative fuels resulted in similar COV of IMEP values when compared to the baseline fuel, along with similar knock index values. This was attributed to the higher octane value which comes with these alternative fuels that offsets the higher combustion temperatures. Ethanol had a reduced burn rate and both ethanol and butanol mixtures had a peak burn rate (percentage of fuel burned per crank angle degree) between 1 and 3 degrees before the baseline Indolene fuel. Both ethanol and butanol mixtures resulted in earlier peak cylinder pressures by about 2-3 degrees.

The researchers at Environment Canada investigated the exhaust emissions produced by a small, four-stroke, two cylinder, closed loop, gasoline engine when run on mixtures of ethanol and iso-butanol [5]. The ethanol and iso-butanol concentrations were E15 and iB16. Unlike most small engines which are carbureted, this engine was fuel injected with an oxygen sensor in the exhaust. This allowed the engine to control the air/fuel ratio. There were five combinations of fuels tested; E0, E10, E15, iB16 and a combination of ethanol and iso-butanol iB8-E10. The combinations of fuels were splash blended on a percent volume basis. These combinations of fuels were run in a 6-mode test multiple times on the same engine and the results were averaged.

Engine torque increased for all alternative fuels with the iso-butanol fuels producing the highest torque values. It was discussed that there is more complete combustion with the oxygenated fuels along with higher octane ratings resulting in higher torque values. Engine and exhaust temperatures were similar with the iso-butanol fuels, where the E10 mixture had a reduction in oil temperature but an increase in exhaust temperature.

E15 and iB8-E10 fuels had a slight increase in BSFC, and E10 and iB16 had a slight decrease in BSFC. These pairs of fuels had similar air fuel ratios which could explain the trends. Thermal efficiency increased for all of the oxygenated fuels in both the weighted and 5-mode tests. The reduced lower heating values of ethanol and iso-butanol helped to offset the mass flow of the fuel when calculating thermal efficiency. This can be understood in the thermal efficiency equation below.

$$\eta_{th} = \frac{3000 * B_p}{LHV * \dot{m}_f} \tag{3}$$

Emissions results were represented in weighted emissions based on the weight at each mode, and the emissions at each individual mode. There was a decrease in weighted CO emissions for all fuel mixtures compared to the E0 fuel. The largest reduction in CO emissions was at mode 1 with the E10 fuel. The added oxygen in these fuel mixtures helped to improve complete combustion, reducing CO emissions. A reduction in THC emissions were also seen with E15 having the largest weighted reduction and E10 having the largest reduction at mode 1. The blend of iB6-E10 was expected to have the largest reduction in THC emissions because it had the highest oxygen content of all the fuels, but the test engine could not maintain RPM at full load which may have resulted in misfiring and higher than expected THC emissions. NO_x emissions only showed significant decreases for the E15 fuel in the weighted analysis which could be because ethanol has the highest latent heat of vaporization of all the fuels. Final greenhouse gas emissions tests concluded that decreases in CO_2 and CH_4 existed for both ethanol and iso-butanol mixed fuels for modal and weighted tests. There were no significant changes in N_2O emissions.

Carbonyl emissions were also an area of interest in this research. It was noted that these emissions are not understood very well nor are their health impacts. 16 compounds were analyzed in this project, with increasing values seen for all oxygenated fuel types. Formaldehyde, acetaldehyde and benzaldehyde saw the largest increases of all the 16 carbonyl constituents and acetaldehyde had the largest variation of these three. It could be seen that the fuels containing ethanol (E10, E15, and iB8-E10) had at least three times the weighted values of E0, with E15 being the highest.

Chapter 4: Experimental Setup

The construction of the dyno cell revolved around a Froude Hofmann AG150-HS low inertia dyno which was used to apply load to the engine and measure engine torque. The eddy current dyno was ideal for high revving engines like the Arctic Cat 600 DSI two-stroke engine, capable of 8,250 RPM. A starter was mounted on the opposite end of the dyno with a billet racing flywheel to turn the engine over. This setup is seen below in Figure 4.1.

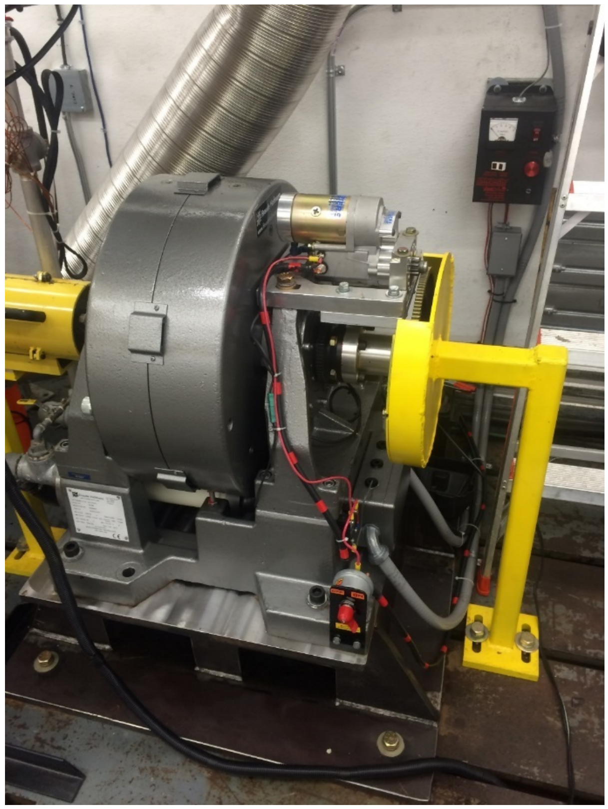

Figure 4.1: Froude Hofmann AG150-HS low inertia eddy current dyno with custom starter

A custom plinth was designed to elevate the dyno off of the bedplate to make engine and dyno maintenance easier. To match the height of the dyno, telescoping elephant feet were used to elevate a steel base plate to which the engine mounted to. Engine mounting holes and slots were machined into the base plate for accuracy and the ability for the engine to be moved around to align the driveshaft. Between the elephant feet and base plate were vibration isolator mounts to reduce vibrations between the engine and mounting equipment. Custom engine mounts were made out of tool steel to provide a rigid body for the engine to mount to during testing. Two types of driveshafts were used during testing. The first driveshaft was a non-telescoping aluminum driveshaft with rubber dampeners and polymer locating pins in each end. This driveshaft was immediately destroyed due to the engines rough idle at 1,650 rpm. It was concluded that the vibration isolators were too soft so during misfire the driveshaft would enter into resonance. A second driveshaft was sourced which was able to plunge the length of 1 inch and contained significantly stiffer torsional material. This driveshaft can be seen below in Figure 4.2.

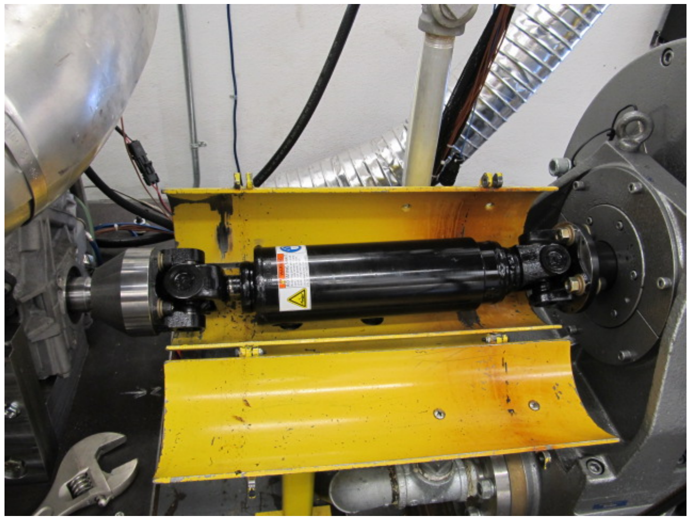

Figure 4.2: Second driveshaft which incorporates a plunging center section

The rubber vibration isolators were also replaced with stiffer rubber isolators to reduce engine movement. A resonance calculation involving the inertia of the engine and dyno was conducted before purchasing the driveshaft to make sure that the torsional stiffness of the driveshaft would not result in resonances that would destroy the shaft at any of the 5 modes of testing. It was calculated that there was a 2nd order resonance at about 3000 RPM, which would only be seen during acceleration and deceleration between modes 4 and 5 [A1]. This driveshaft was longer than the original driveshaft and used a u-joint on each end. Because of the u-joints, the driveshaft needed to be run at an angle of 1 degree to reduce u-joint failure. This angle was calculated with trigonometry and was measured with a dial indicator before installation to make sure actual measurements were within 0.001”. The engine used for this testing was an Arctic Cat 600cc DSI two-stroke engine shown in figure 4.3. This engine utilized a fuel injection system and a unique piston design which contained a slot in the piston skirt for the injector to spray fuel through. This was done to provide more fuel and extra bottom end lubrication at high engine RPM’s [6]. A temperature sensor located in the exhaust system was a safeguard prevent any damage to the engine or exhaust system because of high exhaust temperatures due to lean air/fuel mixtures.

Figure 4.3: Cylinder cut-away of the Arctic Cat 600cc DSI two-stroke engine [6]

The stock hood was used to keep the OEM air box along with the OEM oil tank and pump necessary to supply the engine with oil. A picture of the engine and dyno set-up can be seen below in Figure 4.4.

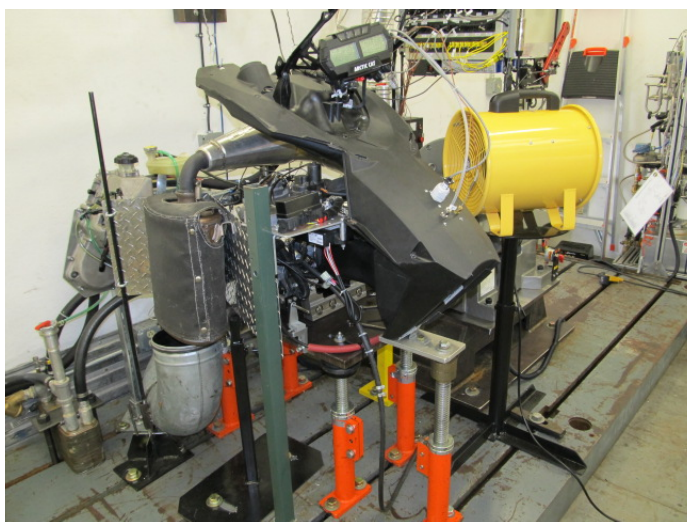

Figure 4.4: Engine setup showing the intake and exhaust configuration

To set this test apart from other analysis, combustion and emissions analysis was to be done on this engine. In order to do combustion analysis, an OEM cylinder head had to be CNC machined to accept the pressure transducers for both cylinders. Figure 4.5 below shows one of the pressure transducers in the magneto side of the engine. A separate sleeve of aluminum was pressed into the cylinder head to seal the water cooling jacket, and then welded to the top the of cylinder head.

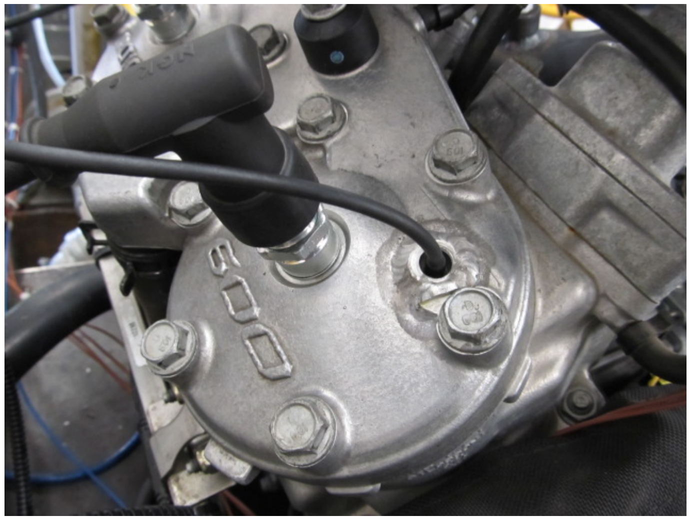

Figure 4.5: PCB pressure transducer installed in the cylinder head of the engine

Combustion analysis was recorded with PCB in-cylinder pressure transducers, model # 115A04 along with an AVL 365C encoder that recorded the degrees of rotation of the crankshaft shown below in Figure 4.6. All of this information was processed by AVL software.

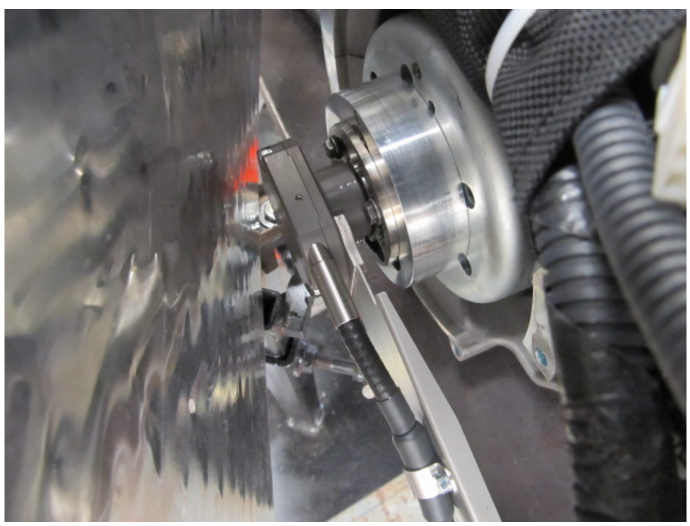

Figure 4.6: Crank angle encoder attached to the magneto

Emissions were recorded using an AVL SESAM emissions bench, which utilized an FTIR analyzer capable of measuring 25 exhaust constituents simultaneously. Emissions were taken directly out of the exhaust via a sample probe and transported to the emissions bench with a heated line. Figure 4.7 below shows the probe location in the exhaust, along with the lambda and temperature sensor.

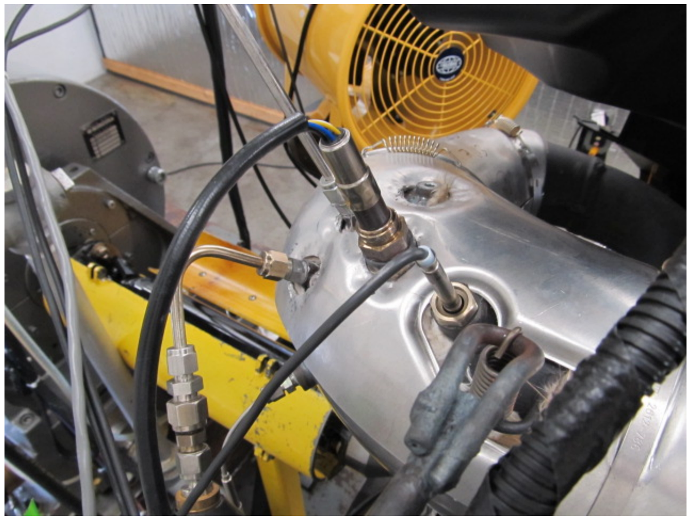

Figure 4.7: Mid-pipe with emissions probe, lambda sensor, thermocouple and factory temperature sensor attached

Fuel blending was required for the current research project. The octane rating was calculated based on the octane number of gasoline and ethanol, and the non-linear fit equation found in the SAE paper produced by the Ford Motor Company [7]. Paragon laboratory measured octane rating to be 92.3 AKI [A2]. This value was between the high and low calculated values. Premium fuel at most gas stations is between 91 and 93 AKI which was required by the manufacture to be used in the test engine.

Octane rating is a critical variable that affects overall engine performance. When conducting engine research, any variables that can be eliminated from the overall project should be investigated. Historically, blending ethanol and gasoline mixtures has been done on percent volume basis. Although the volume concentration of ethanol will be correct when splash blending because the relationship is linear, the octane rating does not follow suit. Research was conducted to examine the resulting octane value (both research octane number (RON) and motor octane number (MON)) of gasoline/ethanol fuel mixtures when blended with different volumes of ethanol.

Plotting octane rating verses a volume percent of ethanol produces a non-linear line. For ease of calculation, a linear relationship based on volume percentage is usually used when calculating the octane rating for a gasoline/ethanol fuel mixture. It could be seen from Figure 4.8 that the linear equation could be off by as much as 2 RON and 3 MON. This may not be good enough when a more accurate octane number is needed when mixing fuel.

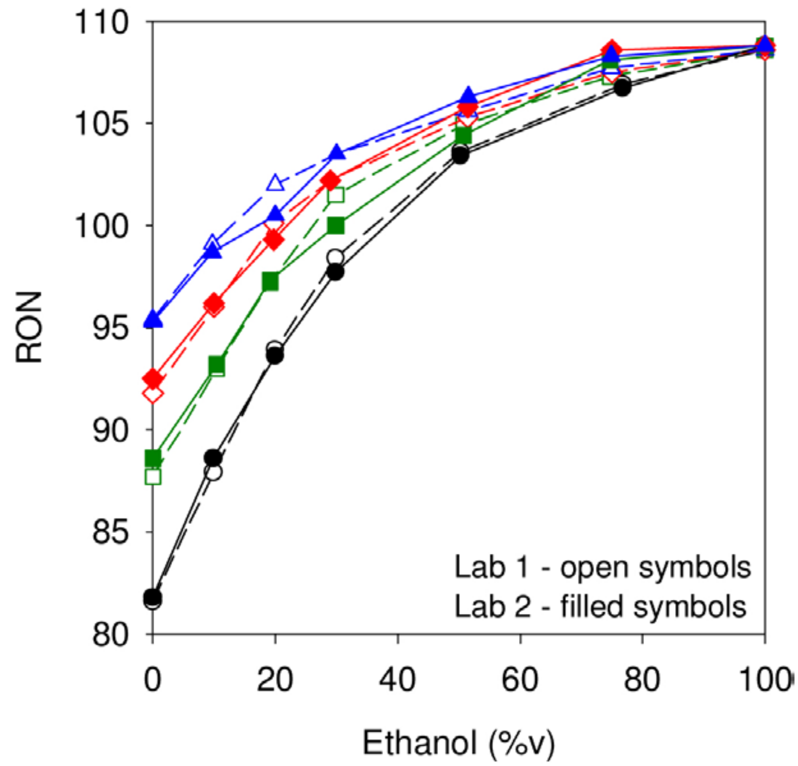

Figure 4.8: RON of ethanol blends in four blend stocks independently determined by four laboratories [7]

This experiment blended a total of 26 different fuels; 5 blendstock fuels with different octane ratings, which were then mixed with different ethanol contents ranging from 10-75 percent ethanol by volume. A more linear curve could be produced when plotting RON or MON values verses a mole percentage of ethanol in the fuel. This mole percentage of ethanol line was used when calculating the non-linear curve fit line. Analysis was done by two different labs to find RON and MON values of the fuel mixtures, so a non-linear curve-fit line could be used which incorporated a calculated scaling parameter seen in equation 2 below. This scaling parameter (P_g) is based on finding a range where the sum of least squared values were at a minimum when looking at the standard deviation of the linear model to the measured RON and MON values plotted verses molar percent of ethanol. The range of P_g for RON is .46-.49 and 1.03-1.21 for MON. These standard deviations were largest where the molar percentage of ethanol was at 50%, which was about 30 percent ethanol by volume. Using this percentage concentration of ethanol, the higher range of values for the scaling parameter is ideal. This non-linear fit equation calculates the octane number of the fuel blend (ON_b) using the gasoline octane number (ON_g), alcohol octane number (ON_a), molar ratio of alcohol (x_a), and the scaling parameter (P_g).

$$ON_b = (1 - x_a)ON_g + x_a ON_a + P_g x_a (1 - x_a)(ON_a - ON_g) \tag{4}$$

This equation was plotted back over the RON and MON measured values verses ethanol percent by volume. This non-linear equation fit the non-linear trend of increasing octane rating verses percent volume of ethanol seen in Figure 4.9. In the end, octane ratings were able to be predicted within 1 octane number of the measured octane rating.

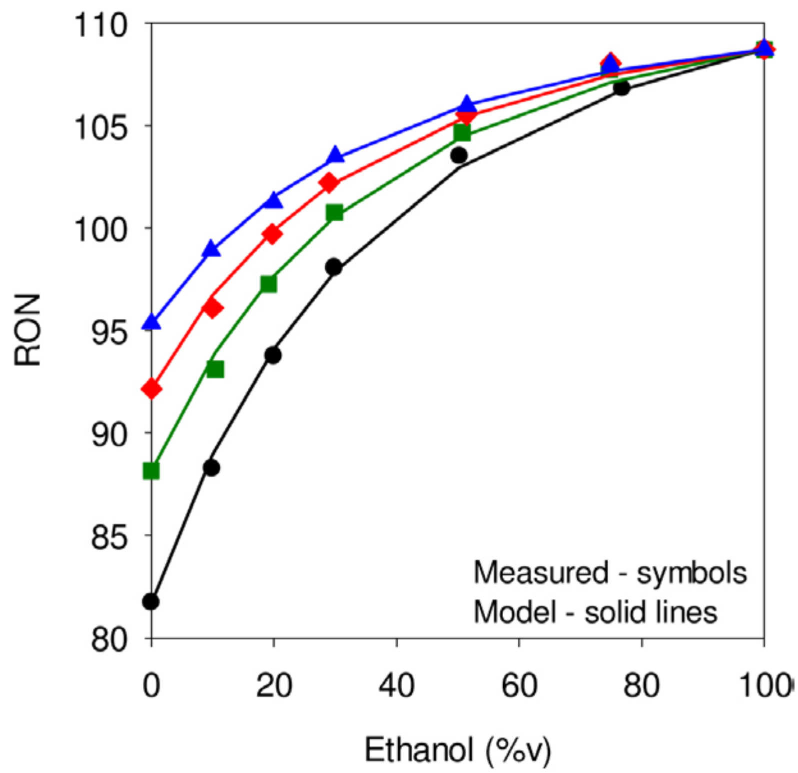

Figure 4.9: RON of ethanol blends in four blendstocks vs. volumetric ethanol concentration and equation (2) model fit [7]

Fuel for this test was pumped from a gas station into 55 gallon drums and then resealed. This was the best way to maintain consistency throughout the testing to rule out any problems concerning the fuel. One drum of 91 E0 fuel, for baseline testing, was filled along with 3 drums of 87 E10. The 87 E10 fuel was blended with pure ethanol on a volume basis to reach a mixture close to 91 E20. The resulting mixture resulted in a fuel with an octane rating of 92.3 [A2] and a volume percentage of 20 percent ethanol. This octane value worked well because of the availability of both 91 E0 and 93 E0 at gas stations. The volume of ethanol was calculated based on testing a sample of the fuel from the 55 gallon drum and adding a known amount of water in a graduated cylinder. Since water bonds with ethanol, the ethanol in the fuel was pulled out of the fuel and combined with the volume of water making a known volume of ethanol and water in the mixture. This divided by the total volume of the mixture produced the ethanol percentage.

A Re-Sol fuel cart was used to maintain a fuel supply pressure of 4 bar and record fuel flow using a Coriolis flow meter. A picture of the fuel cart is shown below in Figure 4.10.

Figure 4.10: Re-Sol fuel cart used for the current research project

A data acquisition box containing the DYNO MAX Pro controllers was used to monitor engine and cell temperatures and pressures along with engine and dyno controls. Power was also supplied to devices such as a lambda sensor and the fuel cart. It was important for everything in the cell to be wired through the DAQ box controller in case of an emergency where an emergency shut off was needed. These emergency shut offs were placed in the cell and control room and would cut power to everything when pressed. The DAQ box also had internal relays that would run engine and dyno cooling, cell exhaust fans, and the engine starter.

The cooling loops consisted of two radiators located outside of the dyno with electric pumps plumbed into each system to flow coolant. Dyno and engine cooling loops were separate with temperatures in each loop being monitored. The engine cooling system was plumbed through a liquid/liquid heat exchanger that was attached to the bedplate. Exhaust gases escaped through a 10 inch diameter duct that was drawn through by a fan on the ceiling of the test cell. The large diameter accompanied with a high flow fan was required because there were times in testing that required sustained operation at mode 1. There was also a room fan that pulled fresh air from the sides of the cell into the room. Depending on the test cell temperature, a combination of a controllable AC unit and heater was used to maintain intake temperatures at 25°C plus or minus 5°C. If colder temperatures were required, an industrial air conditioner (reefer) was used to supply intake temperatures. Both units had the ability to be plumbed into the engines intake. The intake pressure was monitored to make sure no boosting was applied while running. The dyno cell was monitored via camera from the control room.

Chapter 5: Results

The results presented include baseline testing, single parameter sweeps, and recalibration to meet E0 lambda and mid-pipe values with E20 fuel. Initial baseline tests involved running a 5-mode test with no recalibration of the ECU and recording emissions, combustion and performance data. Single parameter sweeps involved maintaining the engine at a single mode speed, and sweeping a single engine parameter (injection time, injection end angle, or ignition timing) recording emissions, combustion, and performance data. Finally recalibration was conducted at two E0 parameters (lambda and mid-pipe temperature) with E20 fuel to see how emissions, combustion and performance was affected with changing engine parameters to meet the E0 values.

5.1 Baseline E0 and E20 Testing

This project started with base-lining the Arctic Cat DSI 600 two-stroke engine with a 91 octane rated fuel containing no ethanol purchased at a local gas station. This baseline was used to compare to the second fuel tested, a gasoline/ethanol fuel mixture with a 92.6 octane rating and an ethanol content of 20 percent by volume. This was done to see how the higher ethanol content fuel would affect emissions, combustion, and performance of the engine.

