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IMPACT OF WATER INGESTION ON COMBUSTION PERFORMANCE IN A SPARK IGNITED INTERNAL COMBUSTION ENGINE

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IMPACT OF WATER INGESTION ON COMBUSTION PERFORMANCE IN A SPARK IGNITED INTERNAL COMBUSTION ENGINE

By

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List of abbreviations

APS LABS	Advanced Power Systems Research Center
BDC	Bottom dead center
BMEP	Brake Mean Effective Pressure
BSFC	Brake specific fuel consumption
CA	Crank angle
CA50	Crank angle at 50% burn
CAC	Charge Air Cooler
COV	Coefficient of Variation
DAQ	Data Acquisition
DWI	Direct Water Injection
ECAM	Exhaust Camshaft
ECU	Engine control unit
EGT	Exhaust gas temperature
EOI	End of injection
ICAM	Intake Camshaft
IMEP	Gross Indicated Mean Effective Pressure
LFE	Laminar flow element
LHU	3 digit RPO code for a particular GM Ecotec engine
LNV	Lowest normalized value
MAP	Manifold absolute pressure
MTU	Michigan Technological University
PN	Part Number
PWI	Port Water Injection

PWM	Pulse Width Modulation
SI	Spark Ignition
SOI	Start of injection
TDC	Top dead center
W/F	Water to Fuel Ratio (mass fraction)
WOT	Wide Open Throttle

Abstract

This study experimentally investigated the effect of unintended water ingestion on a spark ignited internal combustion engine. Testing was performed on a 2.0L inline four cylinder turbocharged engine. Port water injection (with two different levels of atomization) was utilized to introduce water into the combustion system. Five speed / load points were tested from 1300 rpm 3 bar BMEP to 3250 rpm 15 bar BMEP. Testing was done with constant air charge as well as constant throttle emulating the in-vehicle scenario. The water to fuel ratio (W/F) was swept until COV of IMEP reached at least 20% and misfires were detected. With the addition of water, initially combustion performance degrades rapidly, however, once water vapor reaches saturation in the manifold, additional water has less of an impact, until ultimately in-cylinder water content is high enough to induce misfire. High engine loads are seen to be more resistant to combustion degradation due to water ingestion.

1 Introduction

In spark ignited turbo-charged engines, it is common for some sort of charge air cooler (CAC) to be placed after the turbocharger and before the intake manifold. A CAC is responsible for reducing the intake air temperature. An example of how a CAC is placed in an intake track of an engine can be seen in Figure 1. The members of the Advanced Light-Duty Engine Consortium led by Michigan Technological University's (MTU) Advanced Power Systems Research Center (APS LABS), (Ford, GM, FCA, and Nostrum Energy) have theorized that there is condensation occurring in the CAC and that this condensation is finding its way into the combustion chamber. The team at the APS LABS has theorized that under several conditions condensation in the CAC will occur and under other conditions the condensation will dislodge and enter the combustion system. When this condensation enters the combustion chamber it can cause adverse effects on combustion, and the one that is of the most concern is combustion degradation. If a vehicle driving down the road experiences condensation entering the combustion process, the vehicle's engine power output will likely decline during such an event. This type of event is classified as short in duration and temporary. The events that cause CAC condensation entering the combustion system is very unpredictable. Another metric of CAC condensation that is largely unknown is the form that the condensation takes as it enters the combustion system. Due to water having a high latent heat of vaporization of 2257 kJ/kg[1] this allows water to effect the engines charge mixture and intern combustion instability. This work investigates these theories, with an interest in discovering the point that water makes the combustion system unstable.

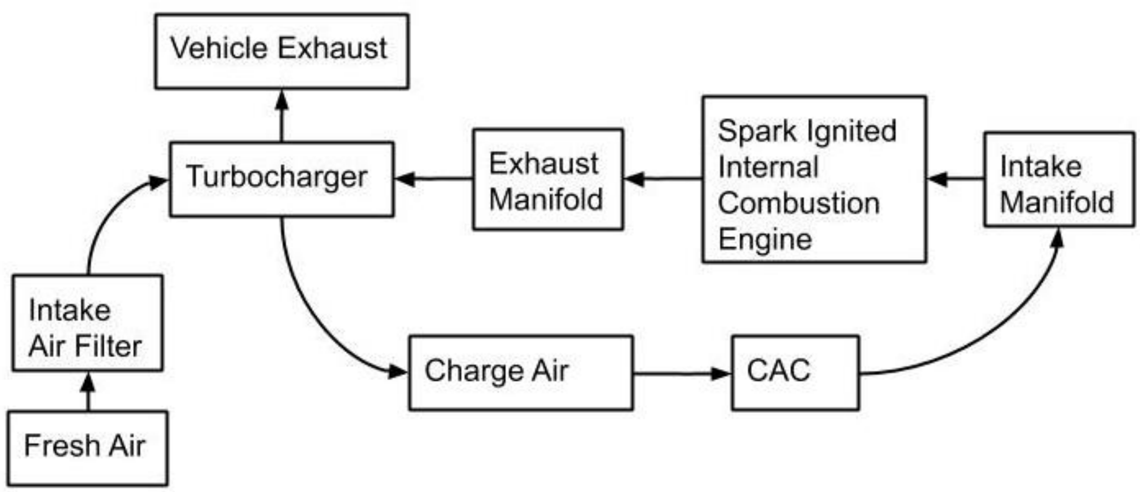


Figure 1 sample schematic of charge air system for a turbo-charged spark ignited engine

2 Literature Review

There have been many published works on how water can benefit engine performance or reduce engine emissions. The sets of literature that will be reviewed in this section are on the effect water has on emissions, and engine performance. Also discussed are the adverse effects that water can have on engine combustion, however, there are few published works on this topic.

2.1 Water Injection for Reduction of Emissions

There have been many studies looking into how water injection effects emissions produced by an internal combustion engine. The following sources reviewed in this section, looked at the W/F ratios previously tested by others. This study will not be focusing on emissions. Harrington [2] saw a reduction in NO emissions with a slight power reduction due to water introduction in combustion. Lanzafame, [3] through experimental data, showed that NO could be reduced 50% with water injection, and that the antiknock index can be raised with water injection. Lestz [4] also saw a 50% NO reduction but an increase in HC emissions with water injection. Li [5] utilized DWI to reduce NOx and CO emissions. Nicholls [6] utilized water injection to reduce NOx emissions, but also saw SFC decrease 0-.75W/F and then return to pre-water injection levels at 1.25 W/F. These studies all similarly found that some of the emissions produced by internal combustion engine can be reduced through water-injection.

2.2 Water Injection for Increased Performance

Due to the high heat of vaporization of water, it has been used to cool the combustion chamber, yielding benefits in knock mitigation. The following sources reviewed in this section, looked at the W/F ratios previously tested by others. The work in this study is not pertaining to water injection for increased engine performance. Worm et al. [7] showed the benefits of water injection with a lower octane fuel at WOT conditions. Boretti [8] utilized simulation tools to evaluate the effects of water injector positioning inside the intake track, and the injector position resulted in varying changes in combustion. Brusca's [9, 10] studies also showed how water injection can increase anti-knock properties of fuel. Cordier [11] showed the effects of water injection on knock mitigation and completed a comparison of the effects of DWI as opposed to PWI.

2.3 CAC Condensation and Condensation Ingestion

This section will go more in-depth on works that have been published on CAC condensation and condensation ingestion in regards to the study completed.

Choi [12] specifically examined the scenario by which water condenses in the CAC and can be subsequently introduced into the combustion chamber. Through an FTIR analysis, the quantity of H_2O present in the exhaust, could be correlated to the amount of water coming into the combustion system. Choi utilized this analysis to quantify the amount of H_2O from the intake air or from condensation that was entering the combustion system. The amount of condensation that would form in a CAC was quantified based on the mass flow rate through the intake. Four air flows were tested that correlated to speed/load points. These points were tested at a relative humidity of 89%, and intake air temperature of 36.5°C. The extremes of the speed load point that were tested are 2000 rpm 7.8 bar BMEP, and 3000 rpm 13.3 bar BMEP, and the corresponding mass flow rates were 0.021 kg/s, and 0.053kg/s, respectively. Approximately 175ml and 275ml of condensation was measured to either be in the CAC at end of test or passed through the engine during a ten-minute test at 2000 rpm 7.8 bar BMEP, and 3000 rpm 13.3 bar BMEP, respectively.

Choi [12] also performed a tip-in test: where either 100ml, 150ml, or 200ml of water was added to the CAC at the start of test. The test run would start at 2000rpm 7.8 bar BMEP then ramped to 3 different higher speed/load points of 2500rpm 9.5 bar BMEP or 2750rpm 11.6bar BMEP or 3000rpm 14.1 bar BMEP for 20 seconds and then return to the 2000rpm 7.8 bar BMEP. This test was intended to understand conditions at which water can move from the CAC into the combustion chamber and cause combustion degradation. The 3000rpm 14.1 bar BMEP point was the only point that experienced misfire with the addition of 150ml and 200ml in the CAC. There were slowburn events observed when 200ml added into the CAC at the 2750rpm 11.6bar BMEP test, and for all water additions to the CAC at the 3000rpm 14.1 bar BMEP test. Choi defined a misfire event as a cycle that has a IMEP of less than 0.5 bar, and a slowburn event is classified as IMEP of 70% of a normal IMEP value.

Like Choi, St-Aubin Ouellette [13], through various experiments, explored how condensation forms in a CAC and how the condensation then moves out of the CAC. However, St-Aubin Ouellette’s experiments also included a study of the effect of the CAC mounting angle in how condensation entrainment is affected. St-Aubin Ouellette’s testing utilized flow benches not including an engine.

Through thermodynamic equations, Tang [14] created curves to predict specific ambient temperatures when condensation will occur in the CAC at a given manifold pressure and ambient air humidity. Figure 2 is an example of one of the two plots Tang published. The region above the curves is where condensation can occur in the CAC.

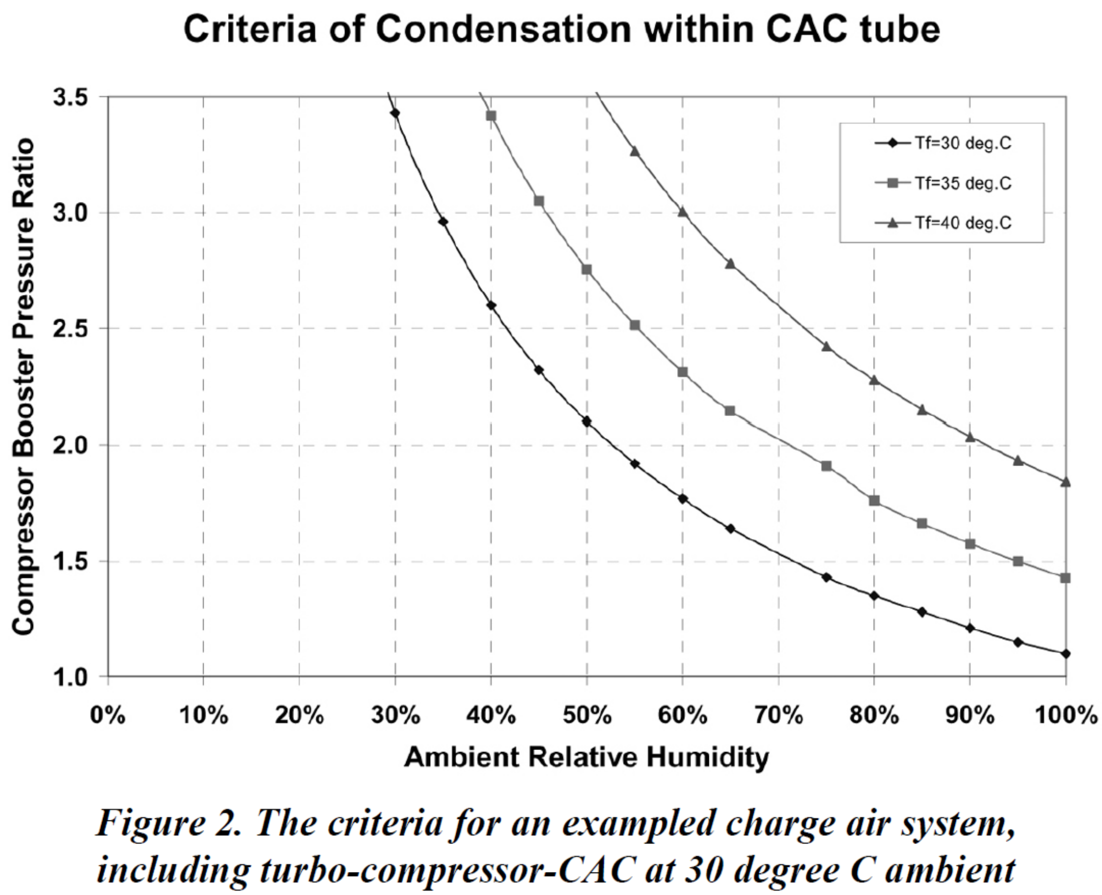


Figure 2 Criteria for condensation inside a CAC tube[14]. See Appendix 9A for copyright licensing information

Cash [15] utilized vehicle data to target the simulation and bench testing of CAC condensation. The study that Cash completed focused on conditions at which condensation will occur in a CAC. A high level take away from this work is that an increase in inlet air velocity will intern increase the condensation inside the CAC. Also, humidity has a major effect on the amount of condensation combined with air charge velocity.

All works reviewed here regarding CAC condensation have looked at how condensation occurs in the CAC or how it moves out of the CAC, with the exception of Choi. There seems to be a gap in study of the effects of water as is it unexpectedly introduced into the combustion system.

2.4 Quantity of Water

Several studies have utilized water injection and reported the quantity of water injected normalized to the amount of fuel being injected. Table 1 summarizes quantities of W/F tested across studies, and the highest W/F in the table is a W/F of 3. Most water injection studies are completed at WOT or high load. Only three studies that were reviewed mentioned COV of IMEP, thus, this is not widely published on water injection.

Table 1 Water/Fuel Quantity

Reference	**Water / Fuel**	**Water effects COV IMEP**	**Speed/Load Tested**
Boretti, Alberto 2013 [8]	0-.07	Not available	1000-5000rpm WOT
Bozza, Fabio 2016 [16]	0-.5	Not available	1500-5500rpm WOT
Brusca, S. 2019 [9]	0-1.5	Not available	CFR engine
Cordier, Matthieu 2019 [11]	0-1	Not available	2000rpm 17bar IMEP; 4000rpm 20bar IMEP;
Falfari, Stefania 2019 [17]	0-.6	Not available	6500 WOT
Harrington, J.A. 1982 [2]	0-1.5	Not available	500,1000,1500 rpm
Hoppe, Fabian 2015 [18]	0-.6	Not available	2000rpm 1.05MPa, 2.26MPa IMEP 3000rpm 1.46MPa IMEP
Lanzafame R. 1999 [3]	0-1.5	Not available	900rpm
Lestz, S. 1972 [4]	0-1.5	Not available	911rpm
Miganakallu, Niranjan 2020 [19]	0-1	Yes, Under 3%	1500rpm 800kPa $IMEP_{net}$
Nagasawa, Tsuyoshi 2020 [20]	0-.6	Yes, Under 9%	2000rpm 1100kPa,1250kPa IMEP
Netzer, Corinna 2018 [21]	0-1	Not available	2500 rpm 16.7bar IMEP
Nicholls, J.E. 1969 [6]	0-1.5	Not available	Unk.
Vacca, Antonino 2019 [22]	0-.5	Not available	2500rpm 15 bar IMEP 4500rpm 20bar IMEP
Worm, Jeremy 2017 [7]	3	Not available	2000-5000rpm at WOT
Zhuang, Yuan 2020 [23]	0-.5	Yes, Under 2.25%	1500rpm at WOT

3 Goal and Hypotheses

The literature regarding water addition into the combustion system has been mainly focused on high load, emission reduction, or both, with very little previous work done studying water injection at partial load. Moreover, previous work has largely been focused on finding the point of maximum benefit of water injection, not probing the detriment of water injection. We seek to fill these gaps in the literature. The goal of this research is to characterize how water ***negatively*** affects the combustion system, pursuant to the light duty consortium's outlined interest in finding the dilution limit of water when combustion is unstable.

Goals:

1. Characterizing the amount of water that causes combustion to be unstable across a range of speed and load conditions representative of customer operation.
2. Characterizing how variations in water atomization affect the limits of combustion stability.

4 Methods

4.1.1 Fuel

This testing was completed using VP Racing Fuel's 'C9' as opposed to fuel from a retail pump. Unlike commercial pump grade fuel, this fuel is held to tightly controlled specifications for each batch allowing for repeatability from barrel to barrel. While being a lower cost option compared to typical test fuels, such as, emissions certification fuels, the VP C9 has a similar characteristic to the emissions certification fuels. A comparison of the fuel specifications can be seen in Table 2. Although there is a difference in the sensitivity for the VP C9 when compared to the certification fuels, this is not seen as being as critical of a parameter compared to differences in boiling point. The difference in the distillation curve are negligible.

Table 2 Fuel Specification Table

Parameter	Tier 2 [24]	Tier 3 [24]	VP C9 [25]	Units
Initial Boiling Point	89	100	91.8	°F
T10	126	129	140	
T50	223	210	217	
T90	317	322	248.4	
Final Boiling Point	406	387	376	
Specific Gravity, 60°F	0.7437	0.7490	0.7225	-
DVPE (EPA equation)	9	8.8	Unk.	psi
RVP	Unk.	Unk.	6.89	
Oxygen	0	3.7	0	mass %
Research Octane Number	96.5	91	98.5	-
Motor Octane Number	88.7	83.5	95	-
AKI (R+M)/2	92.6	87.3	96.8	-

4.1.2 Water Ingestion Delivery

For test points outlined in Table 4, water injection is started from no water and then water injection is increased until the combustion becomes unstable. The point that combustion has become unstable was decided to be quantified as 20% COV of gross IMEP or Misfire. For this testing, a cylinder misfire is classified as a IMEP <0.

Equation 1 COV of IMEP

$$\mathrm{COV_{IMEP}}\% = \frac{\text{Standard Deviation(IMEP)}}{\text{Mean(IMEP)}} \times 100$$

4.1.3 Constant Throttle, Constant Spark, Constant Fuel Flow Rate

This test strategy is designed to replicate an in-vehicle condition that would occur during the event of CAC condensation getting dislodged and entering the combustion system. A CAC condensation event is thought to be short in duration and the engine control system is unlikely to correct for such event. Therefore, throttle, spark and fuel flow rate were held constant. This testing strategy was completed by varying parameters in the engine ECU. The spark table was modified to keep the spark position constant throughout the water sweep. The closed loop lambda correction was turned off to keep the fuel rate into the engine constant during the water sweep. Throttle position was held constant at the position required to achieve the target set point with zero water flow.

4.1.4 Constant Air Charge, Constant Spark, Constant Lambda

This test strategy is designed to examine the limitations of the combustion system. This test strategy also utilized the functionality of the engine ECU for lambda and spark control. The controller had the closed loop lambda correction turned on, and the spark table was set to hold a constant spark position. Constant air charge was achieved by correcting the throttle position based on the LFE reading.

4.1.5 Cam Phase Strategy

Two cam strategies were tested in this study: a min COV and best BSFC position. The best BSFC position was selected due the similarity of the cam strategies that vehicles will be running out in the field. The BSFC position allows for the best conversion of the fuel to torque output. A min COV cam position was selected to show the effect when the engine was at max stability. These BSFC cam and COV cam positions were selected from some pretesting on this engine platform. The pretesting involved completing a full factorial of the cam position of the intake and exhaust cams at the speed load point that were selected to be studied. An initial coarse factorial sweep was completed by testing cam position in increments of 10°. After the initial sweep, a targeted cam positioning sweep was completed in areas of interest. Examples of the cam position testing can be seen in Figure 3, and Figure 4. Figure 3 shows BSFC values for the 1750 rpm test point for the cam sweep that was completed, and the lowest value tested point was selected to be the BSFC cam position. The COV cam sweep for the 1750 rpm test point can be seen in Figure 4, and similarly the lowest COV cam position was selected from the figure.

