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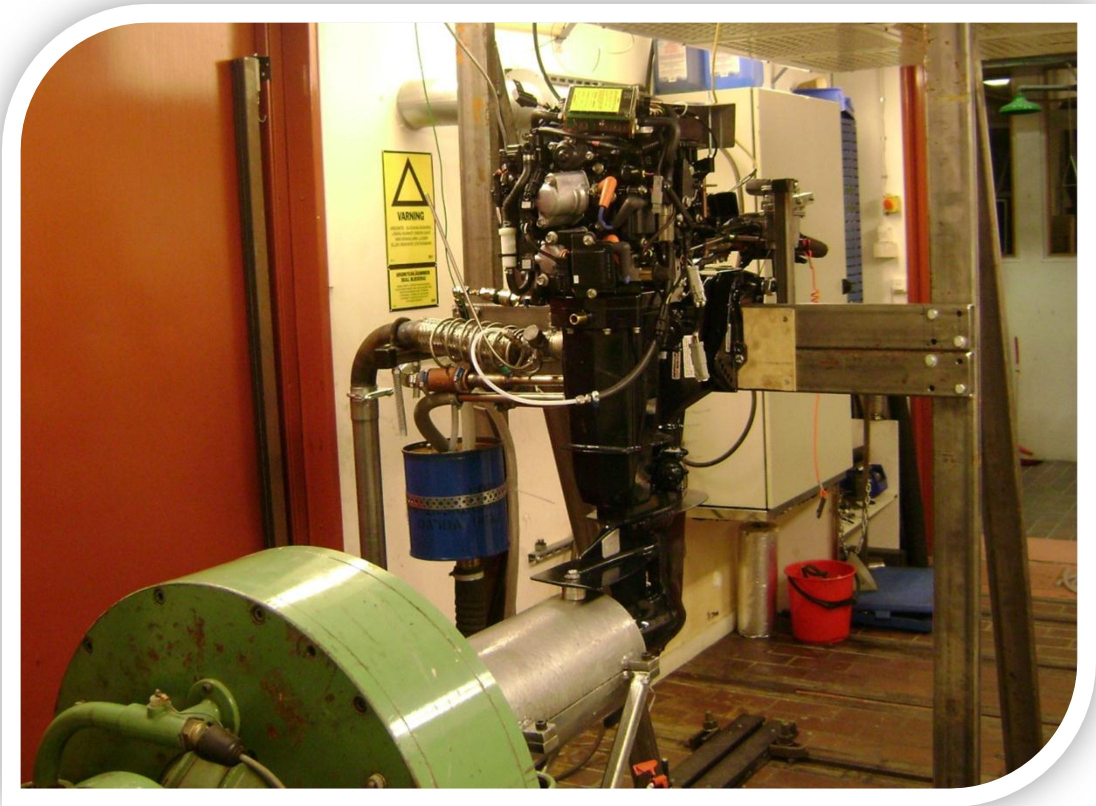

Direct injection system for a two stroke engine

Master’s Thesis in the Automotive Engineering Master's programme

JOSE RODOLFO I. CERVANTES TREJO

Department of Applied Mechanics
Division of Combustion
CHALMERS UNIVERSITY OF TECHNOLOGY
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ABSTRACT

Direct injection is becoming an important option to further optimize internal combustion engines. In the case of two stroke engines, the opportunity of dramatically reducing the HC emissions by almost eliminating short circuit while scavenging, makes this system even more attractive. For small two stroke engines, like the ones used to power chain saw tools, the need of a high pressure fuel pump (its size and electricity needs) makes that the use of direct injection becomes not feasible.

By using a self pressurized (boosted) direct injector, the opportunity of using direct injection opens for these small engines since a high pressure pump is not needed. The Evinrude E-TEC™ outboard engine uses a direct injection system which makes use of this kind of injectors. The Evinrude E-TEC™ outboard engine is analyzed in a test rig to find out the characteristics of the injector and the combustion modes the engine runs at different speeds and loads. The emissions of the engine are measured, along with the power, speed, combustion chamber pressure trace, bsfc, and voltage and current developed by the injector.

Key words:

Direct injection, boosted injector, self pressurized injector, E-TEC, Evinrude, test rig.

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1 Introduction

1.1 Purpose

This thesis is based on the interest of an outdoor power products company in implementing a direct injection system in the two stroke engine of a chainsaw product. The present thesis work aims to the first steps in the process of this implementation.

1.2 Background

The possibility of direct injecting a two stroke engine, as it is going to be seen later, represents many advantages in fuel consumption and emissions, being an important opportunity for a chainsaw application. As the space in such application is limited, direct injection has the important disadvantage of needing more space compared to indirect injection and to carburetor systems. The need for a high pressure pump which is big in size, and the requirement of this pump for a high electric energy supply, which forces to equip the engine with a big generator and a big battery; seriously compromises packaging, cost and weight.

In the beginning of 1990´s Ficht GmbH developed a boosted direct injector for use with two stroke engines. This injector was unique in that it did not required a high pressure pump but was still capable of generating enough pressure to inject into a closed combustion chamber. The way this injector works is described in chapter 4.. In 1995 Outboard Marine Corporation (OMC) licensed Ficht technology and introduced it on a production outboard engine in 1996. The engine manufacturing portion and brands (Evinrude Outboard Motors and Johnson Outboards), including the Ficht technology, were purchased by BRP in 2001.

In 2003, Evinrude introduced the E-TEC™ system, an improvement to the Ficht fuel injection[1]. In 2004, Evinrude received the EPA Clean Air Excellence Award for their outboards utilizing the E-TEC™ system. The E-TEC™ system has recently also been adapted for use in performance two-stroke snowmobiles.

1.3 Objectives

To obtain operation information of the Evinrude E-TEC™ outboard engine's injection system on a dynamometer test rig under idle, medium and heavy load conditions.

Execute a spray characterization on the Evinrude direct injector to find out its main characteristics as size of the cone, penetration, speed and diameter of the drops.

[1] Strauss Sebastian, Zeng Yangbing and Montgomery David. *Optimization of the E-TEC™ Combustion System for Direct-Injected Two-Stroke Engines Toward 3-Star Emissions*. SAE 2003-32-0007.

1.4 Approach

Operation parameters are going to be measured in the Evinrude engine in a test rig. Among this parameters are the operational requirements of the injector (current and voltage supplied). The spray characterization phase involves using the operational characteristics of the injector gathered during rig testing.

2 Two stroke engines

In order to put in context the relevance of the work done in the thesis, a brief description and further comparison of the two and four stroke engine is developed in the first part of this chapter, and then a higher detailed description of the two stroke engine over its most relevant details is presented.

An internal combustion engine (ICE) is a kind of engine which produces work by a combustion process of a fuel-air mixture which takes place inside the body of the engine itself. Other types of engines are electrical, steam, or exothermal.

There are different architectures of internal combustion engines. The type of engine which is the main focus of this work is the reciprocating engine, and it is widely used in many applications, as automotive, marine, tool driving, or electrical power generation. Those with a rotative architecture, which are a different type of ICE, are out of the scope. Two fuels used in the reciprocating engine are mainly used, and define the thermodynamic characteristics that the engine is going to have. The diesel engine injects diesel fuel in a mass of air which is at high pressure and temperature; this inflame the fuel injected, increasing the pressure in the engine. The gasoline engine uses a mixture of fuel and air which is detonated with a spark to increase pressure and obtain work from it. The former is the engine referred to in this work.

In the reciprocating engine there is a rotating crank shaft which delivers the power to a driveline. This crank shaft is connected to a reciprocating piston via a connection rod. The piston is moving inside a cylindrical chamber where the combustion process takes place. The high pressure developed by the combustion process pushes the piston in the direction of the crank shaft transforming the linear force of the piston to an angular force on the crank shaft. The piston moves linearly back and forth along the space available in the cylinder as the crank shaft rotates in one direction. The position when the piston is furthest from the crank shaft is called Top-dead-centre (TDC). The position when the piston is nearest the crank shaft is called Bottom-dead-center (BDC). The furthest part of the cylinder is closed to prevent pressure leakages which may reduce the force that the pressure puts in the surface of the piston.

2.1 Two and Four Stroke Engines

The reciprocating engine is divided in two types: Two-stroke and Four-stroke. In the two-stroke the combustion of the fuel-air mixture takes place every stroke the piston travels towards the crank shaft, from the TDC to the BDC; this is called a power stroke. When the piston approaches the BDC position, ports which are originally blocked by the piston, unblock to allow the change of burned gases with fresh air-fuel mixture in a process called scavenging. See Figure 2-1.

One of the ports has the mission of letting the burned gases to go away from the combustion chamber to the atmosphere, it is called *exhaust port*. The other one, called *scavenging port*, has the task of letting in the fresh fuel-air mixture in to the combustion chamber. The change of gases takes place at the same time, while the piston is in the BDC. Once the piston goes to the TDC back again, the ports are covered and the fresh fuel-air mixture is compressed towards the closed combustion chamber. When the piston reaches the TDC, the spark detonates the air-fuel mixture increasing the pressure in the combustion chamber and forcing the piston again in the BDC direction.