All of the data in this section is reported in a percent difference format. To provide means of evaluation, a positive percent difference value means that the E20 fuel is a larger value than the E0 fuel. A negative percent difference value means the E20 fuel is a smaller value than the E0 fuel.

Figure 5.1 shows the impact of E20 on lambda, in percent difference. An increase in the value of lambda means that the air/fuel mixture is leaner. Depending on the mode, some lambda values were still richer than stoichiometric, even with an increase in lambda using E20 fuel. Running the engine on E20 fuel resulted in a leaner mixture for all 5 modes. This is because E20 fuel contains an additional element of oxygen in the fuel molecule. This trend in lambda can be seen in the literature review where testing utilized a direct-injected two-stroke marine engine when increasing the amount of ethanol in the fuel [4].

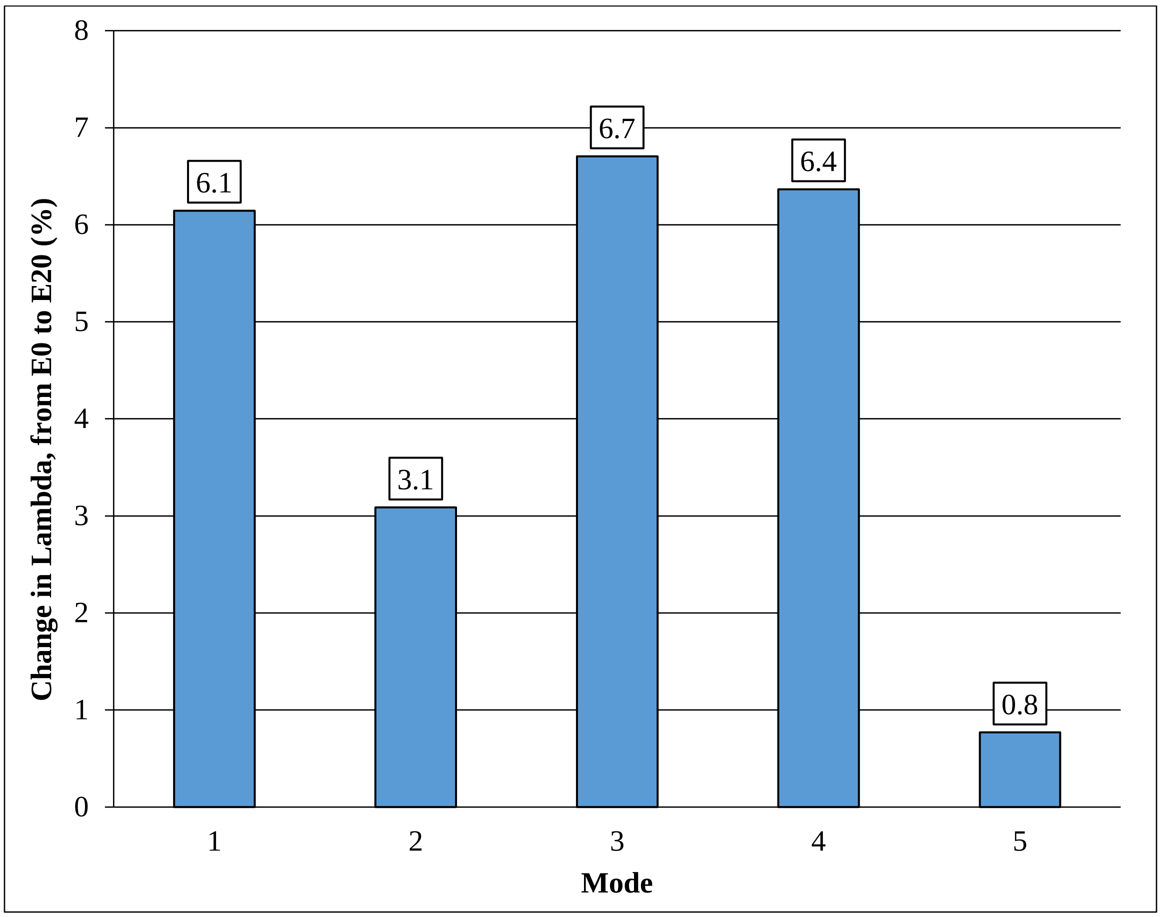

Figure 5.1: Percent change in lambda from E0 to E20

Observed torque increased at mode 1 with E20 fuel because of a more complete combustion due to the oxygen in the ethanol, as shown in Figure 5.2. Torque at modes 3 and 4 varied due to not maintaining the mode torque value. These torque values were still within a half foot-pound of the testing point torque. Mode 5 was negated from Figure 5.2 because this mode is recorded at engine idle, producing a 0 torque reading.

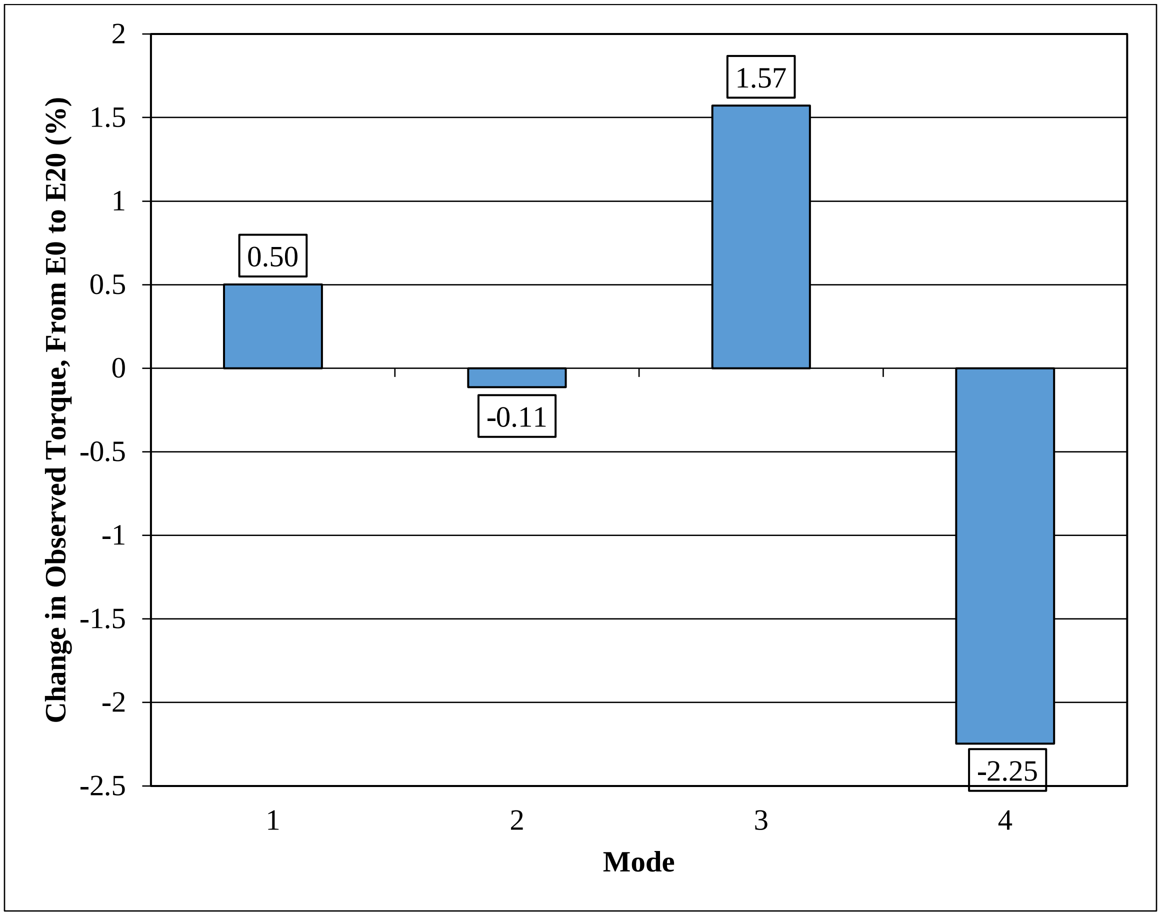

Figure 5.2: Percent change in observed torque from E0 to E20

Mid-pipe temperature was monitored in the exhaust expansion chamber. This temperature was important to monitor because too high of a temperature would ignite any excess fuel in the exhaust causing another ignition in the exhaust system harming the internal components of the muffler. It can be seen in Figure 5.3 below that mid-pipe temperature was higher in Modes 1 and 2 with the E20 fuel, and then produced lower temperature values for modes 3 through 5. Highest mid-pipe temperatures will be produced at stoichiometric air/fuel ratios. Modes 1 and 2 were still fuel rich, but leaner than the E0 mixture shown in Figure 5.1 resulting in higher mid-pipe temperatures. Modes 3 and 4 were leaner that stoichiometric on E0 and thus became even leaner with E20 fuel. This resulted in combustion with excess oxygen slightly reducing exhaust temperatures. Finally, mode 5 had a large reduction in exhaust temperatures with E20 fuel. Ethanol fuel has a much higher latent heat of vaporization than gasoline (840 kJ/kg vs. 350 kJ/kg) resulting in more cooling in the combustion chamber as the fuel evaporates [2]. This initially cooler charge reduces exhaust temperatures at idle.

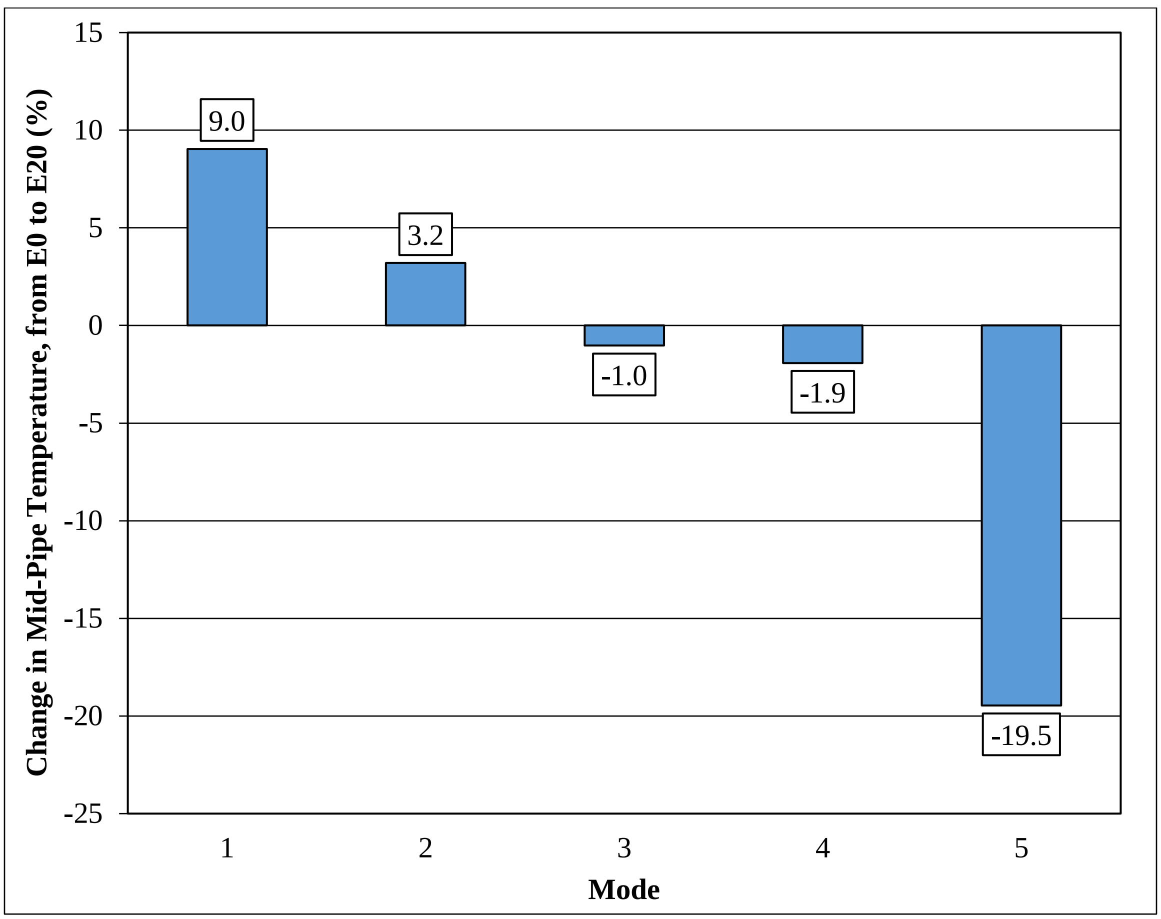

Figure 5.3: Percent change in mid-pipe temperature from E0 to E20

Brake specific fuel consumption (BSFC) is displayed in Figure 5.4. The engine consumed more fuel per power value for all displayed modes except mode 1. This reduction in BSFC is aided by the slight increase in power produced at mode 1. Once again mode 5 was not shown because there is zero power recorded at idle.

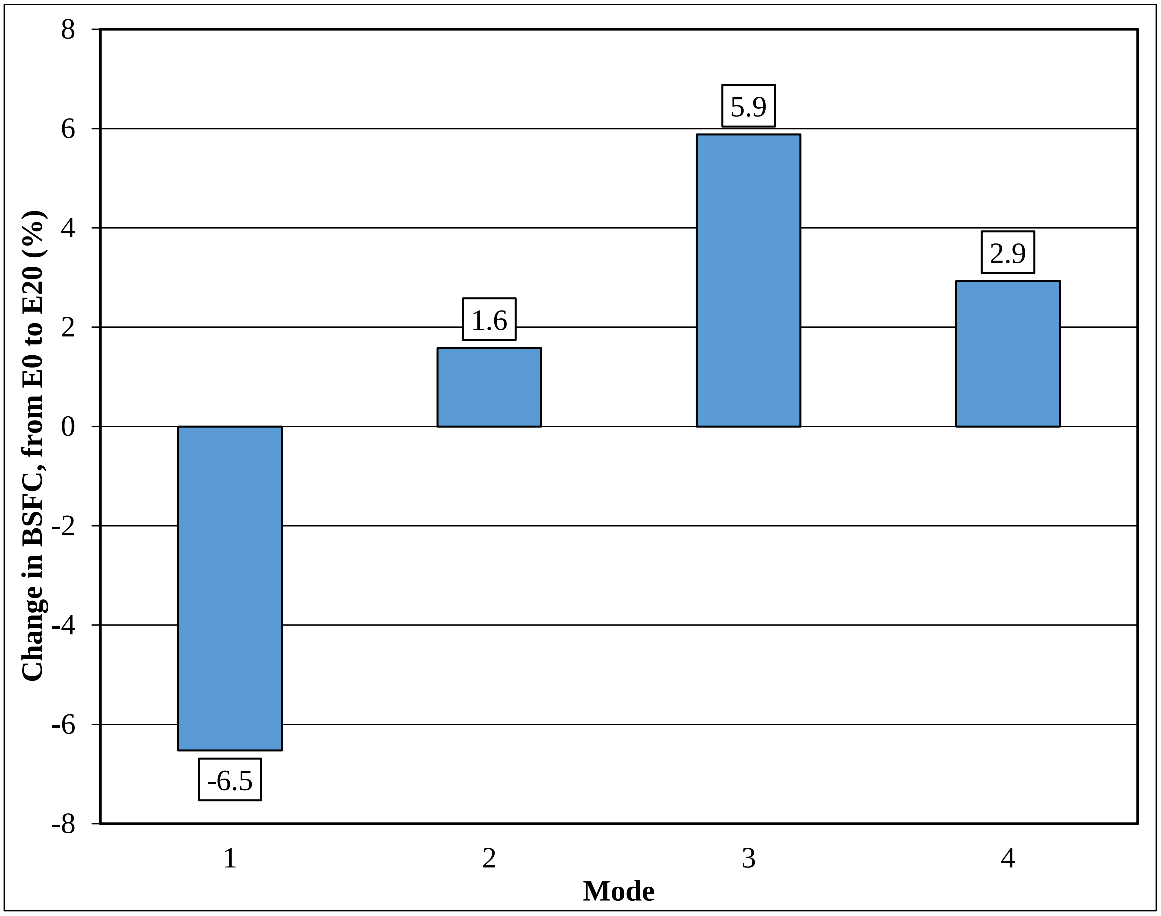

Figure 5.4: Percent change in brake specific fuel consumption from E0 to E20

Carbon monoxide (CO) is one of two regulated exhaust constituents in the snowmobiling industry, along with total hydrocarbons (THC). Carbon monoxide emissions are shown below in Figure 5.5. CO emissions were reduced in all five modes by using E20 fuel, as expected. Carbon monoxide emissions are affected most by air/fuel ratio, where CO is reduced as the mixture becomes leaner [2].

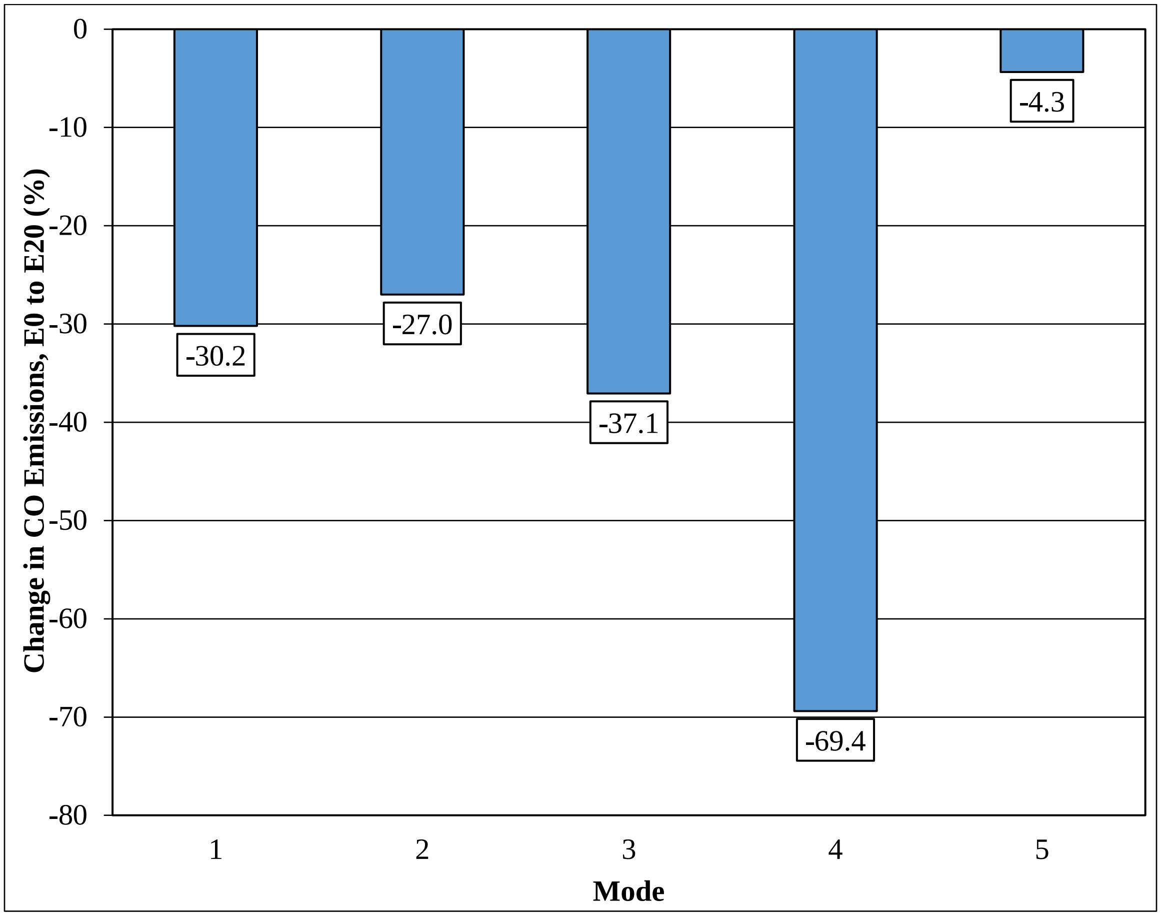

Figure 5.5: Percent change in CO emissions from E0 to E20

Emissions results for carbon dioxide can be seen below in Figure 5.6. CO_2 values usually oppose CO emissions because CO_2 emissions are produced at higher exhaust temperatures which are usually the result of running at a lambda value of 1, or the stoichiometric air/fuel ratio. This figure looks similar to the mid-pipe temperature figure mentioned above.

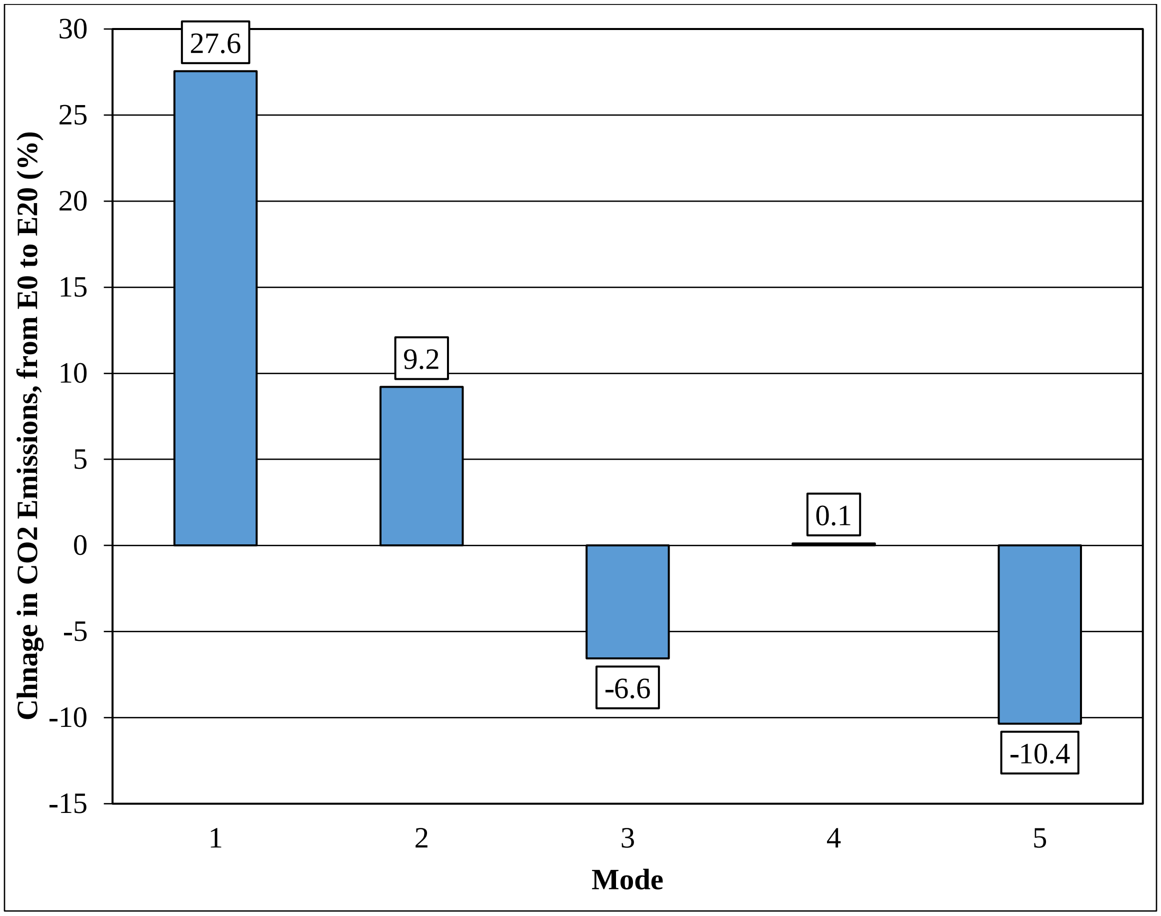

Figure 5.6: Percent change in CO_2 emissions from E0 to E20

Nitrogen monoxide (NO) emissions can be seen below in Figure 5.7. NO emissions are produced mostly in the post flame gasses where there is high temperature and pressure [2]. There were increases in NO emissions in modes 1 and 2. This trend of higher values in modes 1 and 2 is similar to the mid-pipe temperature trend in Figure 5.3 where there were higher temperatures noted for E20.