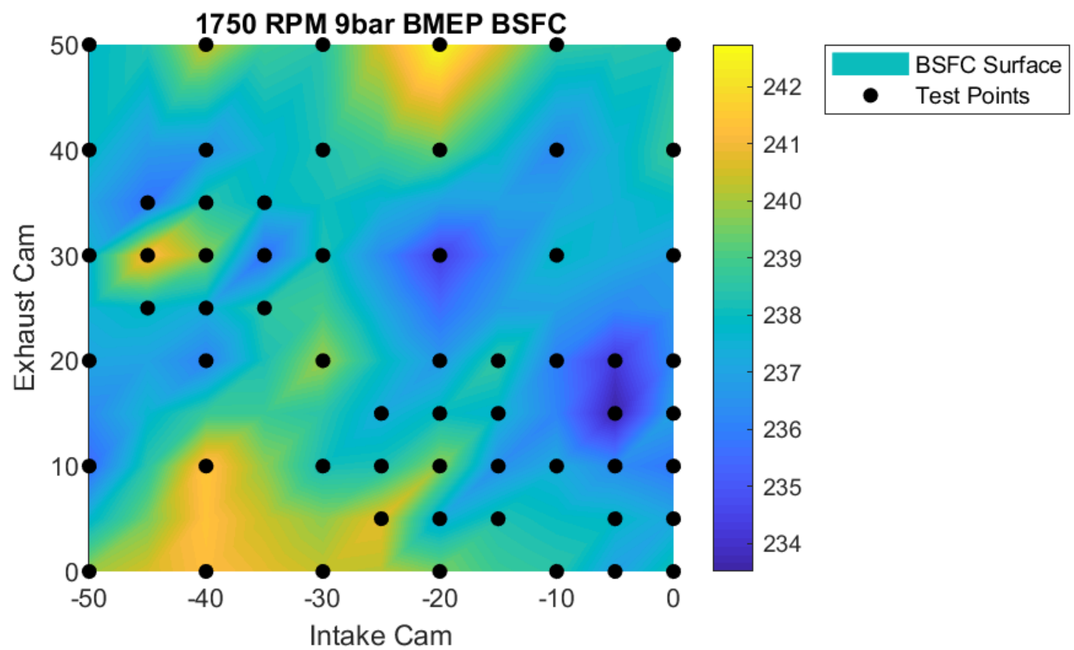


Figure 3 Example plot showing the cam sweeps completed for BSFC cam position

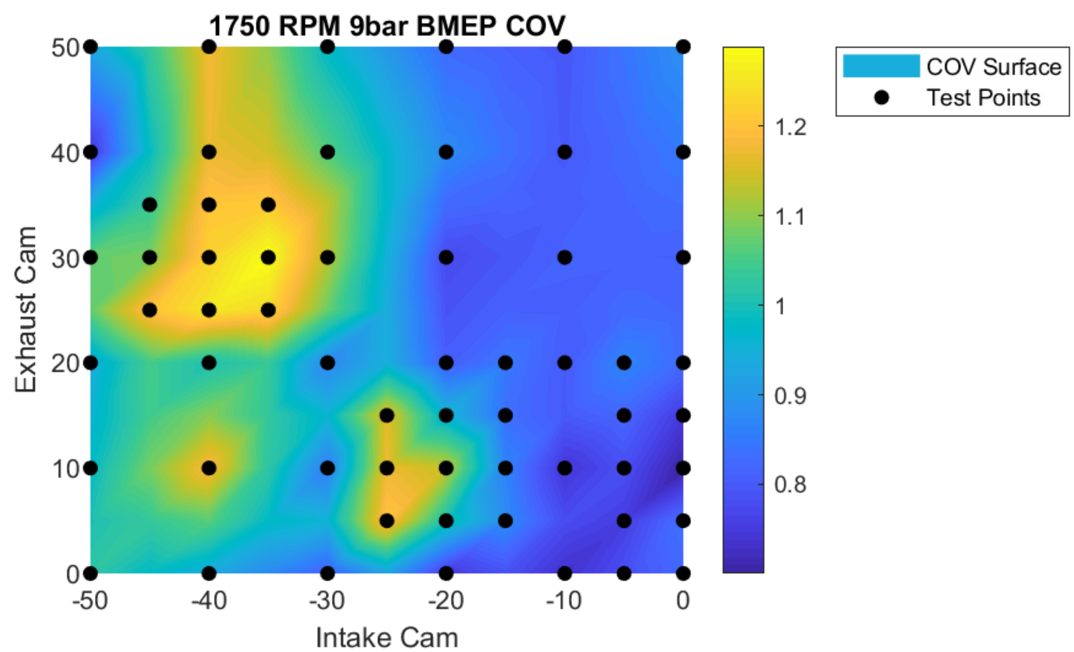


Figure 4 Example plot showing the cam sweeps completed for COV cam position

4.1.6 Water Spray Strategy

With the dislodging of CAC condensation being very unpredictable, this study utilized water injection as the method of introducing water into the combustion system. Water injection allows for repeatability in the testing and better comparison between speed/load points tested. Each of the speed/load points have a different intake air flow rate. If water were introduced into the CAC, it would be hard to quantify the actual amount of water that enters the combustion system due to flow dynamics in the intake track. PWI was the best option to be able to quantify the amount of water that was entering each of the cylinders. Water injection also allows for the water distribution between cylinders to be equal. Table 3 summarizes the decision process behind utilizing water injection for this study. The two water injection spray strategies were tested in this study. The way that each of the strategies will be referred to is as "Atomized" and "Non-Atomized". These two spray strategies were utilized to see the difference that water introduction can have on the combustion system.

Table 3 pros and cons of port water injection verses introducing water into the CAC

Port Water Injection		**Introducing Water In CAC**	
Pros	**Cons**	**Pros**	**Cons**
Each cylinder gets equal amount	Not identical to field issue	Represents field issue	Uneven water distribution
Quantifiable amount per cylinder			Not quantifiable per cylinder
Very repeatable			Water transport dynamics may lead to erratic water delivery (measured flow may not equal ingested mass due to system filling / emptying)

4.1.6.1 Atomized Test Setup

For the atomized spray test setup, the injectors were installed in the manifold with no restrictions to allow for an optimum spray pattern. The right side of Figure 6 shows an example of the stock injector spray.

4.1.6.2 Non-Atomized Test Setup

Non-atomized spray was accomplished with a restriction in front of the injector to coalesce the fine, high velocity, spray streams into a single low velocity stream. The part in Figure 5 was created to take a production Bosch EV14 KxT spray pattern and turn it into a single stream. The injector fits into the top of Figure 5 similar to how a port injector is fitted into an intake manifold, and then the bottom fits into the injector port on the intake manifold. With the spray disruptors installed the water rail was spaced out a distance to allow the injectors and spray disruptors to seal properly. Figure 6 shows the results of the spray disrupters verse the stock injector spray pattern. The view in Figure 6 is looking from the cylinder toward the intake valves. The spray disrupters in this bench test were able to achieve large droplets compared to the small droplets that came from the stock injectors.

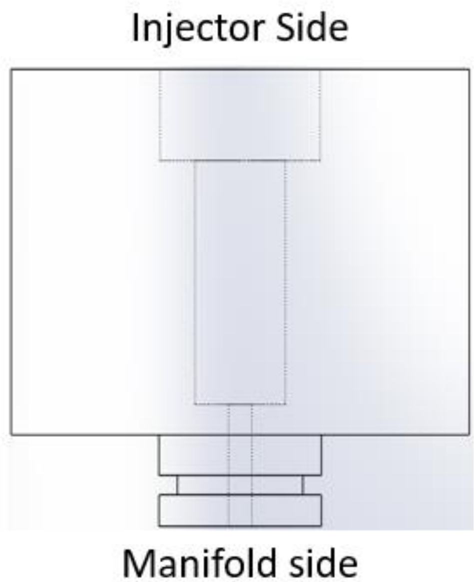


Figure 5 injector spray disruptors


Figure 6 right image atomized spray, left image non-atomized spray

4.2 Test Matrix

The data that was recorded in this study was completed at steady state conditions. All the data recording started after the engine “stabilized” for each of the test conditions that were being tested. No throttle transient or water injection transient were recorded in the data set. By recording the engine at a steady state condition, this allowed for increased test repeatability. Table 4 is the testing matrix as decided by the light-duty consortium to present enough test points to create an understanding of water ingestion in non-optimum conditions. These test points are also speed/load cases that are not usually tested with water injection. In Table 4 the 1250 rpm test point only has one cam position, and this is due to the test point being close to a WOT condition. The single cam position was to ensure the constant air charge could be maintained while the water sweep was performed.

Table 4 Testing matrix

RPM	BMEP (bar)	Cam Position	Spark, Air, Fuel Test Strategy
Atomized Spray			
1300	3	I-8,E38	Constant Spark, Air, Lambda
		Best BSFC	Constant Spark, Throttle, Fuel Flow
		I-28,E8	Constant Spark, Air, Lambda
		Min COV	Constant Spark, Throttle, Fuel Flow
1750	9	I-5,E15	Constant Spark, Air, Lambda
		Best BSFC	Constant Spark, Throttle, Fuel Flow
		I0,E10	Constant Spark, Air, Lambda
		Min COV	Constant Spark, Throttle, Fuel Flow
3250	15	I-30,E10	Constant Spark, Air, Lambda
		Best BSFC	Constant Spark, Throttle, Fuel Flow
		I-45,E45	Constant Spark, Air, Lambda
		Min COV	Constant Spark, Throttle, Fuel Flow
2500	8	I-50,E35	Constant Spark, Air, Lambda
		Best BSFC	Constant Spark, Throttle, Fuel Flow
		I-0,E20	Constant Spark, Air, Lambda
		Min COV	Constant Spark, Throttle, Fuel Flow
1250	10	I-22, E16	Constant Spark, Air, Lambda
		Vol. Eff.	Constant Spark, Throttle, Fuel Flow
Non-Atomized Spray			
1750	9	I-5,E15	Constant Spark, Air, Lambda
		Best BSFC	Constant Spark, Throttle, Fuel Flow
3250	15	I-30,E10	Constant Spark, Air, Lambda
		Best BSFC	Constant Spark, Throttle, Fuel Flow
2500	8	I-50,E35	Constant Spark, Air, Lambda
		Best BSFC	Constant Spark, Throttle, Fuel Flow
1250	10	I-22, E16	Constant Spark, Air, Lambda
		Vol. Eff.	Constant Spark, Throttle, Fuel Flow

5 Experimental Setup

5.1 Engine Setup

The engine used for this testing was a GM LHU. The GM LHU is a 2.0L turbocharged 4-cylinder engine with Direct Fuel Injection (DI) and Dual Independent Variable Cam Phasing (DIVCP) and is representative of many engines that are currently being produced for the light-duty passenger car market. Characteristics that make this engine an ideal candidate for this testing is the dual overhead cams with variable phasing. Table 5 is the specifications for the GM LHU.

Table 5 Engine Specifications

Bore (mm)	86.0
Stroke (mm)	86.0
Compression Ratio (-)	9.2
Connecting Rod Length (mm)	145.5
Wrist Pin Offset (mm)	0.8
Number of Cylinders	4
Firing Order	1-3-4-2
Fuel Delivery	Direct Injection
Water Delivery	Port Injection
Oil Cooling	Oil to Building Process Water (closed loop temperature control)
Charge Air Cooler	Air to Building Process Water (closed loop temperature control)

5.1.1.1 Crankcase Ventilation

The engines crankcase was ventilated to atmosphere via a large catch can. The catch can was designed to allow the oil and water venting with the air from the crankcase to be separated before the air is vented to atmosphere. This also enabled the ability to monitor the amount of water venting out of the crankcase, but the amount of water that is coming into the combustion from the crankcase is unknown.

5.1.2 General Engine Operation

5.1.2.1 Oil Cooling and Engine Coolant

On this test setup the water that traveled through the oil cooler was separate from the engine coolant. The water that travels through the oil cooler is limited by a valve that is controlled by closed loop system to regulate the engine oil temperature. During testing the oil control system was set to maintain an oil temperature of 90°C.

During testing there was some evidence that water was entering the crankcase. This evidence was only present in the oil fill port and dipstick, but when the oil drain plug was cracked, there was no evidence of water entrainment. In order to prevent further water entrainment in the oil, an oil drying procedure was followed. Once water injection testing on the engine had been complete for a day, the engine oil was heated via engine operation to above 100°C. The engine oil was held at this elevated temperature for a minimum of ten minutes to promote the removal of water from the engine oil. The procedure of promoting the removal of water from the engine oil was critical due to the high W/F that were tested in this study. This procedure also helps maintain the oil from reaching a critical water content. When the engine oil becomes entrained with water, the oil can lose its lubricity and that could be catastrophic to the engine due to the lack of engine protection.

The engine that was under test, utilized the stock coolant circuit and maintained the engine coolant thermostat. The engine coolant was controlled via similar closed loop system as the oil system. This system allows the coolant temperature that was entering the engine to be maintained. During this testing, the coolant inlet temperature was set to 75°C.

5.1.2.2 Intake Air Temperature

The CAC that was used for this testing was a water to air cooler. The water that was flowing through the cooler was regulated to maintain a constant cooler out air temperature. This system was setup similar to the oil cooler configuration. For this testing, the cooler out temperature was set to 25°C for MAP below 100kPa and set to 35°C for MAP above 100kPa.

5.1.3 Water Injection Setup

In Figure 7 is a schematic of the water system that was used in this study. This section will discuss the different parts of the water system seen in the schematic.

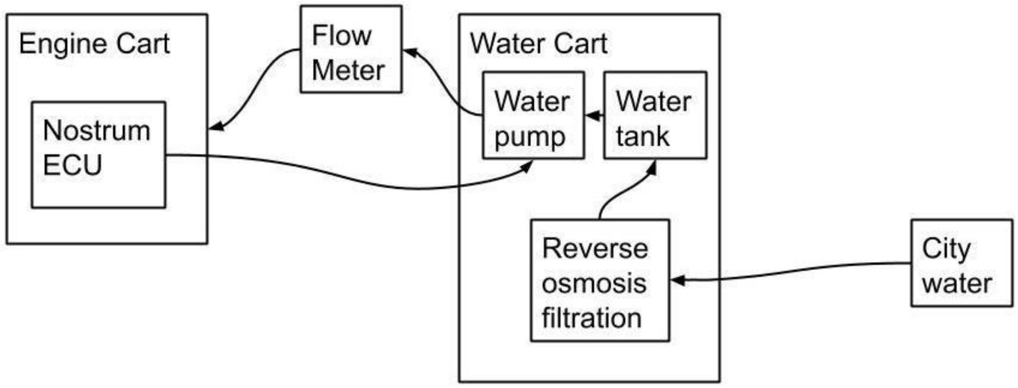


Figure 7 Diagram of Water delivery

5.1.3.1 Water Cart

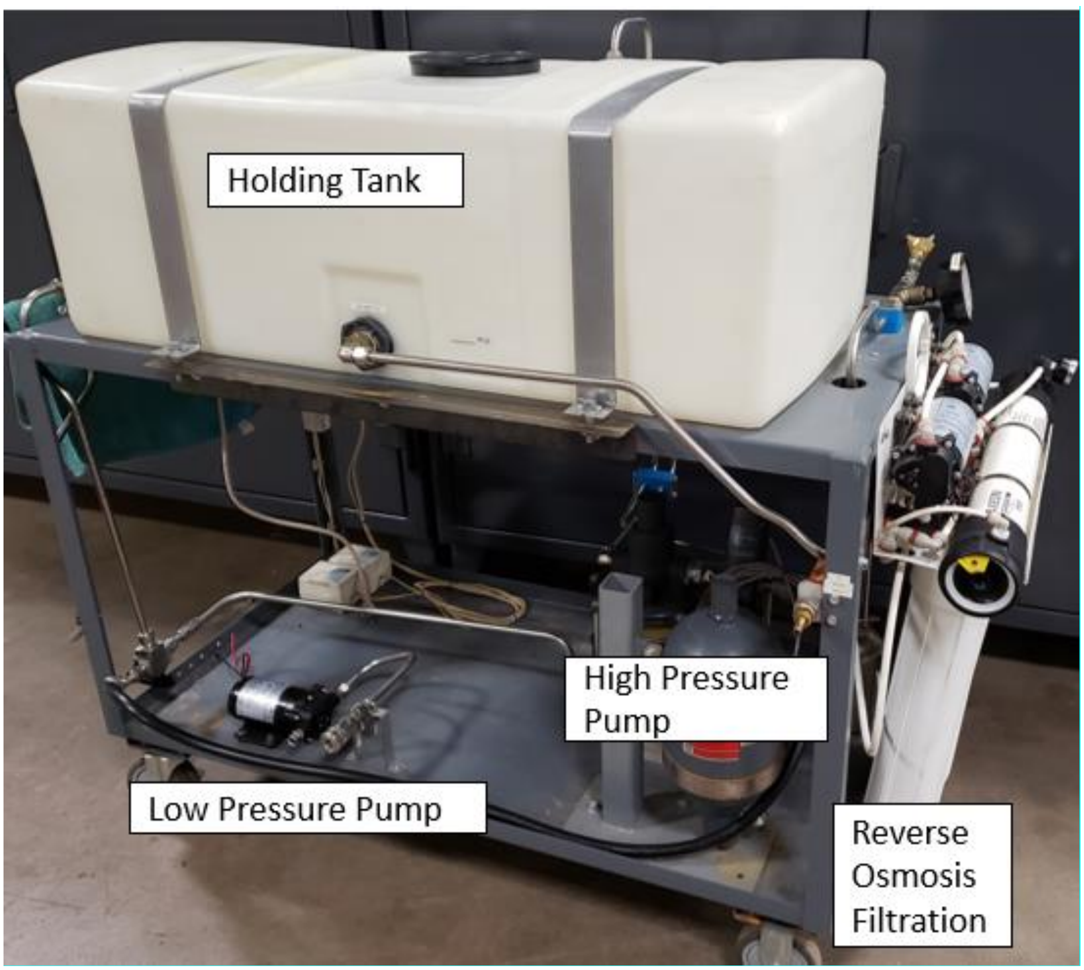


Figure 8 picture of the water cart

The water that is being supplied to the engine cart is coming from a water cart. An image of the water cart can be seen in Figure 8. The water cart utilizes a reverse osmosis filtration system. The filtered water is then fed into a 150L holding tank. The water from the tank is then fed to one of the two water pressure production systems on the cart. The two water production systems can be classified as high- or low-pressure system. The low-pressure water pump seen in Figure 8 was responsible for the low-pressure water injection demands and was mainly configured for PWI. The low-pressure system was exclusively used in this study. The high-pressure pump that is seen in Figure 8 is capable of suppling water pressure for a DWI system. The Nostrum ECU is responsible for regulating the water pressure by adjusting the PWM to low-pressure water pump. The water rail pressure was regulated to 400kPa and under boosted condition the water pressure increased proportionally to the amount of boost.

5.1.3.2 Water Injection

The water injection intake manifold setup was used on previous testing done by Worm [7] and can be seen in Figure 9. There were two different injectors used in testing one set of Bosch EV14 KxT with a high flow rate and a low flow rate set of Bosch EV14 KxT. The part specifications for the injectors can be seen in Table 7. Table 6 shows what injectors were used for each of the test points. Except for the 2500 rpm test point all the test points only used one set of injectors for the water sweeps. The 2500 rpm test point needed the low flow injectors for a W/F of 0.5 only.

On the 1300 rpm test point, it was discovered through testing that the low flow injectors did not have significant turndown. Due to the way the intake manifold was fabricated, only extended tip injectors could be used, and this limited the selection of injectors that could be used for testing. So being that the small injectors had four outlet holes two of these were welded shut to create an even lower flow injector. All testing with the low flow injectors in their stock configuration was completed before the modification. These modified low flow injectors are labeled as “welded” in Table 6.

The injectors were targeted at the septum between the two intake valves. The two circles seen in Figure 10 denote the approximate spray size/location, the large circle is for the high flow injectors and the smaller circle is the low flow injector, and the red dot is the approximate center of the injector spray. The welded injectors are not represented in Figure 10 due to a lack of manufacture spray characteristics due to the modification. This figure is showing the injector spray was targeted to allow equal water distribution on both intake valves.

Figure 9 Water injection manifold, water rail and injectors

Table 6 injectors used for each of the test points

RPM	BMEP (bar)	Water Injector Used
1300	3	Welded
1750	9	Large
1250	10	Large
3250	15	Large
2500	8	Large*

*small injectors used for the lowest W/F point on 2500 test point only

Table 7 injector specification used in this study [26]

	High Flow Injectors	Low Flow Injectors	Welded Injectors
Manufacturer	Bosch	Bosch	Bosch
Part No.	0 280 158 040	0 280 158 038	0 280 158 038
Flow rate/min	670g/980cm3	237g/347cm3	Unk.
Spray Type	Conical Spray	Conical Spray	Unk.
Housing	KxT	KxT	KxT
Spray Angle	30°	20°	Unk.
Bent Angle	0°	0°	Unk.
Delta	0°	0°	Unk.
Resistance	12 Ohm	12 Ohm	12 Ohm

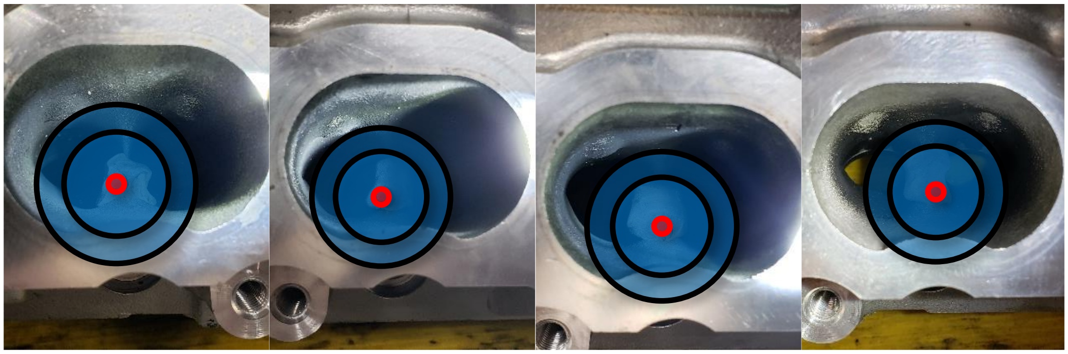


Figure 10 Approximate injector targeting for both injectors

For all the testing in this study, the water injection strategy was anchoring SOI at 90°CA before TDC firing. An EOI strategy for water injection was not used due to the way the water system ECU was configured. The water ECU was only configured for SOI anchoring. The SOI position was selected to allow the maximum injection on a closed intake valve. This injection strategy also allows the same duration for the manifold to reach the water saturation point. Figure 11 shows the SOI location of the water injection events, and the intake valve displacement for extremes of the cam phasing. There was one limitation to the injection system that was discovered during testing, and that was that the injector signal would be clipped at TDC gas exchange. This was only an issue at the high W/F for the 3250 rpm test point.