Figure 2-1 The two stroke engine

The four-stroke engine has one power stroke every two turns of the crank shaft. This engine has one stroke dedicated to the exhaust of the burned gases out of the combustion chamber, and another stroke dedicated to the admission of the fresh fuel-air mixture in to the cylinder. The ports through which the gases flow are not covered and uncovered by the piston. Instead, two valves, generally at the top of the cylinder, open and close accordingly to the admission and exhaust strokes. See Figure 2-2. In the first stroke of the cycle, when the piston is approaching the BDC, the admission valve opens to allow fresh air-fuel mixture go into the cylinder. In the second stroke, when the piston has reached BDC and changes direction to TDC once again, the admission valve closes to allow for the compression of the mixture. Once the piston has reached TDC, the mixture is inflamed and the pressure of the burning gases pushes the piston in the direction of the BDC. When the piston reaches the BDC for the second time the exhaust valve opens to let the burned gasses escape from the cylinder, allowing the piston to eliminate it almost completely when the piston is at the TDC back again.

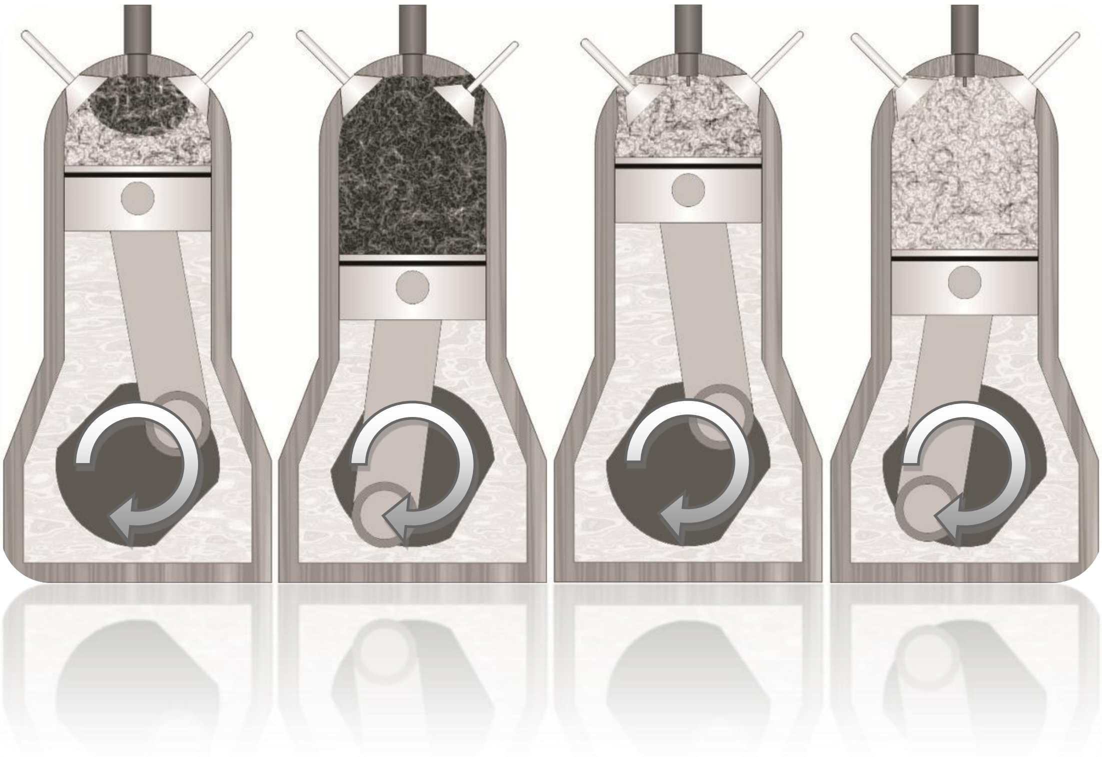


Figure 2-2 The four stroke engine

There are two main differences between the two types of engines. The first one is that, since the two-stroke has a power stroke every turn of the crank shaft and the four-stroke has it every two turns of the crank shaft, the overall power developed by the two-stroke is the double than the power of the four-stroke engine. It also means that a two-stroke engine needs half of the size than a four-stroke to have the same power.

The second one is that the gas interchange in the two-stroke is less efficient that the one in the four-stroke. Since the four-stroke does admission in one stroke and exhaust in another, the gases doesn´t mix and there is no way that fresh fuel-air mixture goes out unburned through the exhaust valve. This makes that the four-stroke has more control over the combustion, being smoother when running and having a better efficiency than the two-stroke. Thus, the advantage of having the double of the power than the four-stroke is only theoretical. Also the problem of the fresh mixture going out through the exhaust valve increases dramatically the emissions of the two-stroke engine, taking it out from the possibilities of being used in applications such as automotive, where these engines have very high emission restrictions.

2.2 Two-stroke scavenging process

As it’s been said, the two-stroke engine changes gases at the same time, in a process called scavenging. In an ideal scavenging process, the fresh fuel-air mixture going in to the combustion chamber should push away the burned gases through the exhaust port whit out mixing with them and whit out letting any particle of fresh mixture to go out as well, filling completely the combustion chamber in which no particles of exhaust gas should remain at all. The real scavenging process is different.

A difference in pressure is needed to be able to move the gases inside the combustion chamber, which means that the pressure in the scavenging port have to be increased in order for the fresh fuel-air mixture to displace the burned gases out of the combustion chamber. In the most simple configuration, the face of the piston which looks to the crank shaft forms a pump with the crankcase. Air is introduced through a manifold in the crankcase by means of the vacuum produced by the piston when increasing the volume inside the crankcase. When the piston reduces the volume, a Check or a Reed valve, or a similar component, blocks the flow of the air back to the manifold, increasing the pressure of the gas inside the crankcase. See Figure 2-3. When the piston unblocks the scavenging port, the pressurized gas in the crankcase flows to the combustion chamber at the other side of the piston, which has a lower pressure. It has to be noticed that, since the exhaust port opens before the scavenging port, the remaining pressure of the combustion is lost, and ideally the chamber is at atmospheric pressure, being the pressure in the crankcase higher and thus flowing the fresh fuel-air mixture to the combustion chamber. In more complex designs, the difference in pressure is achieved by different devices, as stepped pistons, scavenging pumps or external compressors such as turbo chargers and super chargers.

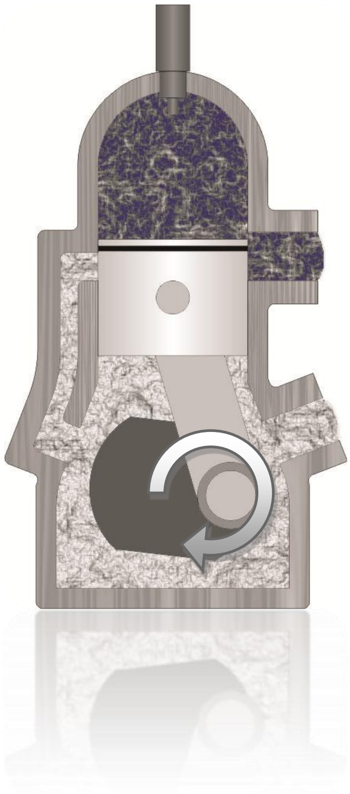


Figure 2-3 Scavenging process in the two stroke engine

The mission of the fresh air-fuel mixture is to replace the burned gases which are spread over all the volume of the chamber. Both, the scavenging and exhaust ports are located at the bottom side of the combustion chamber, as them have to be unblocked by the piston near the BDC and blocked again on the cylinders way to the TDC. See Figure 2-4. On entrance to the combustion chamber, the fuel-air mixture has to travel to the top of the cylinder and back again to displace the burned gases. In such a complex move, mixing between burned and fresh gases happen, depending on the amount of turbulence generated in this movement. There may be also places of the chamber which the fresh gases cannot access in the time available before the closing of the exhaust port. This makes that some burned gases remain in the cylinder to face the next combustion process, lowering the efficiency of it.

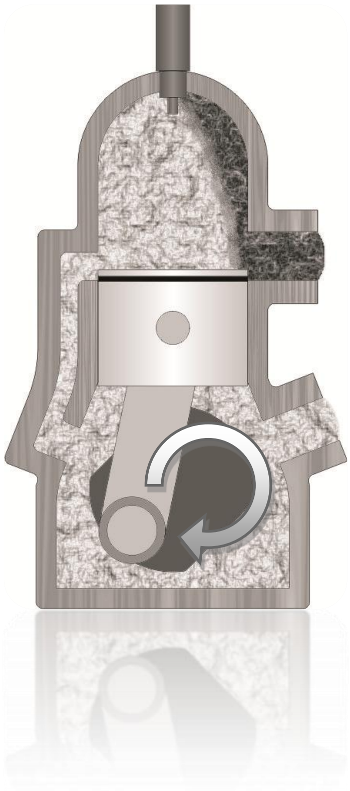


Figure 2-4 Mixing process during scavenging

Also, if the scavenging and exhaust ports are facing one each other at opposite sides of the combustion chamber, the fresh gases will tend to cross the cylinder straight to the exhaust port, going out from the combustion chamber unburned; as the natural tendency of the pressurized gases is to keep going in a straight path. See Figure 2-5. This problem leads to high fuel consumption and high HC emissions.