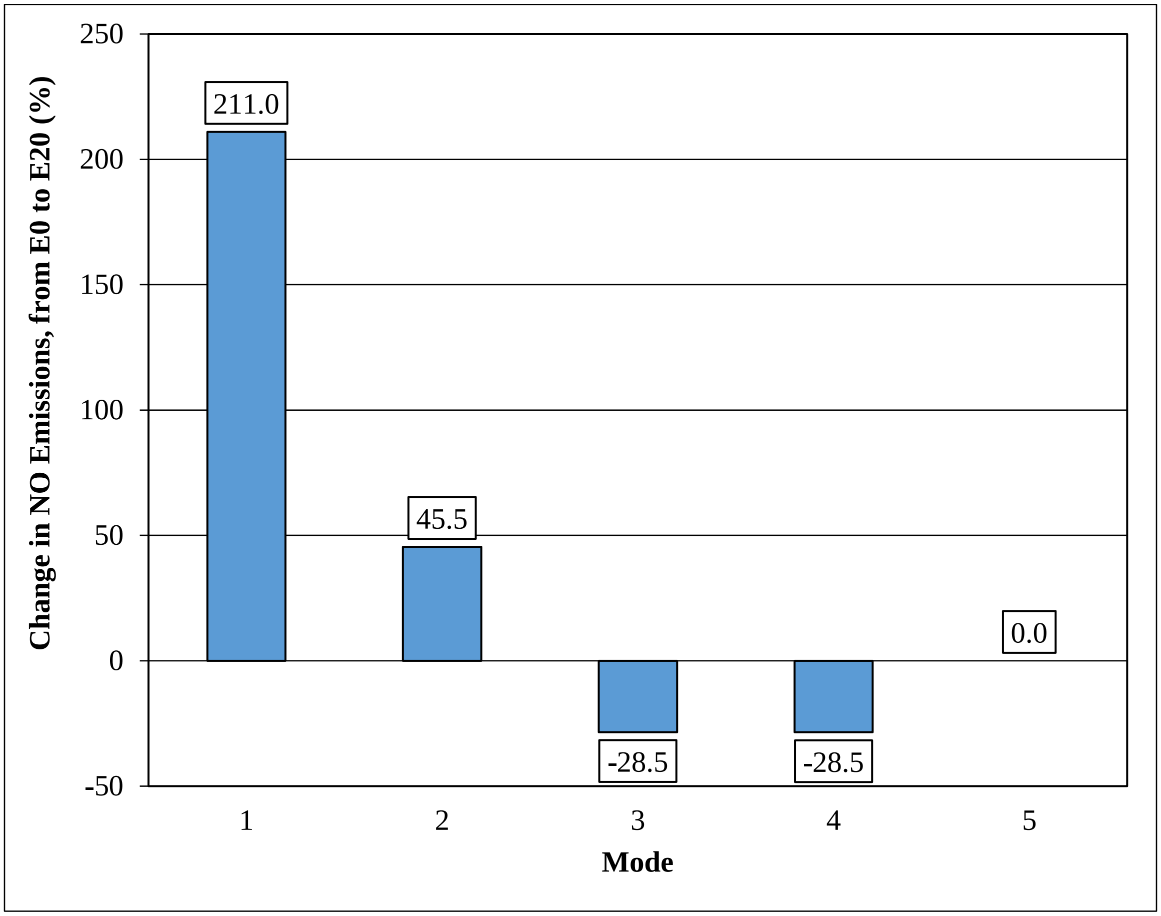

Figure 5.7: Percent change in NO emissions from E0 and E20 fuels

Total hydrocarbon (THC) emissions can be high in two-stroke engines because of the requirement of injecting oil to maintain proper lubrication in the rotating assembly and piston rings. Hydrocarbon emissions come from incomplete combustion, so having a richer air/fuel mixture will result in excess fuel in the combustion chamber not being completely combusted [2]. Running a leaner mixture aids in more complete combustion [1] and reduces THC emissions. This trend can be seen in Figure 5.8 below.

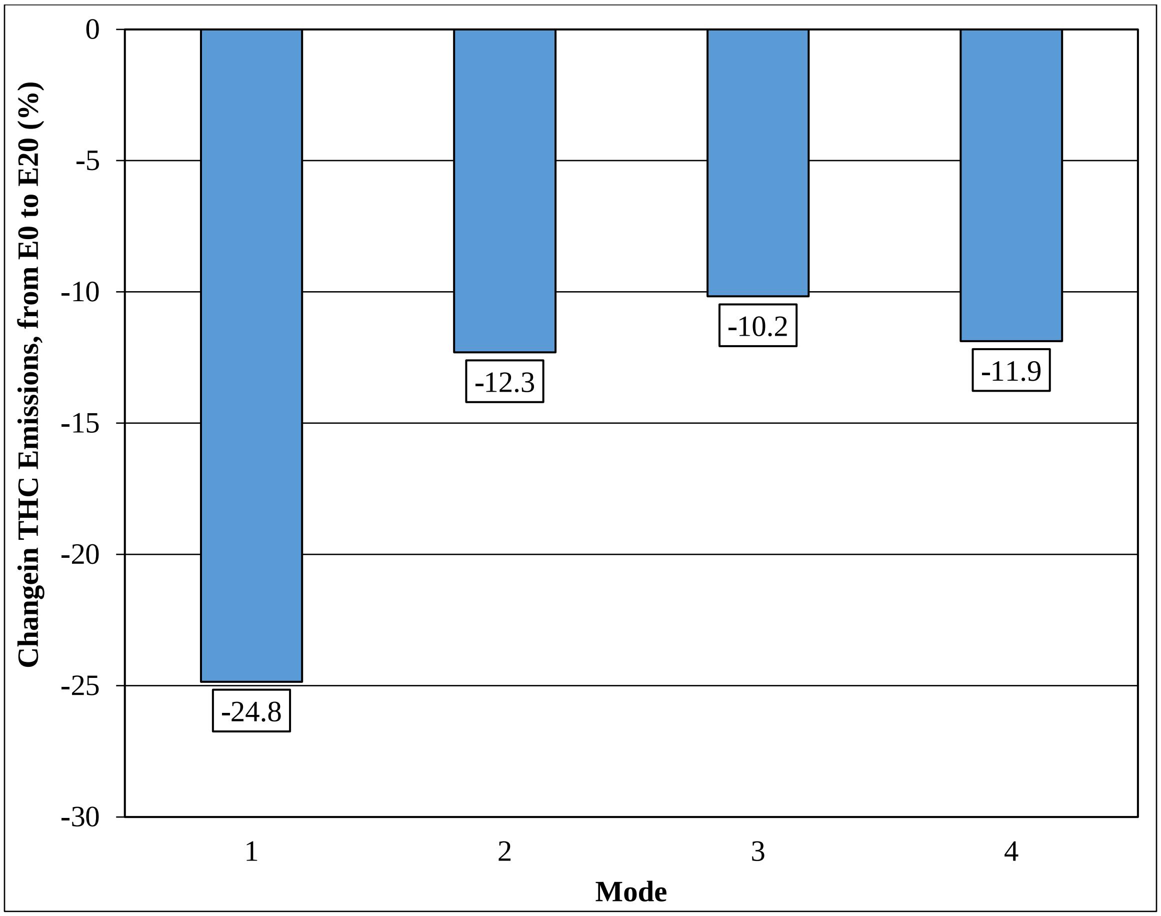

Figure 5.8: Percent change in THC emissions from E0 to E20

There is little emissions research that looks into aldehyde emissions. These emissions are known carcinogens and are harmful to humans and the environment [1]. Both formaldehyde and acetaldehyde are formed when hydrocarbon chains are bonded with oxygen. Formaldehyde and acetaldehyde emissions are displayed in this report in Figure 5.9 and Figure 5.10. It can be seen that there is an increase in both emissions for all modes when using E20 fuel because of the excess oxygen in the fuel.

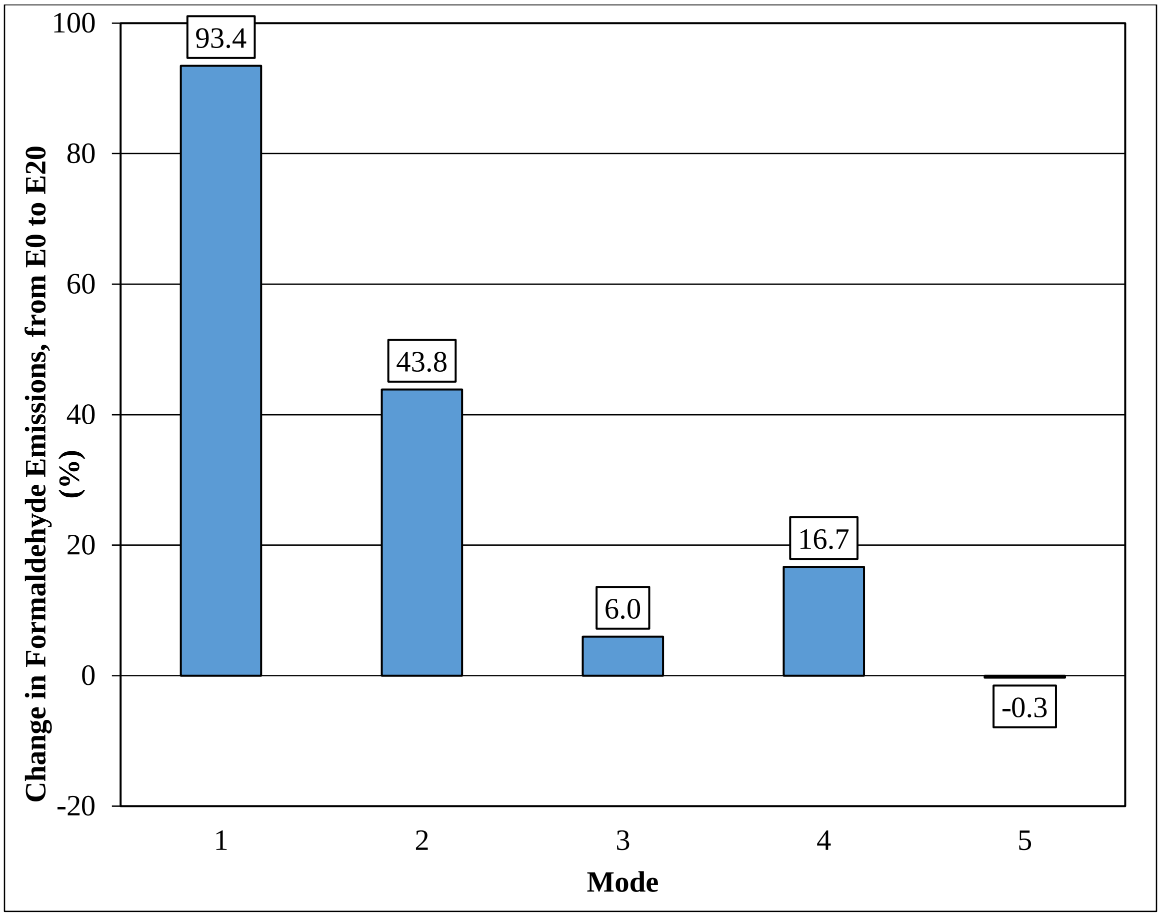

Figure 5.9: Percent change in formaldehyde emissions from E0 to E20

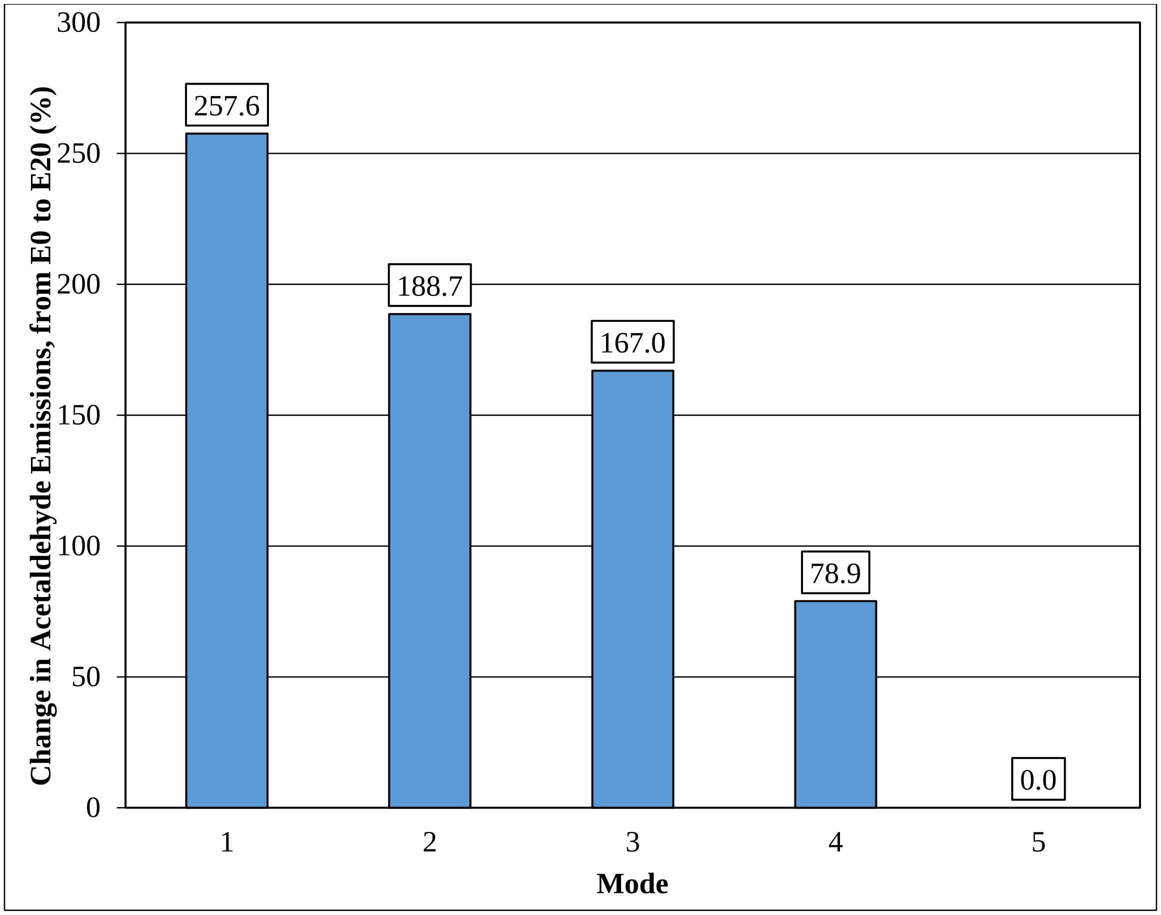

Figure 5.10: Percent change in acetaldehyde emissions from E0 to E20

Combustion equipment utilized pressure recordings over 360 degrees of crank rotation to compute combustion variability, or Coefficient of Variation in Indicated Mean Effective Pressure (COV of IMEP). A more stable combustion will have a low COV in IMEP value. Figure 5.11 shows the E20 fuel having a more stable combustion in modes one and two. Higher octane fuel is more resistive to knocking, reducing the chances of uncontrolled auto ignition produced by high temperatures and pressures.

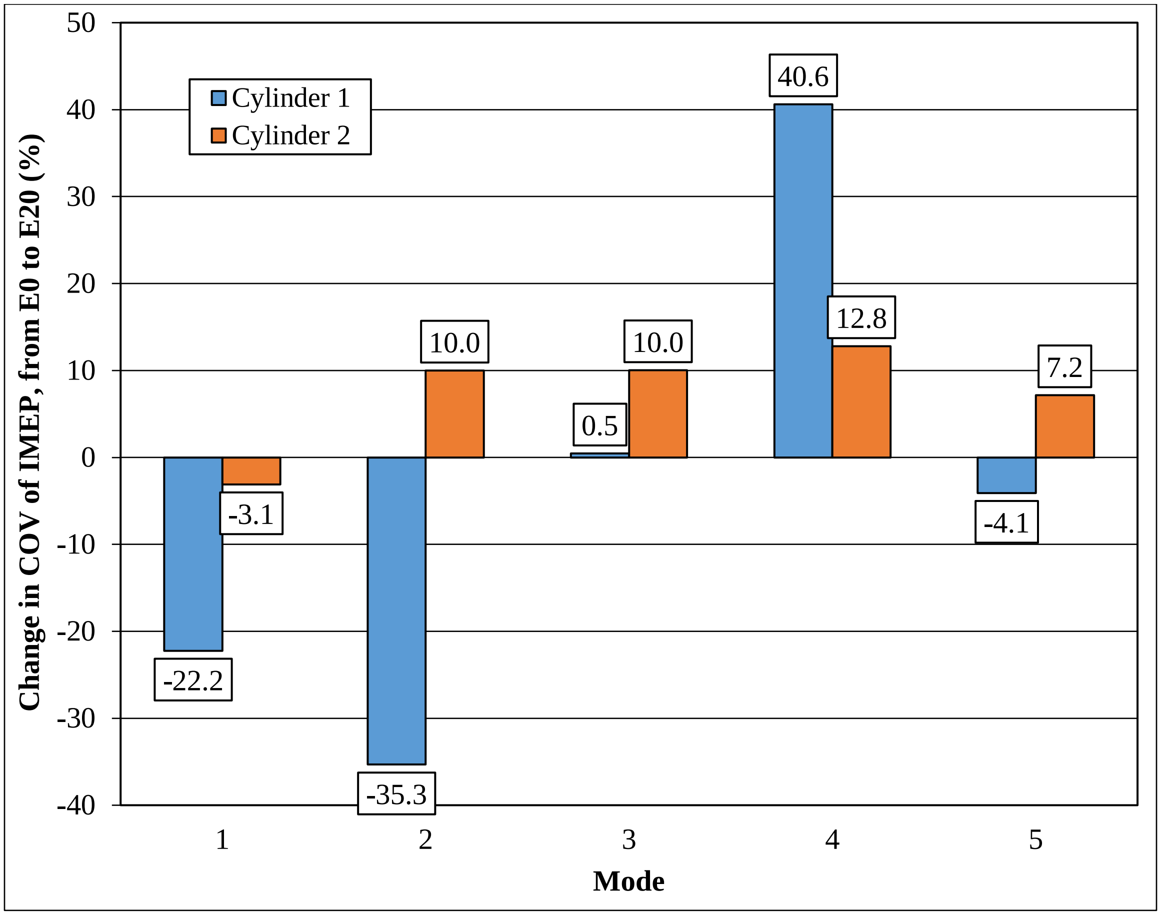

Figure 5.11: Percent change in CoV in IMEP from E0 to E20

Burn duration is calculated by looking at in-cylinder pressures over the range of crank angle degrees in the combustion and expansion stroke. Figure 5.12 shows E20 having a longer burn duration than E0. This is due to the ECU advancing timing for E20 fuel to reduce mid-pipe temperatures, and ethanol having a more complete combustion resulting in a longer burn than E0.

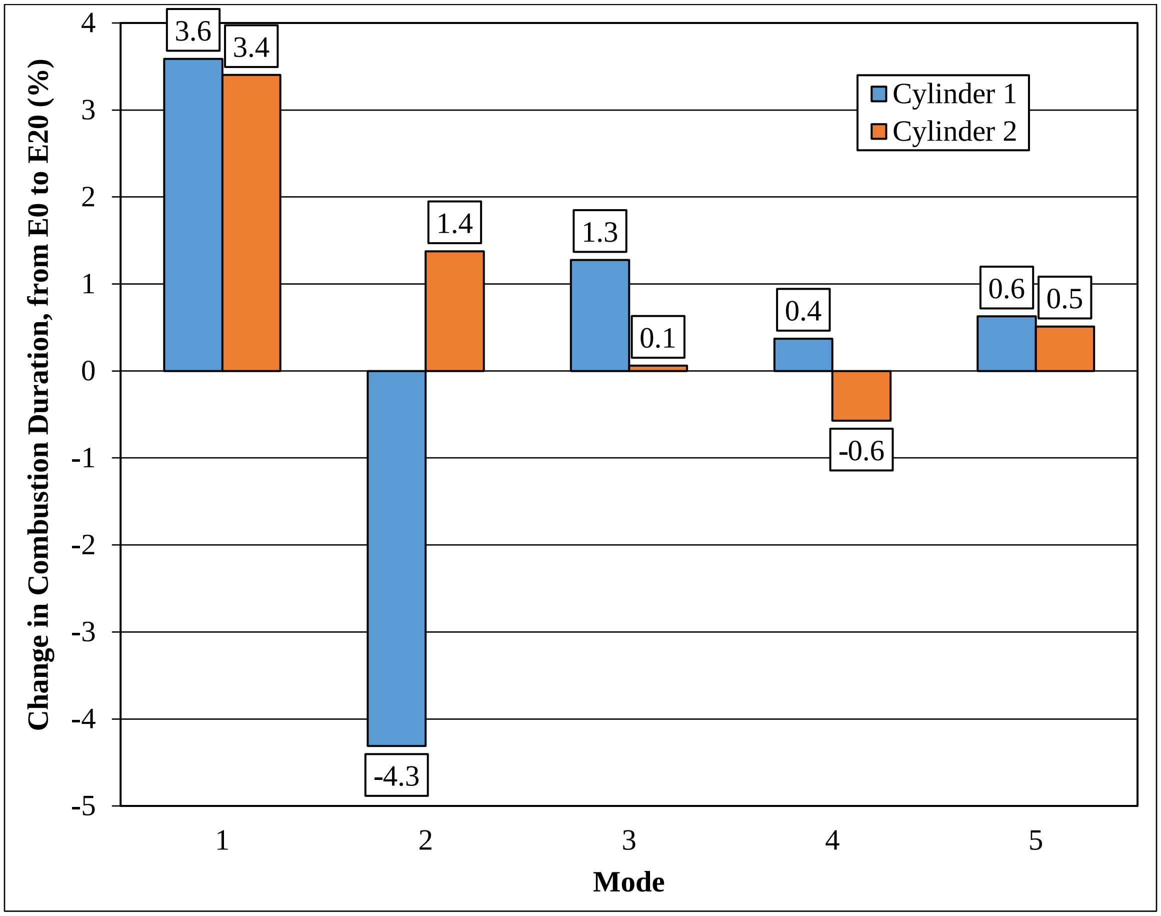

Figure 5.12: Percent change in combustion duration from E0 to E20

5.2 Single Parameter E20 Sweeps

The second part of this project involved performing single parameter sweeps at each of the 5 modes while running the engine on E20 fuel. There were three parameters, or major "levers", that were adjusted in the ECU. These parameters included injection time, injection end angle, and ignition timing. Injection time is defined as how long the fuel injector was held open, in milliseconds, allowing fuel to flow into the cylinder. Injection end angle was the point when the fuel injector closes which corresponds with degrees of crankshaft rotation after top dead center. This engine has a unique piston design where an open slot is located in the piston skirt allowing the fuel injector to inject fuel through the piston to the bottom end for additional lubrication at high RPMs. For large advances in injection end angle, the fuel injector is injecting while the piston is moving down past the injector, on top of the piston, and then while the piston is moving up towards top dead center. For large reductions in injection end angle, the fuel injector is mostly injecting through the piston skirt and on piston rings as the piston moves from top dead center to bottom dead center and then finishing the injection on the top of the piston. Due to the unique piston and injection design, injection end angle emissions trends were difficult to explain and would require more research to do so. Ignition timing was the degree in crank rotation before top dead center when current is sent to the spark plug to ignite the air/fuel mixture in the cylinder. One parameter was swept over multiple values while maintaining the other two constant to see the effect that each parameter had on engine performance, emissions, and combustion.

The sweeps were conducted in all modes, but mode 1 data is displayed in this report because it shows the most prominent change in the results. The swept parameter range was not always met at mode 1 because of too high of exhaust temperatures due to enleanment, poor combustion causing misfire, or the observed torque value was too low causing the test to be stopped.

An anomaly can be seen in the single parameter sweep figures at an injection time of -8 percent. At this percent injection time, the air/fuel mixture was too lean not allowing proper combustion resulting in misfire. This hypothesis is supported by an increase in COV of IMEP from the baseline calibration at -8 percent injection time shown in Figure 5.13 along with the emissions results not following fuel lean trends.

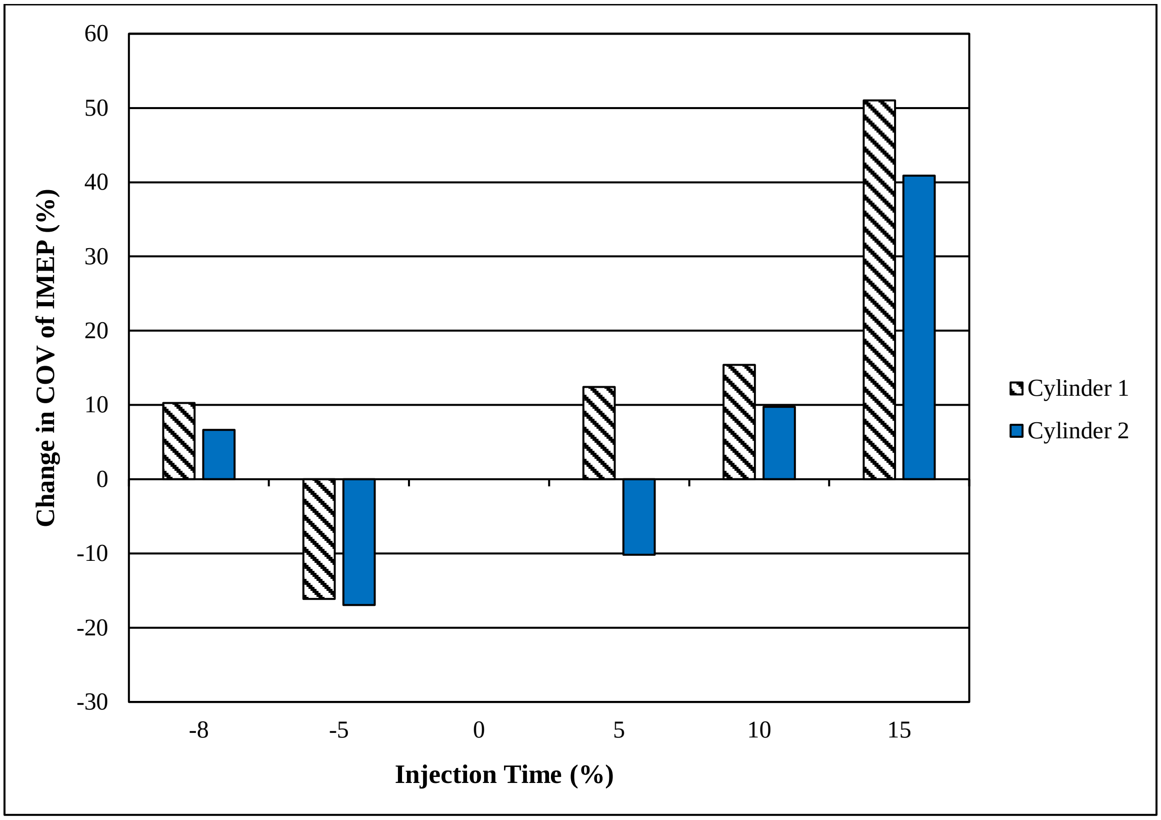

Figure 5.13: Percent change in COV of IMEP from stock injection time

Lambda is most affected by the injection time as shown in Figure 5.14. Because lambda is a ratio of the actual air/fuel mixture divided by the stoichiometric air/fuel mixture, adding and taking away fuel shows the most change of all the sweeps. The decrease in lambda at -8 percent injection time is due to misfire. Fuel that was not completely combusted due to misfire is left over and combusted again during the next cycle. This leftover fuel adds to the next incoming charge making it a more fuel rich mixture.