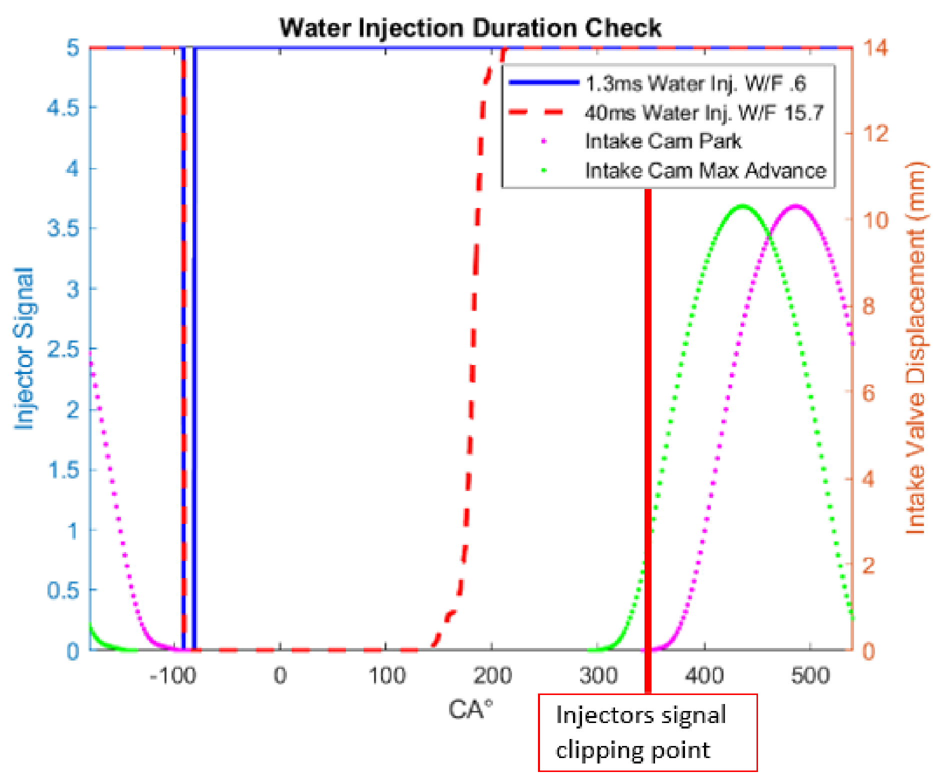
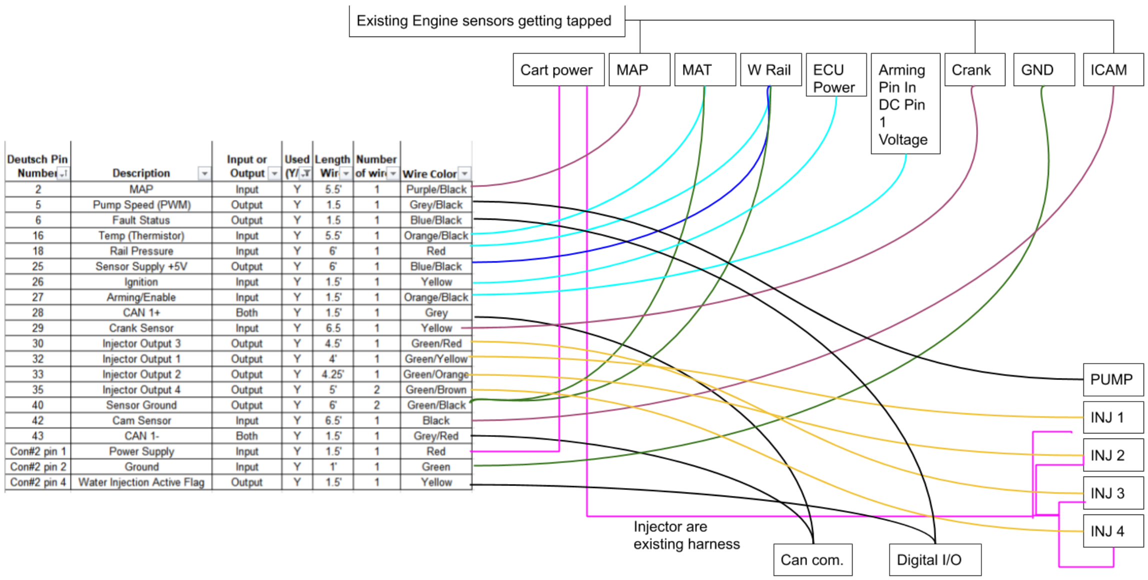


Figure 11 water injection strategy plot

5.1.3.3 Water Injection Control

The water injection control is completed by utilizing a second ECU that was supplied by Nostrum Energy. This ECU is designed to be a production intent setup to be modular with existing engines. The ECU is a Pi Innovo controller that has been configured for this application. The ECU utilized existing engine sensors such as MAP, intake cam position sensor, and crank position sensor. A couple of sensors were also added such as a MAT, and water rail pressure transducer. Figure 12 shows how the Nostrum ECU is wired to the engine and shows the inputs and outputs as they were wired.



Deutsch Pin Number	Description	Input or Output	Used (Y/	Length Wir	Number of wir	Wire Color
2	MAP	Input	Y	5.5'	1	Purple/Black
5	Pump Speed (PWM)	Output	Y	1.5	1	Grey/Black
6	Fault Status	Output	Y	1.5	1	Blue/Black
16	Temp (Thermistor)	Input	Y	5.5'	1	Orange/Black
18	Rail Pressure	Input	Y	6'	1	Red
25	Sensor Supply +5V	Output	Y	6'	1	Blue/Black
26	Ignition	Input	Y	1.5'	1	Yellow
27	Arming/Enable	Input	Y	1.5'	1	Orange/Black
28	CAN 1+	Both	Y	1.5'	1	Grey
29	Crank Sensor	Input	Y	6.5	1	Yellow
30	Injector Output 3	Output	Y	4.5'	1	Green/Red
32	Injector Output 1	Output	Y	4'	1	Green/Yellow
33	Injector Output 2	Output	Y	4.25'	1	Green/Orange
35	Injector Output 4	Output	Y	5'	2	Green/Brown
40	Sensor Ground	Output	Y	6'	2	Green/Black
42	Cam Sensor	Input	Y	6.5'	1	Black
43	CAN 1-	Both	Y	1.5'	1	Grey/Red
Con#2 pin 1	Power Supply	Input	Y	1.5'	1	Red
Con#2 pin 2	Ground	Input	Y	1'	1	Green
Con#2 pin 4	Water Injection Active Flag	Output	Y	1.5'	1	Yellow

Figure 12 Wiring Schematic for Nostrum ECU on Engine Side

5.1.4 Sensors

5.1.4.1 Thermocouples

All the thermocouples are K-type. Temperatures measured are: oil gallery, EGTs, pre and post turbine, pre and post compressor, throttle body inlet, intake manifold plenum, coolant in and out, fuel in, and water in.

5.1.4.2 Cylinder Pressure Transducers

Pressure transducers are referenced to an Omega 0-50 psi (0 – 3.4 bar) absolute pressure transducer located in the intake manifold. The cylinder pressure transducers are referenced to the MAP sensor at BDC gas exchange. [27] The in-cylinder pressure transducers that were used in this study are listed in Table 8, and they are 5mm face sealing transducers with a 1 mm passage into the cylinder with no thermal protection.

Table 8 Pressure Transducers in Engine

	Cylinder 1	Cylinder 2	Cylinder 3	Cylinder 4
Transducer	AVL GH12D	PCB 115A04	AVL GH12D	AVL GH12D

5.1.4.3 Data Acquisition

Two data acquisition systems were used in this study. One of the systems was for the non-crank angle resolved measurements such as thermocouples. The DAQ utilized for this was an NI PXIe-1078 utilizing the NI Veristand 2018 software. The other acquisition system was for crank angle resolved measurements for combustion analysis. A 360 pulse per revolution BEI model XH20DB-37-SS360-ABZC-28V/V-SM encoder was used for crank angle resolved measurements. TDC location was determined utilizing a Kistler Type 2629DK TDC probe. The combustion analysis was completed using A&D Redline II CAS system running the software version 3.7 of A&D CAS.

5.1.4.4 Pressure Transducers

The pressures that are being measured on the engine are MAP, oil gallery, barometric, exhaust pressure pre-turbine, water rail, and fuel pressure.

5.1.4.5 Flow Meters

The water and fuel flow into the engine are measured by Emerson Micro Motion Mass Flow meters model CMFS015H520N0A2ECZZ. Air flow into the engine was measured by a Meriam LFE and a Meriam MDT500 transmitter models Z50MC2-4 and ZMDT500-10-38-MT, respectively. The LFE is placed before a 200L surge tank that is in the intake track before the intake air reaches the engine cart. The surge tank is used to reduce the pulse in the intake stream that are caused by the engine. The humidity of the air that is entering the engine was not directly measured, but the weather humidity was recorded with each of the data sets.

5.1.4.6 Test Cell Equipment

The dynamometer that was in the test cell that was used for all of this testing was a GE AC dynamometer rated at 375kW model 5TKF44SDC03AQ04. Dynamometer torque measured by a PCB load cell model 1403-13A.

5.1.5 Engine Control

An aftermarket ECU was utilized due to limitation in the production ECU from the GM LHU as found by Worm [28]. The aftermarket ECU used was Bosch Motorsports MS6.3. Some of the many functions this ECU offered were cam control, closed loop lambda correction, and spark timing.

6 Results

This section will be reviewing the results of the testing that was complete as a part of this study.

6.1 Individual Cylinder Results

All cylinder results were reviewed during the data processing of the testing, and during this review large variations in an individual cylinder were sought out. Figure 13, and Figure 14 are an example of the results from all four cylinders. These figures show that all four of the cylinders trend closely together. The rest of the results that will be presented will be an average of the three cylinders due to a transducer cable failure during testing. There is one exception, the IMEP LNV plots are the lowest value from all the cylinders.

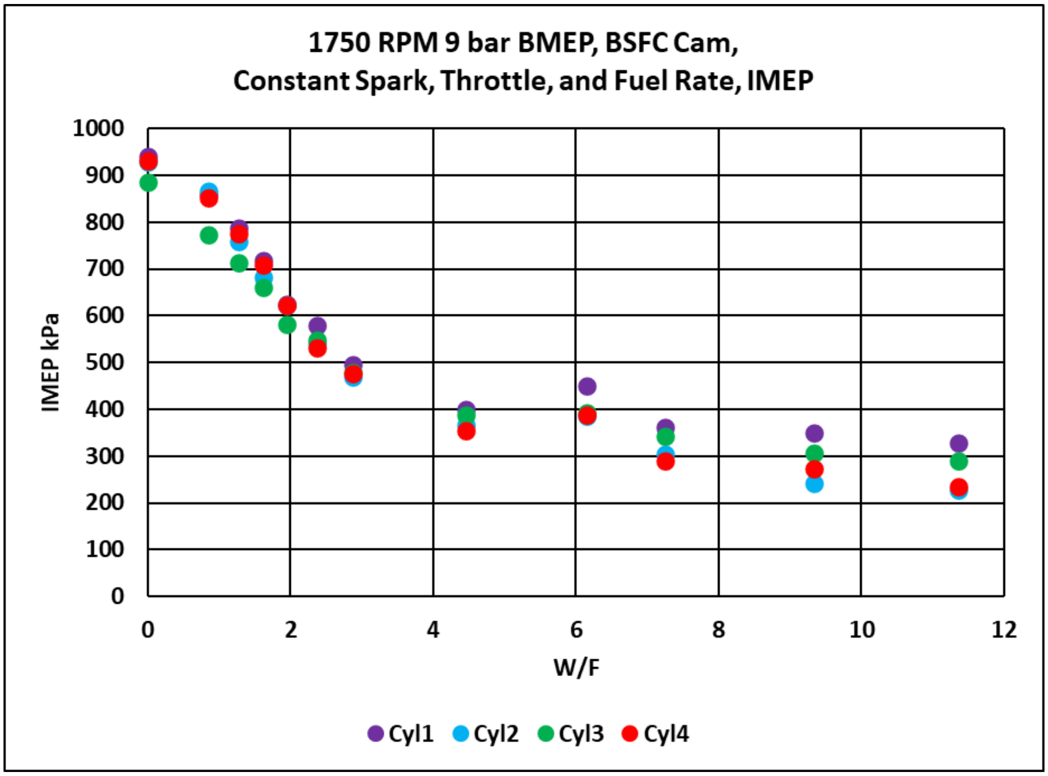


Figure 13 1750 rpm 9 bar BMEP 4 cylinders results example IMEP plot

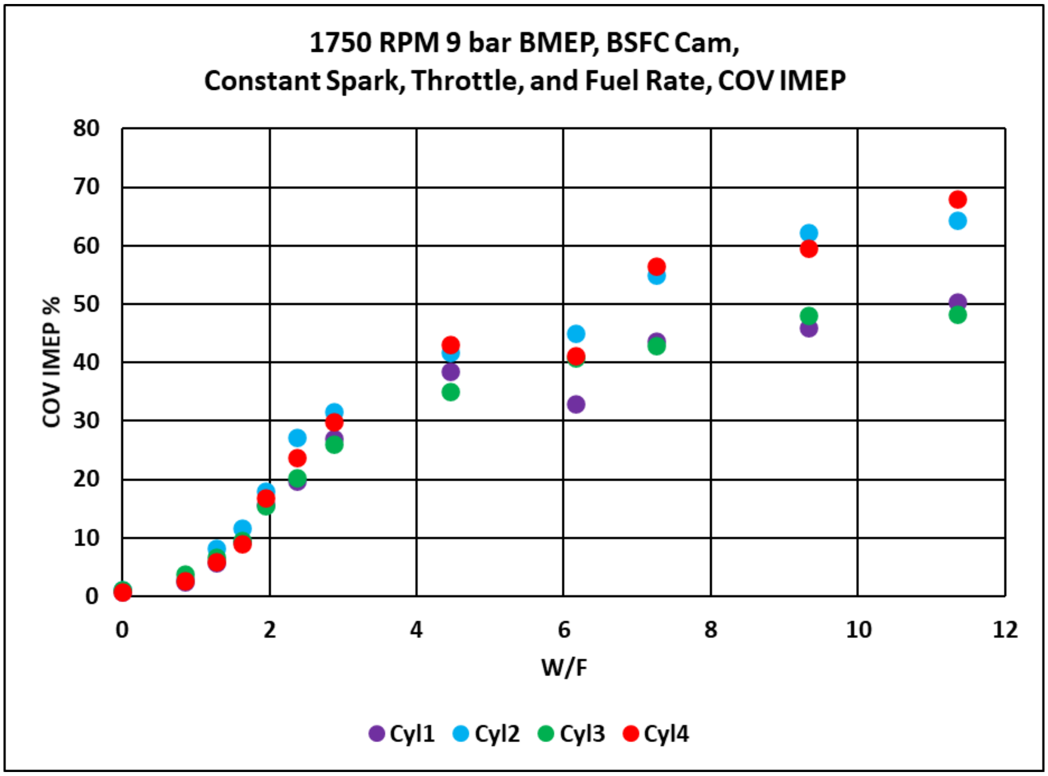


Figure 14 1750 rpm 9 bar BMEP 4 cylinders results example COV of IMEP plot

6.2 Atomized testing

The atomized testing utilized the injectors referenced in section 5.1.3.2. The plots seen in this section will be engine average values for the different test methods. Each of the test points will have four sets of data present seen in Table 9. The 1250 rpm test point will only have two sets of data present because only one cam position was tested.

Table 9 data sets to be presented

BSFC Cam Position	Constant Air Charge, Constant Spark
	Constant Throttle, Constant Spark, Constant Fuel Rate
COV Cam Position	Constant Air Charge, Constant Spark
	Constant Throttle, Constant Spark, Constant Fuel Rate

Figure 15 shows COV of IMEP plotted for the 1750 rpm test point. Both cam positions and air charge configurations have similar trends. The constant throttle test trends to a higher COV than the constant air charge at W/F above 7. This may be due to the added instability due to the nature of the constant fuel rate and its interaction with the addition of water. The COV of IMEP curve starts to plateau at a W/F of 4 for the 1750 rpm test point.

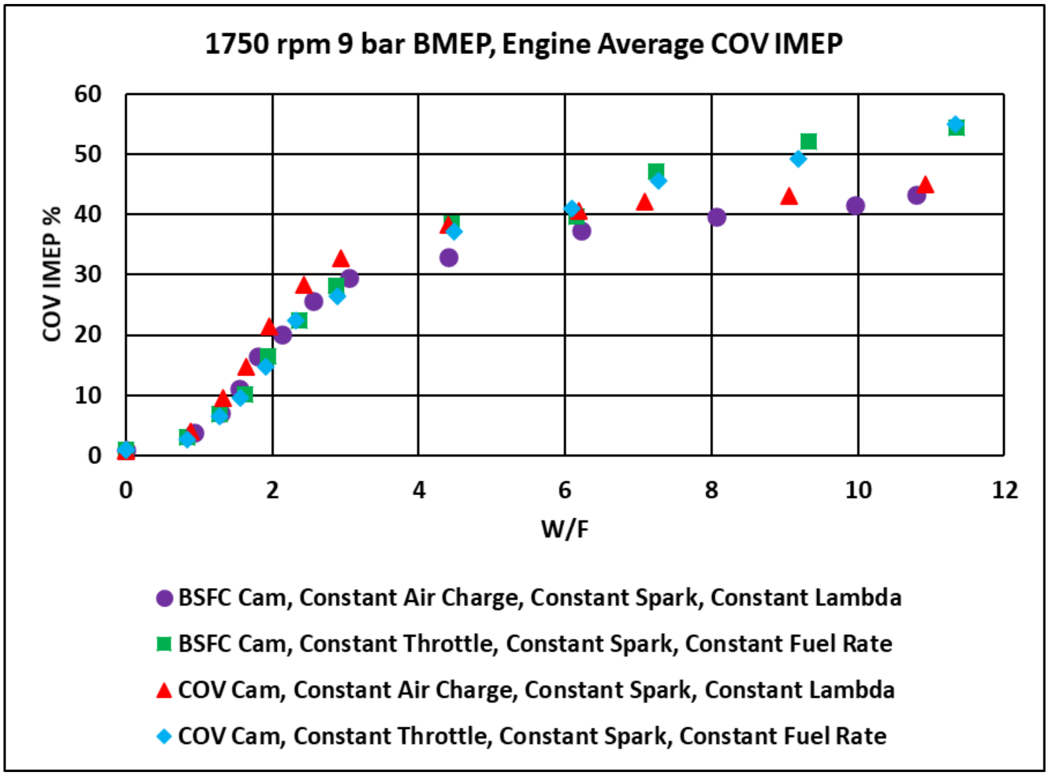


Figure 15 1750 RPM test point atomized spray COV of IMEP plot

Figure 16 is the COV of IMEP plot for the 2500 rpm test point. This has a similar result as the 1750 test point, but COV of IMEP rises faster at lower W/F. This is due to the slightly lower load at the 2500 rpm test point. This test point also sees the rise rate of COV decrease at a W/F of 3.

The 1300 rpm test point is the lowest load point presented in this study and plot for COV of IMEP and is presented in Figure 17. This test point utilizes the welded low flow injectors. This test point is the least resilient to combustion degradation caused by the addition of water into the combustion system.

Figure 18 are the plots of COV of IMEP for the 3250 rpm test point. Due to the high engine speed and load, and water injection system limitations, the plateau effect that is seen at the other test point is not present here.

Figure 16 and Figure 17 see that the min COV cam has a slightly lower COV when compared to the best BSFC cam. This is likely due to the COV cam having more combustion stability initially than the BSFC cam, and during the water sweep the initial stability is more resilient to the combustion degradation.

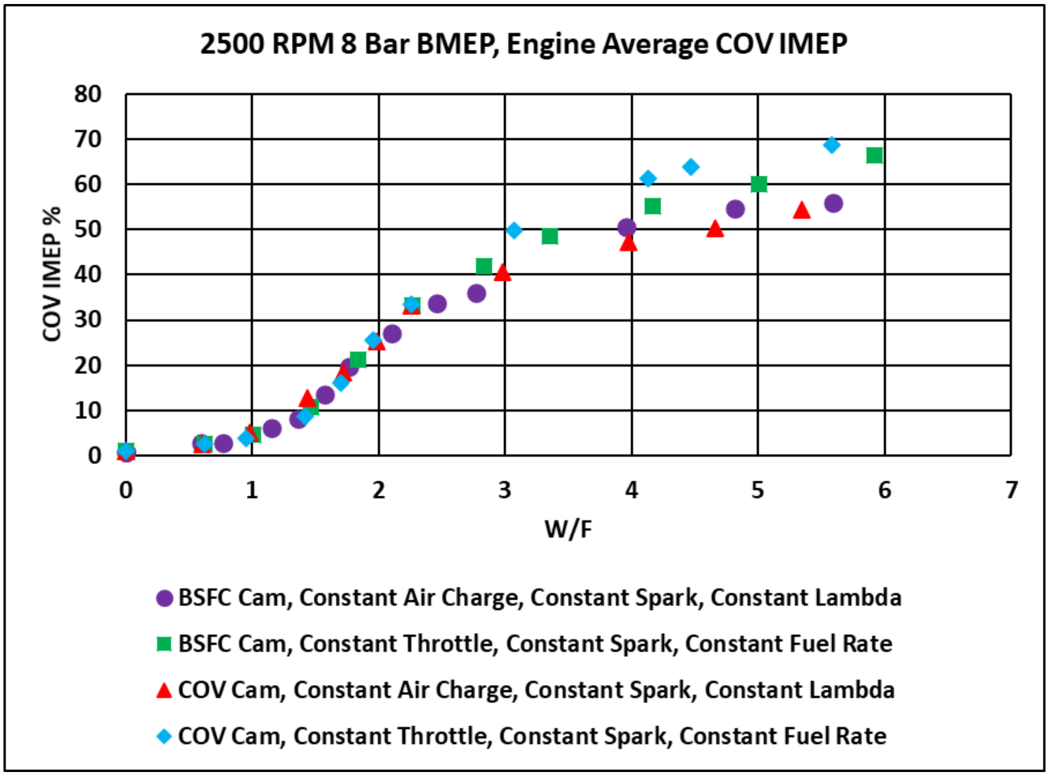


Figure 16 2500 RPM test point atomized spray COV of IMEP plot

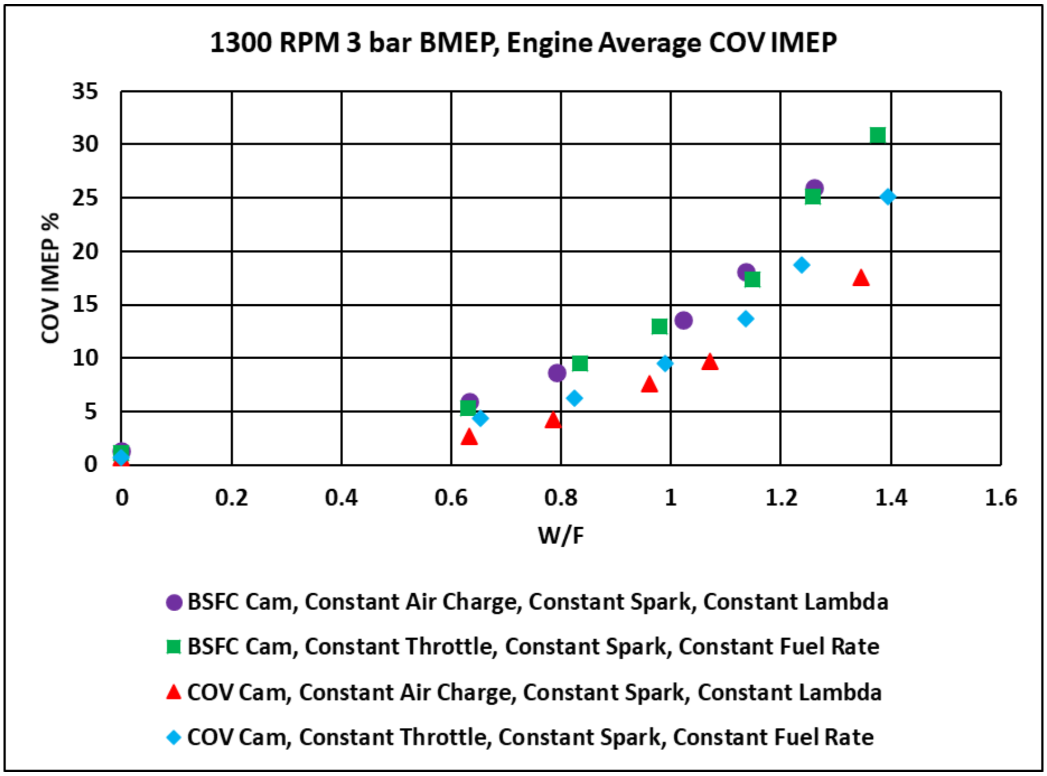


Figure 17 1300 RPM test point atomized spray COV of IMEP plot

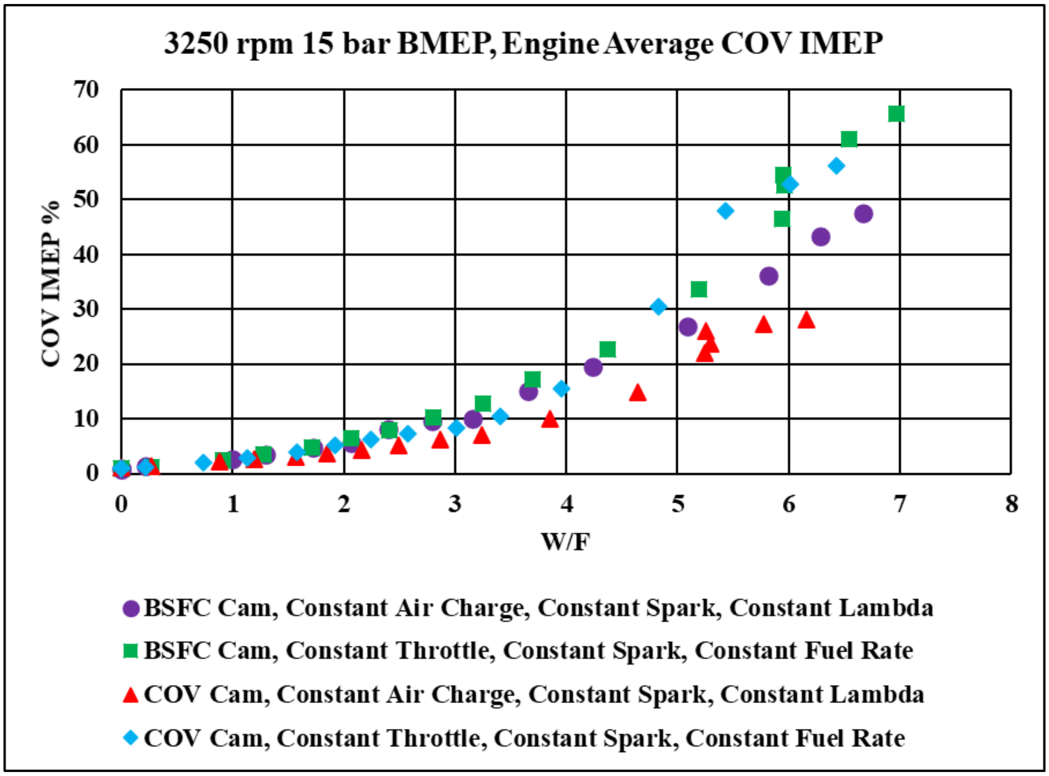


Figure 18 3250 RPM test point atomized spray COV of IMEP plot

As seen in the COV of IMEP plot, there was some discussion that some of the test points reached a plateau point. The cause of this plateau point is theorized to be the point when air in the intake runner reaches the saturation point. The air coming into the intake has a limit to the amount of water that can be absorbed based on various intake pressures, and temperatures. Figure 19 has temperature sweeps for 5 different pressure points that equate to the maximum W/F that is allowable before water vaporization stops in the intake runners. For each of the pressure curves the fuel rate is held constant from the 25°C starting point.