Figure 2-5 Short circuit during scavenging

The efficiency of the scavenging process depends mainly on the design of the combustion chamber. Different locations of the ports can force the path of the fresh gases in a way which reduces mixing and prevents them to go straight to the exhaust port. See Figure 2-6. Cross scavenge is the simplest arrangement of the ports, in which fresh gases go in at one side of the cylinder, and burned go out at the opposite one. The design of the top of the cylinder must be done in a way that it blocks the gases from short circuiting to the exhaust port, and to force them to the top of the cylinder. This arrangement has good scavenging at medium and low speeds, but high fuel consumption at high speeds[2]. In Loop Scavenge, both ports are in the same side of the cylinder, the exhaust above the scavenging one. This arrangement avoids the fresh gases to go straight to the exhaust port, and makes that the fresh gases reach it only after having travelled through all the combustion chamber. This arrangement has good scavenging at wide open throttle but low scavenging and high fuel consumption at part throttle operation[3]. In Uniflow arrangement, instead of using ports at the bottom of the cylinder, the exhaust is relocated to the top of the combustion chamber behind valves like the ones used in the four-stroke engines. This makes that the fresh gases doesn't have to change direction in order to clean up the cylinder from burned gases, they only travel through the cylinder from the bottom to the top. This arrangement can produce an annular shaped area of burned gases unscavenged, but this problem can be overcome by producing a swirling motion in the incoming gases. This arrangement produces a very good scavenging at wide open throttle[4], but it loses

[2] John B. Heywood, Eran Sher. *The Two-Stroke Cycle Engine.* Society of Automotive Engineers. 1999. Pag 52, Table 2-1.

[3] John B. Heywood, Eran Sher. *The Two-Stroke Cycle Engine.* Society of Automotive Engineers. 1999. Pag 52, Table 2-1.

[4] John B. Heywood, Eran Sher. *The Two-Stroke Cycle Engine.* Society of Automotive Engineers. 1999. Pag 52, Table 2-1.

the simplicity of the conventional two-stroke engines because it needs complex mechanisms to operate the valves.

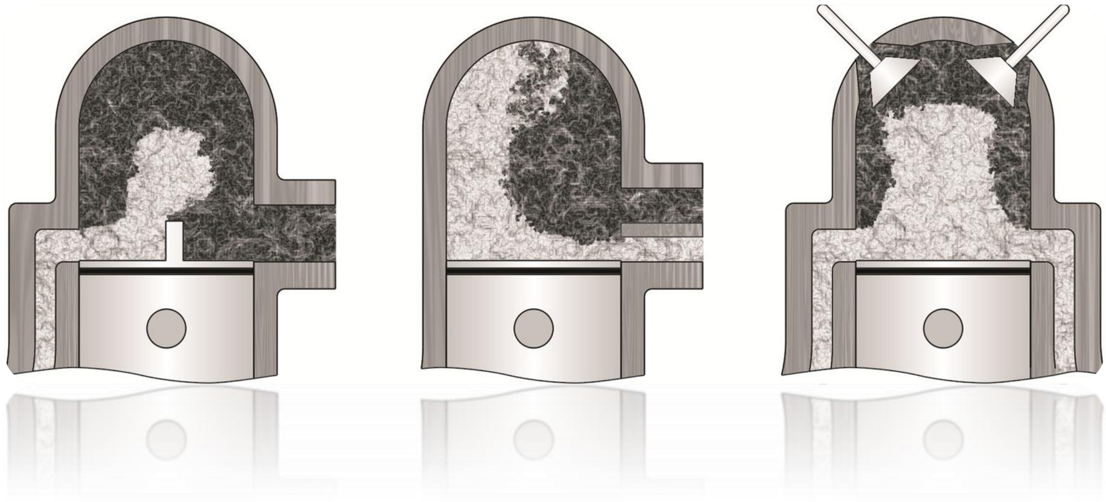


Figure 2-6. Different arrangements of scavenging ports. a) Cross scavenge. b) Loop scavenge. c) Uniflow.

3 Direct Injection

3.1 Injection types

The gasoline spark ignited engine has two types of injection system.

- Indirect injection or port injection
- Direct injection

The indirect injection system places the fuel in the air manifold, outside the combustion chamber. See Figure 3-1. This arrangement produces a premixed fuel-air mixture, which means that the fuel and the air are already mixed when they enter the cylinder. This has as consequence that, in normal conditions, the flame originated by the ignition of the spark plug will grow from the center (position of the spark plug) to the walls. When the flame collides with the walls, it transfers an important amount of heat to them; this heat cannot be converted to pressure and will go away as waste through the cooling system.

In abnormal conditions an auto ignition phenomenon, known as Knock, may occur. While the flame develops, the burned gasses behind it increase their pressure, compressing the unburned gasses in front of the flame and increasing their temperature, pressure and density. If the temperature is high enough, it will cause a detonation of these gasses, which release the major part of its energy, if not all, faster than the gases being ignited by the flame. This phenomenon can cause a serious damage to the engine.

For the two stroke engine specifically, it will cause the problem of short circuiting described in the previous chapter, in which an unburned amount of fuel-air mixture goes straight to the exhaust port, increasing emissions and fuel consumption, reducing the efficiency of the engine.

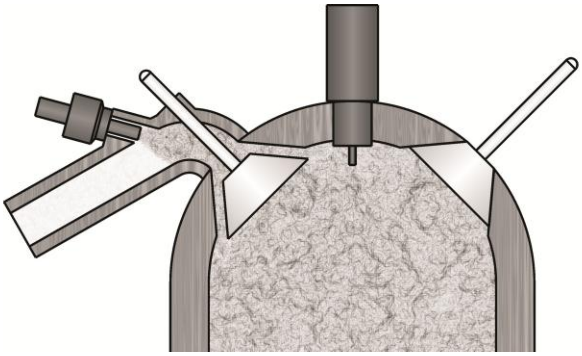


Figure 3-1. Indirect injection, or port injection.

The direct injection system introduces the fuel directly in the combustion chamber, which means that only air enters the cylinder through the manifold and the mixture of air and fuel takes place inside the combustion chamber. See Figure 3-2. This system has the possibility of running in the premixed mode previously described if the injection of the fuel is done during the admission stroke, and in a mode called stratified if the fuel is injected at the end of the compression stroke which means that the fuel burns while being injected.

Stratified combustion has the advantage of avoiding knock, since there is no air-fuel mixture in the front of the flame. Gasoline needs to have a special property of anti ignition to avoid knocking when the pressure inside an engine's combustion chamber is very high, but in the case of stratified combustion it is not needed; even more, different kinds of fuels can be used by the same engine.

A second advantage of stratified combustion is that, since the front of the flame doesn't reach the cylinder wall, less amount of heat is transferred to them, increasing the work transferred to the piston. The third advantage of this type of injection is that the combustion can be controlled by the amount of fuel being injected, living the opportunity of not using a throttle valve at all and to reduce significantly pump losses. This also gives better control of the flame in the combustion chamber than premixed mode, as in premixed mode the flame propagation depends heavily on the air movement inside the combustion chamber which is not always the same, and in stratified mode the flame is always near the injector and controlled by the injection of the fuel and the shape of the piston.

As for the two stroke engine the big advantage is that, since the fuel is being injected when the exhaust port is closed, the short circuit problem is totally overcome. This characteristic is very important since this problem has not permitted the two stroke engine to be further used in the automotive industry due to the high emissions it produces.

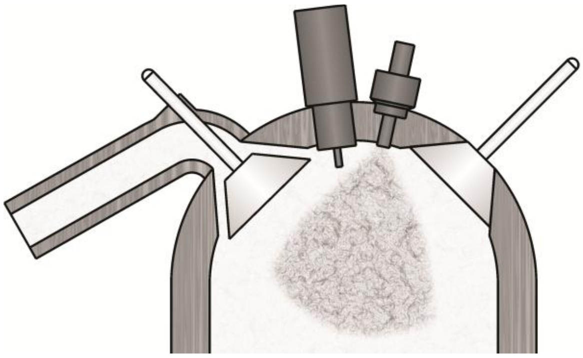


Figure 3-2. Direct injection.

Another big and important difference between both combustion modes is that flame propagation is faster in premixed mode than in stratified. This makes that premixed mode can reach higher engine speeds, developing higher power than stratified mode. This difference makes that, in practical applications, an engine needs to use both combustion modes. Stratifying at low and mid revs makes that the operation of the engine is smoother and the fuel consumption is improved. Premixing at high revs will develop a higher power

The final difference to be mentioned in this work is the fuel pressure the injector needs to have on each injection type. Indirect injection runs on premixed mode, and in this mode the gasoline have time enough to mix with the air. Direct injection runs also in stratified mode, in which the gasoline has to mix with the air very fast to be able to ignite when going out from the injector. This requires the direct injector to produce a thinner spray than the indirect injector (smaller drops of fuel in the jet spray). To achieve this, the direct injection system needs a higher pressure in the fuel rail, and thus a high pressure fuel pump, which increases energy consumption, weight and cost. There are some other strategies to implement a direct injection system, like a dual fluid injector, or a hammer type of injector. An example of the last one is going to be presented in section 4.