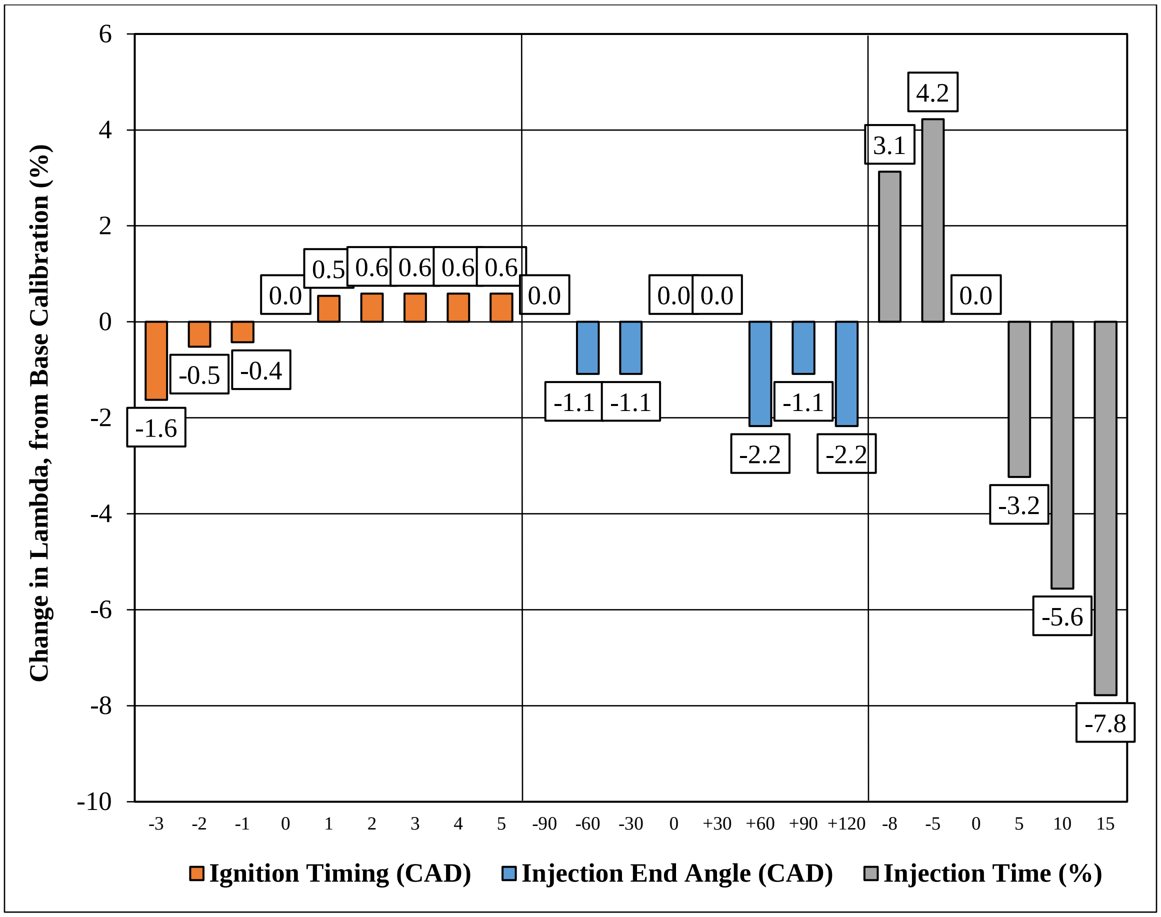

Figure 5.14: Percent change in lambda for the single parameter sweeps of ignition timing, injection end angle, and injection time

Engine torque was affected by both the injection time sweep and ignition timing sweep. Figure 5.15 shows that by increasing or decreasing the OEM calibration values decreased the observed torque value. Over the duration of the test, intake temperatures did rise which can lead to lower observed torque values. All intake temperatures were within the required temperature of 25°C plus or minus 5°C otherwise the test would be rejected.

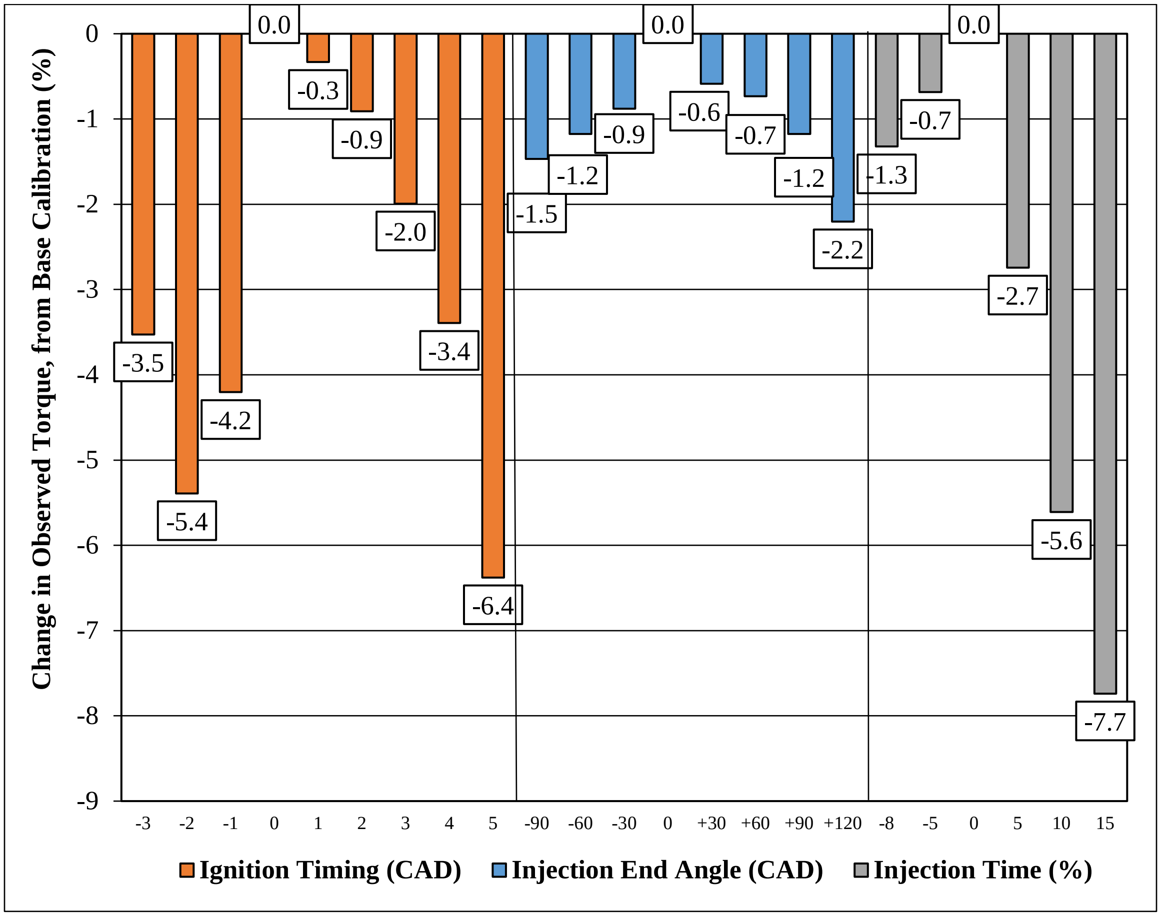

Figure 5.15: Percent change in observed torque for the single parameter sweeps of ignition timing, injection end angle, and injection time

Similar to torque, mid-pipe temperature was most affected by sweeping the injection time and ignition timing. Figure 5.16 below shows these similar trends. A two-stroke engine does not contain valves, allowing a small portion of the incoming air/fuel mixture to pass through the cylinder and out the exhaust port making the mid-pipe temperature affected by both the unburned fuel after combustion and the new air/fuel mixture coming into the cylinder. A richer air/fuel mixture will let more raw fuel into the exhaust cooling it as it evaporates. Evaporating fuel also cools the combustion chamber resulting in lower combustion temperatures below stoichiometric conditions. A leaner but still fuel rich mixture, which is seen at mode 1, increases the mid-pipe temperature because the highest combustion temperatures are seen at the stoichiometric air/fuel ratio. Retarding the ignition timing results in a later combustion which results in the hot combustion gases to exit into the exhaust thus increasing the temperature.

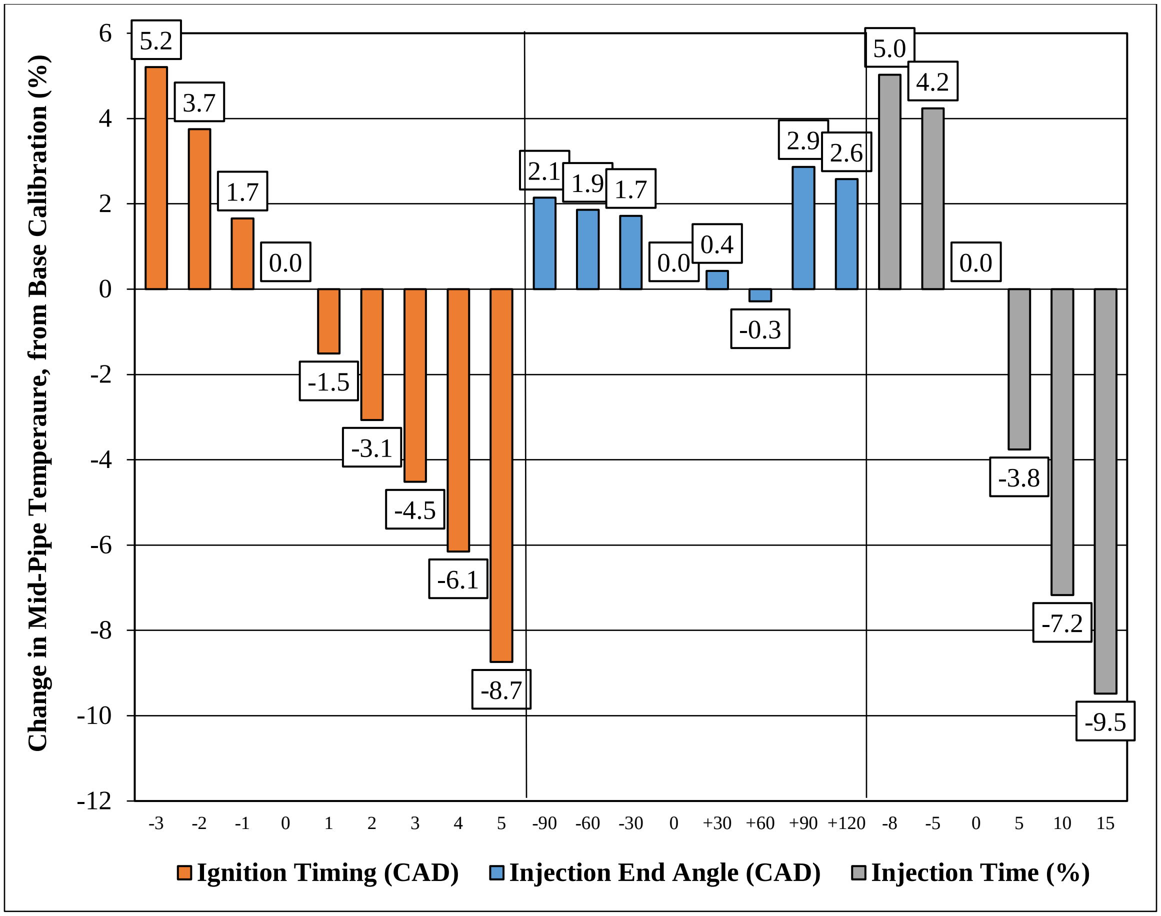

Figure 5.16: Percent change in mid-pipe temperature for the single parameter sweeps of ignition timing, injection end angle, and injection time

Brake specific fuel consumption is affected by the fuel consumed by the engine and the output power; with a lower value being beneficial for fuel economy. Increasing the injection time increases the fuel consumed raising the BSFC value, which is shown in Figure 5.17 when sweeping the injection time parameter. Changing the ignition timing resulted in reduced observed torque, which also increased the BSFC.

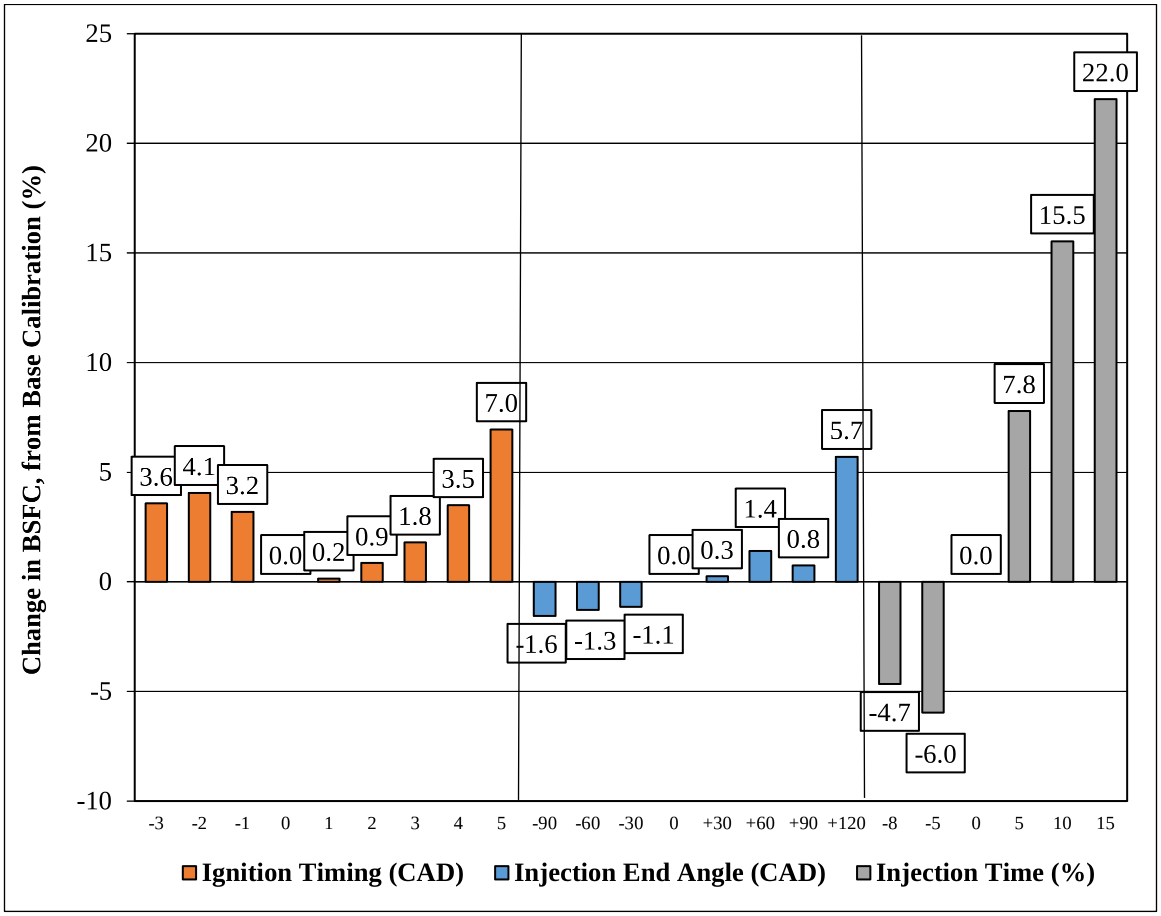

Figure 5.17: Percent change in brake specific fuel consumption for the single parameter sweeps of ignition timing, injection end angle, and injection time

Running single parameter sweeps helps to show how each emissions constituent are affected by the changing engine parameters. Carbon monoxide emissions are mostly affected by air/fuel ratio. This is backed up by the data displayed in Figure 5.18 showing injection time having the largest effect on CO emissions. Injection end angle and ignition timing do not show signs of significantly affecting CO emissions.

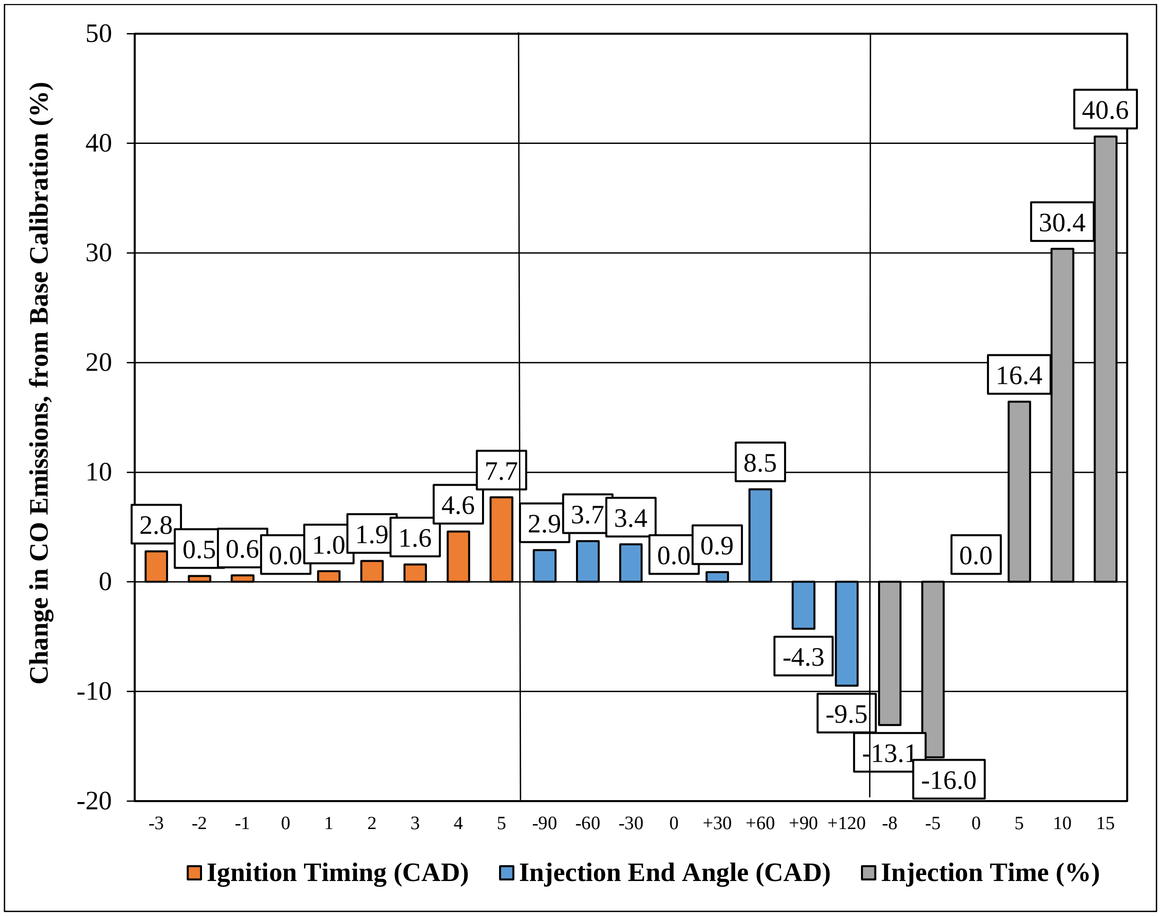

Figure 5.18: Percent change in carbon monoxide emissions for the single parameter sweeps of ignition timing, injection end angle, and injection time

Carbon dioxide emissions are most affected by the injection time parameter sweep; CO_2 requires high temperatures and excess oxygen to form, that result from a more fuel-lean mixture. Figure 5.19 shows that advancing the ignition timing can reduce CO_2 emissions, while retarding ignition timing increases CO_2 emissions. Adjusting injection time and ignition timing also greatly affected mid-pipe temperature which contributed to CO_2 formation.

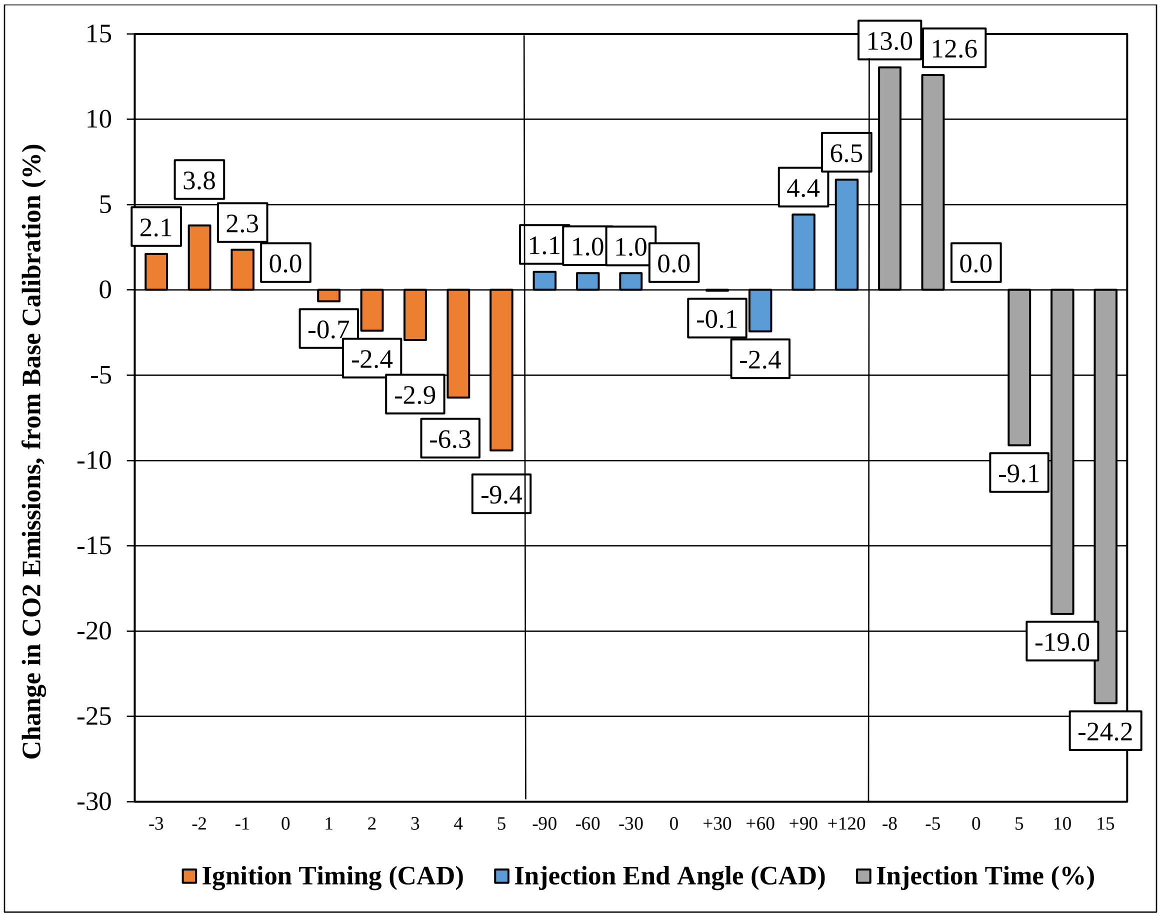

Figure 5.19: Percent change in carbon dioxide emissions for the single parameter sweeps of ignition timing, injection end angle, and injection time

Nitrogen oxide emissions are most affected by the fueling shown in Figure 5.20, where the injection time parameter sweep has the largest change in percentage. Increasing the amount of fuel in the air/fuel mixture decreases NO emissions, and decreasing the amount of fuel creating a leaner air/fuel mixture increases NO emissions. Increasing or decreasing ignition timing reduces peak cylinder pressures where NO emissions are produced. Combustion instability was seen at -8 ms of ignition timing resulting in unstable combustion and a reduction in NO emissions.