For example, the 1750 rpm test point was run at an approximate MAP of 100 kPa, and the plateau effect is seen at an approximate W/F of 4. Utilizing Figure 19, the approximate air temp in the intake runner is 70C°. Another test point to look at is the 2500 rpm test point. It has an approximate MAP of 90 kPa, and the plateau effect is at an approximate W/F of 3. According to Figure 19, the air temperature where the water is being injected can be estimated to be 60C°. There is some temperature gradient present as the intake air passes from the last thermocouple in the intake track to the combustion chamber. In Figure 19, the air temperature of where the water is being injected can be estimated. However, water vaporization will not only take place in the intake but will also occur in the combustion chamber.

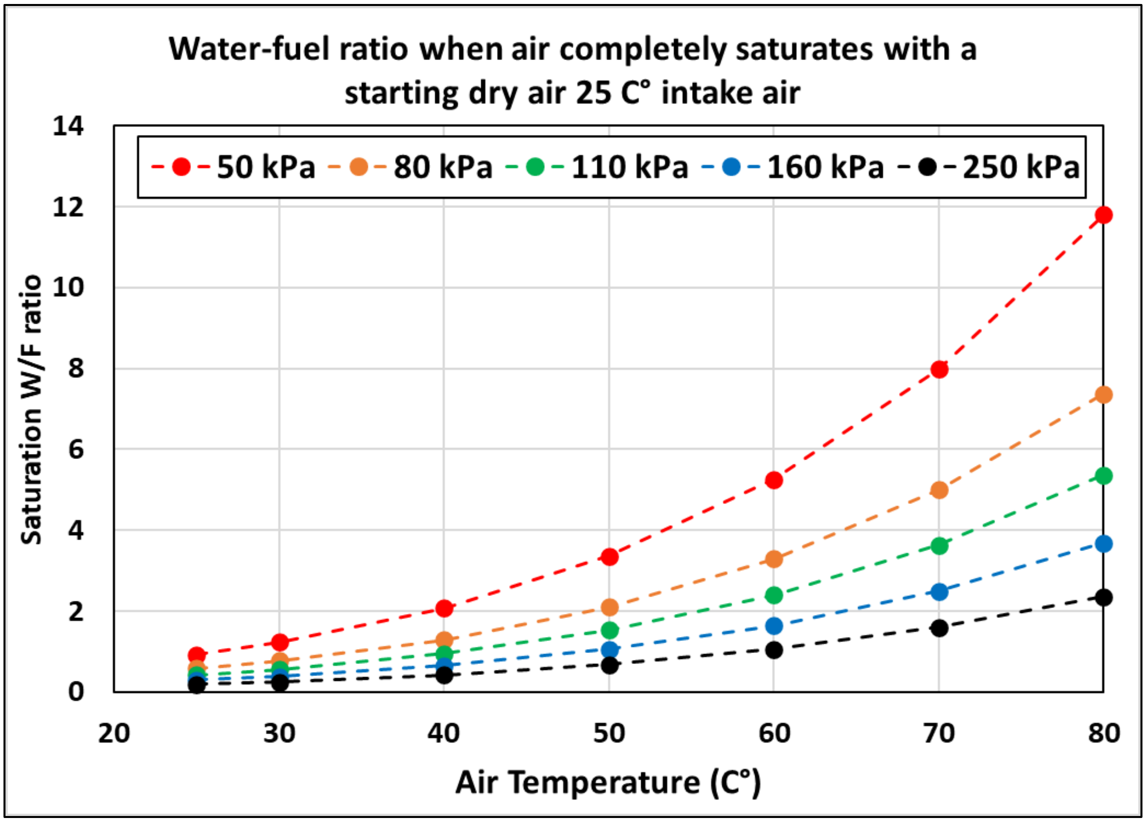


Figure 19 W/F air saturation plot starting with dry air

Figure 20 is a misfire plot for the 1250, 1750, 2500, and 3250 rpm test points. All these test points are constant throttle, spark, and fuel rate. At the high COV of IMEP seen in Figure 15 through Figure 18 misfires were not seen at similar high rates. Misfires can be considered as a non-gaussian distribution, and this could be a reason that more misfires were not captured. There is work currently being completed at the APSLABS regarding misfire distribution.

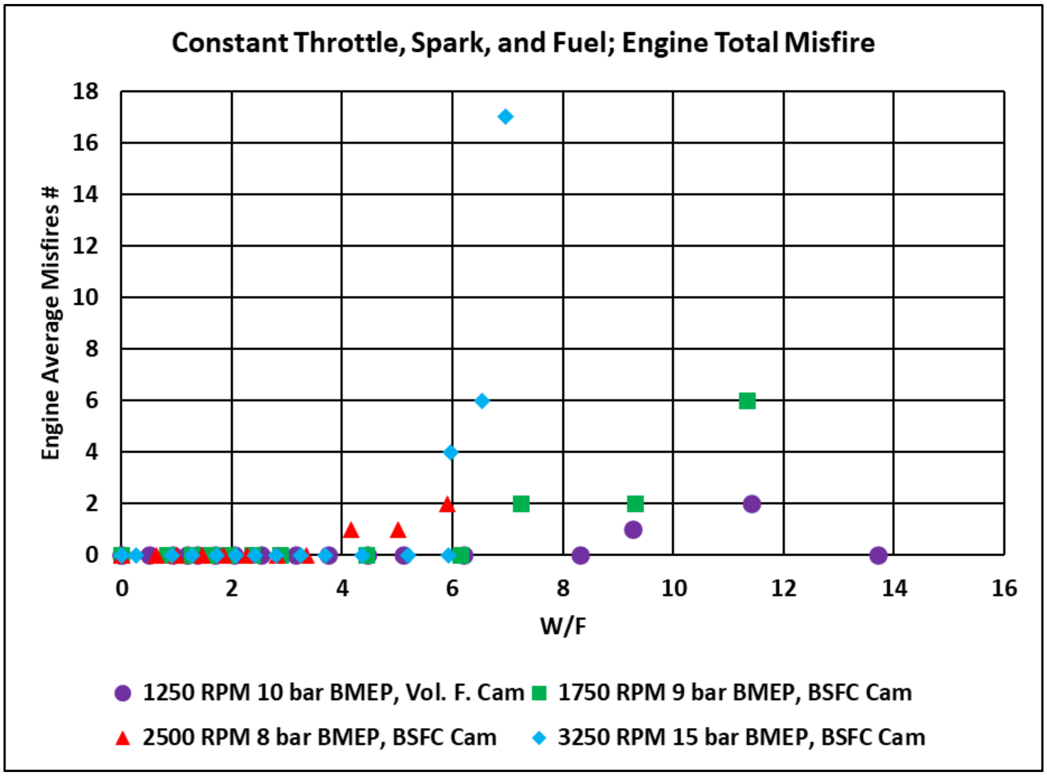


Figure 20 misfire plot for four of the speed load points

Figure 21 and Figure 22 are both IMEP plots for 1750 and 2500 rpm test points respectively, and these plots decrease to a similar plateau point. These plateau points correlate to a similar trend that is seen in the COV plots. Also, in Figure 25 the IMEP plot, for the 1250 rpm test point seems to reach a slight plateau around a W/F of 6.

Figure 23 and Figure 24 are the IMEP plots for the 3250 and 1300 rpm test points, and both of these test points have similar trend shapes. The 1300 rpm test point did not see a plateau point, and this would likely be seen if the test were continued out to a higher W/F. For the 3250 rpm test point the plateau point is not observed. One theory is that due to the high speed/load point water vaporization in the intake was limited. The test point had a MAP that was approximately 160kPa, and according to Figure 19 the max vaporization in the intake is a W/F of 1. Water vaporization is also time dependent and with the high engine speed the water that is injected is likely going into the cylinder as a liquid. Another thought is that this test point will have the highest in cylinder temperatures of all the points tested in this study. This increase in cylinder temperature is conducive to increased vaporization of water inside the combustion chamber.

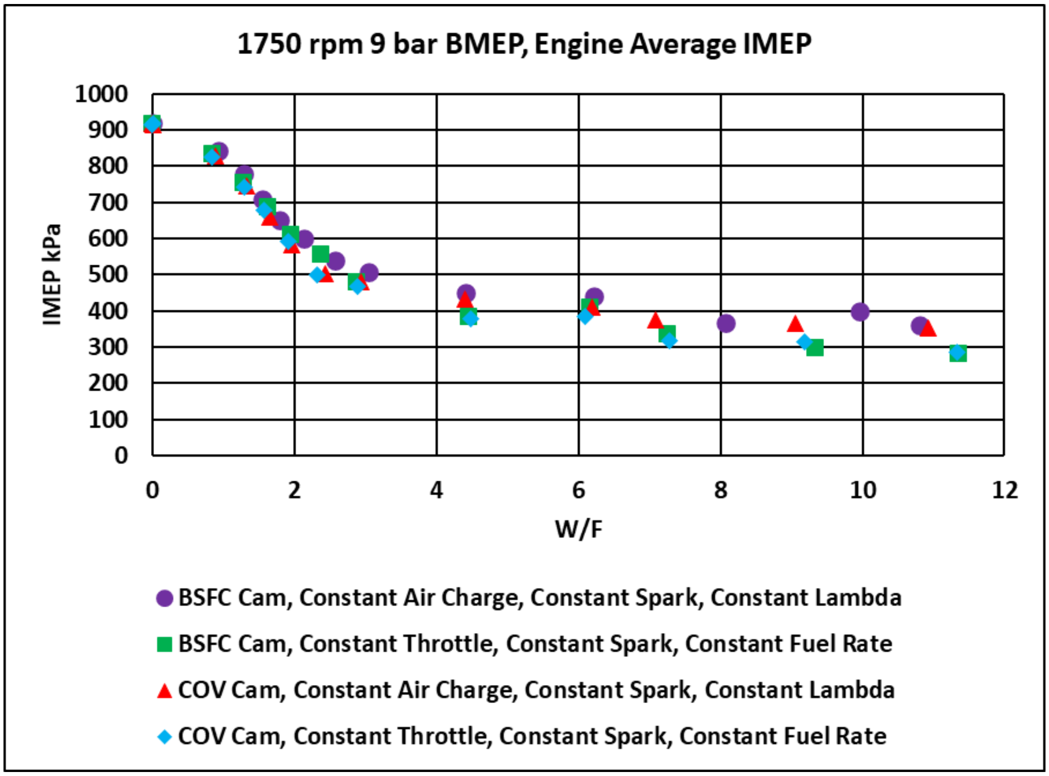


Figure 21 1750 RPM test point atomized spray IMEP Plot

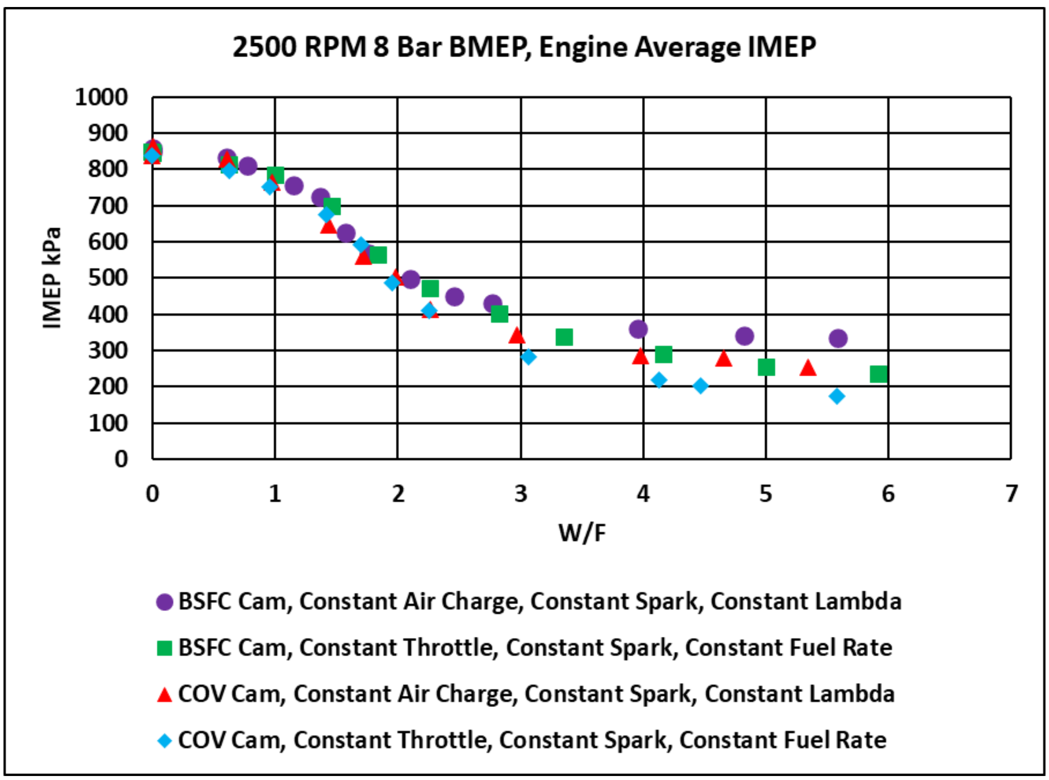


Figure 22 2500 RPM test point atomized spray IMEP plot

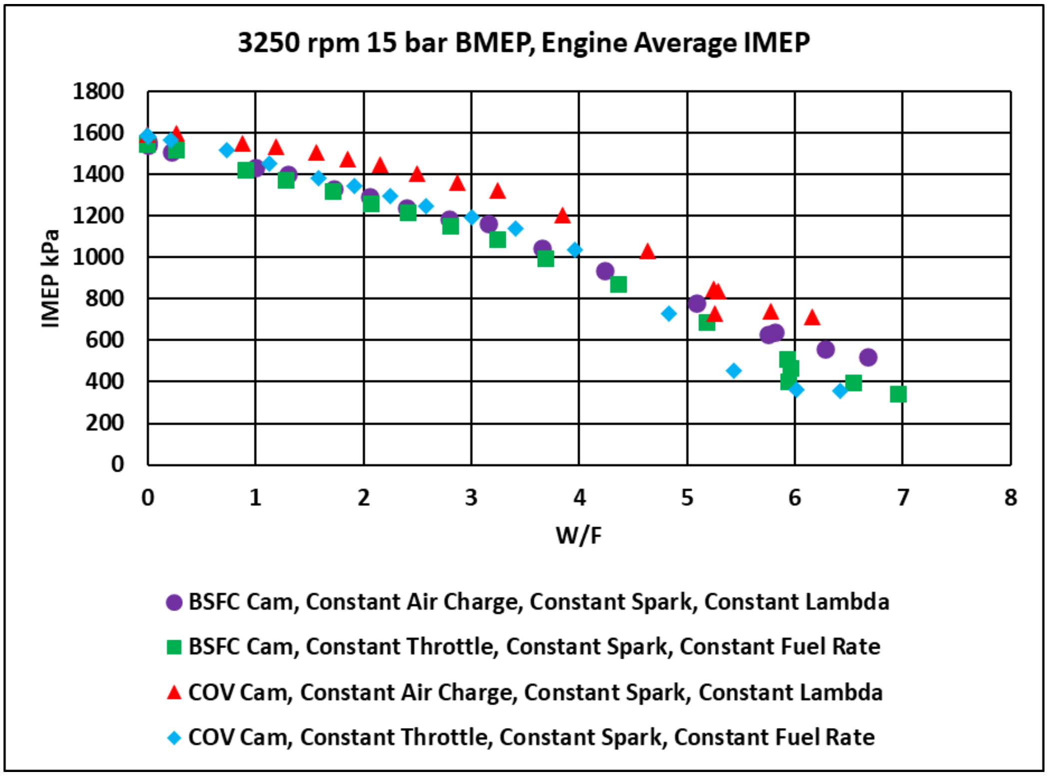


Figure 23 3250 RPM test point atomized spray IMEP plot

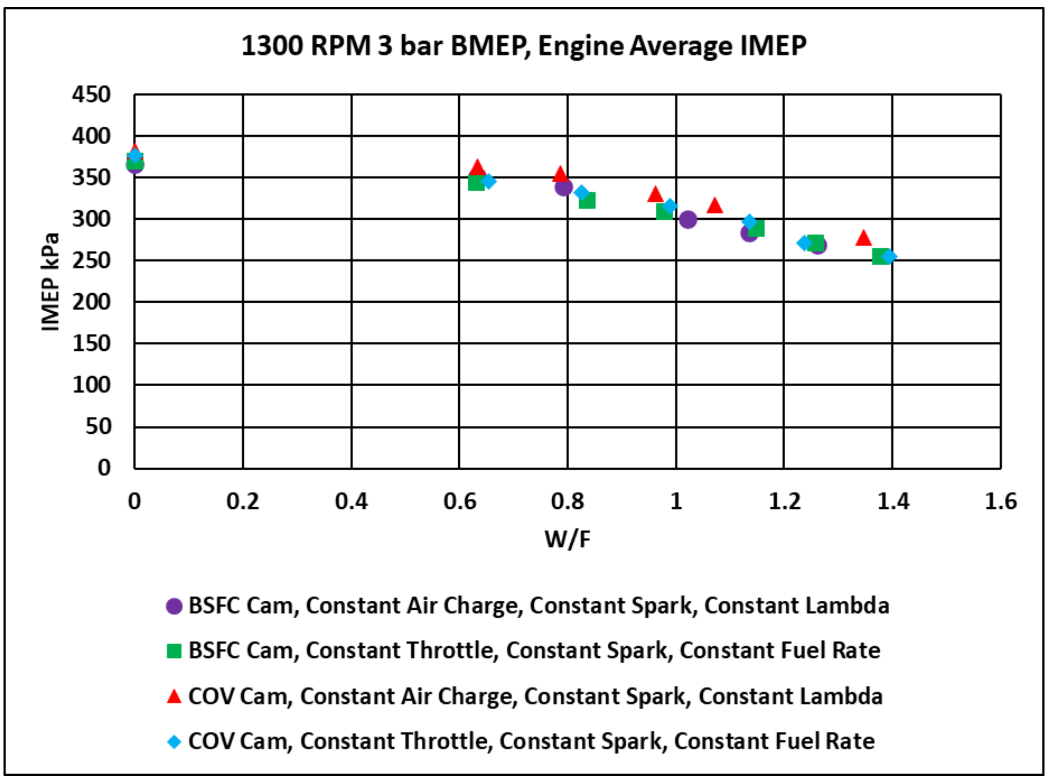


Figure 24 1300 RPM test point atomized spray IMEP plot

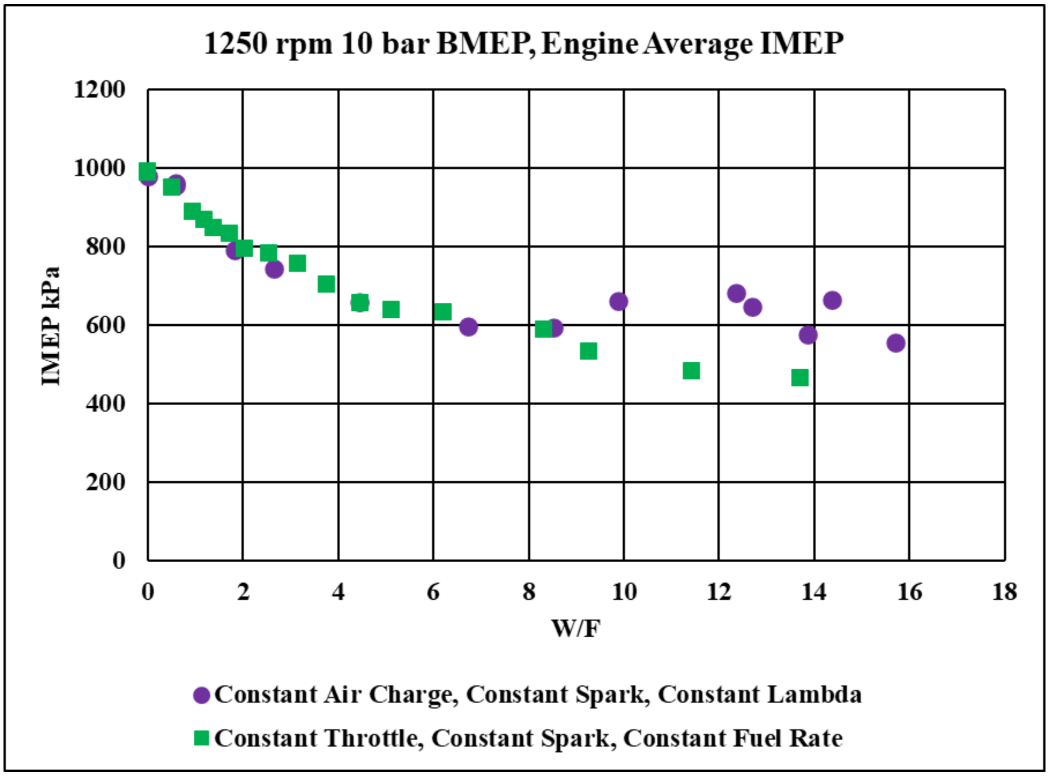


Figure 25 1250 RPM test point atomized spray IMEP plot

Figure 26 and Figure 27 are the lambda plots for the 1250 and 1750 rpm test points, respectively. Seen in the lambda plots, the constant throttle test trends richen as W/F increases. After the initial decreases in lambda due to a displacement in air charge the lambda plot then returns. As a reminder, the constant throttle test water sweeps are run at a constant fuel flow rate that is set at a W/F of 0. This decrease in lambda is due to water vapor displacing the air charge, and the subsequent increase is likely due to reaching a maximum saturation point in the intake.