3.2 Fuel Injectors

Attending to the injection mechanism, the injectors can be divided in two types.

- Solenoid actuated
- Piezoelectrically actuated.

The solenoid actuated injectors use a solenoid to actuate the needle which allows the fuel to be injected in to the combustion chamber. The piezoelectric actuated injector uses an arrangement of piezoelectric crystals to operate the needle. This crystals change their dimensions when an electric voltage is applied to them. The main difference between them is that piezoelectric injectors have a higher actuation speed than solenoid ones. This speed increment can be used to produce two injections during the same stroke, as to change from stratified to premixed mode in a softer way. It is also possible for them to vary the needle lift, changing the flow of the fuel.

4 The Evinrude E-TEC™ Outboard Engine and its Injector's Characteristics

The factor which makes the engine being analyzed so interesting is the fuel injector it has. The engine is direct injected, having all the characteristics described in the previous chapter. But this injector is different from the common direct injectors described.

This injector is solenoid activated, but it is able to develop the fuel pressure needed for the injection by itself: it doesn't need a high pressure pump like the common direct injection systems. The E-TEC™ injector is based in the FICHT™ development of self pressurized injector (boosted). The system needs a low-pressure fuel pump providing 2.5 bar[5].

"To actuate the injector, an electromagnetic field accelerates a steel armature through a short free stroke until it impacts a fuel column, thus generating fuel pressure which opens a spring loaded outwardly opening nozzle. During the first part of the injection event, the fuel pressure is increased due to the additional conversion of the initial armature kinetic energy into pressure. The fuel pressure during the second stage of the injection process is solely driven by the electromagnetic force."[6]

Figure 4-1 The Evinrude E-TEC™ Engine

The limit of pressure the injector can create is 50 bar. The driving voltage of the injector is 40 V. The time that the injector is fed with this voltage determines the time of the injection. The injector develops a current of up to 10 amperes, depending of the time that the injector is fed with voltage. The coil of the injector reaches its current limit at 1.5 mSec. An image from the injector is shown in Figure 4-2. Figure 4-3 shows an image of the injector with the main case disassembled.

[5] Strauss Sebastian, Zeng Yangbing and Montgomery David. *Optimization of the E-TEC™ Combustion System for Direct-Injected Two-Stroke Engines Toward 3-Star Emissions*. SAE 2003-32-0007.
[6] Strauss Sebastian, Zeng Yangbing and Montgomery David. *Optimization of the E-TEC™ Combustion System for Direct-Injected Two-Stroke Engines Toward 3-Star Emissions*. SAE 2003-32-0007.

Figure 4-2 Injector size

Figure 4-3 Injector Construction

5 Methodology

As it has been mentioned in chapter 1, the main objective of this thesis work is to obtain information of the engine's operation and to characterize the injector's spray.

The steps taken to achieve these objectives are shown in the following list.

1. Selection of the parameters to be measured during the normal operation of the engine.
2. Installation of the engine in the test cell. This step involved hard work, as some adaptations to the engine had to be done. Also the sensors to be used to measure the parameters were installed during this step. The biggest issue with the sensors was the set up of the data acquisition system. The pressure sensor and the angle encoder gave many problems.
3. Setting up of the experiments. This means to decide the points at which the measurements are going to be taken. Points determined by different loads (torque) at different speeds.
4. Design of the data structure. As many points were to be measured, and there were to be many parameters by each point, a data structure had to be decided in order to access the information fast and in an ordered way during the data manipulation.
5. Performing of the measurements in the test rig.
6. Setting up of the injector in the spray characterization chamber, in order to characterize the injector's spray.
7. Performing of the spray characterization.
8. Analysis of the information (results, analysis and conclusions).

5.1 Parameters selection

The information to be gathered from the engine was mainly defined by the thesis sponsor, and it was mainly the combustion modes the engine runs at. Other aspects of the engine's operation the sponsor was interested in, were the pressure inside the combustion chamber, the power and the torque developed by the engine. As the combustion mode the engine is running at does not comes straight forward from a sensor, but from an analysis that has to be done from other parameters, this ones were specified as a first step. The parameters to be measured are shown in Table 1.

Table 1. Parameters measured from the engine

Crank angle position.	This is determined by an angle encoder. The signal is fed directly in to a computer.
Combustion chamber's pressure trace.	This measurement comes from a pressure sensor installed on the engine's head and which is in direct contact with the gasses inside the combustion chamber. The signal is continuously fed in to a computer, and matched with the angle encoder signal.

Injection timing.	The current going in to the injector is continuously monitored and matched with the angle encoder signal. Then, the pulse of current shown in a plot reveals the injection timing.
Ignition timing.	This is analog to what is done with the injection timing, but monitoring the current being fed in to the coil.
Fuel consumption.	An instrument is used which weights the amount of fuel used along a determined period of time. From here, the fuel consumption in g/sec is straight forwardly obtained. Also, the bsfc is calculated using the measured mass over the average power delivered by the engine during that same period of time.
Exhaust gasses' lambda value.	It is measured using a lambda sensor very common in the automotive industry. However, it is not useful when measuring this engine, since as it is a two stroke engine, the short circuit phenomena modifies the properties of the exhaust gases. For the specific case of the Evinrude engine, as it is short circuiting only air, the lambda value goes very lean (up to 4 when idling).
Throttle's angle.	A sensor locating the position of the throttle was not used. Instead, since the throttle is controlled using a servo and thus the position of the throttle is directly related to the dial controlling the servo, it was calculated from the dial position. The angle of the throttle when full closed and full open was measured and related to the position of the dial at the same points. From there, the angle of the throttle is derived.
Emissions	They were measured using an emissions measuring instrument. There were measurements for HC, NOx, NO, NO_2, CO and CO_2.
Engine speed	This was provided by the brake's instruments. Since the brake was connected to the engine's gearbox, as it is going to be described in section 5.2, it has to be converted to the real engine's speed using the gear ratio of the gear box.
Engine torque	This was provided by the brake's instruments. The case is analog to the engine speed. It had to be converted to the real engine's torque using the gear ratio of the gear box.
Engine power	This was derived from the speed and torque information.
Engine bmep	This was derived from the power, using the formula $bmep = \frac{P}{V\,n}$; P=power, V=engine's displacement, n=engine's speed.

Also, in order to control the injector in the spray characterization rig, the injector's electrical parameters had to be measured first during the normal operation of the engine.

5.2 Test Rig Set Up

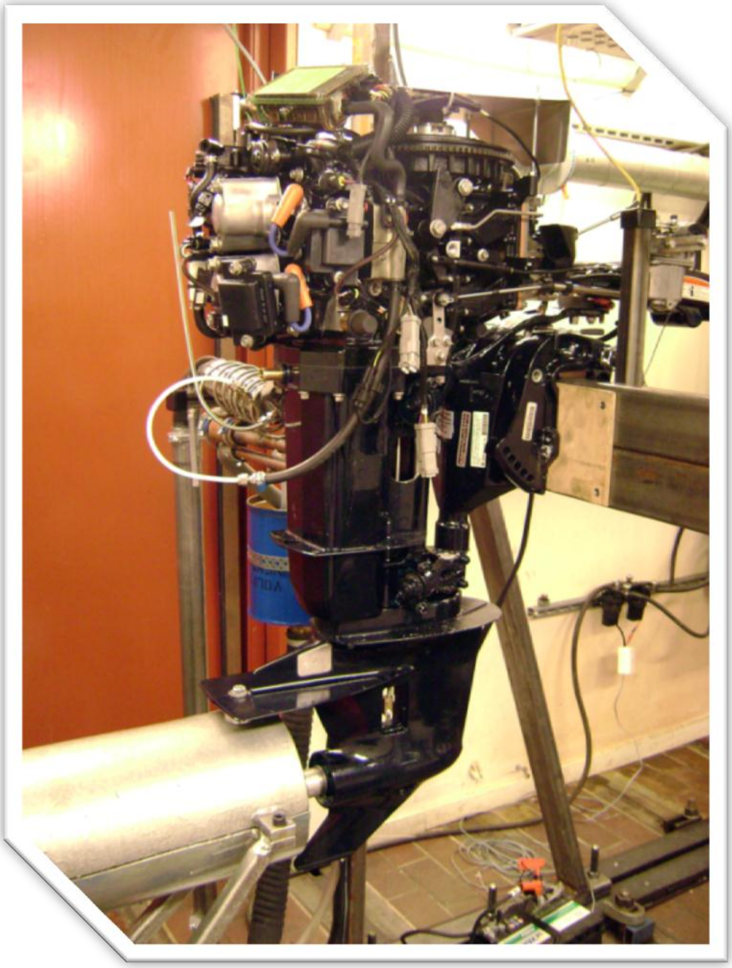


Figure 5-1 Engine on rig

The outboard engine was connected to the measuring brake through the propeller shaft. It was decided to mount the engine complete, to try to preserve as much conditions the engine was designed to as possible. For this reason, all the brake measurements such as brake power and bmep have the gear box included, in other words, the losses of the gear box have not been calculated.