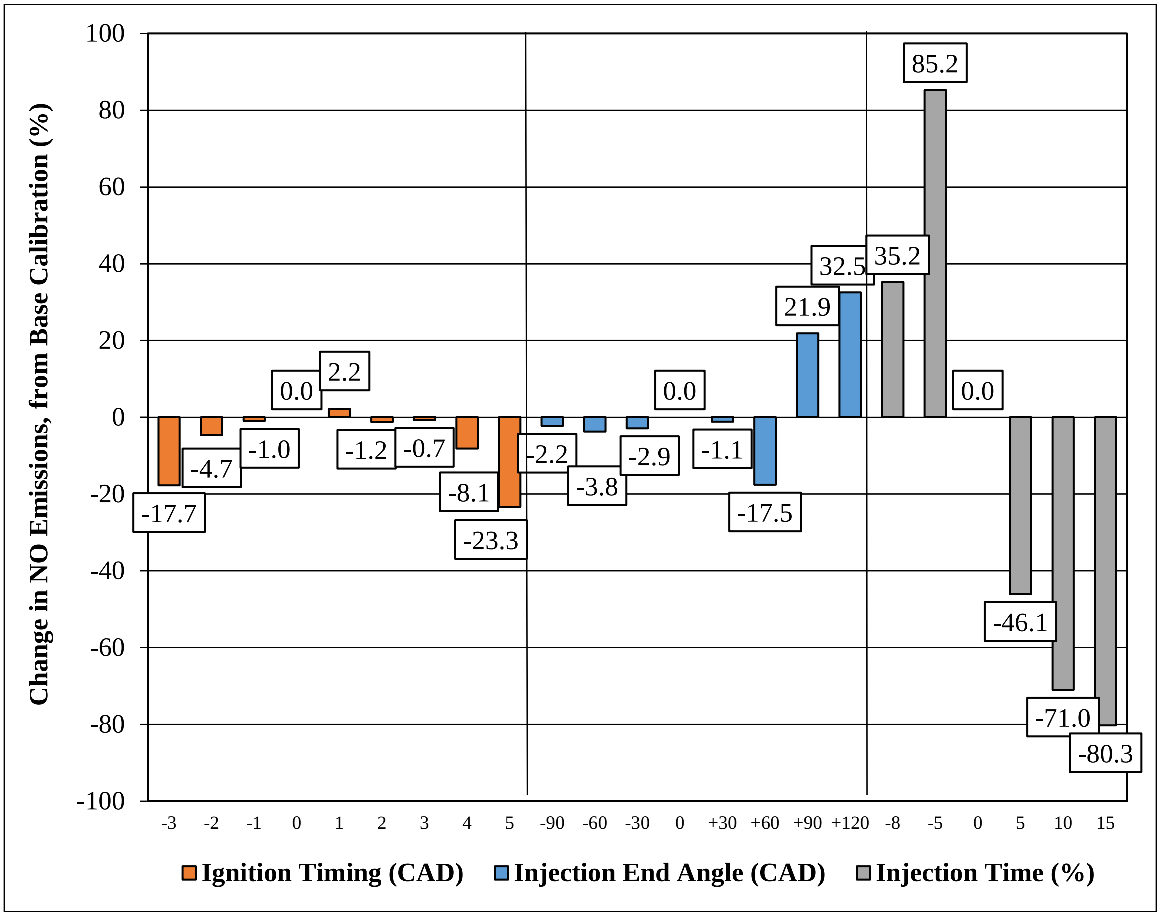

Figure 5.20: Percent change in nitrogen oxide emissions for the single parameter sweeps of ignition timing, injection end angle, and injection time

Total hydrocarbon emissions are categorized as unburned fuel in the exhaust. Excess unburned fuel is the result of increasing the injection time creating a fuel rich mixture or by advancing the ignition timing (in mode 1) resulting in incomplete combustion and excess unburned fuel. Shortening the window for more complete combustion by retarding ignition timing (in mode 1) or taking away fuel reduces THC emissions. These trends can be seen below in Figure 5.21.

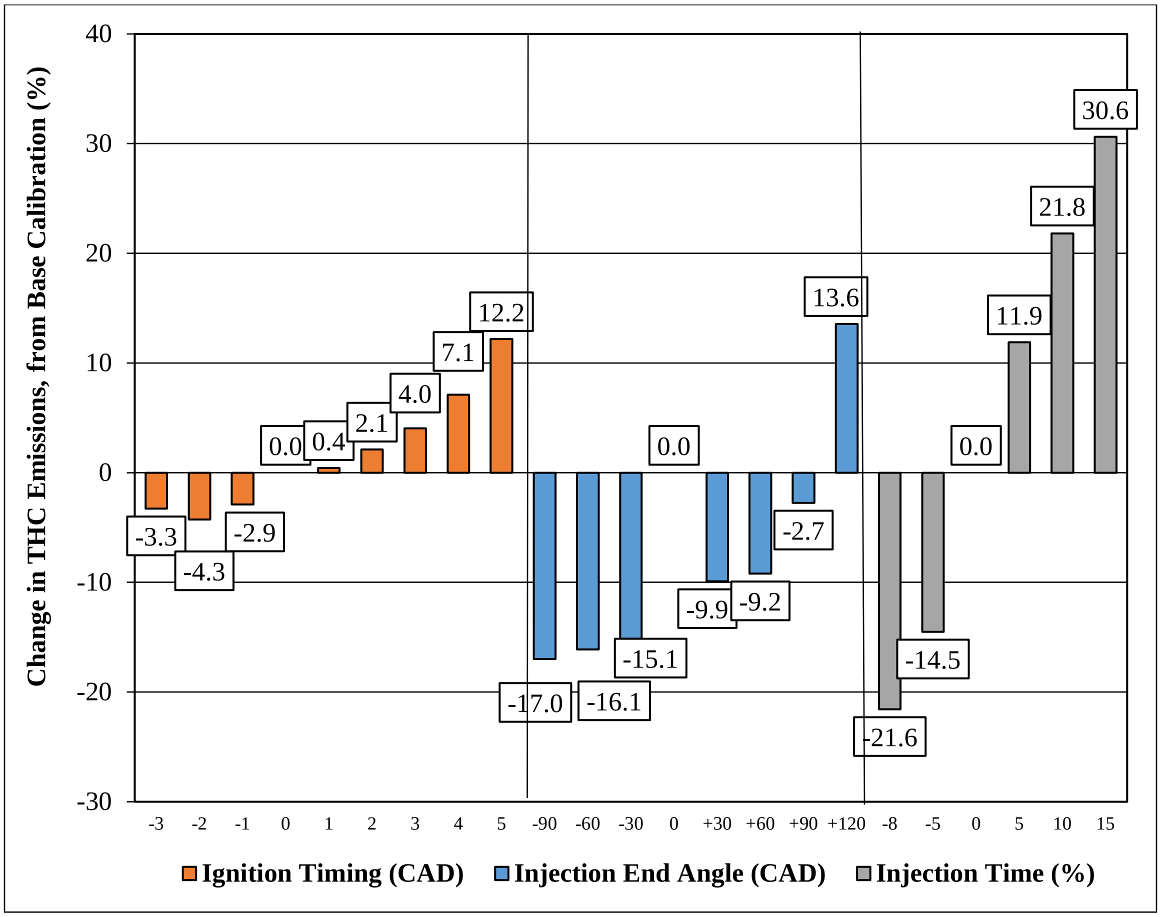

Figure 5.21: Percent change in total hydrocarbon emissions for the single parameter sweeps of ignition timing, injection end angle, and injection time

Formaldehyde emissions show similar trends to the mid-pipe temperature results. Figure 5.22 shows that adding fuel or advancing the timing reduces formaldehyde emissions, while taking away fuel or retarding timing increases formaldehyde emissions.

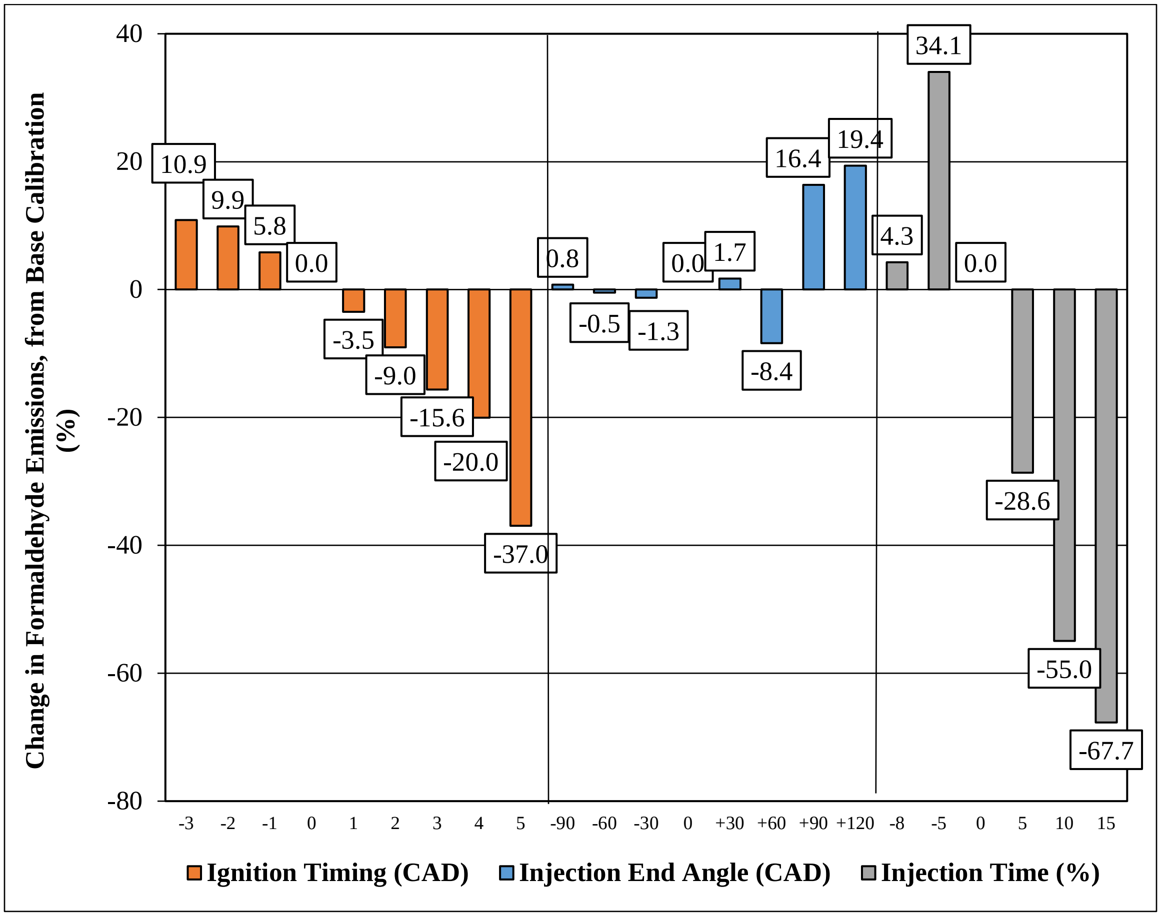

Figure 5.22: Percent change in formaldehyde emissions for the single parameter sweeps of ignition timing, injection end angle, and injection time

It can be seen below in Figure 5.23 that acetaldehyde emissions acted similarly to formaldehyde emissions.

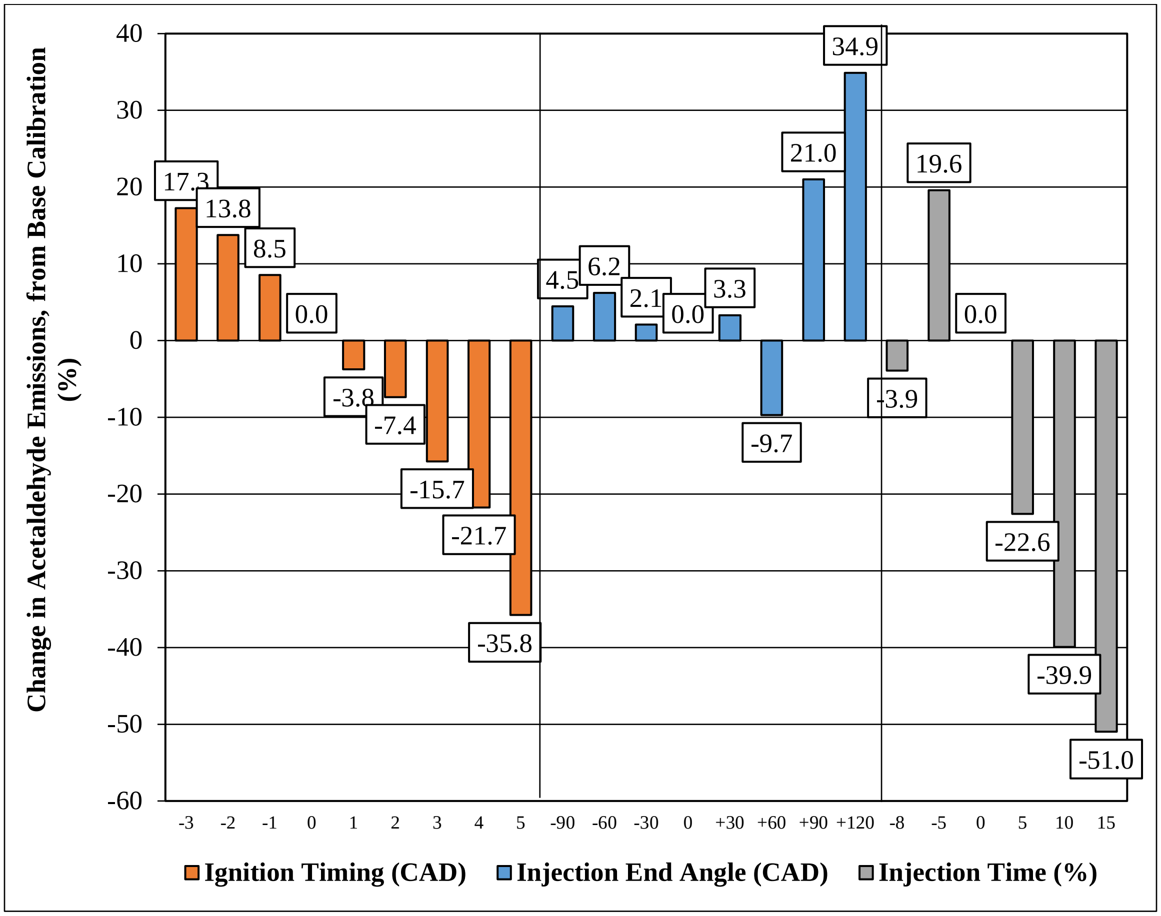

Figure 5.23: Percent change in acetaldehyde emissions for the single parameter sweeps of ignition timing, injection end angle, and injection time

5.3 Matching E0 Lambda values using E20 Fuel

One objective of this project was to determine how E20 fuel at E0 lambda baseline values would affect performance and emissions for each mode. In order to do this, recalibration at each mode was required. Lambda is affected by the amount of fuel and air entering the combustion chamber, so only injection time was increased to match the E0 lambda values. Table 2 below shows the percent increase in fuel required to meet E0 lambda values. All other parameters were held constant and controlled by the ECU.

Recalibration included increasing the injection time to create a richer air/fuel mixture for the E20 fuel. Modes 1, 2 and 5 remained fuel rich of stoichiometric and modes 3 and 4 remained fuel lean of stoichiometric. Figure 5.24 below shows that the E20 recalibration test lambda values were within 1 percent of the baseline E0 test.

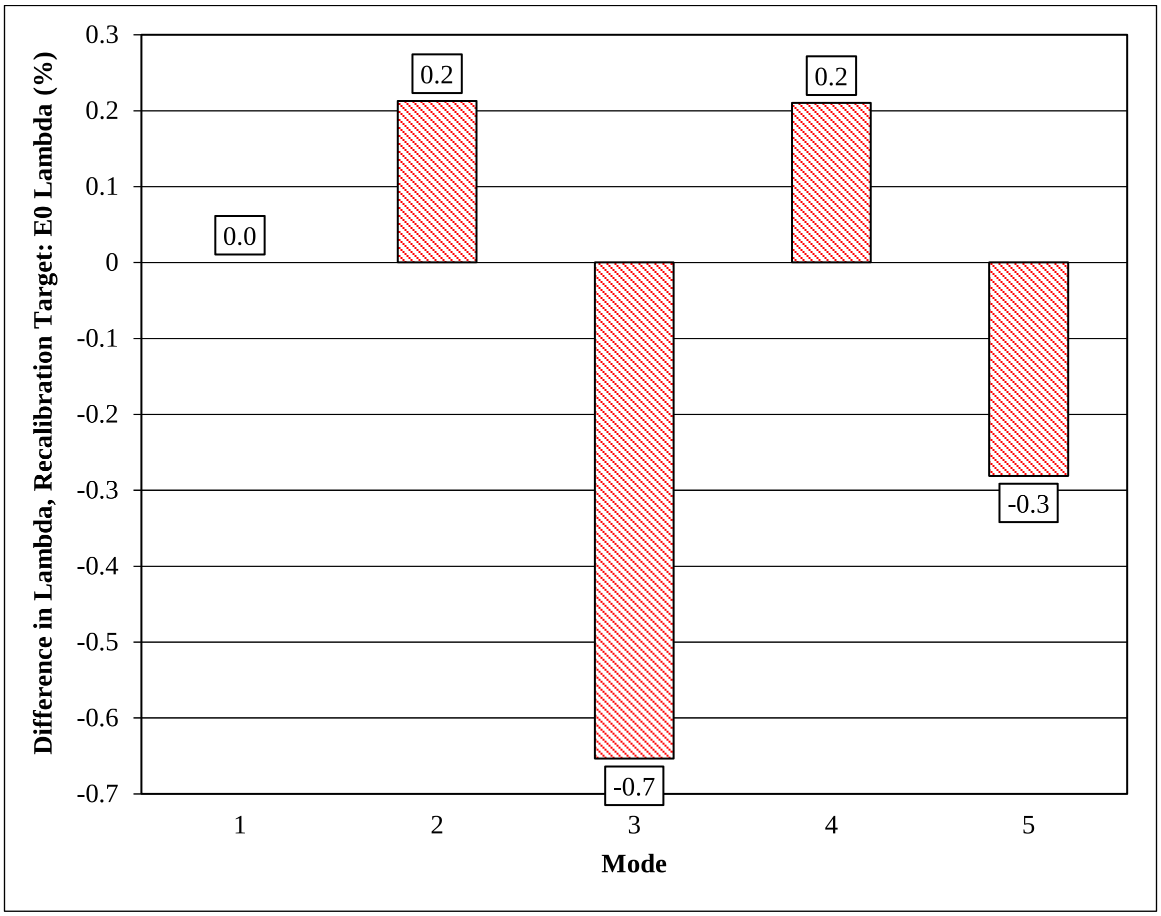

Figure 5.24: Difference in lambda values at each mode for the recalibration test targeting E0 lambda

An increase in observed torque was seen at mode 1 after recalibration involving the addition of more fuel to reduce lambda. Modes 2 through 4 maintained torque values within 0.5 ft-lbs. Mode 5 was neglected from Figure 5.25 because mode 5 is at engine idle where no torque is produced.

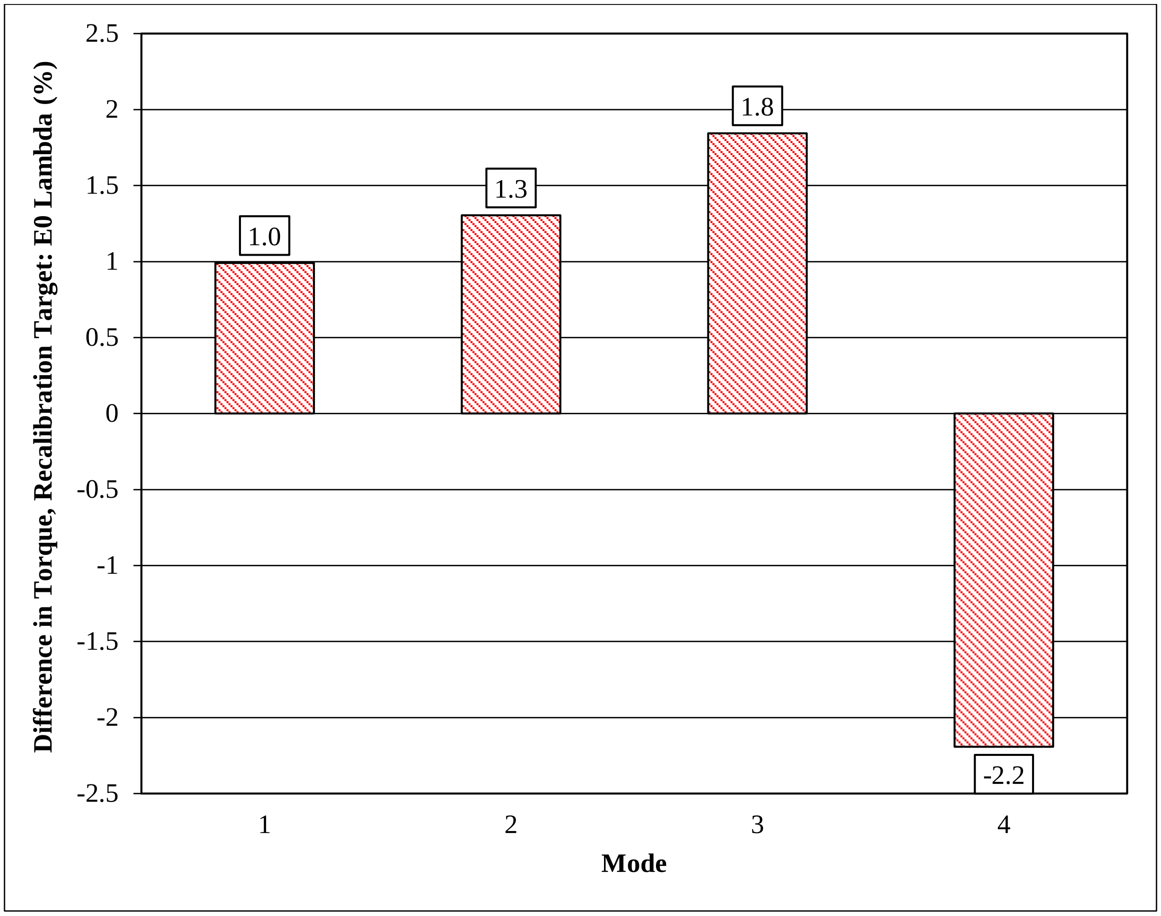

Figure 5.25: Difference in observed torque values at each mode for the recalibration test targeting E0 lambda

The addition of fuel reduced mid-pipe temperatures when compared to the percent difference values when no recalibration for E20 was performed. The utilization of injecting excess fuel to cool the exhaust is seen below in Figure 5.26. Keep in mind the mixtures at mode 3 and 4 were still leaner than stoich even though more fuel was added.

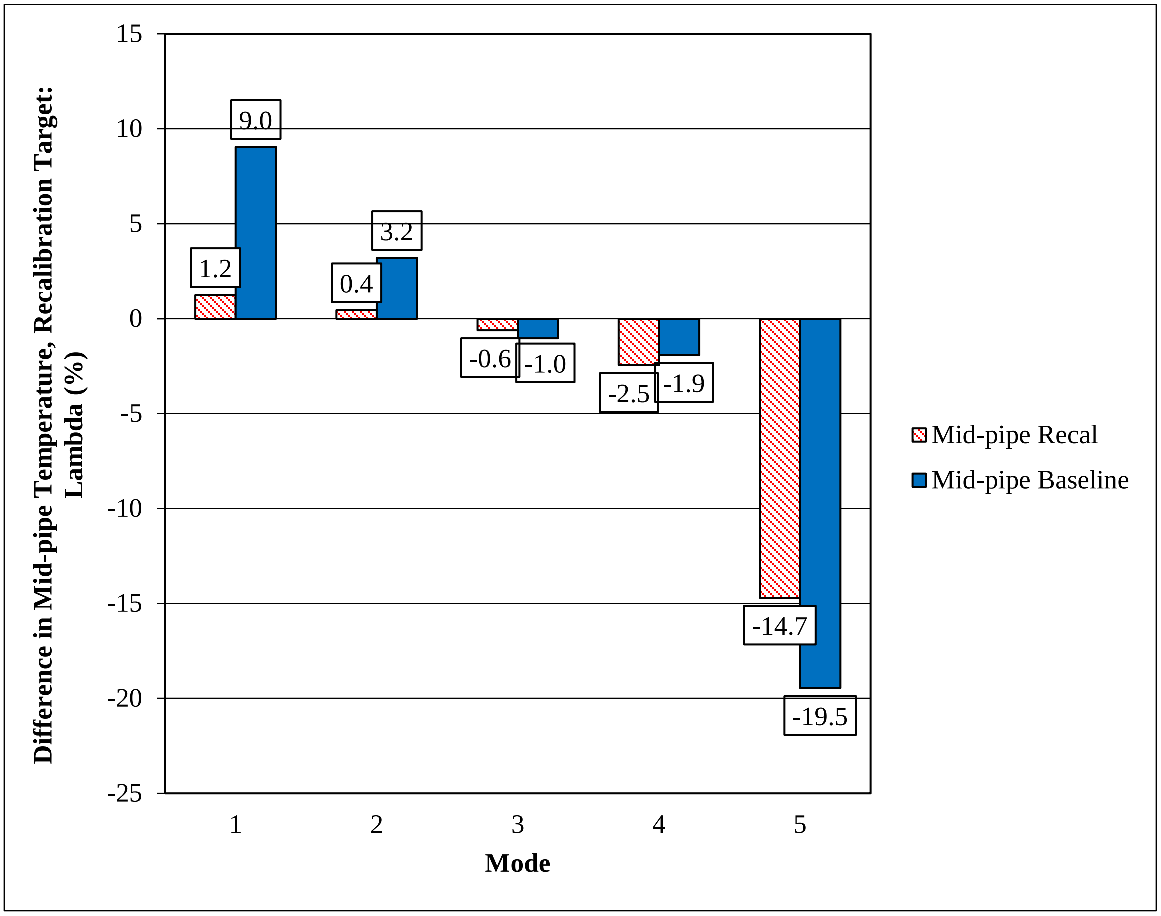

Figure 5.26: Difference in mid-pipe temperature values at each mode for the recalibration test targeting E0 lambda

Brake specific fuel consumption values increased for all modes except mode 2. An increase in torque along with a lower fuel flow value at mode 2 resulted in the outlier. With the addition of more fuel to reduce lambda, fuel flow values increased resulting in increases in BSFC. Figure 5.27 below shows the BSFC trends, along with an outlier at mode 2.