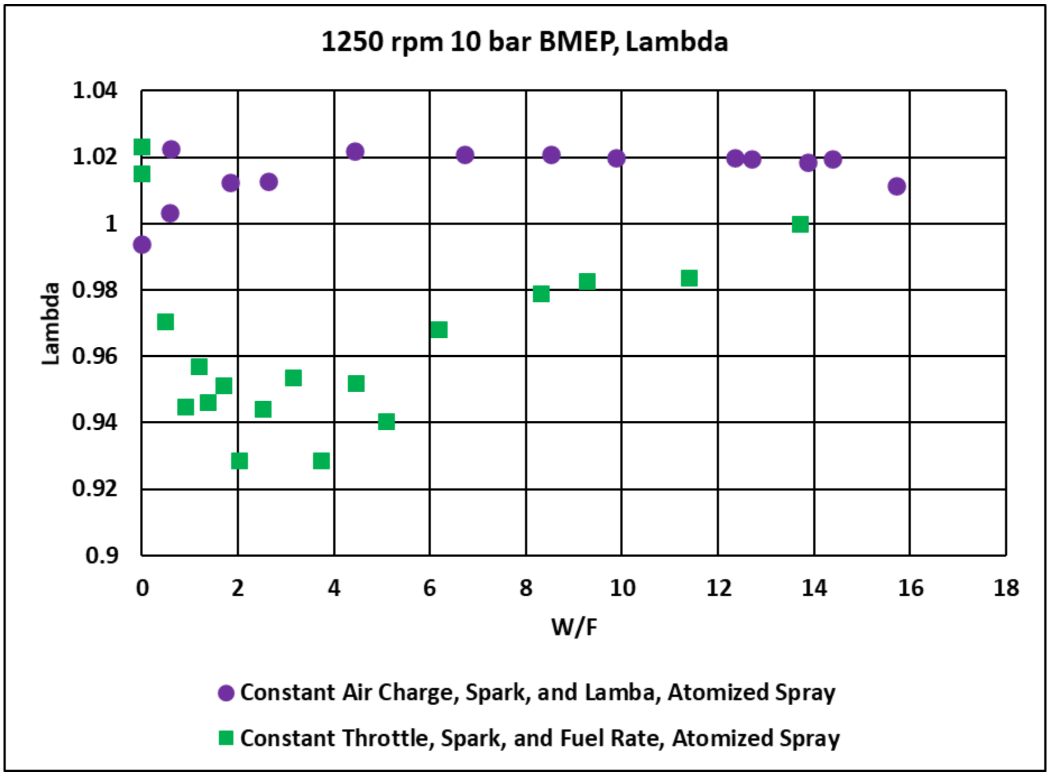


Figure 26 Lambda plot for 1250 rpm test point

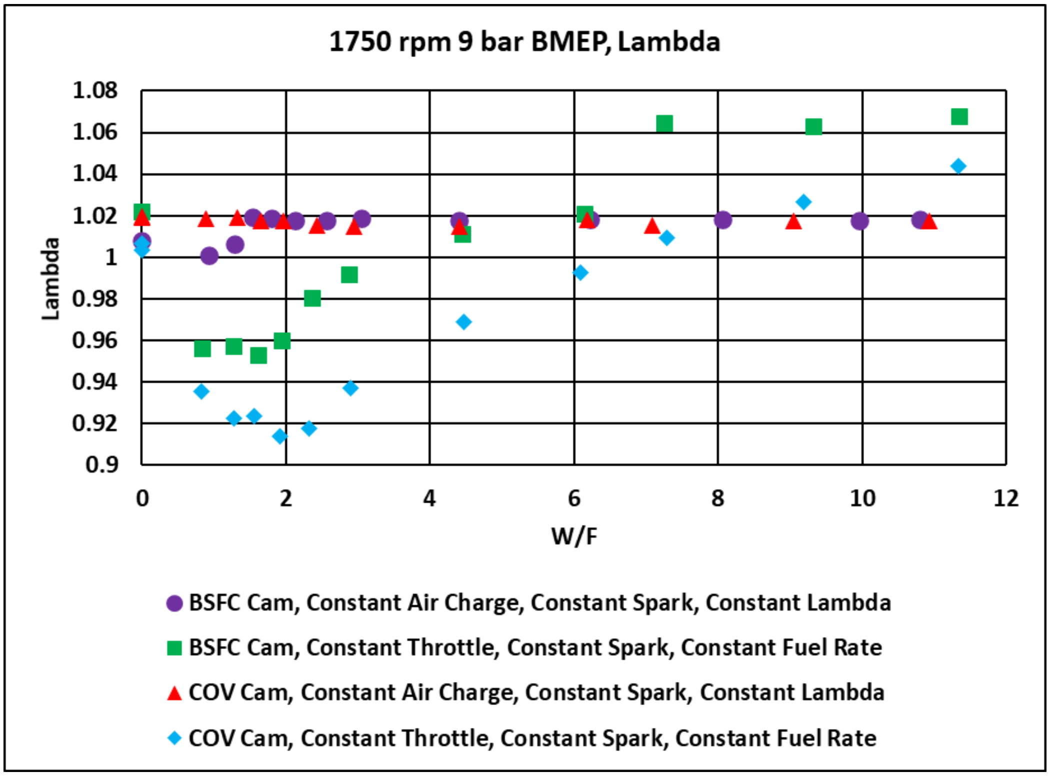


Figure 27 Lambda plot for 1750 rpm test point

Figure 28 to Figure 32 show EGT as a function of W/F for all the test points. It is interesting to note that for all conditions except 3250 RPM, initially with water addition, EGT increases to a maximum before decreasing. The initial increase in EGT is likely due to the retarding combustion phasing. Recall for all the test conditions spark timing was held constant, and the water vapor acts as a dilutant, leading to a retarding of combustion phasing. The retarding of combustion phasing can be seen in Figure 33 through Figure 36 which shows plots of CA50. Beyond the peak of the EGT, the cooling effects due to latent heat vaporization of the water addition becomes dominant thus reducing EGT. The 3250 test point saw only a decrease in EGT, due to the engine speed/load at the test point not allowing for the retarding of combustion to increase the EGT. The high engine speed having a high mass flow out of the engine was not conducive to increasing EGT. Water vaporization is time dependent and at the high engine speed the time for vaporization in the intake is reduced. This test point is likely seeing the vaporization of the injected water into the exhaust thus causing the decrease in EGTs. On another note, the CA50 plots reach similar plateau points as the IMEP, and COV of IMEP plots that correspond with their test points. Once the maximum water vaporization occurs the rate of combustion degradation decreases.