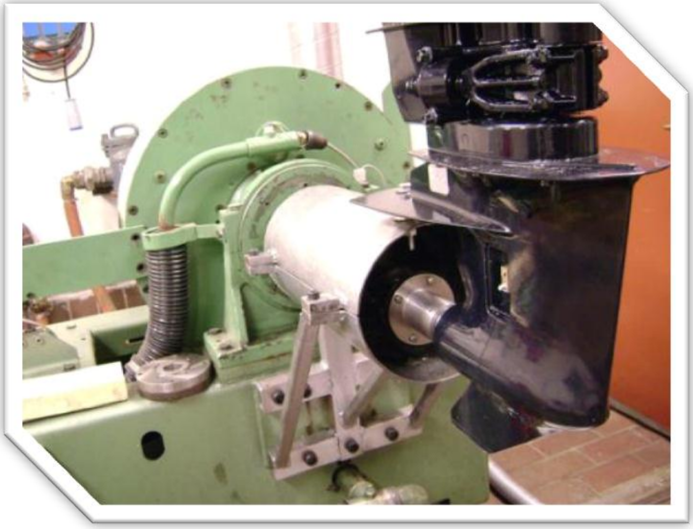


Figure 5-2 Gearbox and brake

The first challenge with the engine was to split the cooling water from the exhaust gasses. The engine is designed to use water from the sea, or the lake it is being used at, and use it as a cooling fluid. Once the water has been used, it is mixed with the exhaust gases in a chamber between the gearbox and the engine, and they are disposed from the engine thru the center of the propeller. As the emissions measurement equipment does not admit any particle of water, the engine had to be dismantled and modified to split water from exhaust gases.

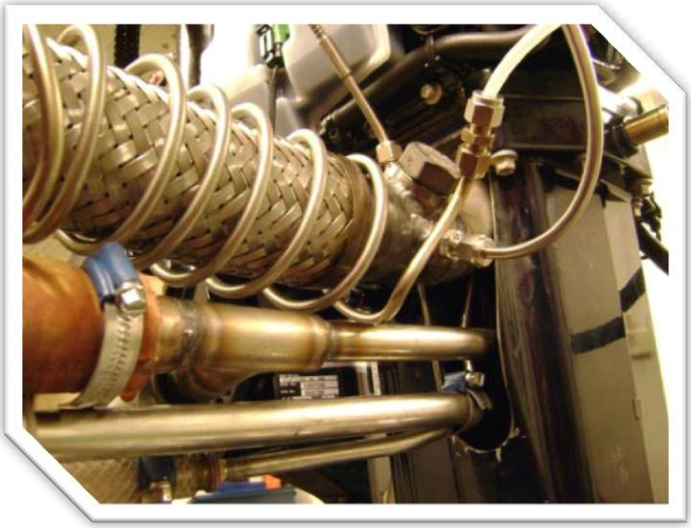


Figure 5-3 Cooling and exhaust separation

The gearbox temperature was a concern. I is normally cooled down by the water it is immersed in. The complexity of needing a container for the cooling water and a shaft going out from it through a sealed hole motivated for a different solution. It was decided to circulate the oil from the gear box in to a heat exchanger. This was not enough when speeding up the engine at high loads, being one of the biggest obstacles when measuring those points. A fast and simple fixture was conducted using a simple bucket, a hose and aluminium paper for the most demanding points. Even though this fixture was not able to maintain the temperature for a long time, it gave enough time to record the data point by point.

Figure 5-4 Cooling of gearbox

The exhaust pipe ends in an open container full of water. This arrangement tries to replicate the counter pressure of the water of the sea. In the beginning, the exhaust pipe was connected to the exhaust system of the test cell. This system sucks the gasses from the engine with a light vacuum. This vacuum pulled more oil from the oil container in the engine, damaging the spark plugs. For this reason, the option of the container full of water was adopted.

Figure 5-5 Exhaust container

The engine had holes aside of the spark plugs blocked by a screw. May be this holes are used in some way in larger engines, but they are not in this engine. A pressure sensor was adapted to fit this hole. The model of pressure sensor couldn't stand the temperature (AVL GM12D), and in a preliminary measurement it stopped operating. A second sensor was installed, with the provision this time of air cooling. A thermocouple was also installed to monitor the effectiveness of the cooling system. The temperature was maintained around 200 C.

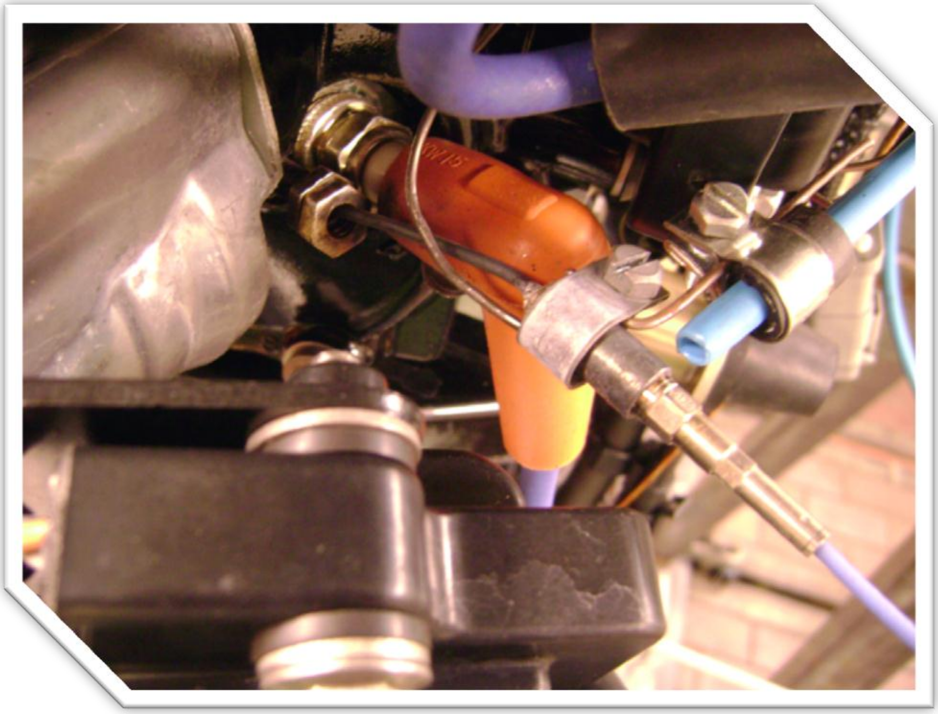


Figure 5-6 Pressure sensor setup

The engine was connected to an AVL Dynamic Fuel Balance to perform fuel consumption measurements.

The ignition and injector timings were measured with a Fluke AC current clamp. The intention with those was only to track the timings, as they are not precise enough as to measure the current of the injector. With this purpose a simple circuit was built to measure current and voltage. It made the current to run through resistors of known values and the voltage on those resistors was measured. Using a differential amplifier the current and voltage was captured in the computer using LabView express. As the differential amplifier was not available all the time for being in use in another project, only selected points were measured with it.

5.3 Experimental Setup

Relevant points for measurement must be selected which represent some interest for the engine analysis. In the beginning it is difficult to select those points as there is not knowledge about the engine's characteristics. For such reasons, some predictions have to be done in order to plan ahead the points to measure.

As a start point, it is possible to calculate the torque at the peak power point by using the technical information of the engine, in which is stated that:

$$P_{max} = 18.4\ kW\ @\ 5800\ rpm$$

From there, a torque of 30 Nm is calculated for that point.

In a preliminary test, by using the dyno's analogical equipment (speed and torque meter), two measurements of the torque at WOT where taken, being **41.86 Nm@2150 rpm** and **46.51 Nm@3225 rpm**. **3000 rpm** is assumed to have the **highest torque**, and a torque of **15Nm@900 rpm** is also assumed. From this information it is possible to build an approximated torque curve, which is shown in Figure 5-7.