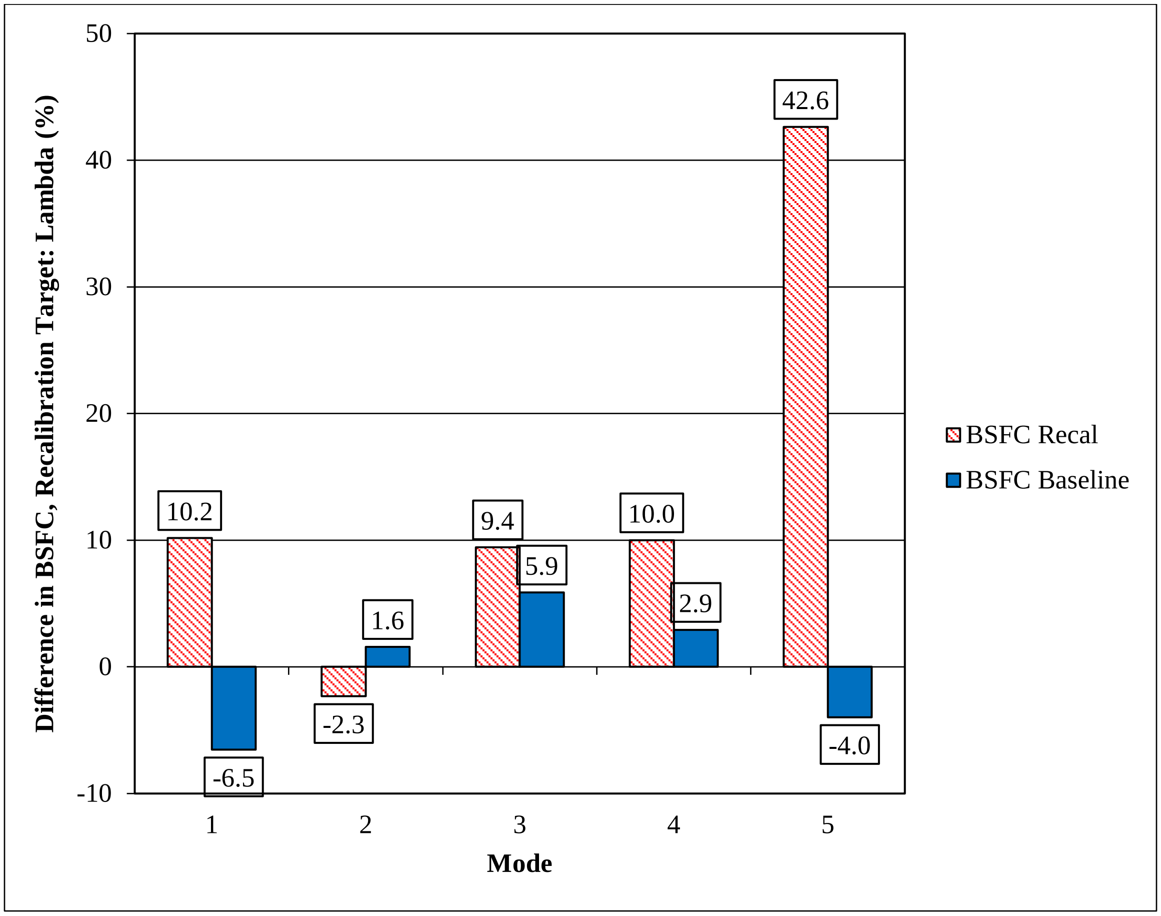

Figure 5.27: Difference in brake specific fuel consumption values at each mode for the recalibration test targeting E0 lambda

Carbon monoxide emissions are controlled by the air fuel ratio; adding more fuel will increase CO emissions. Figure 5.28 shows that modes 3 and 5 using the E20 recalibration produced more CO emissions than the baseline E0 test. CO emissions were still reduced for modes 1, 2 and 4 even with the addition of fuel. When comparing the E20 baseline test to the recalibration E20 test, the recalibrated E20 test produced higher CO emissions for all modes.

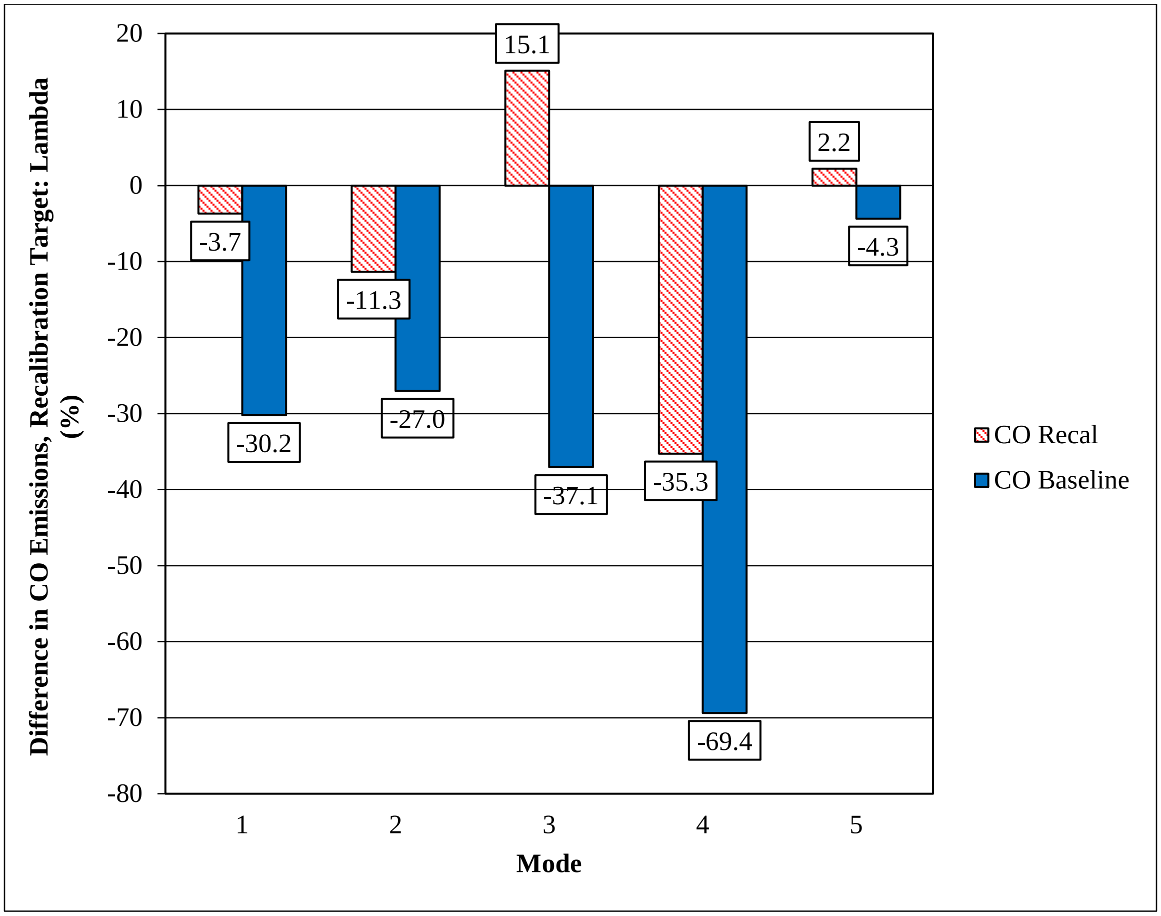

Figure 5.28: Difference in carbon monoxide values at each mode for the recalibration test targeting E0 lambda

With the addition of fuel to meet E0 lambda values, CO_2 emissions were reduced by about 3 times the amount from the E20 baseline test for mode 1. There was a slight increase in CO_2 emissions for modes 2-4 seen in Figure 5.29.

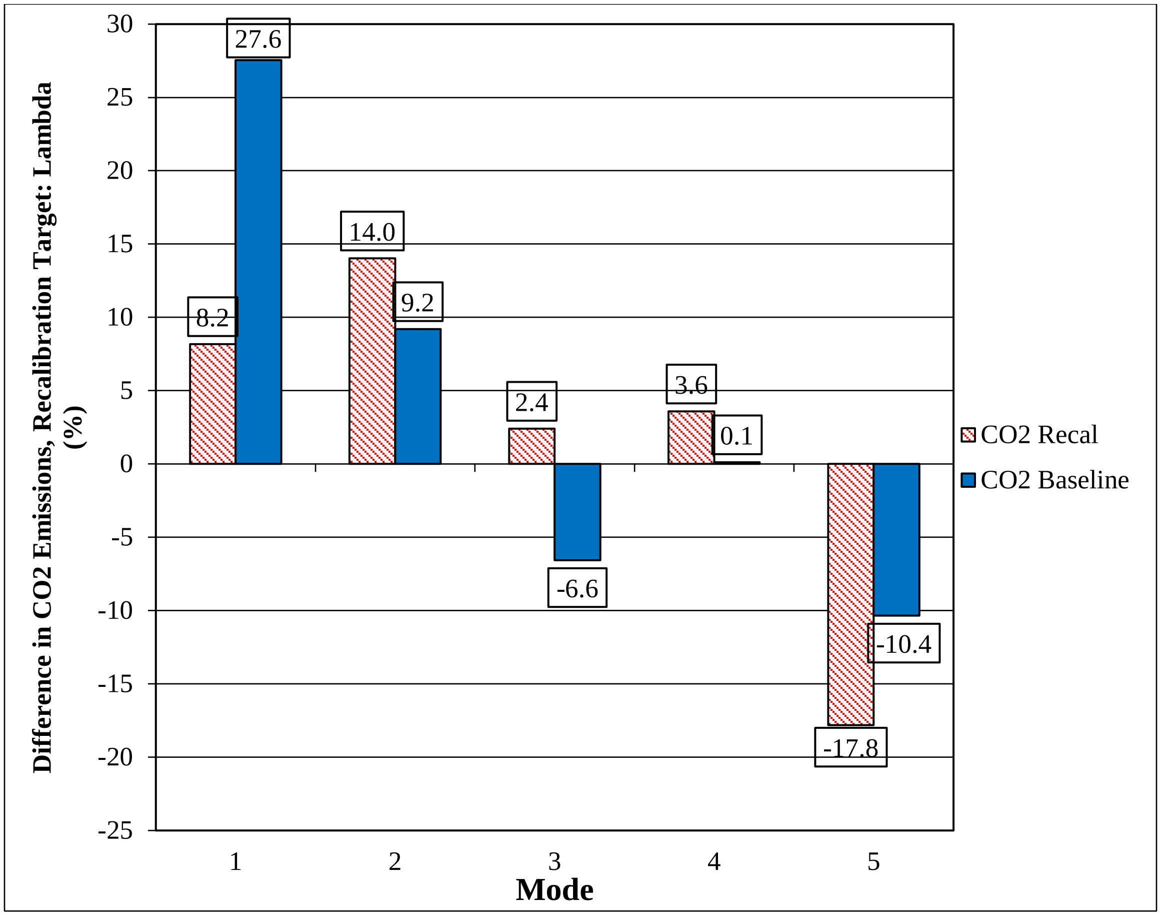

Figure 5.29: Difference in carbon dioxide values at each mode for the recalibration test targeting E0 lambda

Similar to CO_2 emissions, there was a reduction of NO emissions at mode 1 compared to the baseline test. This trend was almost a 90 percent decrease in NO emissions and can be seen below in Figure 5.30. Modes 2-4 saw an increase in NO emissions compared to the baseline E20 test because the addition of fuel brought the air/fuel mixture closer to stoich.

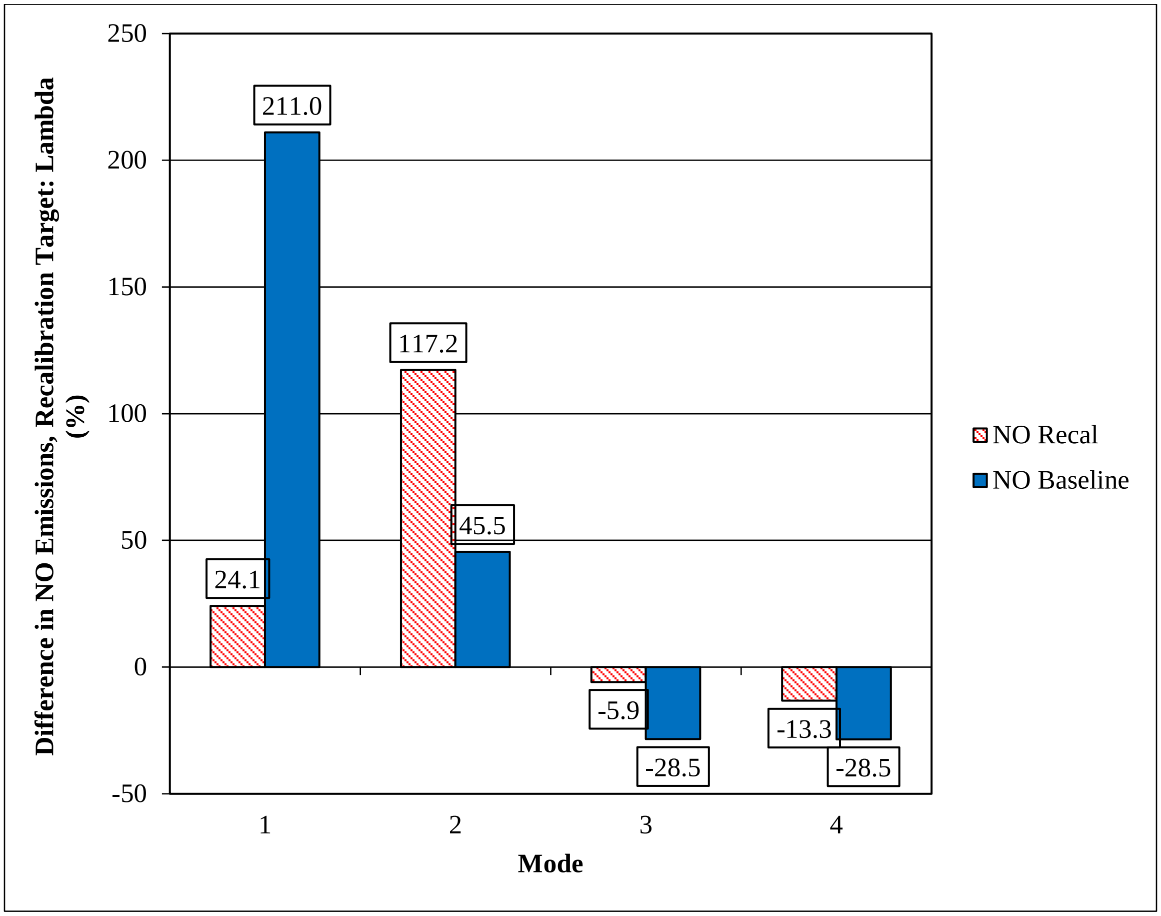

Figure 5.30: Difference in nitrogen oxide values at each mode for the recalibration test targeting E0 lambda

Mode 1 emissions produced more THC for E20 with the recalibration than the E0 due to the addition of extra fuel to reduce lambda for the test; this is show in Figure 5.31. Baseline data for E20 produced no THC emissions greater than E0 values. An increase in THC emissions was seen for modes 3 and 4 compared to the E20 baseline test while a reduction in THC emissions occurred at mode 2.

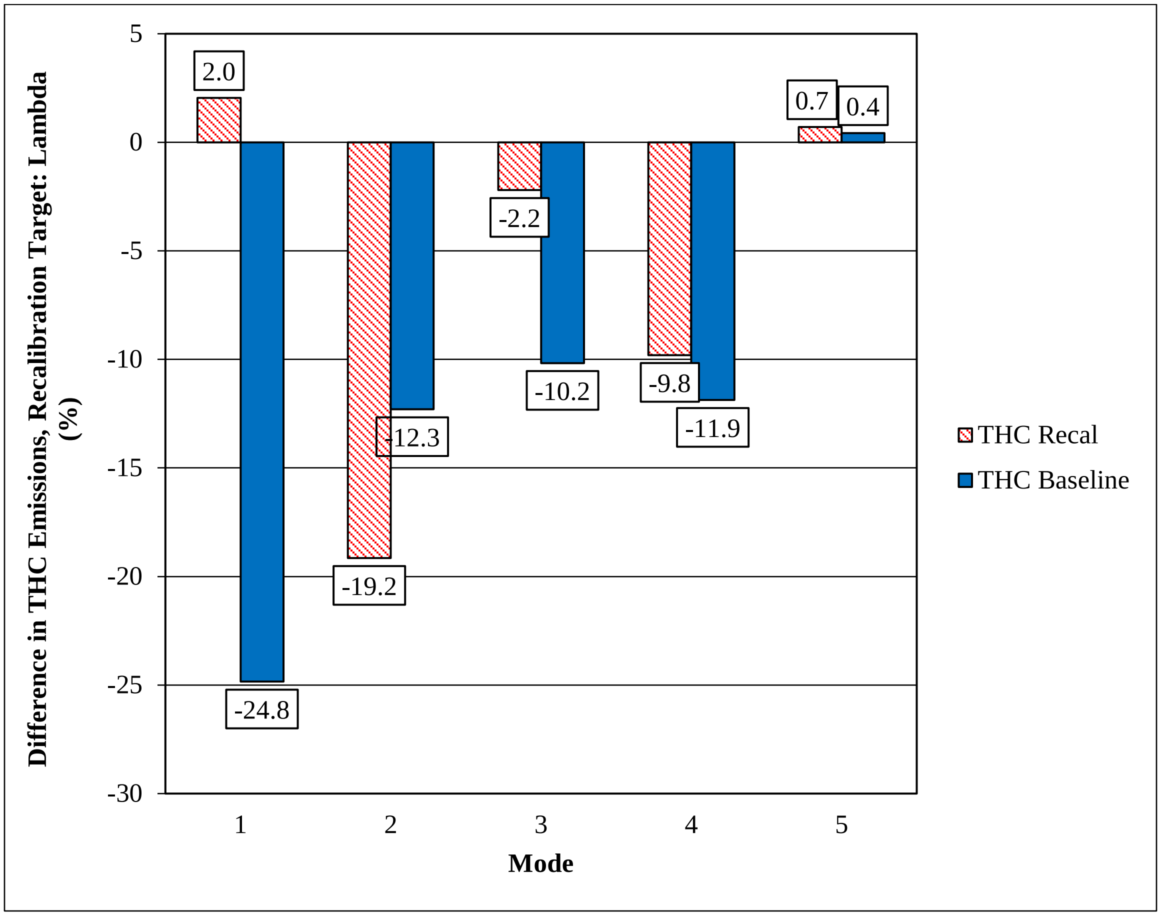

Figure 5.31: Difference in total hydro-carbon values at each mode for the recalibration test targeting E0 lambda

A reduction in formaldehyde emissions was seen at modes 1-4 with the largest reduction at mode 1 compared to the baseline E20 test. These reductions in formaldehyde emissions can be seen below in Figure 5.32. Although the percent change in these emissions is large, the actual change in formaldehyde emissions is less than 250 parts per million (ppm) at mode 1. These formaldehyde emissions values decrease in ppm as RPM decreases.

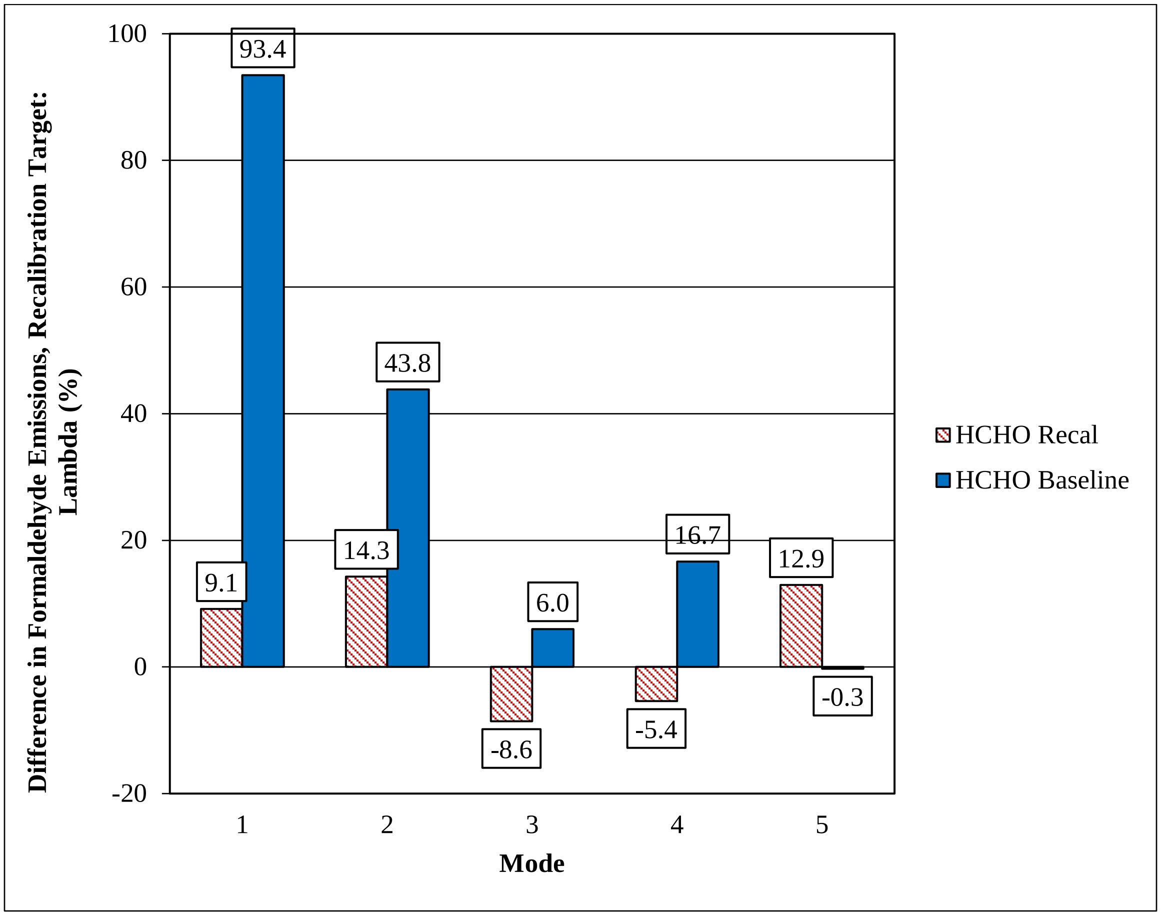

Figure 5.32: Difference in formaldehyde values at each mode for the recalibration test targeting E0 lambda

Similar to formaldehyde emissions, acetaldehyde emissions saw the largest reduction at mode 1, with reductions in modes 2 and 4 compared to the baseline test. Figure 5.33 shows an increase of 300 percent for acetaldehyde emissions at mode 3 compared to the E20 baseline. Once again, these large increase in acetaldehyde emissions is due to the very small overall amount of this constituent produced in the exhaust.

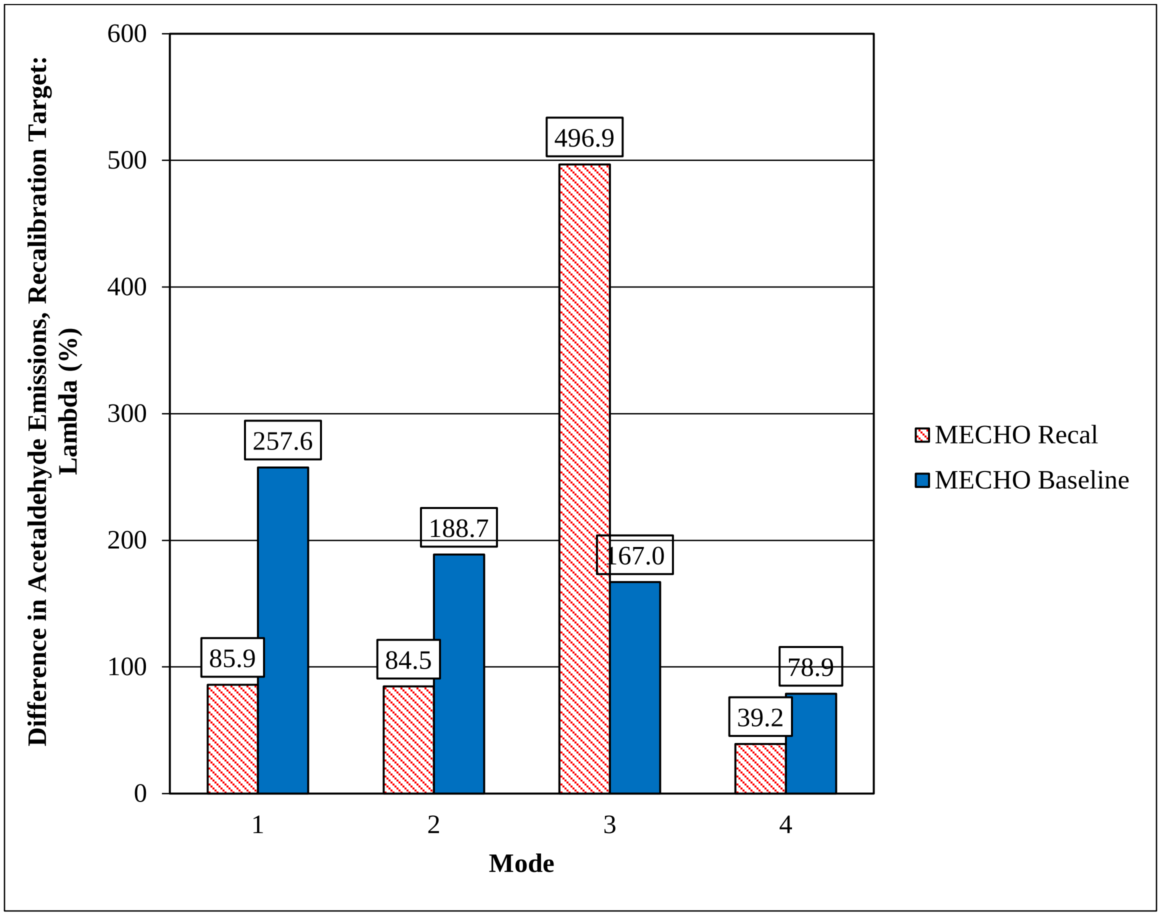

Figure 5.33: Difference in acetaldehyde values at each mode for the recalibration test targeting E0 lambda

5.4 Matching E0 Mid-Pipe Temperature using E20 fuel

Another test that was conducted matched E20 mid-pipe temperatures to the E0 baseline mid-pipe temperatures. This test intended to see how mid-pipe temperatures form emissions with different fuels if the temperatures are similar. Changing both ignition timing and injection time affected mid-pipe temperature. For this test, mid-pipe temperature was met by first adjusting the amount of fuel injected into the engine, and then adjusting the spark timing to “fine tune” the mid-pipe temperature. Figure 5.34 shows the percent difference of the mid-pipe temperatures for the E20 recalibration compared to the baseline E0 mid-pipe temperatures.