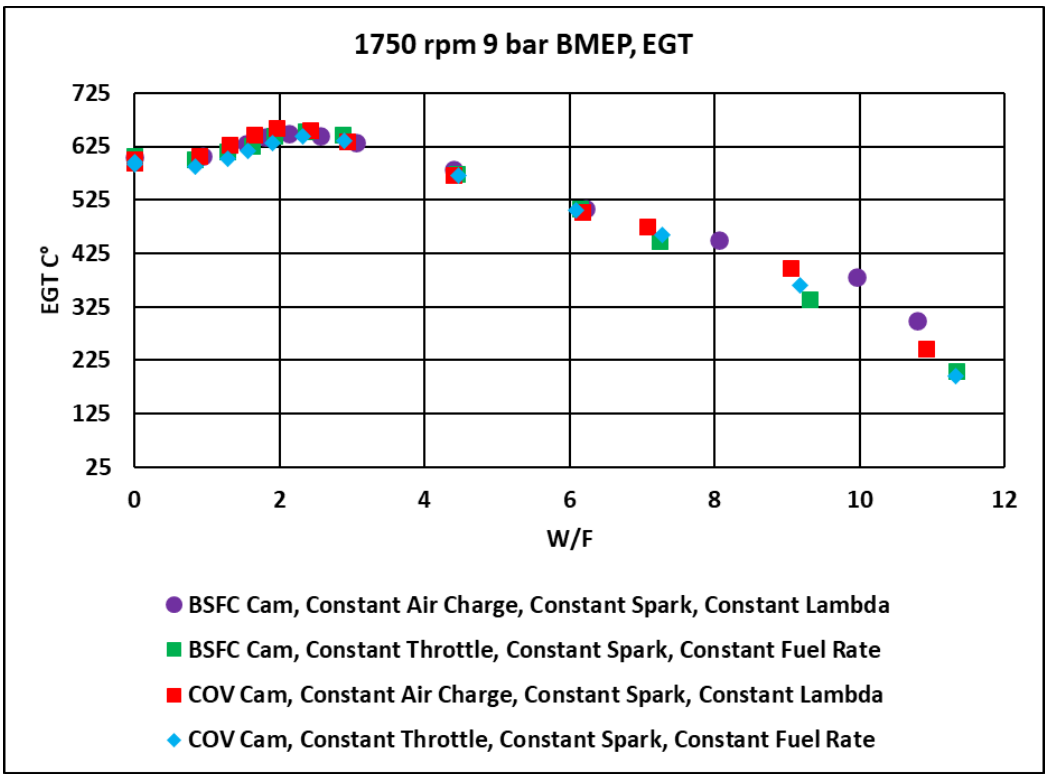


Figure 28 1750 RPM test point atomized spray EGT Plot

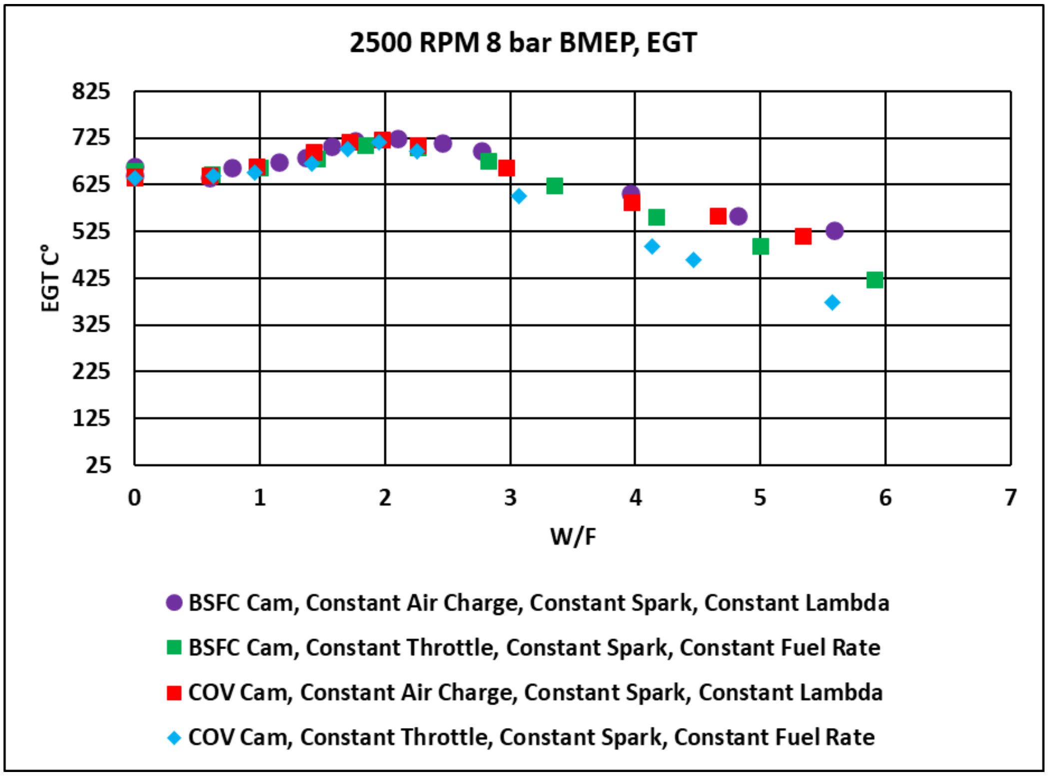


Figure 29 2500 RPM test point atomized spray EGT plot

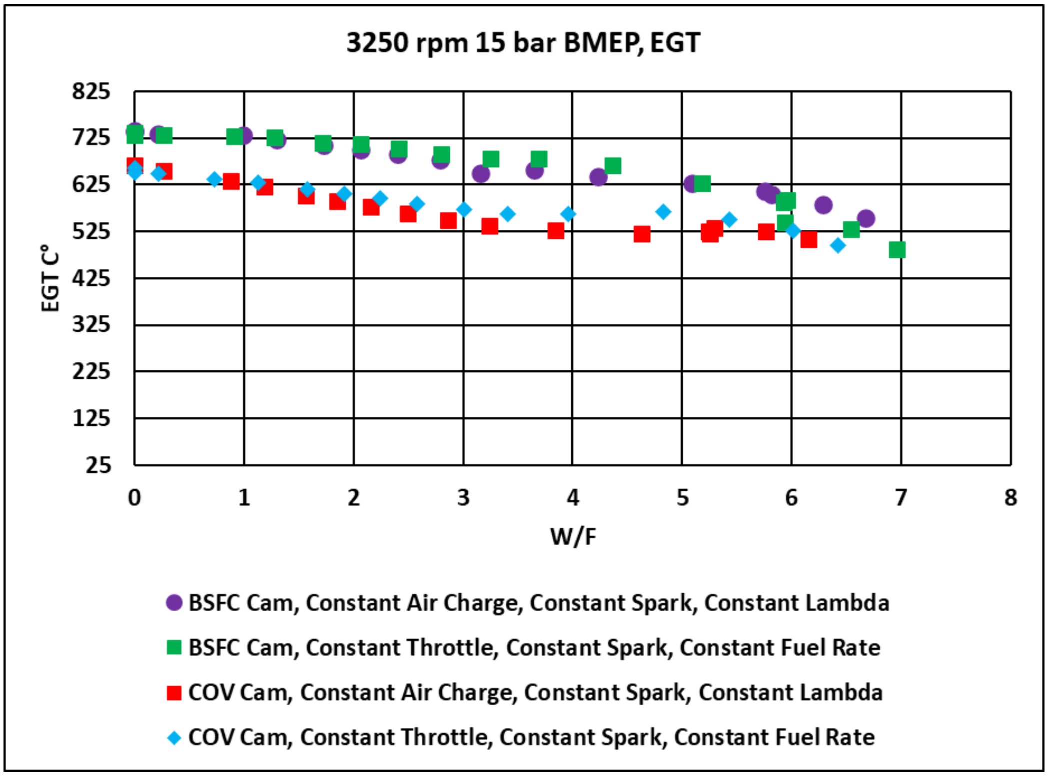


Figure 30 3250 RPM test point atomized spray EGT plot

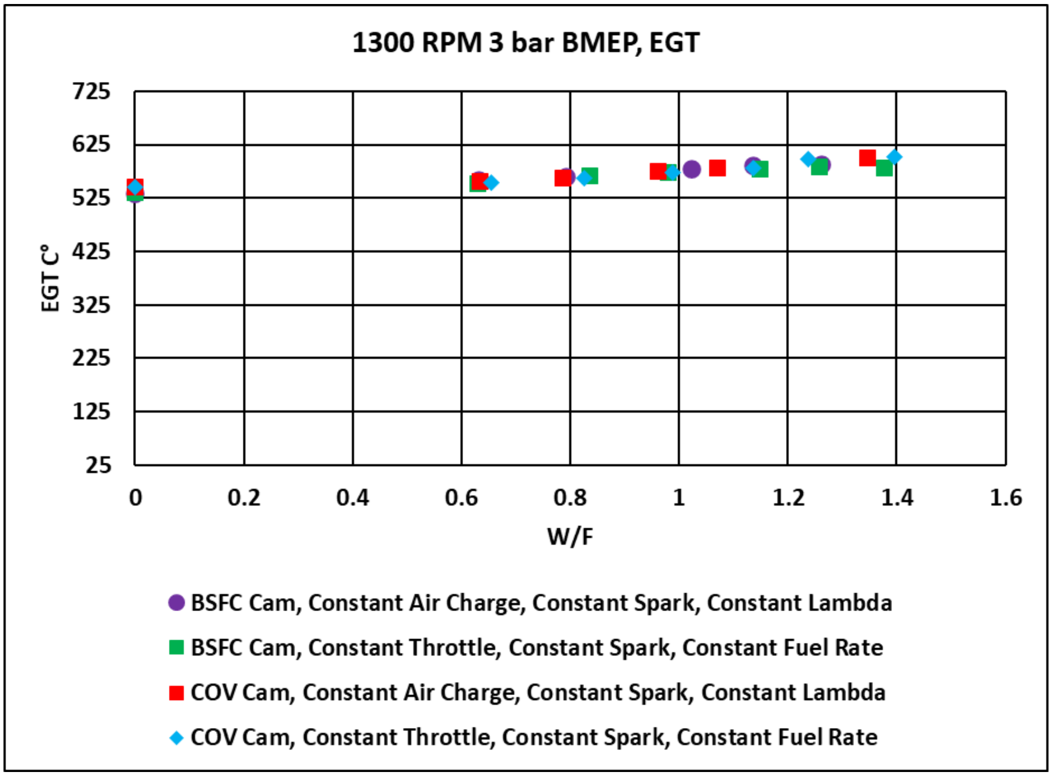


Figure 31 1300 RPM test point atomized spray EGT plot

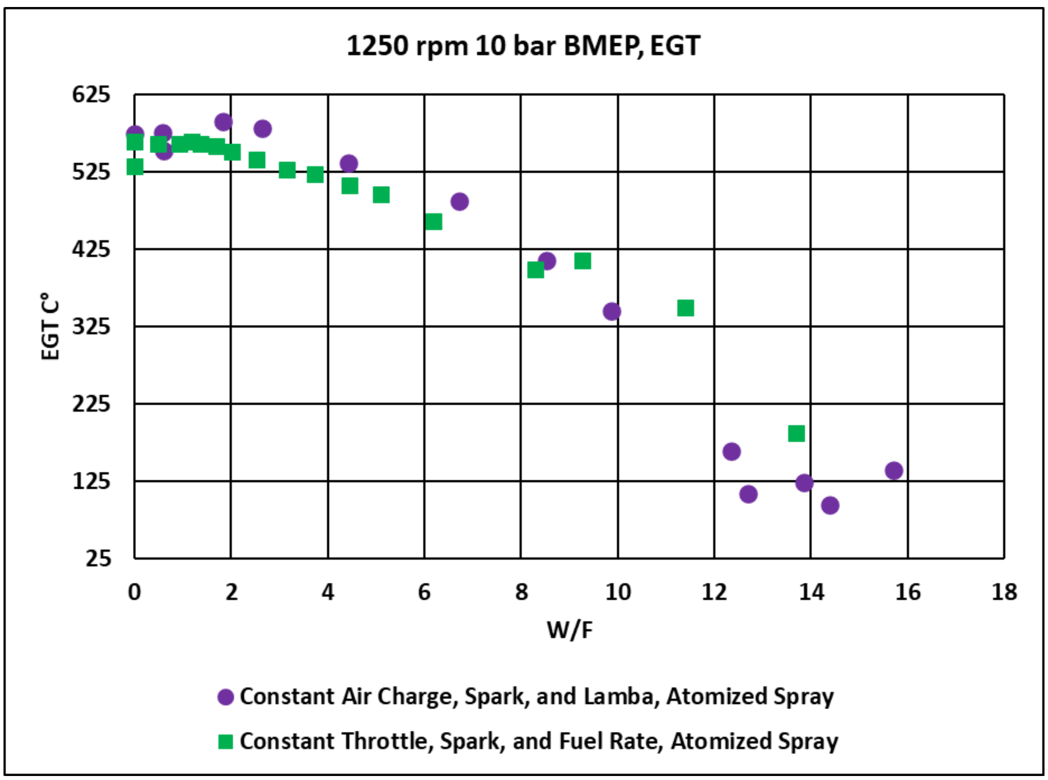


Figure 32 1250 RPM test point atomized spray EGT plot

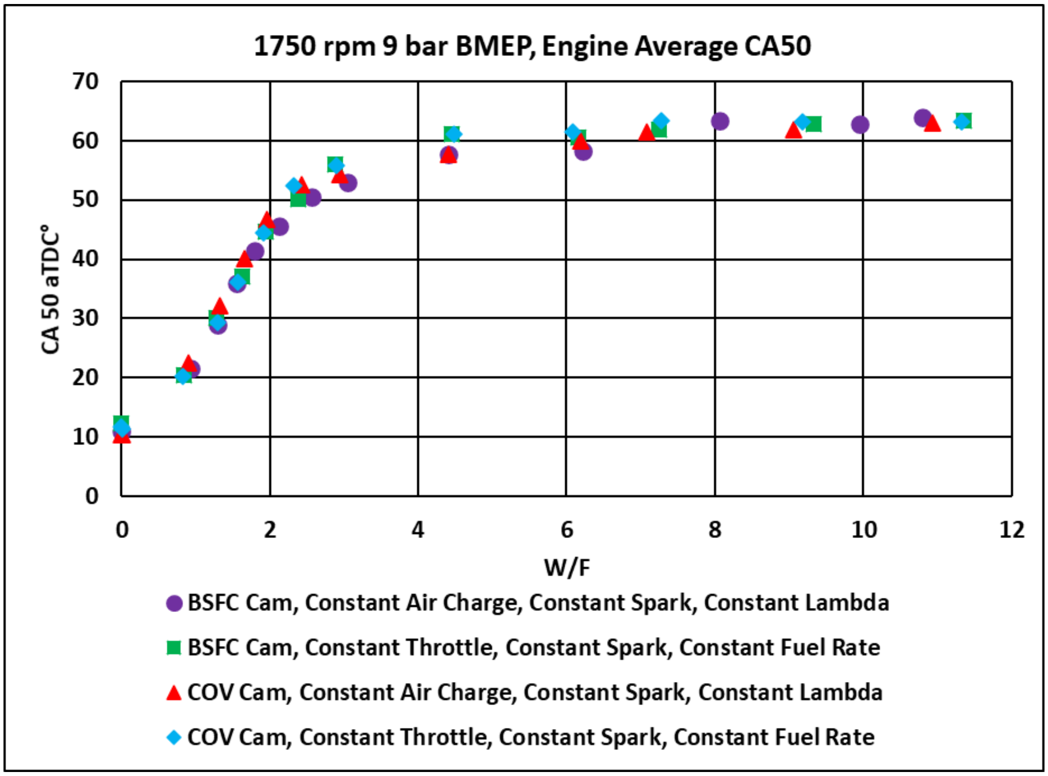


Figure 33 1750 RPM test point atomized spray CA50 plot

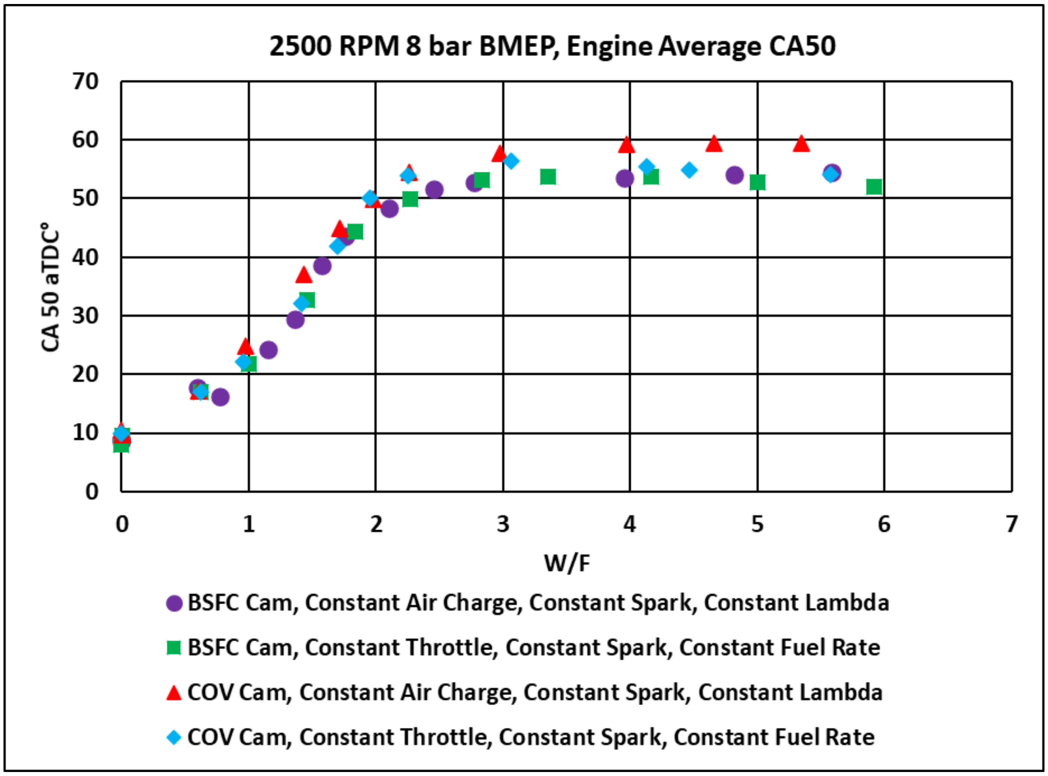


Figure 34 2500 RPM test point atomized spray CA50 plot

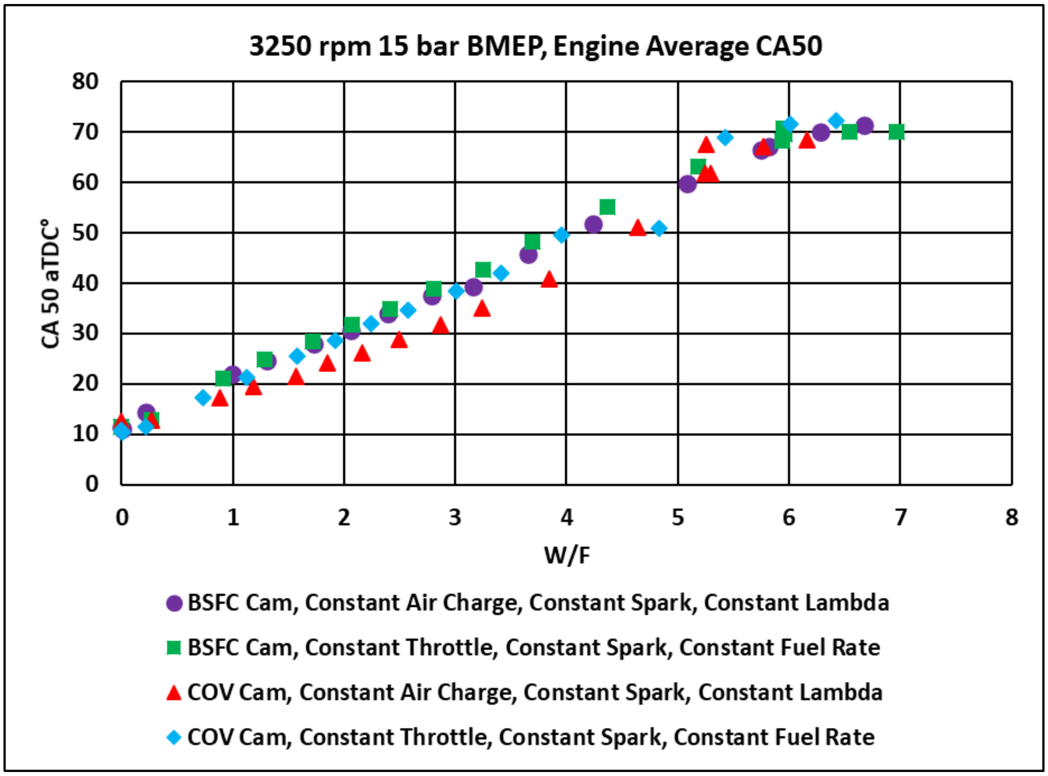


Figure 35 3250 RPM test point atomized spray CA50 plot

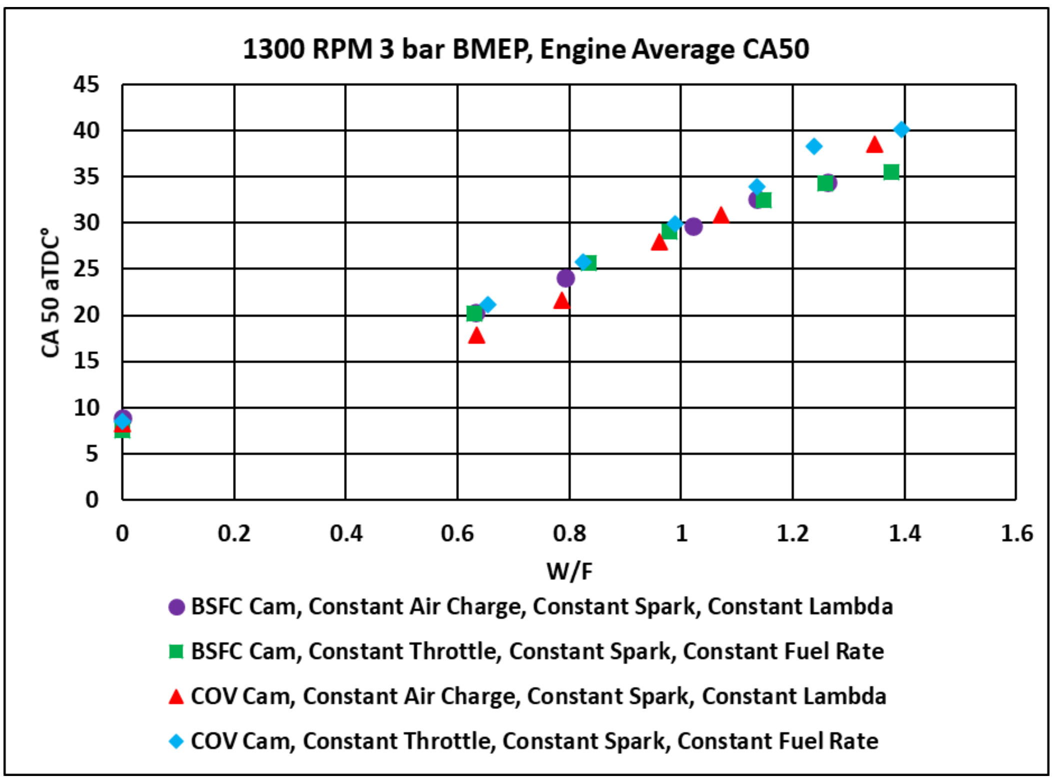


Figure 36 1300 RPM test point atomized spray CA50 plot

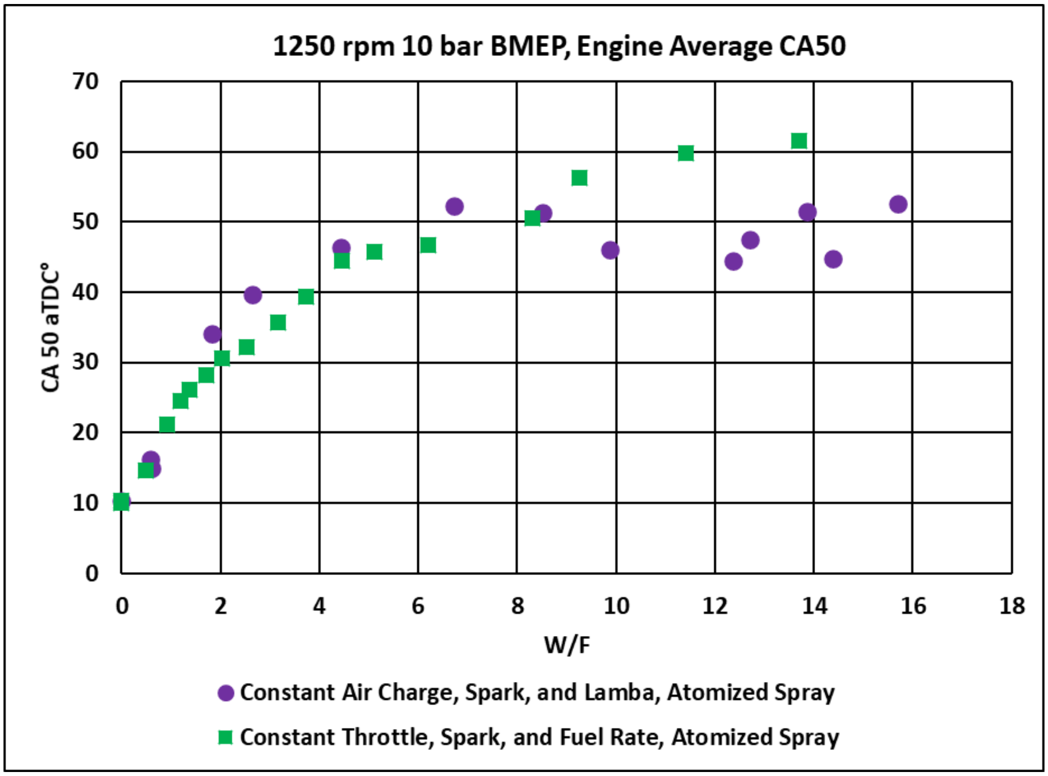


Figure 37 1250 RPM test point atomized spray CA50 plot

An interesting phenomena is seen in **Error! Reference source not found.**Figure 38 and Figure 40, as the W/F is increased the MAP increases, even with constant throttle. This may seem counterintuitive as one might expect the latent heat of vaporization of the water to decrease charge temperature in the manifold, thus decreasing MAP. This MAP increase is likely due to the addition of water in the form of vapor into the intake and the explanation for the pressure increase could be explained by Dalton's law of additive pressures. "Dalton's law of additive pressures: The pressure of a gas mixture is equal to the sum of the pressures each gas would exert if it existed alone at the mixture temperature and volume" [1]. This addition of water in the form of vapor adds an additional pressure part to the intake and intern increases the MAP. Figure 39 shows a subset of the data from Figure 38, but with the calculated MAP increase based on the partial pressure of the water vapor overlaid on the plot. The calculated values closely match the tested values thus supporting the MAP increase theory. The slight over-estimation of the MAP at a W/F of likely due to the decrease in charge temperature as a result of the water heat of vaporization, as this was not accounted for in the calculated pressure.

Equation 2 Daltons Law of Partial Pressure

$$\boxed{\begin{matrix}\text{Gas A}\\ P_A\end{matrix}} + \boxed{\begin{matrix}\text{Gas B}\\ P_B\end{matrix}} = \boxed{\begin{matrix}\text{Gas mixture}\\ P\end{matrix}}$$

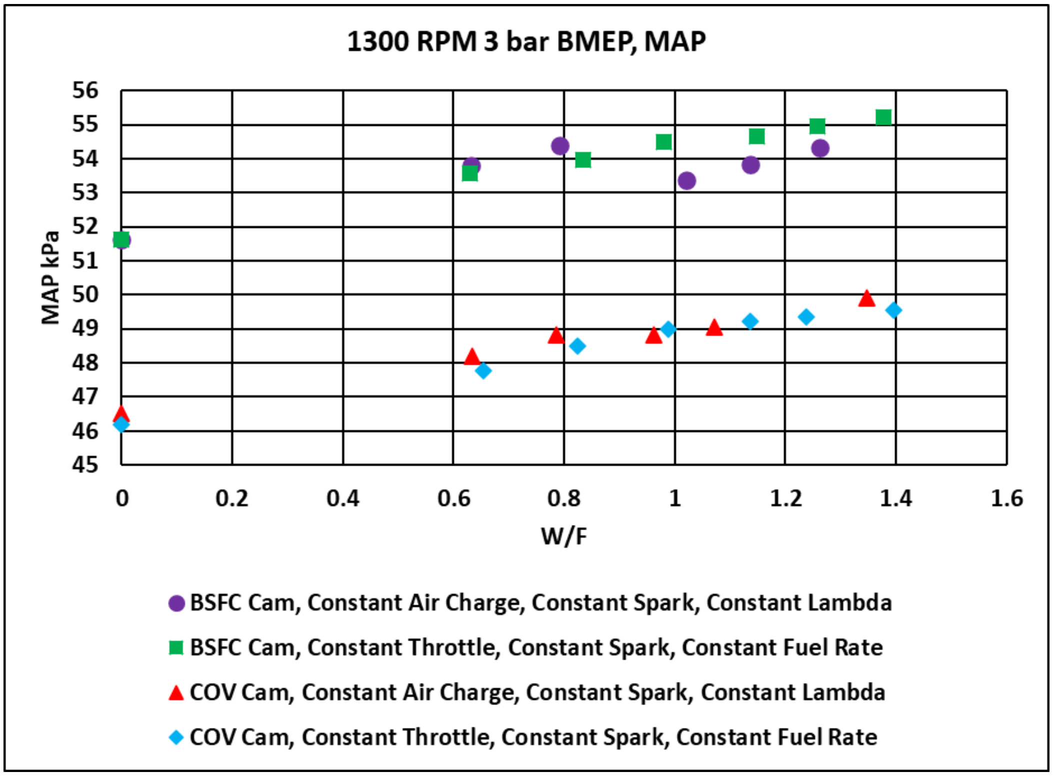


Figure 38 1300 RPM test point atomized spray MAP plot

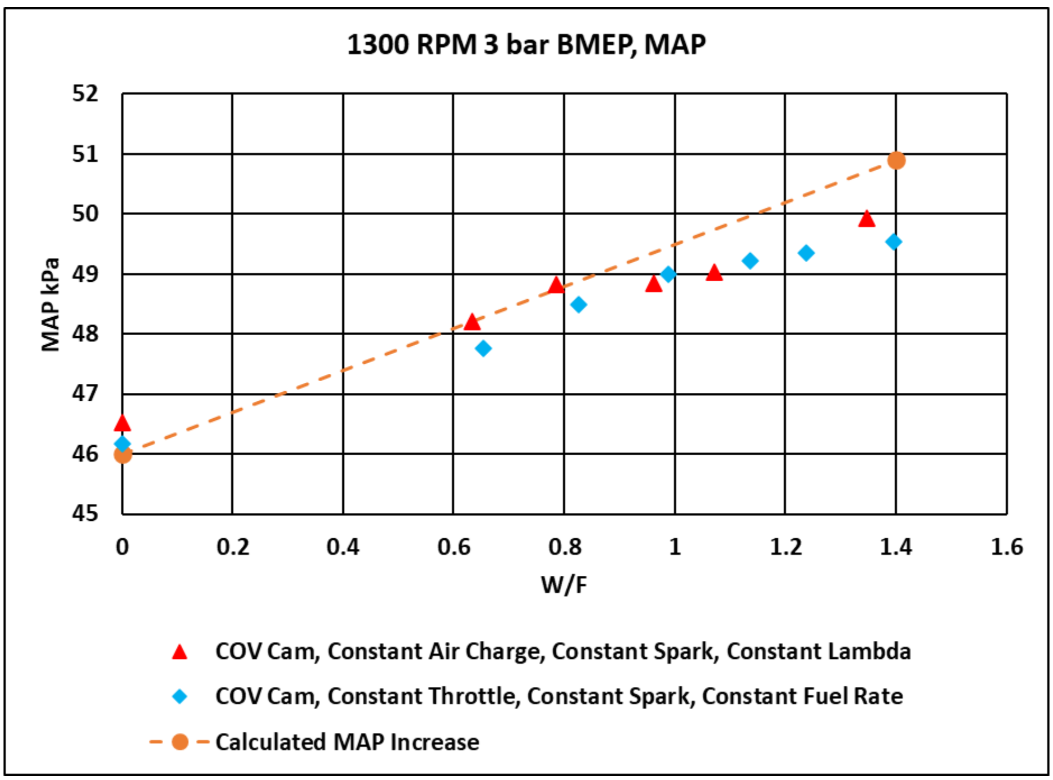


Figure 39 MAP plot showing calculated MAP increase

The 2500 rpm test point (Figure 40) reaches a MAP maximum at a W/F of 3, and then decreases. A potential explanation for this is that once saturation occurs in the manifold, additional water will not further increase the partial pressure of water vapor, thus not support further increase in MAP. However, additional water may still cool the charge, albeit not through phase change, rather through absorbing heat from the valves and port walls. This continued cooling effect (without further increase in vapor pressure) is thought to explain the non-monotonic MAP trend seen in Figure 40.

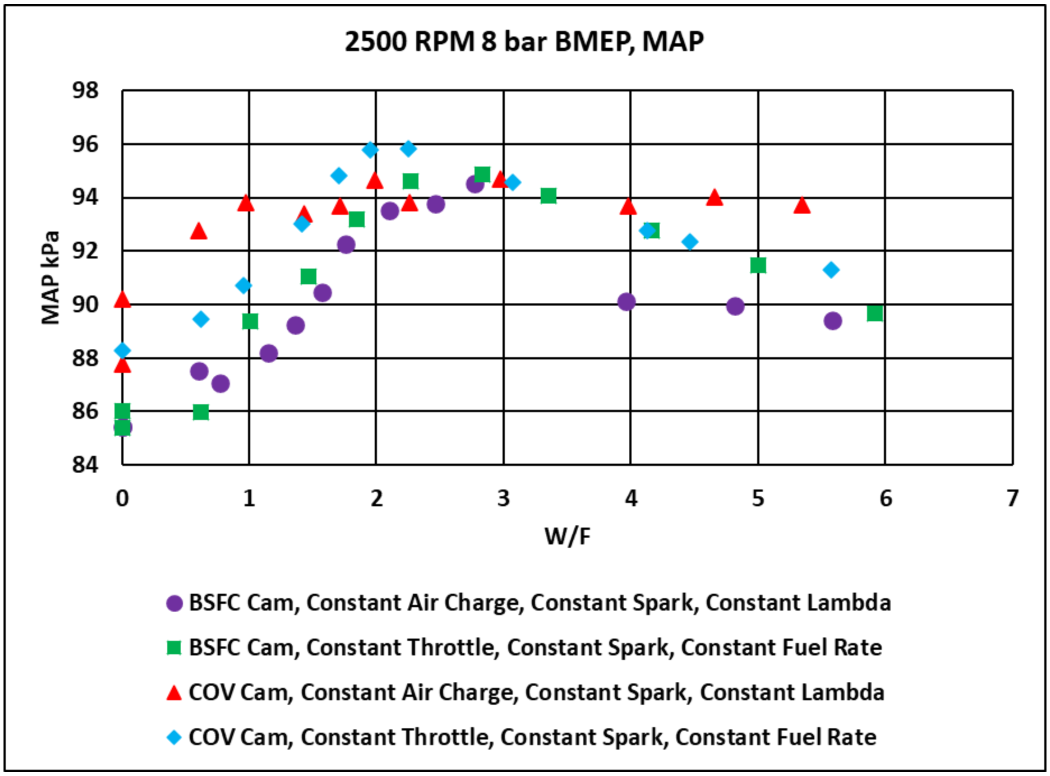


Figure 40 2500 RPM test point atomized spray MAP plot

In this next set of plots there will be a comparison of the 1250, 1750, 2500, and 3250 rpm test points to show how varying speed and load effects the ability of water to degrade combustion. Figure 41 is the LNV of IMEP plot of the four speed load points. The 2500 and 1750 test points have loads of 8 and 9 bar BMEP, respectively. These two test points seem to follow a similar trend in Figure 41, and this may be due to a load phenomenon.

Seen in Figure 42 is the IMEP water sweep plot for the same four test points previously mentioned. One interesting phenomenon that is seen is that the IMEP trends for the four test points ends at a similar IMEP value. Each of the water sweeps end at a IMEP of 200-450 kPa. The part that is interesting is that the drastic difference in the speed/load point to then having a proximity of IMEP value at high W/F. This may be a factor of combustion or engine dynamics. Figure 43 quantifies the W/F to a water volume per cylinder per cycle for the four test points previously mentioned. This adds a better understanding of the volume of liquid water being introduced into the engine.