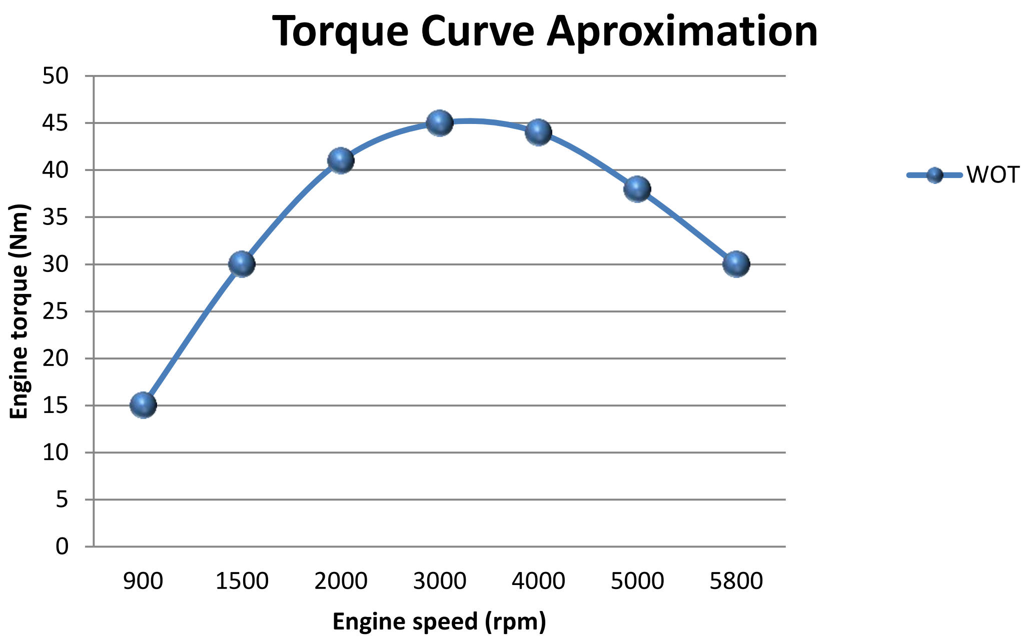


Figure 5-7 Torque curve approximation

As the time needed to take the measurements from each point is not known, a set of priority points is selected; this in the case that it becomes not possible to measure all the points. It was decided to measure all the points in the case that time would be available.

In general it is possible to say that the points of interest are those of low load in a low speed area and those with a high load in the high speed area. Those points are shown in Figure 5-8.

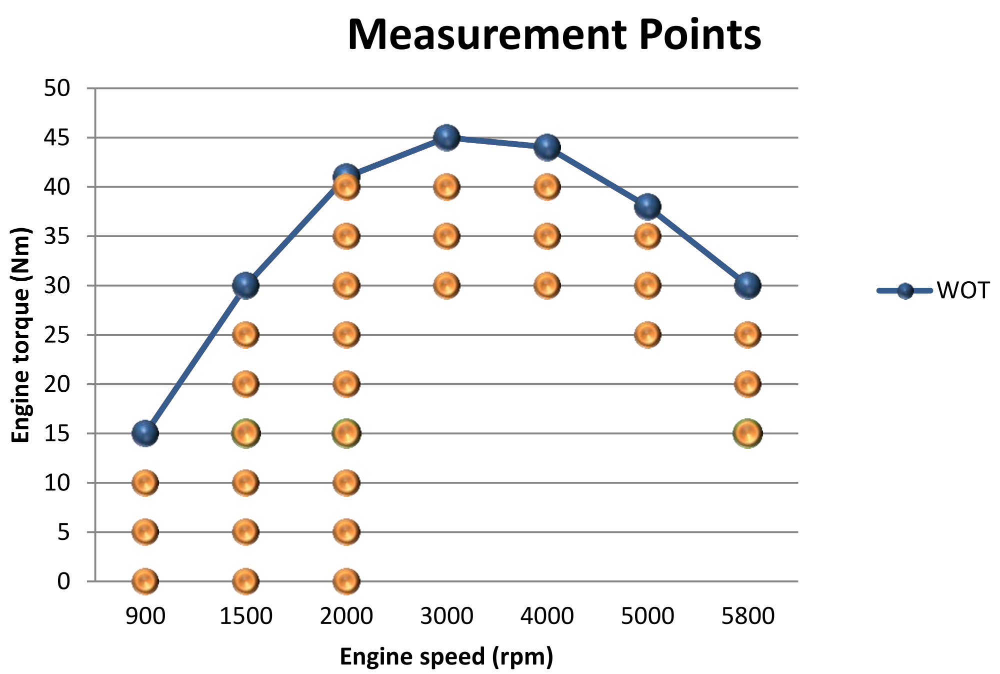


Figure 5-8 Measurement points

During preliminary tests, it has been identified that the engine runs lean in the low revs area having a lambda value is as high as 4, and it decreases as speed and load increase. Further observation of the engine has shown that the throttle doesn't open until a certain load and speed has been reached. It has been observed that the speed of the engine while the throttle is still closed can go as high as **8 Nm@3000 rpm**, or **11 Nm@2500 rpm**.

As this frontier where the throttle starts opening is interesting also to analyze, some measurements have been done to include it in the predicted torque curve. This line is added to the curve in Figure 5-9.

It is also needed to say that the lambda values seen on the lambda measurement device are not real. It will always show lean values because a two stroke engine short circuits clean air in to the exhaust port. This will make that the amount of air will increase in relation to the fuel burned, making the reading something different from what is going on in the combustion chamber. The real lambda values are calculated from the CO and CO2 measurements. This is explained later.

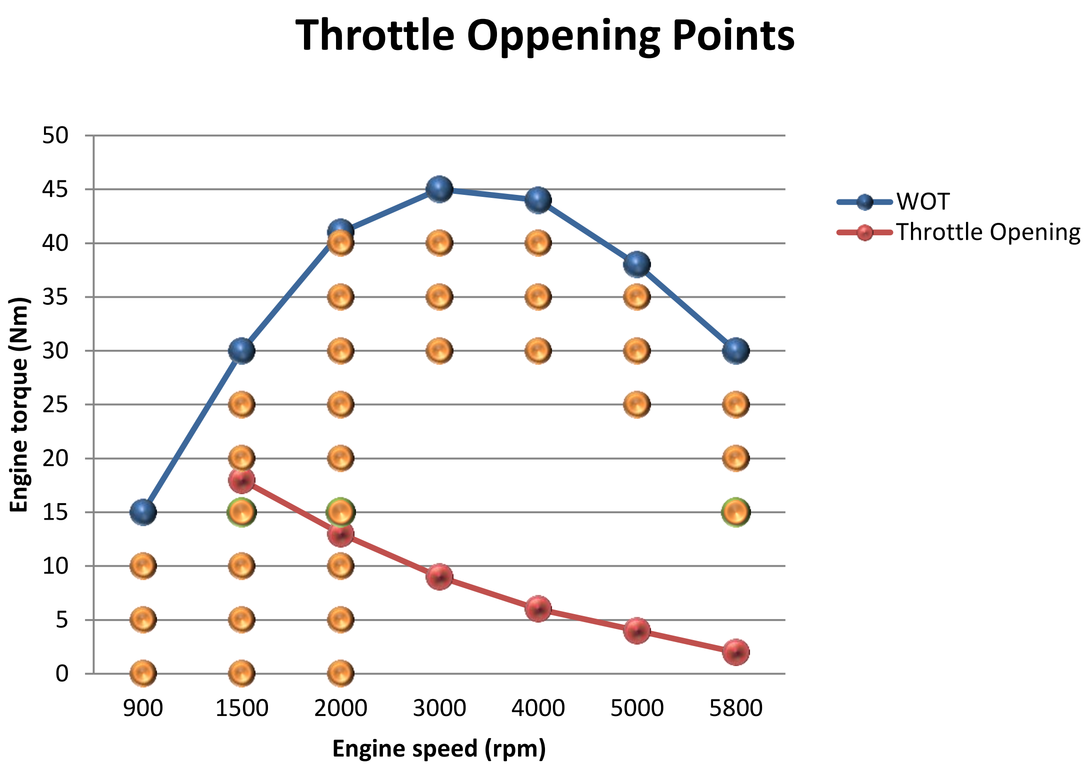


Figure 5-9 Throttle opening points

In this plot the throttle opening curve is showing the frontier at which the engine starts working at lambdas near to 1, while the area below the curve is an area in which the engine runs lean, driven only by the changes of mass of fuel and injector and ignition timings.

There is one more curve that has to be considered. Since this engine has been conceived for a marine application, attention has to be put in the power consumed by the propeller. This curve can be approximated as well. From the Propeller Handbook[7] there comes the relation:

$$PHP = C_1 x RPM^n$$

n = 2.7 for average boats

PHP = Propeller horse power

RPM = Engine speed

C1 = Constant

From here, the torque relation can be found:

$$\tau = C1 x RPM^{1.7}$$

The engine's operation manual mentions that "the correct propeller for your boat, under normal load conditions, will allow the engine to run near the midpoint of the RPM operating range at full throttle"[8]. Considering 3000 rpm as the middle of the operating range, and using τ=45 Nm, C1 is found to be 55.22x10^{-6}. In this way, the propeller's torque curve is approximated to Figure 5-10.