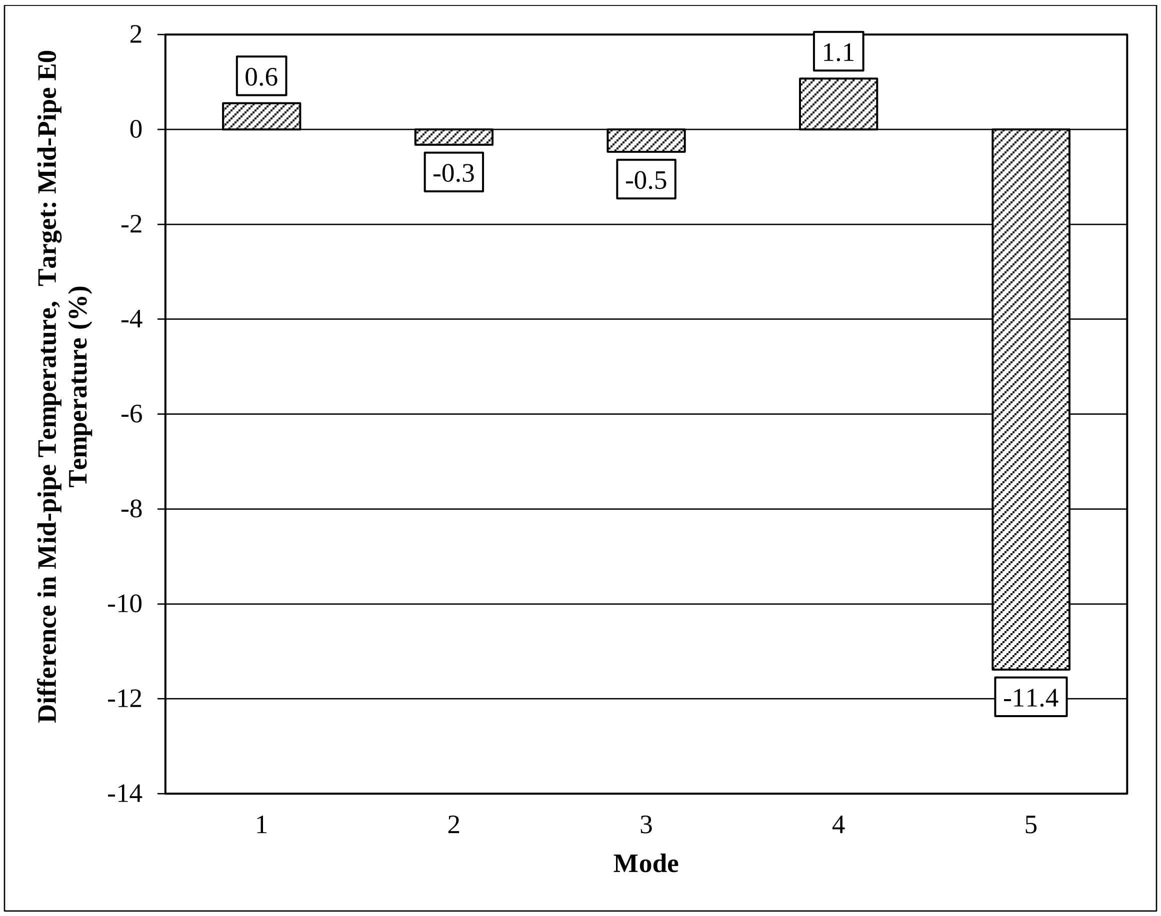

Figure 5.34: Difference in mid-pipe temperature at each mode for the recalibration test targeting E0 mid-pipe temperatures

Even though mid-pipe temperature was met by adding or subtracting fuel, lambda values did not match those from the previous test where the objective was to match E0 lambda values. It can be seen below in Figure 5.35 that modes 1, 4 and 5 were leaner than E0 lambda values, and modes 2 and 3 were more fuel rich. Modes 1 and 2 were still richer than stoichiometric after calibration and modes 3 and 4 were still leaner than stoichiometric. This is the same as the stock calibration.

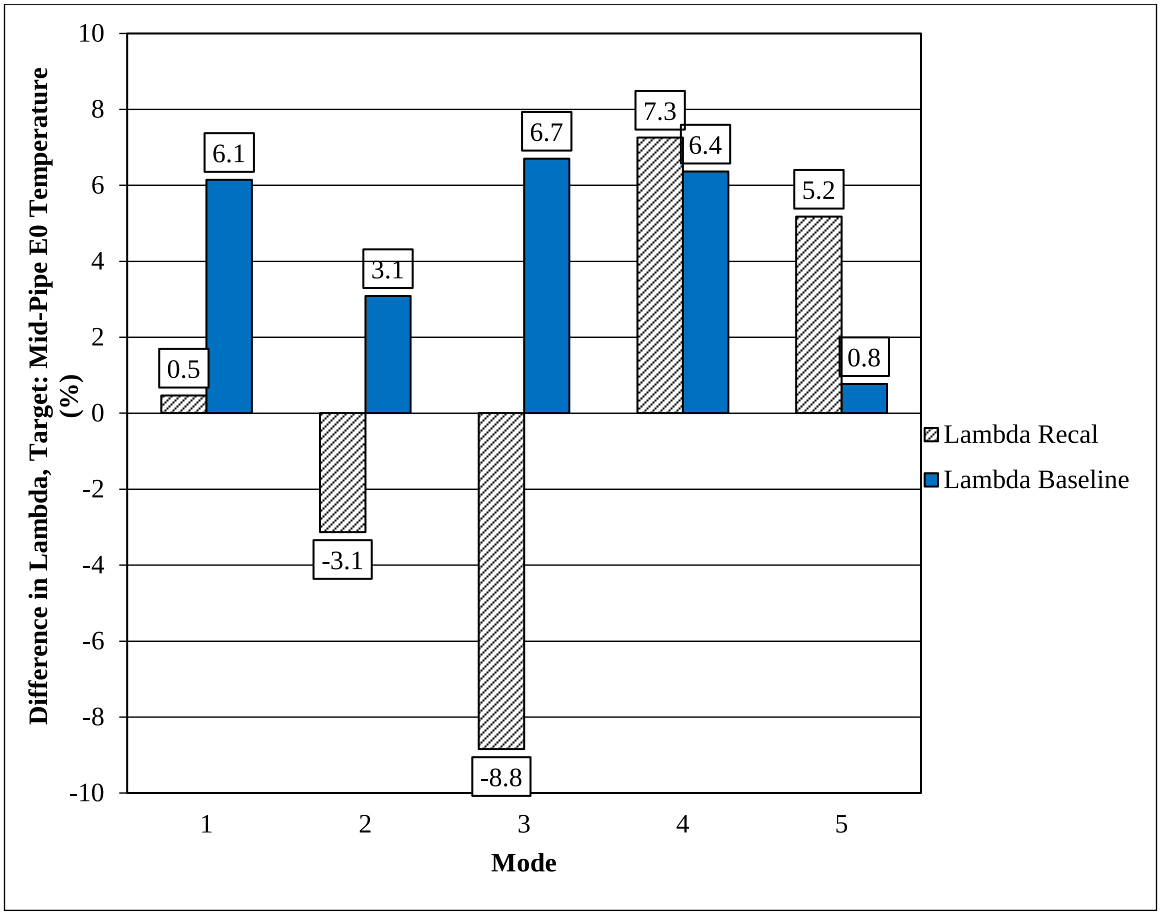

Figure 5.35: Difference in lambda values at each mode for the recalibration test targeting E0 mid-pipe temperatures

E0 torque was lower than manufacture spec resulting in a larger percent difference in torque at mode 1. The mid-pipe temperature recalibration did produce more torque than the manufacture spec. Torque was similar for modes 2-5 because they are held at a specific value by the dyno. Torque are mode 5 was not shown because the torque value at idle is 0 ft-lbs. Figure 5.36 shows these torque values in percent difference.

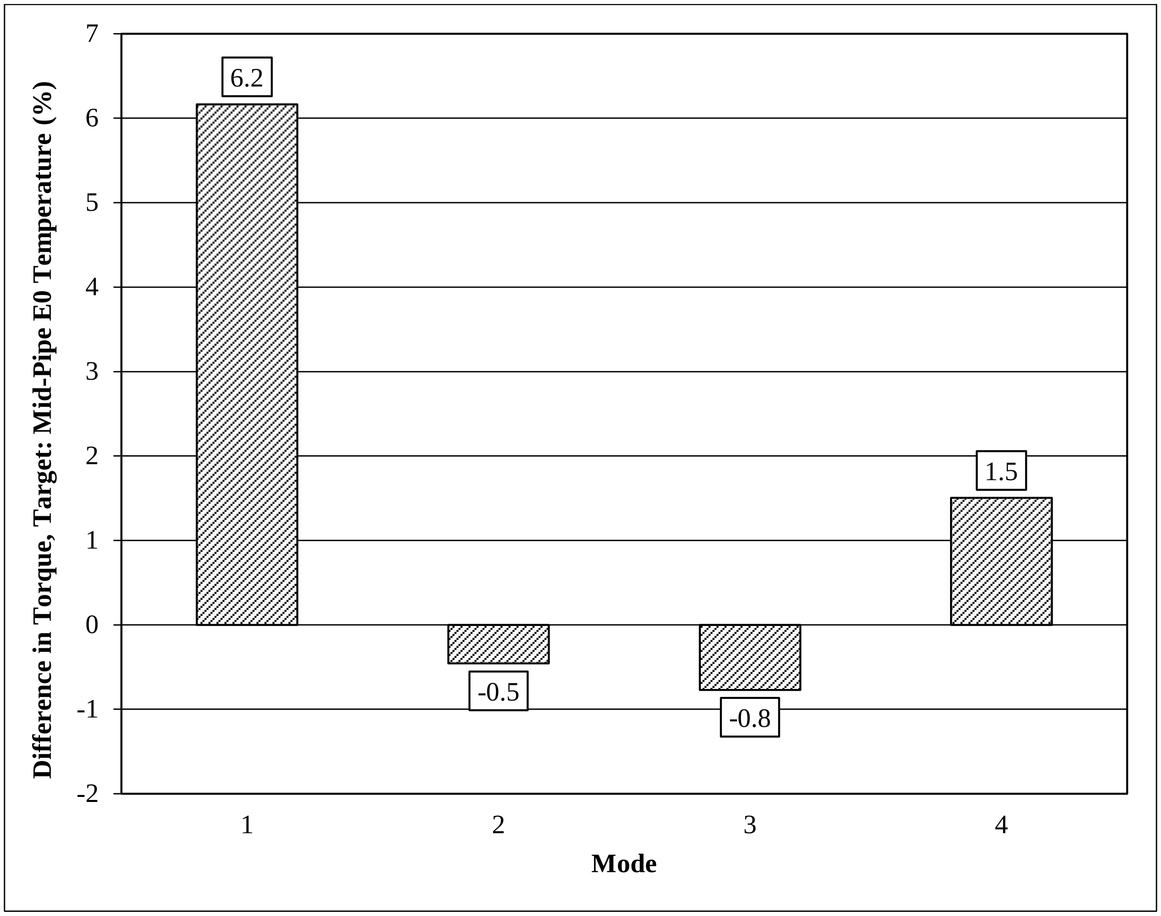

Figure 5.36: Difference in torque values at each mode for the recalibration test targeting E0 mid-pipe temperatures

Brake specific fuel consumption is displayed below in Figure 5.37. This figure supports an increase in fuel flow for modes 1-3 compared to the E0 baseline test with little increase in power.

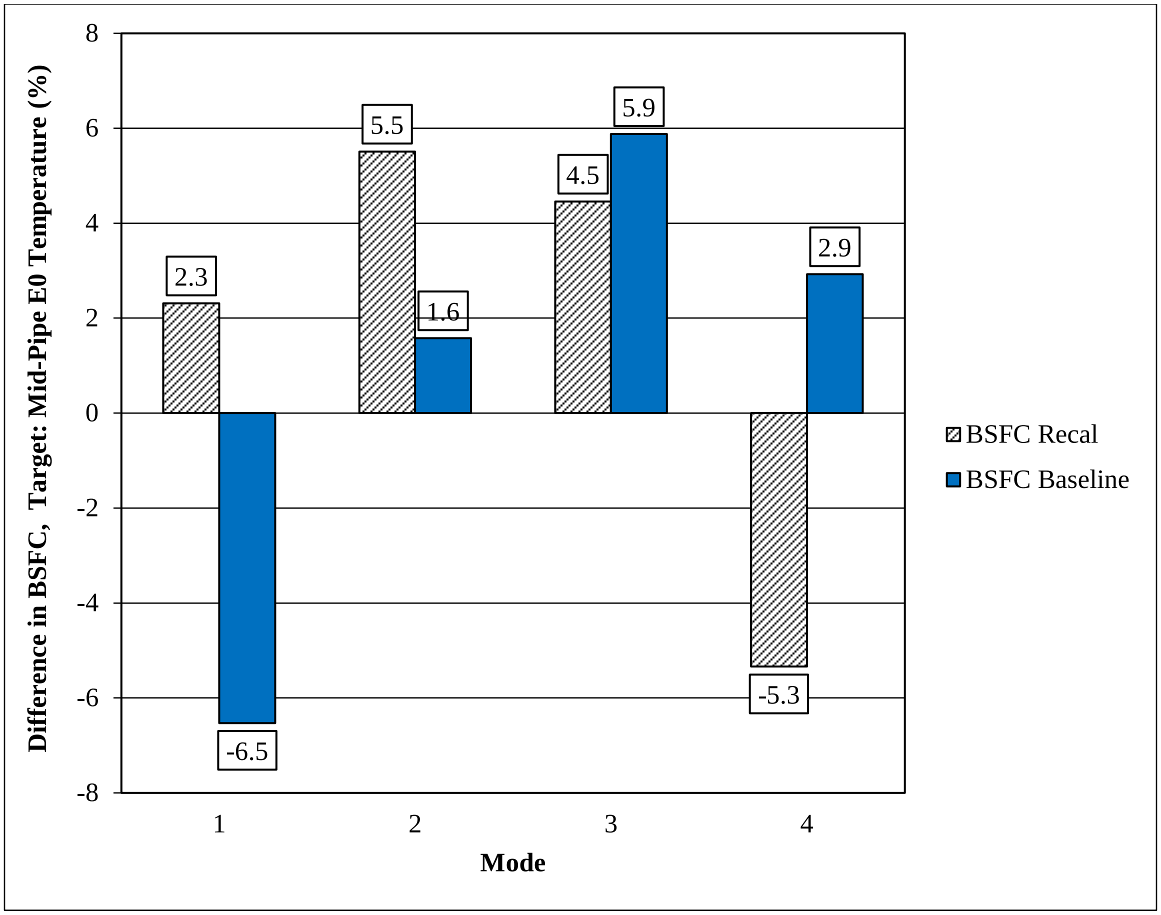

Figure 5.37: Difference in brake specific fuel consumption values at each mode for the recalibration test targeting E0 mid-pipe temperatures

Carbon monoxide emissions are shown below in Figure 5.38. Carbon monoxide emissions follow lambda trends where an increase in lambda decreases CO emissions, and a decrease in lambda shows an increase in CO emissions. There was an overall increase in CO emissions for modes 1 through 3 compared to the E20 baseline test.

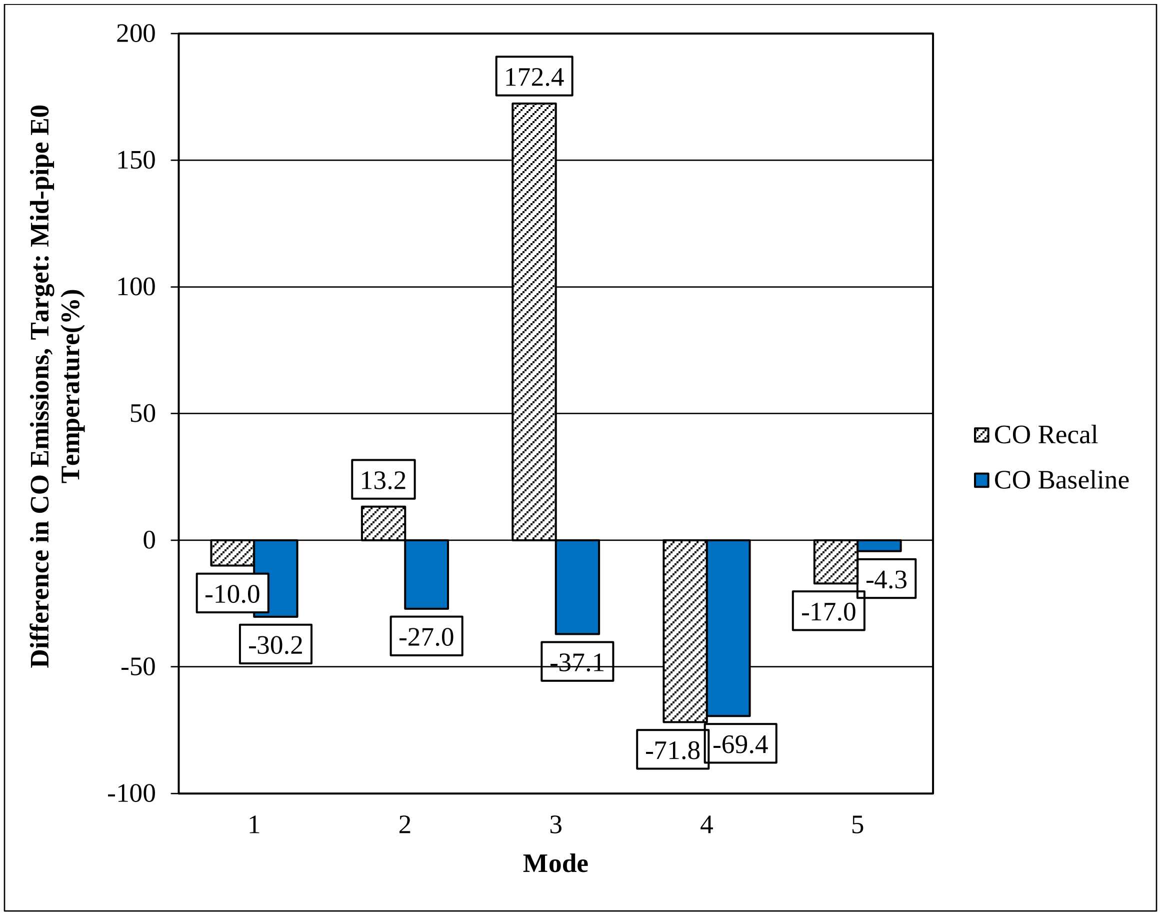

Figure 5.38: Difference in carbon monoxide emissions at each mode for the recalibration test targeting E0 mid-pipe temperatures

Carbon dioxide emissions show opposite trends of carbon monoxide emissions. Figure 5.39 shows where ever there was an increase in CO emissions, a decrease in CO_2 emissions ensued and vice versa except for mode 3. Modes 1 and 2 showed an overall decrease in CO_2 emissions compared to the E20 baseline test and a slight increase in CO_2 emissions for modes 3 and 4 compared to the E20 baseline test.

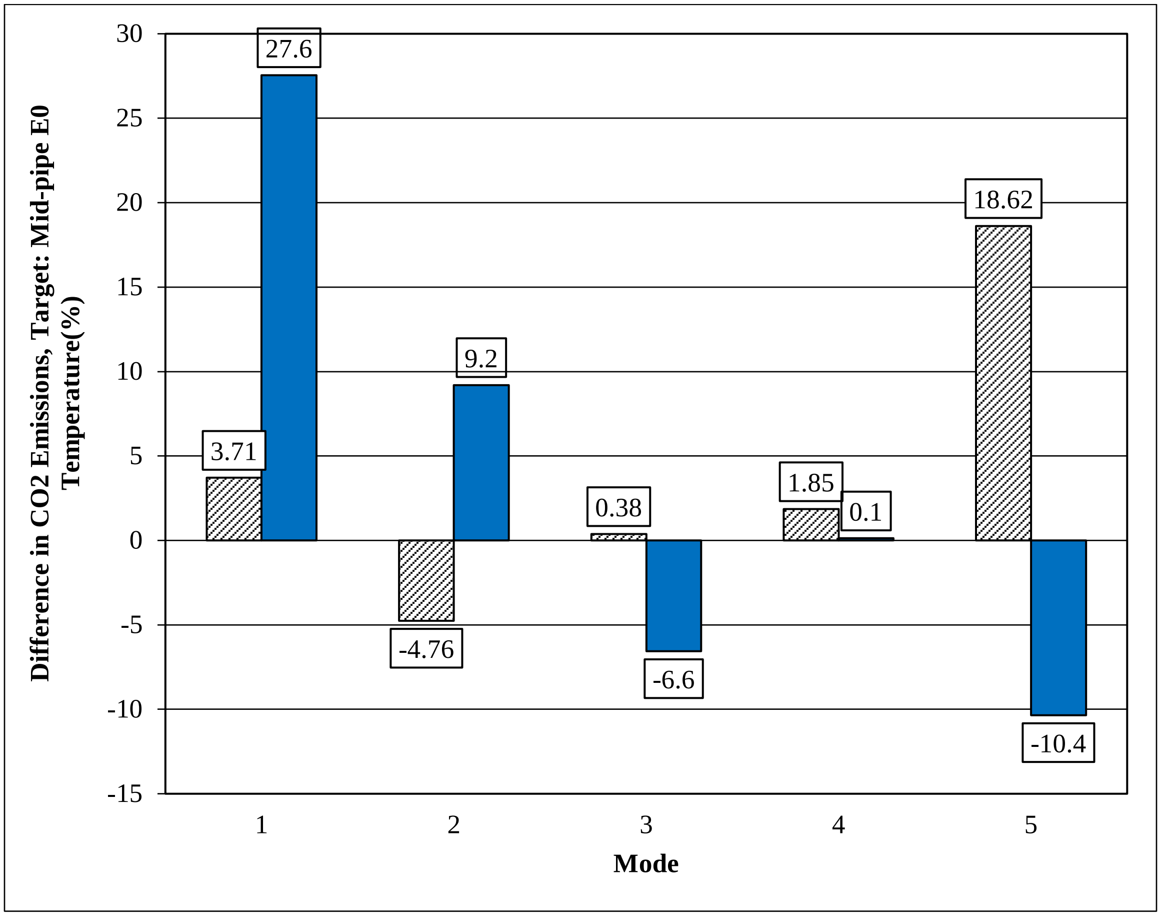

Figure 5.39: Difference in carbon dioxide emissions at each mode for the recalibration test targeting E0 mid-pipe temperatures

Figure 5.40 below shows an increase in nitrogen oxide emissions in mode 1, but a decrease in NO emissions for modes 2 through 4. Compared to the E20 baseline, there was a reduction in NO for all modes after the E20 mid-pipe temperature recalibration. There were no recorded NO emissions at idle (mode 5).