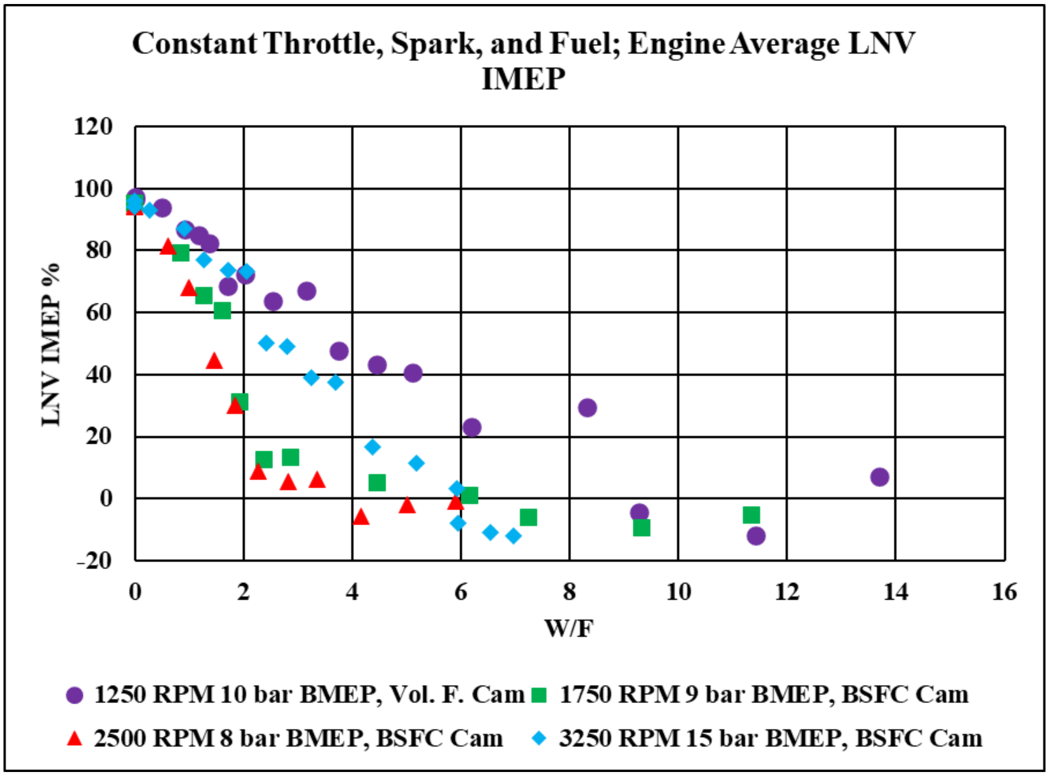


Figure 41 Four speed load comparison LNV IMEP plot

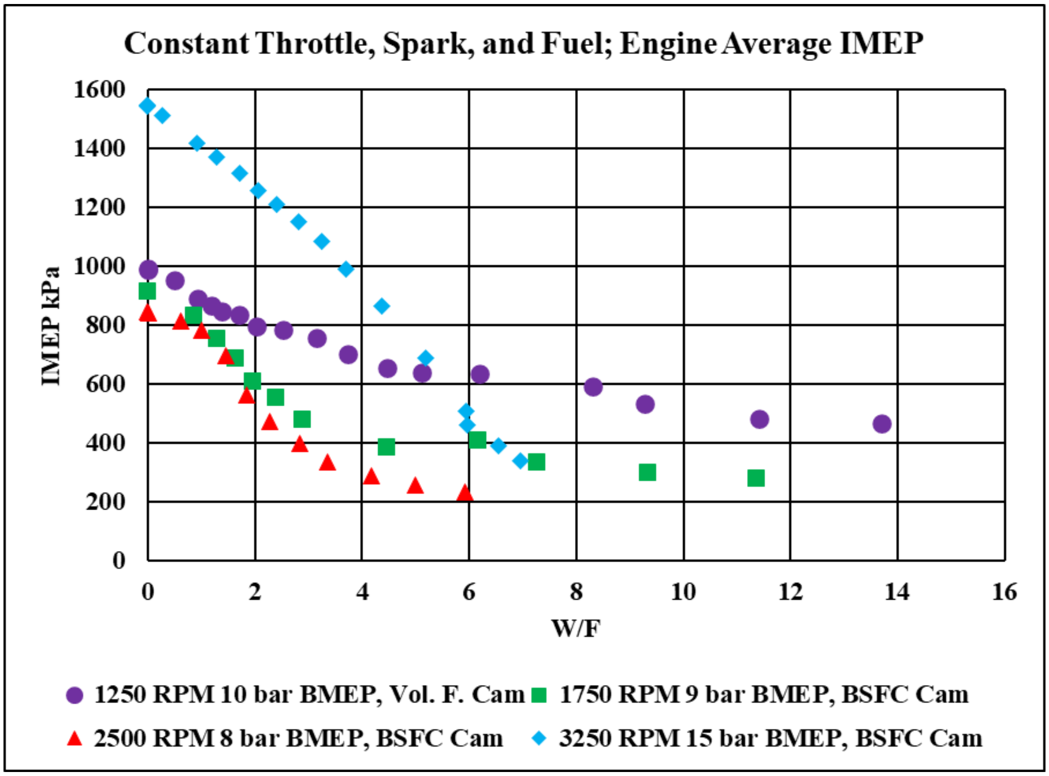


Figure 42 Four speed load comparison IMEP plot

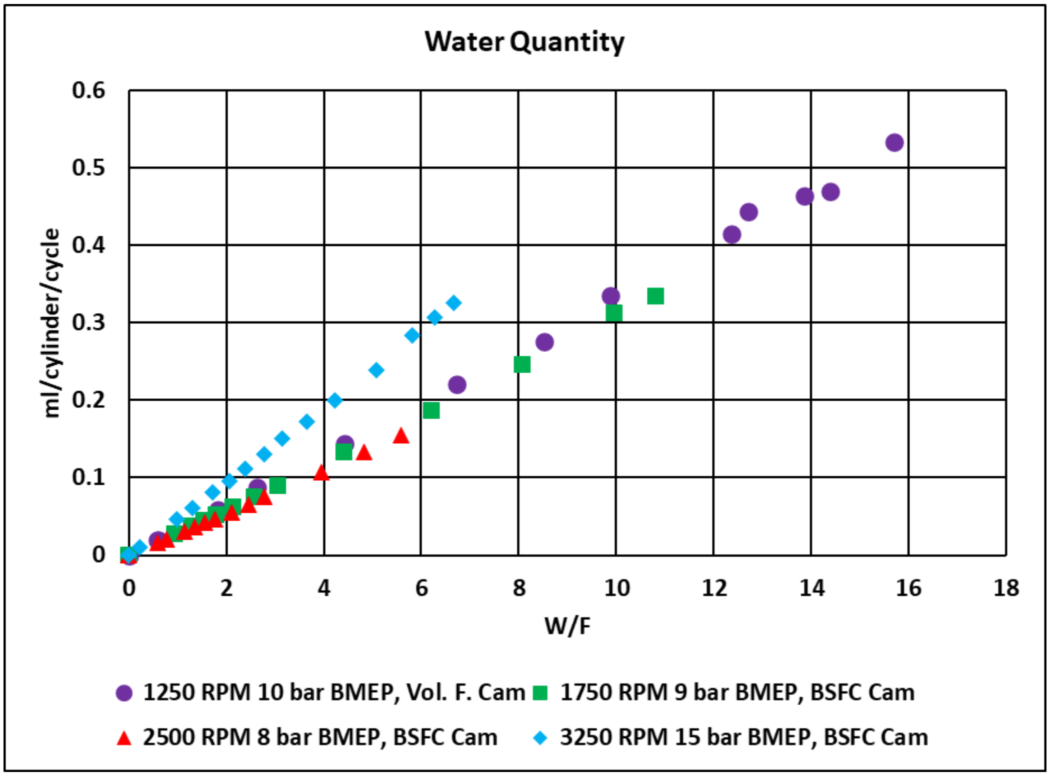


Figure 43 Four speed load comparison water quantity per cylinder per cycle plot

6.3 Non-Atomized testing

The non-atomized testing utilizes the hardware discussed in section 4.1.6.2. As mentioned in the testing plan the non-atomized testing was only completed for the 1250, 1750, 2500, and 3250 test points. Figure 44 through Figure 47 show the COV of IMEP plots for the non-atomized testing. These plots show in some capacity that the non-atomized spray sees less combustion degradation being that the non-atomized spray has a lower COV of IMEP when compared to the atomized spray. This is likely due to the non-atomized spray not being able to vaporize as affectively as the atomized spray. When water vaporization occurs, it displaces the air charge, and this intern effects combustion performance. The difference in the vaporization rate is likely due to the droplet size that is coming out of the injectors. The non-atomized spray seen in Figure 6 has a large difference droplet size from the atomized spray. The non-atomized and atomized testing were completed on separate days so there is some humidity difference in the data. These differences in humidity are minor and should not have a drastic effect on the results.

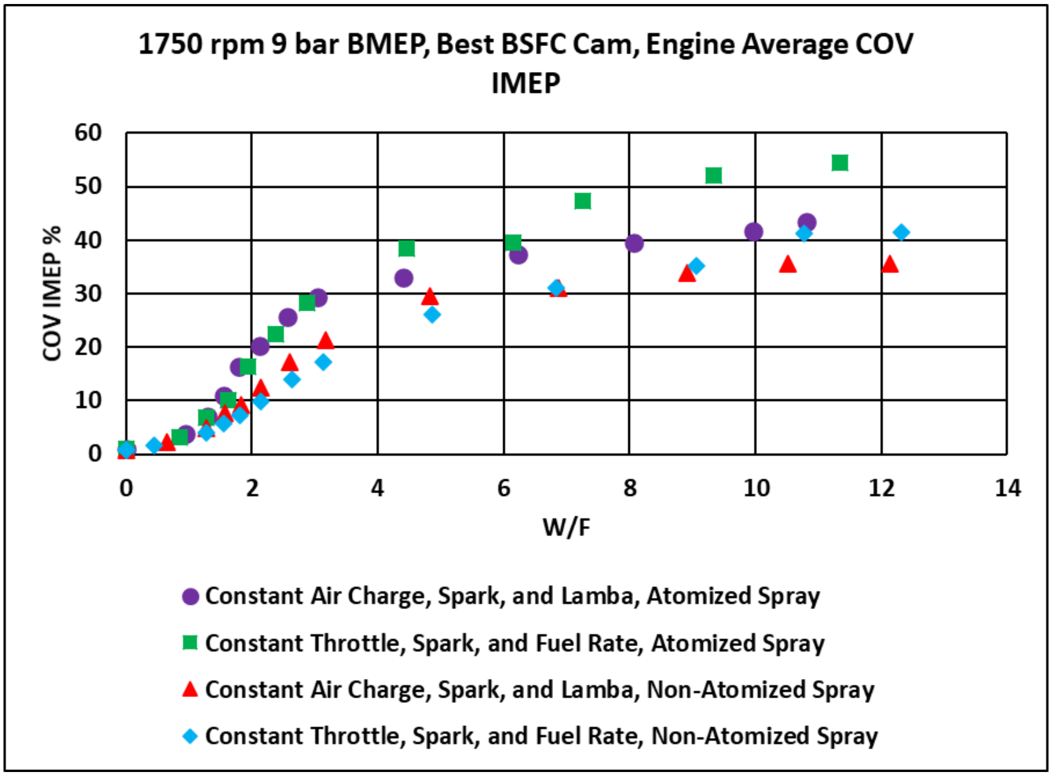


Figure 44 1750 RPM test point atomized spray and non-atomized spray COV IMEP

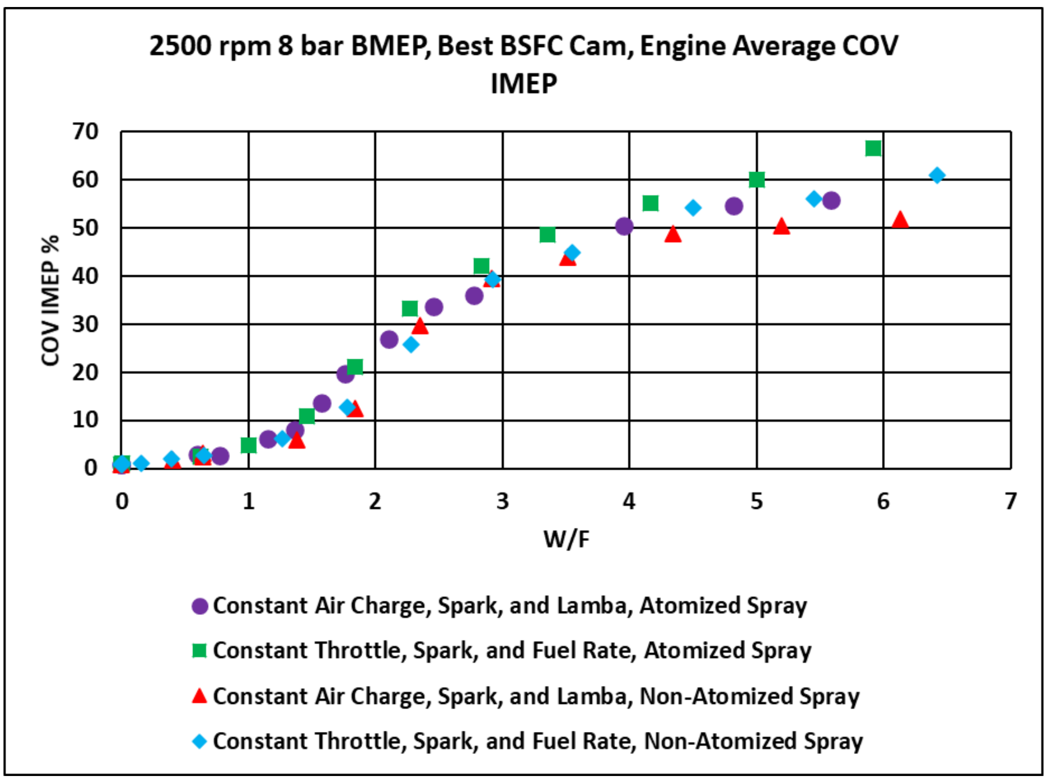


Figure 45 2500 RPM test point atomized spray and non-atomized spray COV IMEP

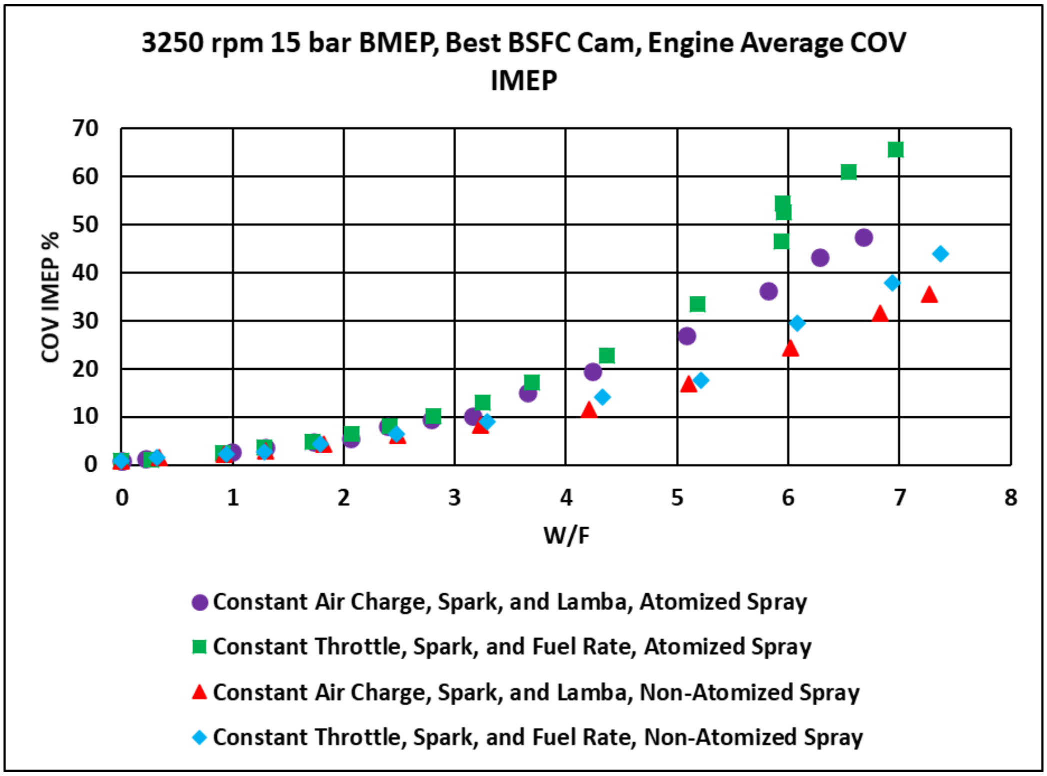


Figure 46 3250 rpm test point atomized spray and non-atomized spray COV IMEP

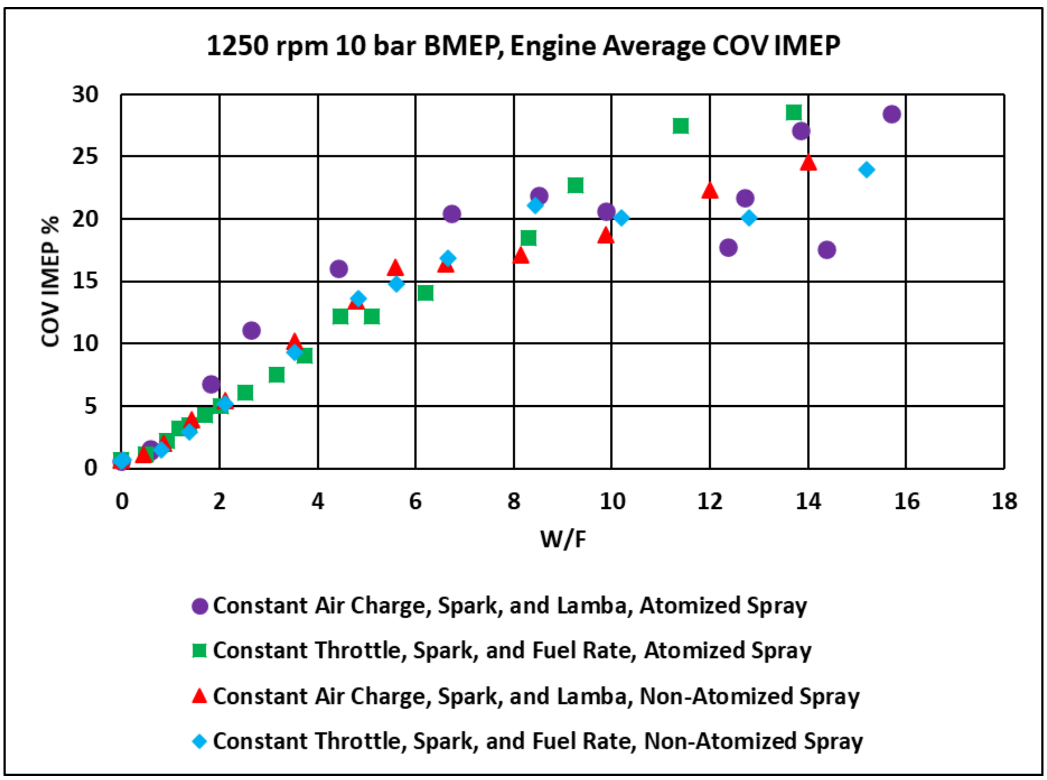


Figure 47 1250 rpm test point atomized spray and non-atomized spray COV IMEP

Figure 48 through Figure 51 shows four examples of how the non-atomized versus the atomized spray performance in the IMEP plots. The IMEP plot shows a similar trend to the COV plot with the non-atomized spray having less combustion degradation. This is a slight contrast with the non-atomized spray preforming better in the IMEP plot. The 1250 rpm test point does not show as large of a contrast for the two different water spray strategies. The lack of difference in the two spray strategies may be due to the engine condition that this test point.