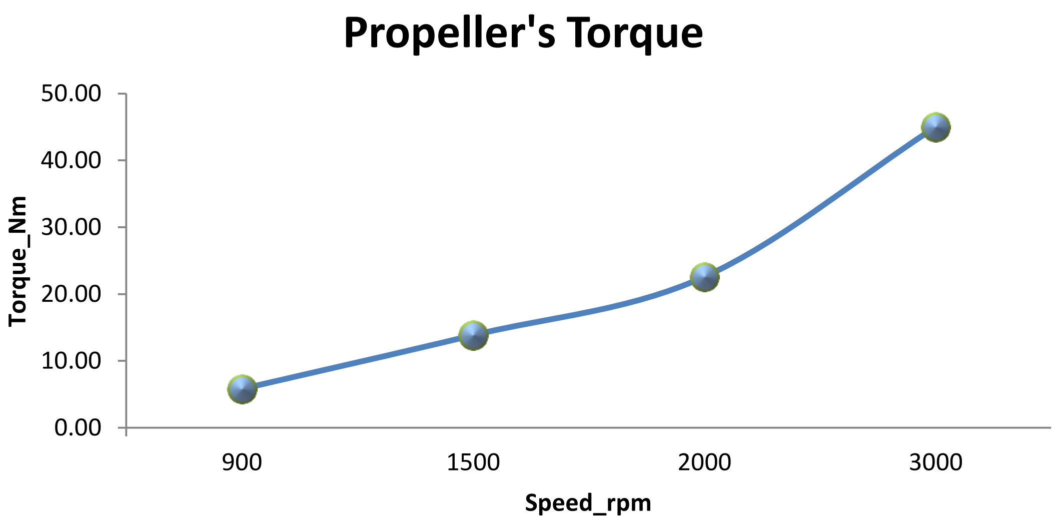


Figure 5-10 Propeller's torque curve

The propeller's torque is then added to the torque map. The complete set of points resulting from the addition of all this curves becomes the Engine Map, which organizes all the measurements that have to be done to the engine; it is shown in Figure 5-11.

[7] Dave Gerr. *Propeller Handbook*. International Marine. 2001. Pag. 4 @ Google Books.
[8] BRP Us Inc. *Evinrude Etec Outboard Operator's Guide.* 2010. Pag. 65.

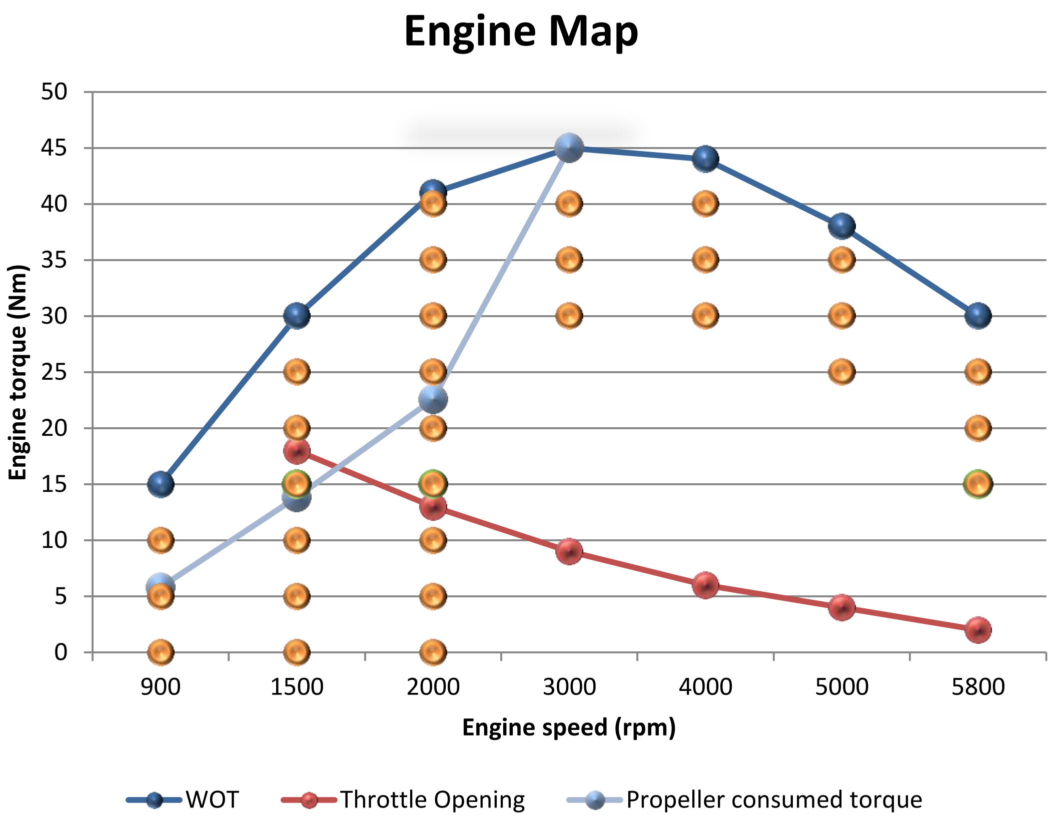


Figure 5-11 Full set of measurement points

Each point is named using a number, in order to easily identify it during the analysis of the data. The numbers assigned to the points are shown in the following Table 2.

Table 2 Predicted measurement points

Measurement points													
	load												
speed	0	5	10	15	20	25	30	35	40	45	wot	Throttle op	Propeller
900	1	2	3								4	5	45
1500	6	7	8	9	10	11					12	13	46
2000	14	15	16	17	18	19	20	21	22		23	24	47
3000	49	50	51	52	53	54	25	26	27		28	29	48
4000							30	31	32		33	34	
5000						35	36	37			38	39	
5800	55	56	57	40	41	42					43	44	

The consecution of numbers is related to the progression of the load in a single engine speed. The points for the WOT, Throttle opening and Propeller's curve are identified separately in the same table.

5.4 Data Structure

Information from 50 cycles will be stored for each measurement point. This is with the intention of averaging the information from those 50 cycles, and being more precise with the measurements. Each cycle will have 15 parameters measured and 5 more parameters calculated from the first ones.

The pressure trace, the spark timing, the injector timing, the speed and the load are refreshed cycle by cycle, while the rest of the data, as emissions of fuel consumption, are captured only one time and being stored the same for all the cycles. The structure of the data is shown in Table 3.

Once there is a data average for each point, the bmep, torque and power curves can be built. Also, from the injector and spark timing information, the injector's operation characteristics can be gathered in order to analyze the mode it is running at (premixed or stratified). The values that are going to be considered for this are the following:

- Spark timing in CAD.
- Injection timing in CAD.
- Injection time in seconds.

Important information is also the current and voltage used by the injector. For this purpose, and for them to be accurately measured, a circuit was built which sent the information to a differential amplifier which allowed the signals to be fed in to the logging computer. The differential amplifier is not available in the lab all the time, so special measurements will be performed for this purposes, gathering the current and the voltage signals for all the loads in steps of 5 Nm for the constant speed of 3000 rpm, and for all the speeds planned for the engine map at a constant load of 20 Nm. For the points considered in the engine map, the current to the injector will be measured using a magnetic clamp with the purpose of detecting when the injection is taking place, as the current reading of this device can be not as precise as the differential amplifier one.

Table 3 Data structure as stored in Matlab

Evinrude	Point	Cycle	Data	
			PresTrace	
			injCurrent	
			sparkTim	
			fuelConsm	
			lambda	
			dialPos	
			bsfc	
			HC	**measured**
			NO	
		a01	NO2	
			NOx	
			CO	
			CO2	
			speed	
			torque	
			throtAng	
			massInjected	**calculated**
			airMass	
	point01		bmep	
			power	
		a02		
		a03		
		.		
		.		
		.		
		a50		
		Data		
Evinrude	point02			
	point03			
	.			
	.			
	.			
	point48			
	bmepTrace			
	torqueTrace			
	powerTrace			
	injParameters			
	injMap			

5.5 Spray Characterization Setup

The spray characterization task was performed using a spray characterization chamber which consists of a sealed volume in which the injector discharges fuel in to a pressurized volume representing the back pressure of the cylinder. The chamber has windows thru which the injector's spray is measured. The spray characterization chamber is shown in Figure 5-12.

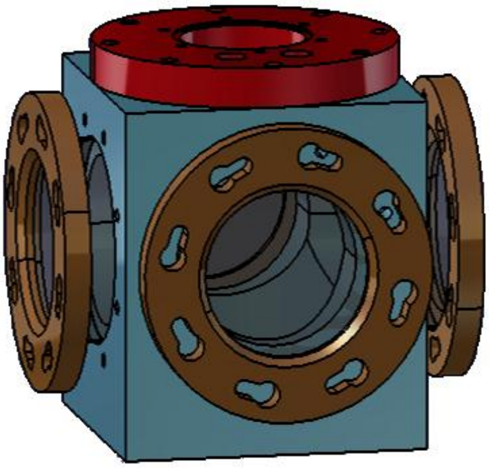


Figure 5-12 Spray Characterization Chamber

As the Evinrude injector has a different volume and shape from the common gasoline indirect and diesel injectors for which the chamber was designed, it was needed to modify the support to which the injector was attached to the chamber. A cut view of the spray characterization chamber is shown in Figure 5-13. There it is possible to see the position the injector has in the chamber.