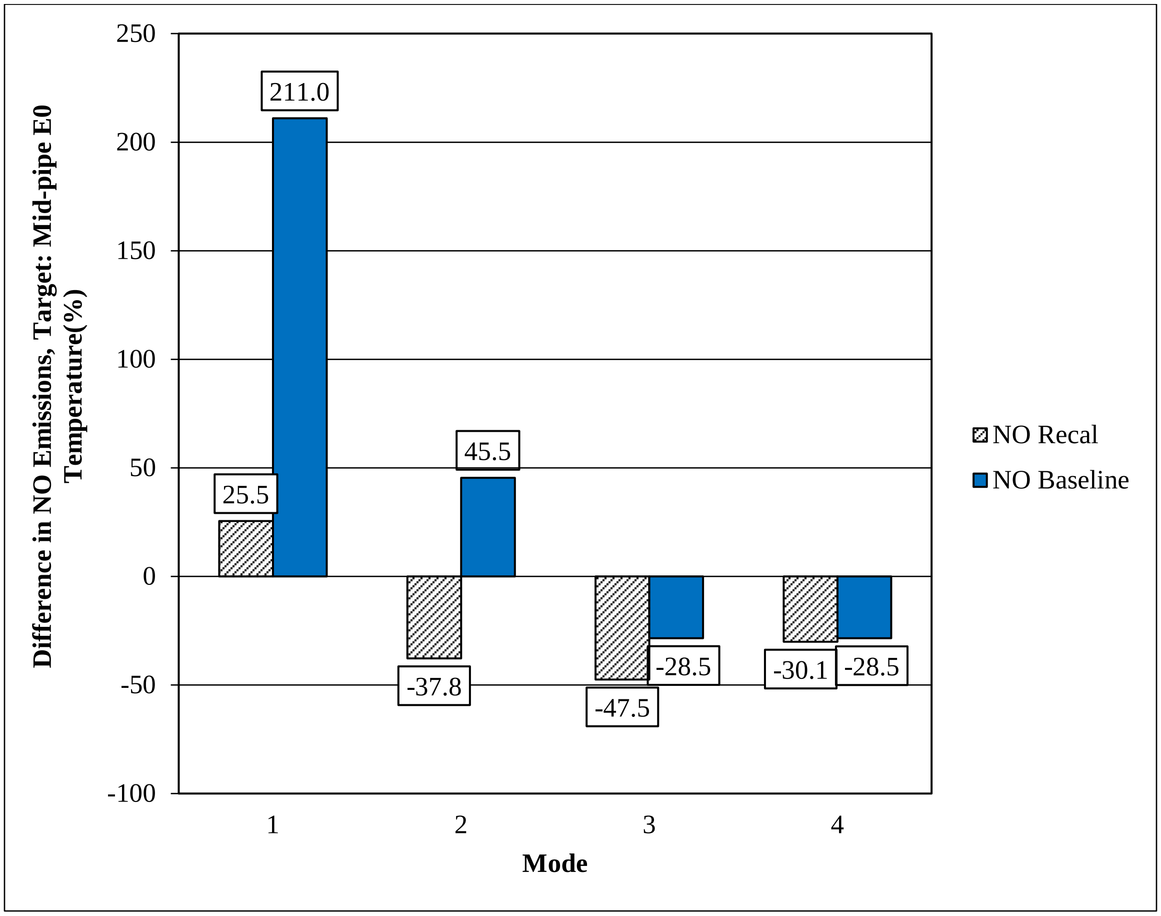

Figure 5.40: Difference in nitrogen oxide emissions at each mode for the recalibration test targeting E0 mid-pipe temperatures

Total hydrocarbon emissions were reduced in modes 2 through 4 compared to the baseline E0 test; this can be seen in Figure 5.41. The THC analyzer is maxed out at mode 5, resulting in an inconclusive reading whether or not there was an increase or decrease. There was an overall increase in THC emissions for modes 1 through 3 compared to the non-calibrated E20 baseline test due to the excess fuel added to meet mid-pipe temperatures.

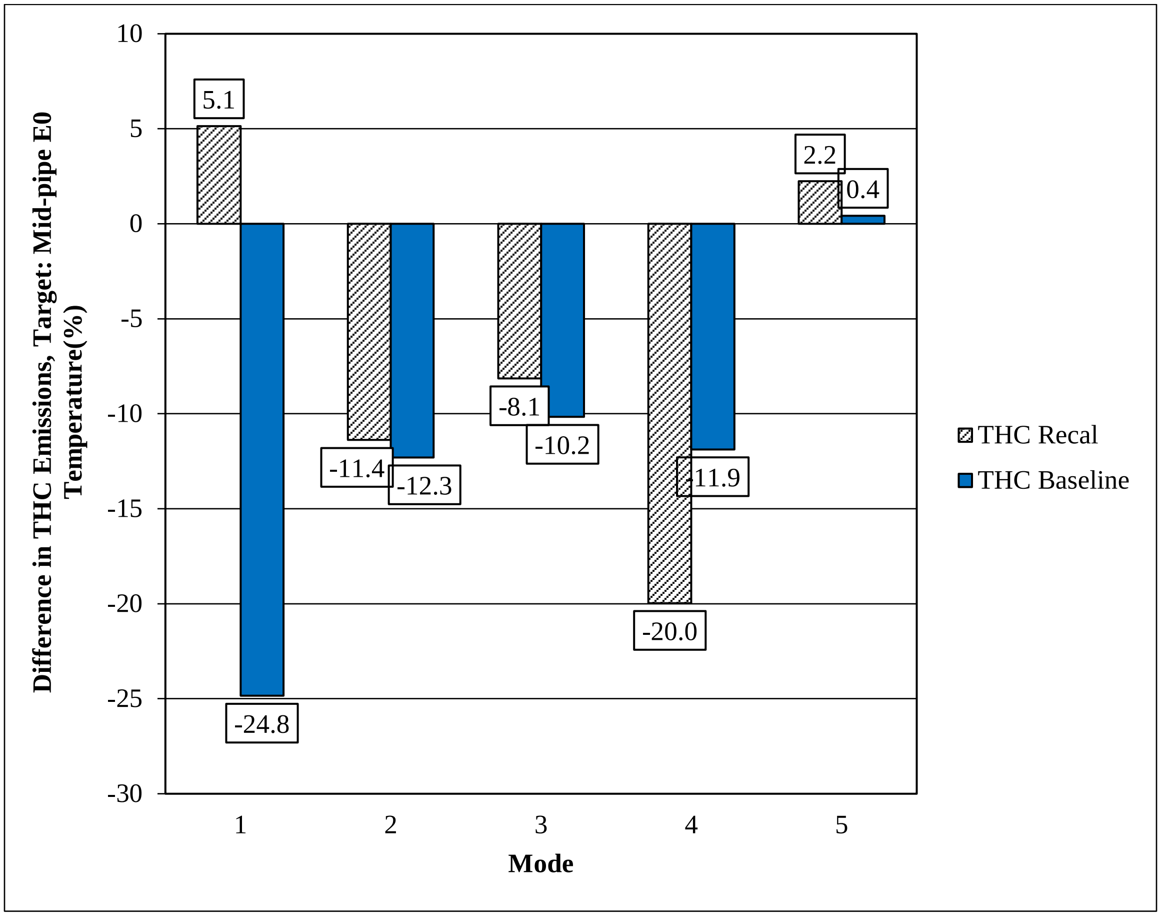

Figure 5.41: Difference in total hydrocarbon emissions at each mode for the recalibration test targeting E0 mid-pipe temperatures

Formaldehyde emissions totals acted similar to lambda values recorded during the 5-mode test. Formaldehyde emissions increased when there was an increase in lambda and decreased when there was a decrease in lambda. Fewer formaldehyde emissions were produced in modes 1 through 3 than the baseline E20 test, and more formaldehyde emissions than the E20 baseline test in modes 4 and 5. This trend in formaldehyde emissions can be seen in Figure 5.42.

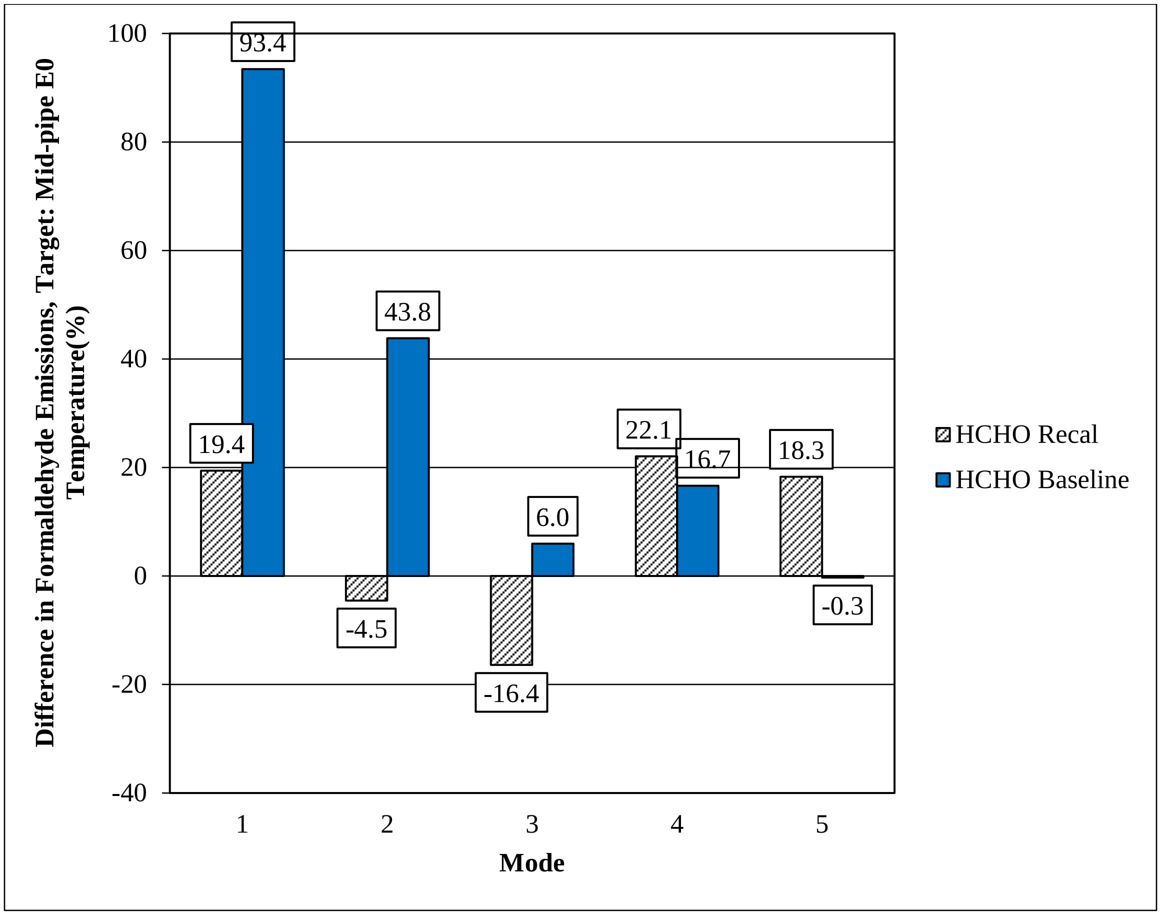

Figure 5.42: Difference in formaldehyde emissions at each mode for the recalibration test targeting E0 mid-pipe temperatures

Acetaldehyde emissions increased for all modes as shown in Figure 5.43, but were lower than the baseline E20 test except for mode 4.

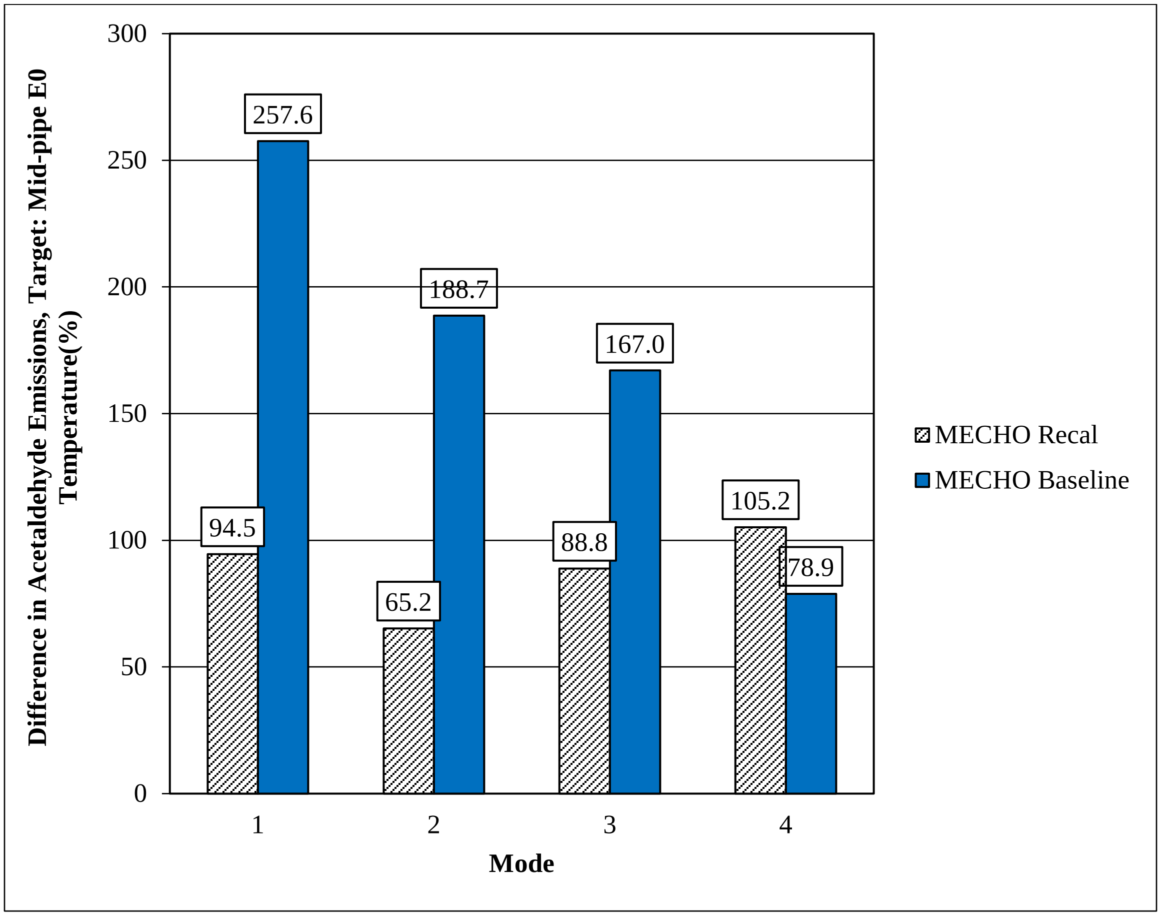

Figure 5.43: Difference in total acetaldehyde emissions at each mode for the recalibration test targeting E0 mid-pipe temperatures

Chapter 6: Conclusions and Future Work

6.1 Conclusions

Running a gasoline/ethanol fuel mixture of 20 percent by volume was very predictable and followed trends similar to previous ethanol research. Each mode in the baseline testing showed an increase in lambda when running E20 fuel. This resulted in a slight increase in torque at mode 1 and increases in mid-pipe temperature for modes 1 and 2. This increase in mid-pipe temperature is due to running closer to stoichiometric, where exhaust temperatures are the highest. Decreases in mid-pipe temperature for modes 3 through 5 were displayed and can be attributed to the high latent heat of vaporization that ethanol has, cooling the incoming air/fuel mixture resulting in lower combustion temperatures. BSFC was increased for modes 2 through 4 because the air/fuel ratio was lower for E20 requiring more fuel. Carbon monoxide emissions decreased for all modes because of the enleanment of E20. Carbon dioxide and nitrous oxide saw increases in modes 1 and 2 which match the higher mid-pipe temperatures found at those modes. Reductions in CO_2 and NO were found at lower mid-pipe temperatures in modes 3 and 4. Total hydrocarbons were reduced for all modes because of the increase in lambda as well as increases for all modes for formaldehyde and acetaldehyde.

Sweeping engine parameters helped to show which exhaust emissions would increase or decrease as changes were made to the ECU. Since lambda is fuel dependent, sweeping only the injection time changed lambda. Mid-pipe temperatures were affected by both ignition timing and injection time. By changing when the air/fuel mixture in the combustion chamber was ignited allowed more or less un-combusted fuel into the exhaust allowing manipulation of mid-pipe temperatures. Similar to adding or taking away fuel allowed mid-pipe temperature to change. Since CO is dependent on lambda, adjusting injection time only affected this exhaust constituent. The other exhaust constituents (CO_2, NO, THC, formaldehyde, and acetaldehyde) showed to be affected by both injection time and ignition timing.

The first recalibration done with E20 fuel was adjusting injection timing to manipulate lambda so it was the same value as the E0 baseline test at each mode. Doing this resulted in a richer mixture for all modes which saw an increase in torque at mode 1 and a reduction in mid-pipe temperatures for all modes compared to the E20 baseline test. Modes 1 and 2 still had higher mid-pipe temperatures than the E0 baseline test. BSFC increased for all modes which follows the trend of decreasing lambda, but there as an outlier at mode 2 where BSFC was reduced by two percent. Even with the addition of fuel to reduce lambda, there was still a reduction in CO emissions at modes 1, 2, and 4, when compared to the baseline E0 test. A slight increase in CO_2 emissions was recorded after recalibration except for mode 1. NO emissions saw a decrease in mode 1 and an increase in mode 2 when compared to the E20 baseline emissions. There was still fewer THC emissions produced after recalibration except in mode 1 where there was a slight increase. Finally, there was a reduction in formaldehyde emissions for modes 1-4.

Adjusting mid-pipe temperature to meet E0 baseline temperatures yielded a similar lambda value at mode 1 as the baseline value, but a reduction for lambda (richer) in modes 2 and 3, and an increase in lambda (leaner) for modes 1, 4, and 5. This led to an increase in CO emissions for modes 2 and 3 where lambda was richer, and a reduction in CO emissions in modes 1, 4 and 5 where lambda was leaner. CO_2 emissions were reduced at modes 1 and 2 though still a positive percent difference value, and slightly increased at modes 3 and 4 when compared to the E20 baseline test. A reduction in NO was found in modes 2 through 4 which were still lower than the E20 baseline test. Similar to the lambda recalibration test, THC emissions were reduced for modes 2 through 4, and increased for mode 1. Formaldehyde emissions had a similar trend to the lambda recalibration test as well, where a reduction was seen from the baseline test where lambda is richer.

6.2: Future Work

This project was created in a short amount of time and was successful but some improvements should be included in future work to the dyno cell for any new engine package being added.

With the location of the test cell susceptible to high changes in temperature, the intake and cell temperature were controlled. This was done using a simple air conditioning unit and heater to maintain intake temperature within an acceptable range of 25°C. Room fans were used to exhaust dyno cell air which was replaced by ambient air from vents leading to the outside. Manual changes had to be done to the AC and heat units in order to make changes in intake temperature. This was a very brute force method and did not do a good job maintaining constant temperatures over the period of the testing with large ambient temperature changes. A much better AC/heating unit should be invested in to maintain incoming air temperature and humidity so that torque and temperature values do not deviate throughout a long testing period. This should lead to recording more constant emissions values.

The accuracy of the fuel pressure required to properly operate the Arctic Cat DSI engine was very high because of the fuel injection and it had a small window (±1 psi) of acceptable fuel pressure. The fuel delivery equipment used for testing performed flawlessly for modes 2-4 maintaining pressure, but lacked the ability to maintain set fuel pressure at wide open throttle. Some manual changes needed to be done to increase the fuel pressure so the window provided by the OEM could be met. This would result in shutting the engine down, over pressurizing the fuel system, and then going back to mode 1 to record data. This required extra time for testing and loss of temperature in the exhaust. Maintaining the correct fuel pressure is important to calculating fuel flows for BSFC values and for engines that are fuel injected. Having incorrect fuel pressure could also result in false emissions recordings because with having fuel injectors opening and closing by values of milliseconds, over/under fueling can result in skewed emissions.

Finally, more testing should be conducted to back up the current research data after test cell improvements have been made. This would include multiple baseline E0 and E20 tests along with more tests comparing matching E0 lambda and mid-pipe temperatures with E20 fuel to see root causes of emissions formation. Analysis of toxic emissions should also be conducted because there is little research done on formation of these emissions constituents.

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[6] J. Bassett. (2013). *Inside Arctic Cat’s 600 CTEC-2 Engine* [Online]. Available FTP: http://www.snowmobile.com/products/inside-arctic-cats-6000-ctec2-engine-1691.html

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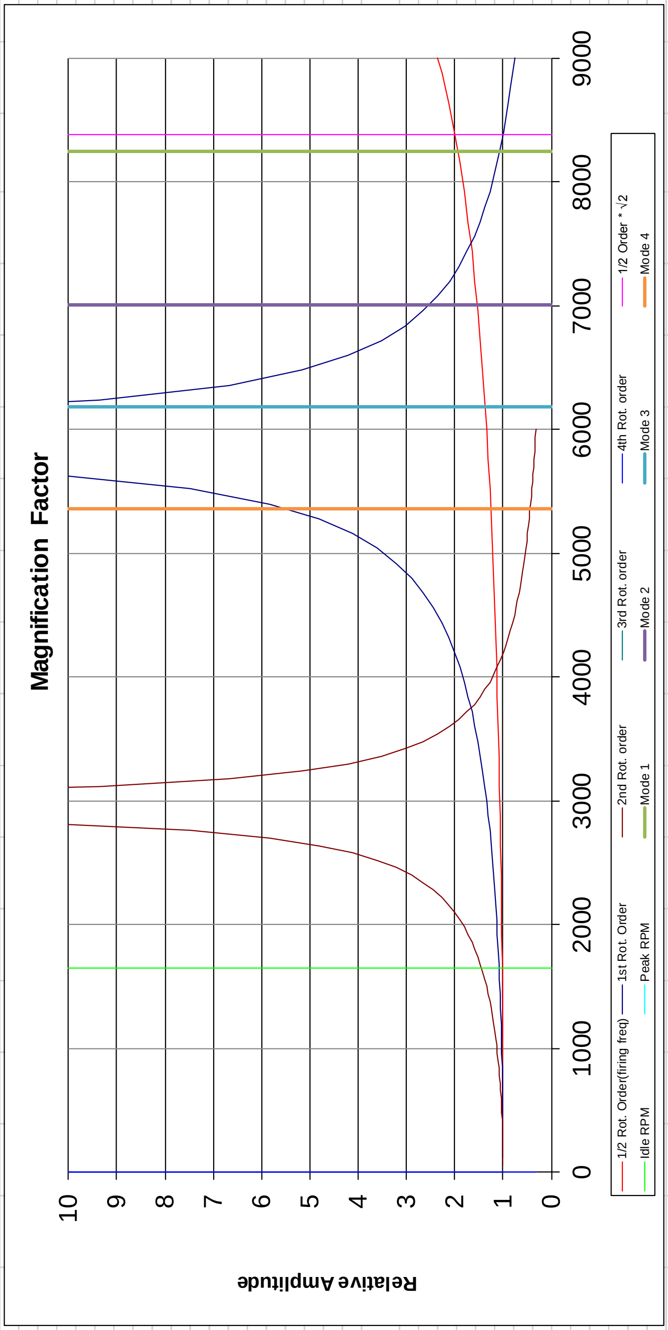
Appendix A.1 – Resonance calculation for a flexible coupling driveshaft



Inputs	
Calculateds	
Natural freq (hz)	98.85702645
Natural freq (RPM)	2965.710793
I1 (kg*m^2)	0.122
I2 (kg*m^2)	0.0145
Torsional Stiffness (Nm/rad)	5000
	950 rpm
Peak Magnification Factor	43.2661221
60% of ωn RPM	1779.426476
Idle RPM	1650
Peak RPM	8250
Gearbox Ratio	1
Max Engine RPM under 60% ωn	1779.426476
Lowest allowable RPM	8388.296852
Gearbox ratios	
Kymco gearbox	3.053
H2 high range w/net forged	3.584
Orders to graph	On(1)/Off(0)
1/2	1
1	1
2	1
3	0
4	0

Appendix A.2 – Paragon Laboratory E20 fuel results

MANY TESTS. ONE LABORATORY.

PARAGON Laboratories

December 14, 2015

Jon Loesche
Michigan Tech University
834 Franklin Street
Hancock, MI 49930

RE: Workorder: 204798

Dear Jon Loesche:

Paragon Laboratories, Inc. received the samples associated with the workorder listed above for the analyses presented in the following report. The analyses pertain only to the aliquot of sample received.

This material is confidential and is intended soley for the person to whom it is addressed. If this is received in error, please contact the number below.

Please note that any unused portion of the sample(s) will be discarded 60 days after sample receipt, unless requested otherwise.

We appreciate the opportunity to assist you. If you have any questions concerning this report, please contact an Account Coordinator at (734) 462-3900.

Sincerely,

Sharon L. Johnson

Sharon L. Johnson
sjohnson@paragonlaboratories.com
Senior Account Coordinator

[A] Paragon Laboratories, Inc. is accredited to ISO/IEC 17025 by A2LA for analytical methods referring to this footnote. (A2LA Cert. No. 2705.01 Expires 3/31/2017)
Paragon Laboratories, Inc. is certified by the Michigan DEQ to analyze Drinking Water. (MDEQ Lab No. 9901 Expires 2/25/2017)

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PARAGON Laboratories

SAMPLE SUMMARY

Workorder: 204798 MICH TECH-120715

Lab ID	Sample ID	Sample Description	Matrix	Date Collected	Date Received	Collector
2047980001	91 E20-GASOLINE/ETHANOL MIX		Gasoline	11/19/2015 14:00	12/7/2015 07:50	Jon Loesche

12649 Richfield Ct. Livonia, MI 48150 P 734.462.3900 F 734.462.3911 W http://www.paragonlaboratories.com

PARAGON Laboratories

ANALYTICAL RESULTS

Workorder: 204798 MICH TECH-120715

Lab ID:	**2047980001**	Date Collected: 11/19/2015 14:00	Matrix: Gasoline
Sample ID:	91 E20-GASOLINE/ETHANOL MIX	Date Received: 12/7/2015 07:50	
Sample Desc:		PO:	

Parameters	Qualifier	Result Units	Dilution Factor	Reporting Limit	Result Qualifier Min	Result Qualifier Max	Analyzed	By
Octanes								
Analytical Method: ASTM D2699 & D2700 [A]								
Research Octane Number		97.6	1	1.0			12/11/2015 13:50	AKL
Motor Octane Number		87.0	1	1.0			12/11/2015 13:50	AKL
AKI Calculation		92.3	1	1.0			12/11/2015 13:50	AKL
Sensitivity		10.6	1	1.0			12/11/2015 13:50	AKL

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Paragon Laboratories, Inc. is certified by the Michigan DEQ to analyze Drinking Water. (MDEQ Lab No. 9901 Expires 2/25/2017)

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