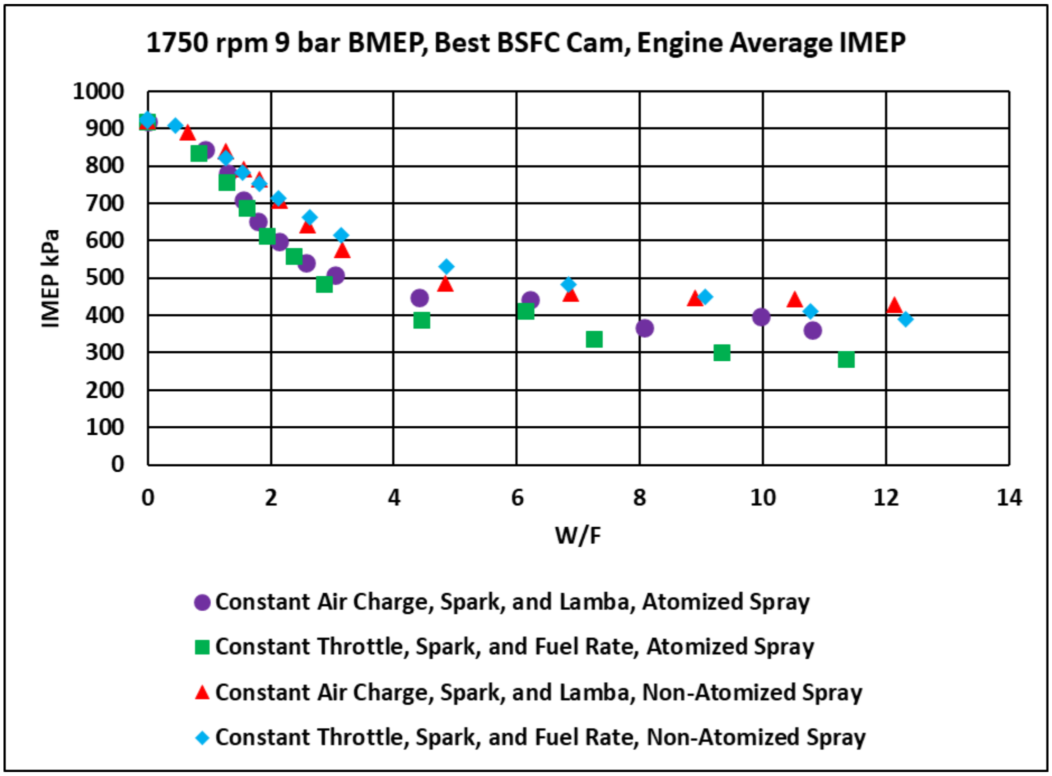


Figure 48 1750 rpm test point atomized spray and non-atomized spray IMEP

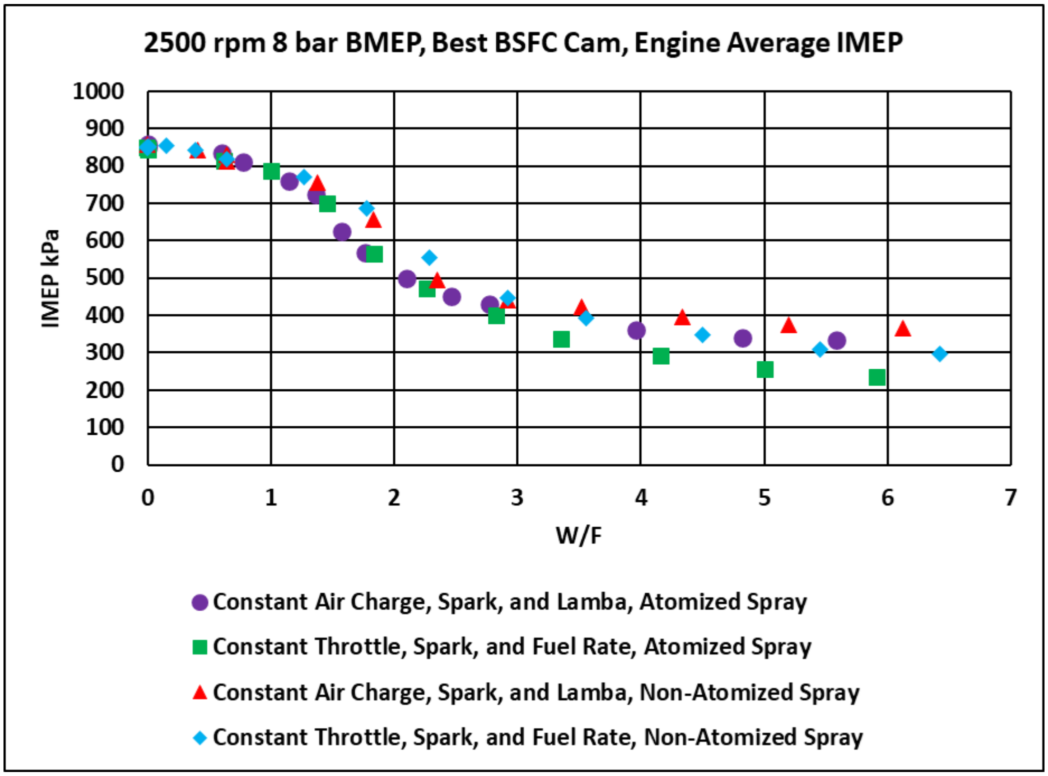


Figure 49 2500 rpm test point atomized spray and non-atomized spray IMEP

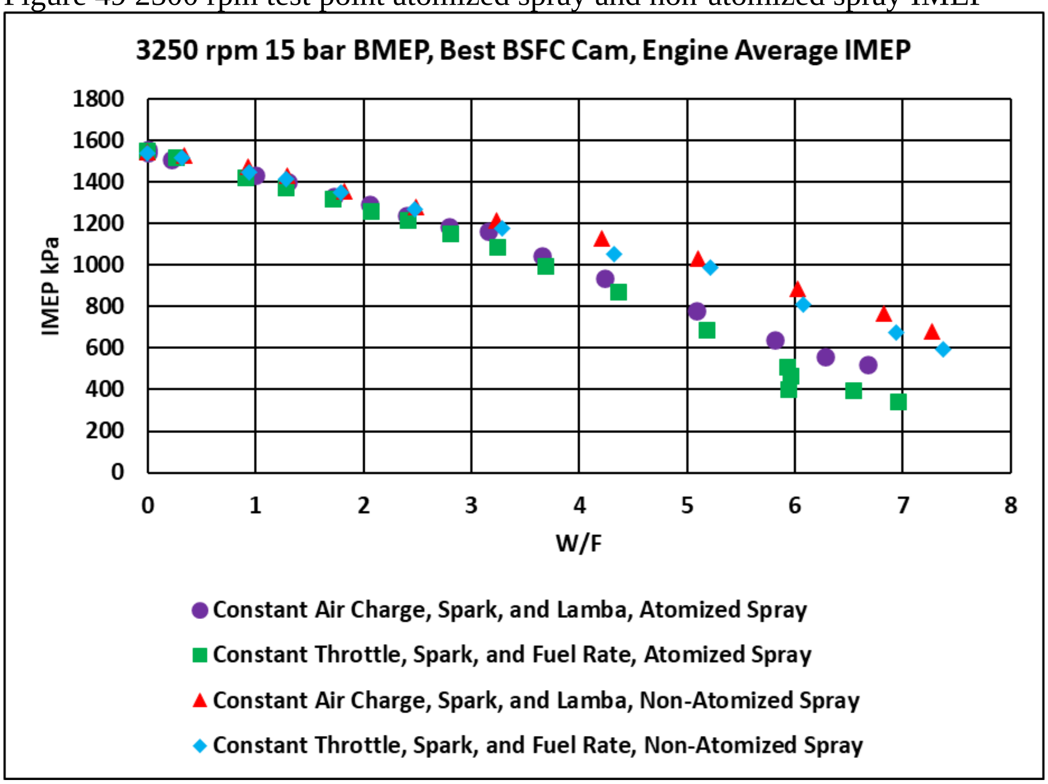


Figure 50 3250 rpm test point atomized spray and non-atomized spray IMEP

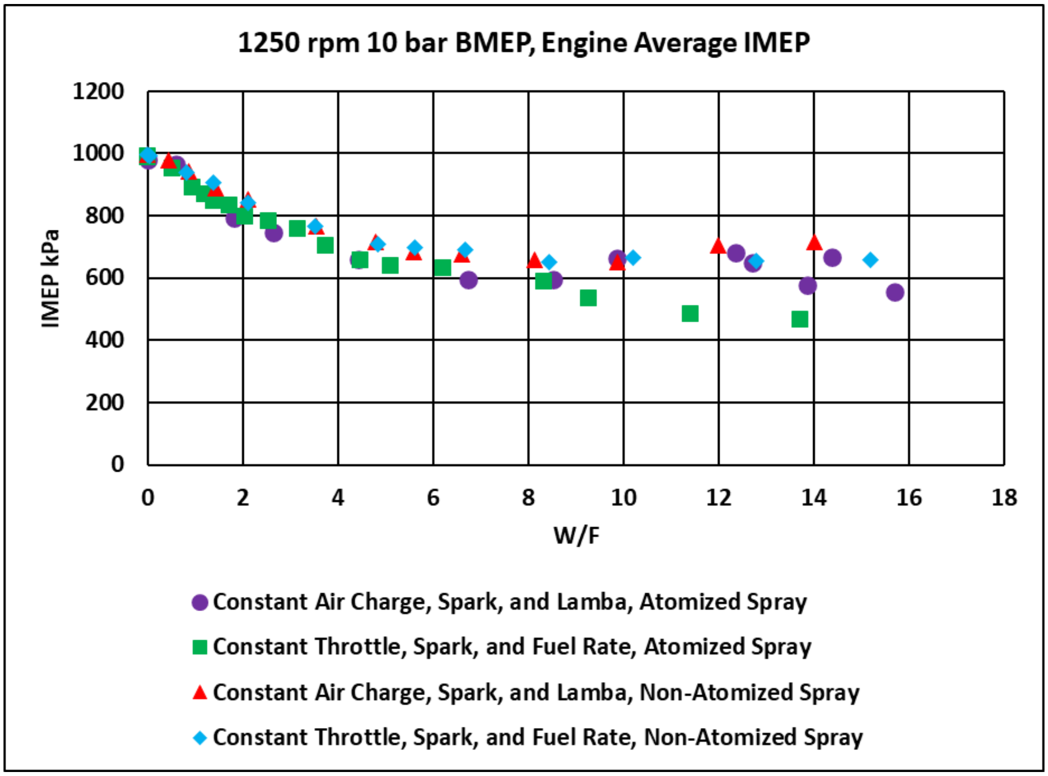


Figure 51 1250 rpm test point atomized spray and non-atomized spray IMEP

Figure 52 and Figure 53 are MAP plots for the 1750 and 2500 rpm test point, respectively. The non-atomized spray testing shows less of a MAP increase when compared to the atomized spray testing. This further supports the theory that the non-atomized spray is not as effective at vaporizing as the atomized spray. With W/F being the metric that normalizes the water amount to the fuel rate and being that the speed load between the two injection strategy is the same, the fueling will be the same based on the fuel strategy.

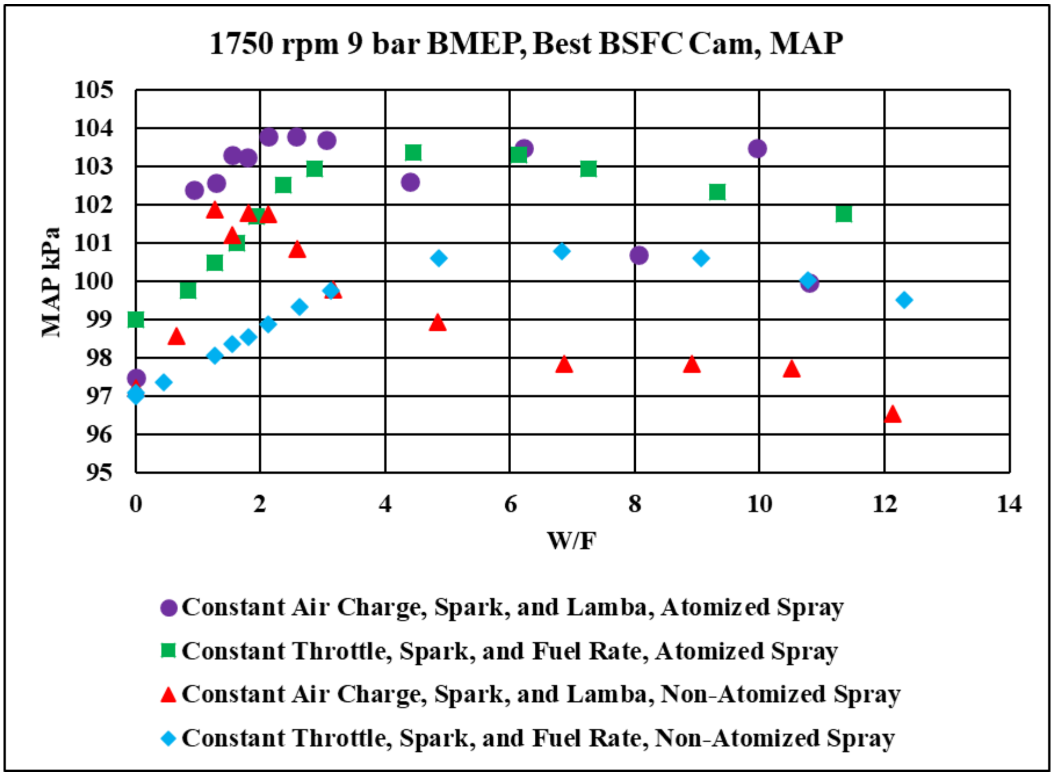


Figure 52 1750 rpm test point atomized spray and non-atomized spray MAP

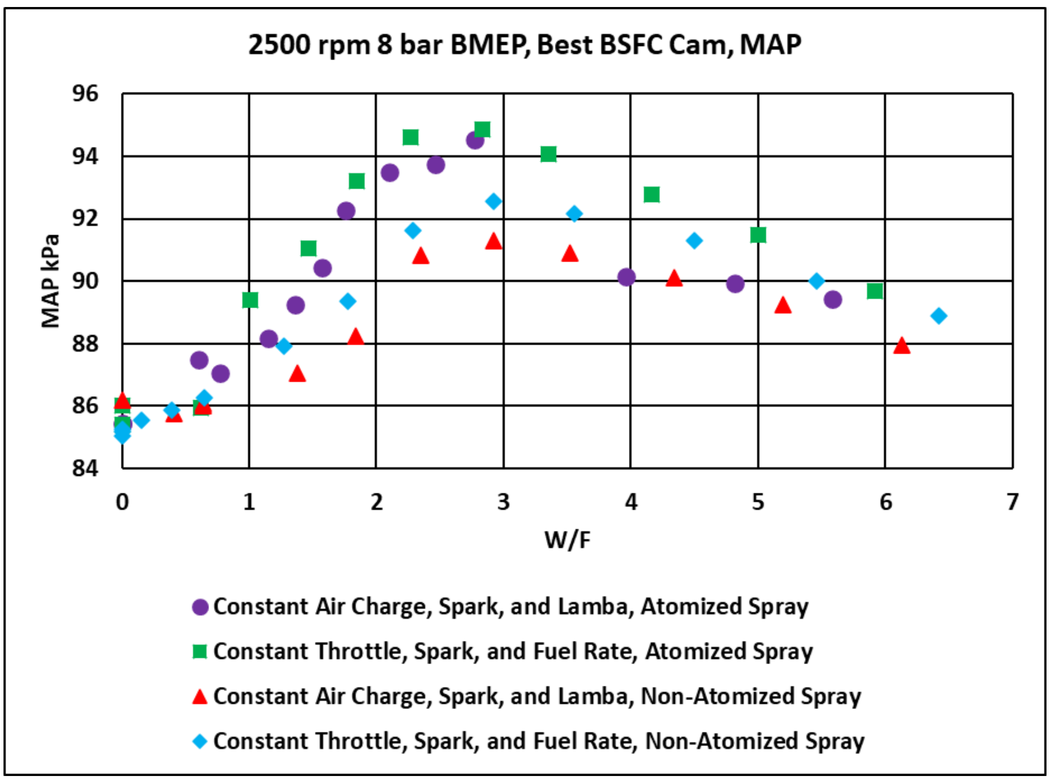


Figure 53 2500 RPM test point atomized spray and non-atomized spray MAP

Table 10 is a summary of the test points where COV of IMEP crosses the thresholds of 5%, 10%, and 20% corresponding to a W/F. In general, the lower the load, the less tolerant the combustion system is to water ingestion.

Table 10 Summary of W/F that COV of IMEP crosses 5%, 10%, and 20%

RPM	BMEP (bar)	Cam Position	Spark, Air, Fuel Test Strategy	W/F at COV 5%	W/F at COV 10%	W/F at COV 20%
Atomized Spray						
1300	3	I-8,E38	Constant Spark, Air, Lambda	0.6	0.82	1.2
		Best BSFC	Constant Spark, Throttle, Fuel Flow	0.6	0.82	1.2
		I-28,E8	Constant Spark, Air, Lambda	0.82	1	1.3
		Min COV	Constant Spark, Throttle, Fuel Flow	0.75	1.1	N/A
1750	9	I-5,E15	Constant Spark, Air, Lambda	1	1.5	2
		Best BSFC	Constant Spark, Throttle, Fuel Flow	1	1.5	2.1
		I0,E10	Constant Spark, Air, Lambda	1	1.5	1.8
		Min COV	Constant Spark, Throttle, Fuel Flow	1	1.5	2.1
3250	15	I-30,E10	Constant Spark, Air, Lambda	1.7	3	4.2
		Best BSFC	Constant Spark, Throttle, Fuel Flow	1.7	2.7	4
		I-45,E45	Constant Spark, Air, Lambda	2.4	3.8	5
		Min COV	Constant Spark, Throttle, Fuel Flow	1.7	3.1	4.2
2500	8	I-50,E35	Constant Spark, Air, Lambda	1	1.4	1.7
		Best BSFC	Constant Spark, Throttle, Fuel Flow	1	1.4	1.7
		I-0,E20	Constant Spark, Air, Lambda	0.8	1.2	1.8
		Min COV	Constant Spark, Throttle, Fuel Flow	0.9	1.5	1.8
1250	10	I-22, E16	Constant Spark, Air, Lambda	1.5	2.2	6
		Vol. Eff.	Constant Spark, Throttle, Fuel Flow	2	4	8.5
Non-Atomized Spray						
1750	9	I-5,E15	Constant Spark, Air, Lambda	1.2	1.8	2.8
		Best BSFC	Constant Spark, Throttle, Fuel Flow	1.2	2.1	3.6
3250	15	I-30,E10	Constant Spark, Air, Lambda	2	3.7	5.5
		Best BSFC	Constant Spark, Throttle, Fuel Flow	2	3.5	5.5
2500	8	I-50,E35	Constant Spark, Air, Lambda	1	1.6	2.1
		Best BSFC	Constant Spark, Throttle, Fuel Flow	1	1.6	2
1250	10	I-22, E16	Constant Spark, Air, Lambda	1.9	3.5	10.5
		Vol. Eff.	Constant Spark, Throttle, Fuel Flow	2	4	8

7 Future Work

1. One thing that was observed during the testing for this study and was not captured, was the transient response of the event when water injection starts. Seen in Figure 54, is an example of a phenomenon that occurs when water injection starts at a high W/F, and the plot contains: misfire count, IMEP, MAP, and start of water injection. When this high W/F injection event starts, combustion degrades sharply to a point of complete misfire, then, rather abruptly, the misfires stop, and combustion "stabilizes". In actuality this transient behavior more closely approximates the anticipated in-vehicle scenario, and thus warrants investigation. For example, one hypothesis is that the still hot metal temperature creates a significant amount of steam, displacing fresh air, and leading to complete misfire for several cycles until the steam is purged from the intake and the metal temperature has reached equilibrium. Further research in this could explore the limits of the combustion system right after water is introduced into the combustion system with the goal of better understanding this abrupt transient trend.

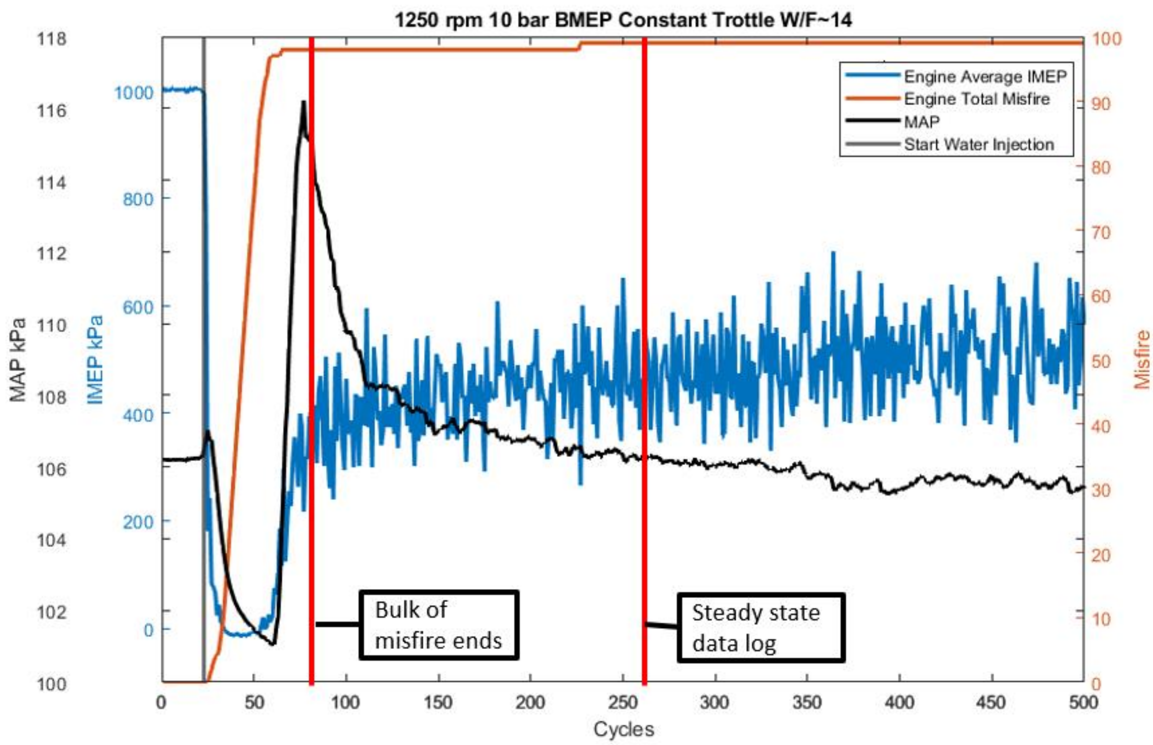


Figure 54 Transient data log example

2. A study looking at the influences of ambient humidity on the tolerance to water ingestion.
3. Another possible future work, would be to develop a model to represent the effects of water ingestion.

4. A study further exploring in cylinder temperatures at the 3250 rpm 15 bar BMEP to gain a better picture of where water vaporization is occurring.
5. Research the effects of fuel sensitivity regarding combustion degradation with water ingestion.

8 Conclusion

1. Seen in Table 10 is a summary of the W/F at which the COV of IMEP of 5%, 10%, and 20% are crossed from this testing. This table summarized that load seems to influence combustion degradation as W/F is increased. For example, at the low load point of 1300 rpm 3 bar BMEP, 20% COV of IMEP was reached at a W/F slightly greater than 1:1. However, greater than 5:1 and even as much as 10:1 were required to degrade combustion to 20% COV of IMEP at the highest loads such as 1250 rpm 10 bar BMEP and 3250 rpm 15 bar BMEP.
2. A plateau effect of the limit of water vaporization in the intake was observed in the data. The plateau point is due to the air in the manifold reaching the saturation point of water. The rest of the water that is not able to be vaporized is being introduced into the combustion chamber as a liquid where it has negligible impact on volumetric efficiency and provides negligible additional dilution.
3. Additionally, a MAP increase was seen in the test points where MAP was below atmospheric conditions. This is likely a result of the increase in partial pressure in the manifold (due to the water vapor) having a larger impact on total pressure than the decrease in charge temperature due to the heat of vaporization of the water. This hypothesis was corroborated as at least plausible with a 0-D calculation.
4. Some additional observation as combustion degrades is that the test points see an EGT increase initially with the addition of water and then decrease past an inflection point. This is thought to be due to an initial retarding of combustion phasing (recall spark timing was constant in this testing) due to the diluent effect of the water vapor. However, beyond a point the fresh charge is saturated, and liquid water in the combustion chamber is likely still vaporizing late in the combustion process (perhaps even into the exhaust stroke at high W/F's), where the impact on combustion phasing is negligible (because water is still liquid at the point of 50% MFB), while the impact on EGT is significant due to the heat of vaporization.
5. It was seen that the non-atomized spray did not affect combustion as drastically as the atomized spray. For example, the COV of IMEP for the non-atomized testing was lower than the atomized testing at the same W/F. This is due to the non-atomized spray not vaporizing in the intake as quickly as the atomized spray. Thus, more of the water enters the combustion chamber as a liquid, where it has negligible impact on combustion.

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