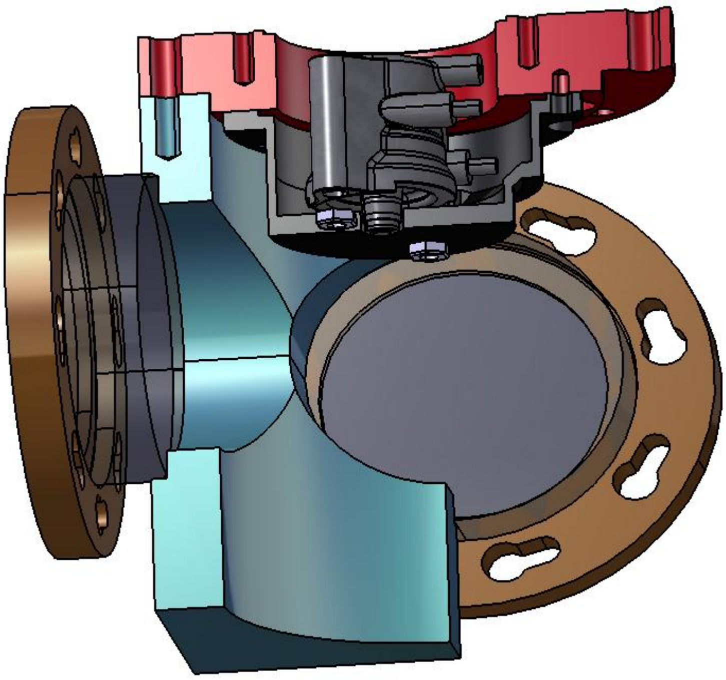


Figure 5-13 Spray Characterization Chamber Section View

The control of the injector was an important issue at this stage of the project. At the combustion lab there is a set of standard injector controllers, but at the beginning it was not sure how to set tem up in order to produce a useful signal. During the first measurements of the current driving the injector, it became clear that those standard controllers were useless as the current they produced didn't match the current required by the Evinrude injector. See section 6.2 for details on the current used by the injector. For this reason, it was needed to build a controller specially designed to drive this injector. It was a main issue as it took long time until the controller was ready to be used.

High speed imaging was used to capture images of the injector's spray. The images were used to measure the penetration and the diameter of the spray's cone. It also allows seeing the spray's behaviour against different back pressures. Also, it is known that the injector is capable of generating its own pressure to inject the fuel in to the combustion chamber, but it is interesting also to find out if the pressure of the pump feeding the fuel in to the injector makes a difference in the spray This was possible to see using high speed imaging as well. The arrangement is shown in Figure 5-14.

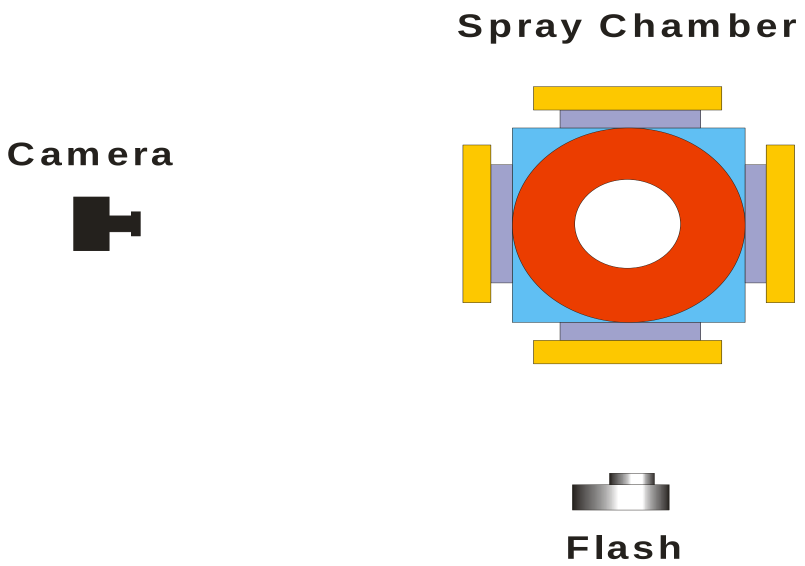


Figure 5-14 High Speed Imaging Setup

To measure the speed and diameter of the drops in to the injector's spray, a phase doppler laser was used. Usually, when performing this kind of measuring, a complete mesh of the spray is measured, but as the intention in this work is to have an estimation of the speed and diameter of the drops, only one point was taken. Figure 5-15 represents the spray in to the combustion chamber. It is clearly visible the source of the spray, the needle, which is set to be the origin to start measuring the spray size.

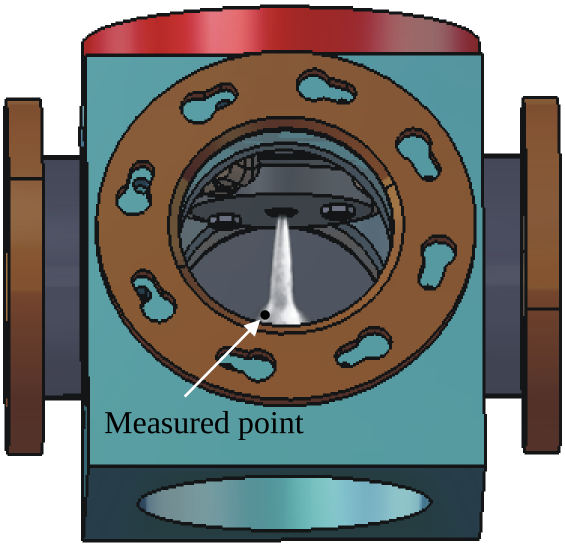


Figure 5-15 Spray in to the combustion chamber

The point selected to measure drop speed and diameter was at 53.5 mm below the surface of the needle, and 17.5 mm out its center. As the spray has a conic shape, it is useless to measure in the center, so the measuring point has to be set on the face of the cone. The representation of the measured point is also shown in Figure 5-15.This point was selected because it is considered that there the spray is completely developed.

The arrangement of the phase doppler instrument is shown in Figure 5-16. The receiver related to the beam has to be at 60º in order to properly work.

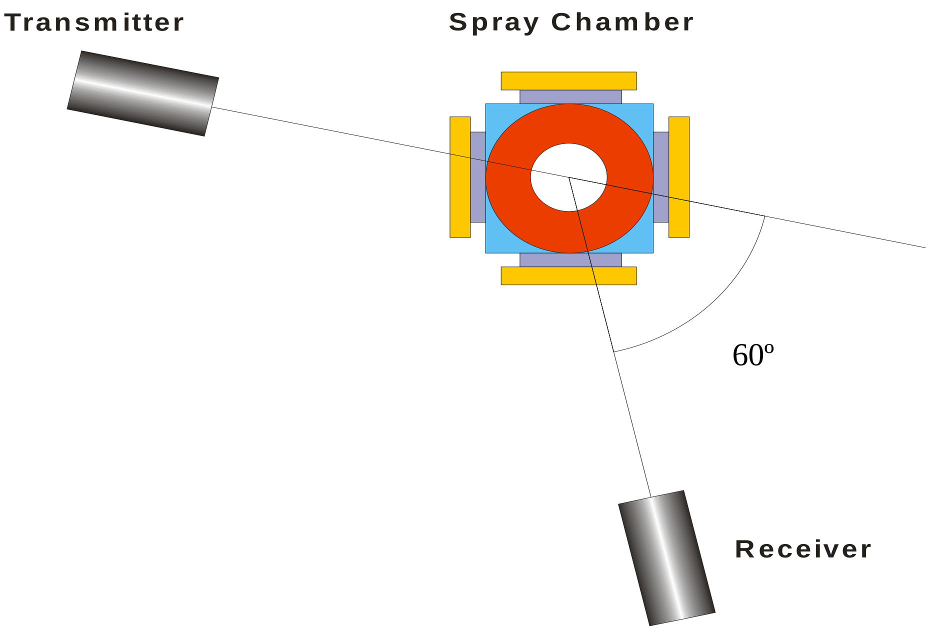


Figure 5-16 Phase Doppler set up

6 Results

This chapter presents the results obtained from the engine characterization in the test rig. The first result is to obtain the real torque-speed curve and to compare it to the curve estimated in the methodology section. The power-speed curve and the imep-speed curve are also presented in this section.

The second section presents the electrical measurements from the injector. As it's been said in the previous chapter, the current and voltage of the injector have been measured in a way which assures high precision, and they are presented compared to the injection time.

In the third section the map of the injector is presented, in which the injection timing is deployed along spark timing and injection time. This is with the purpose of identifying the combustion mode in which the engine is running at (homogeneous or stratified).

The fourth result presented is composed by the plots for the specific fuel consumption and the emissions, as to identify the consumption and emissions performance of the engine through its normal use along the propeller curve.

6.1 Torque-Speed curve

In the approximated torque curve described in chapter 5, Figure 5-7, the maximum torque was assumed to be 45 Nm@3000 rpm. According to it the propeller curve was set to have its peak at the same point. In the real curve the maximum torques in every speed were found to be higher than expected, and the maximum engine torque was found to be 52.5 Nm@4000 rpm (among the speeds that were measured). The torque-speed curve is shown in Figure 6-1.

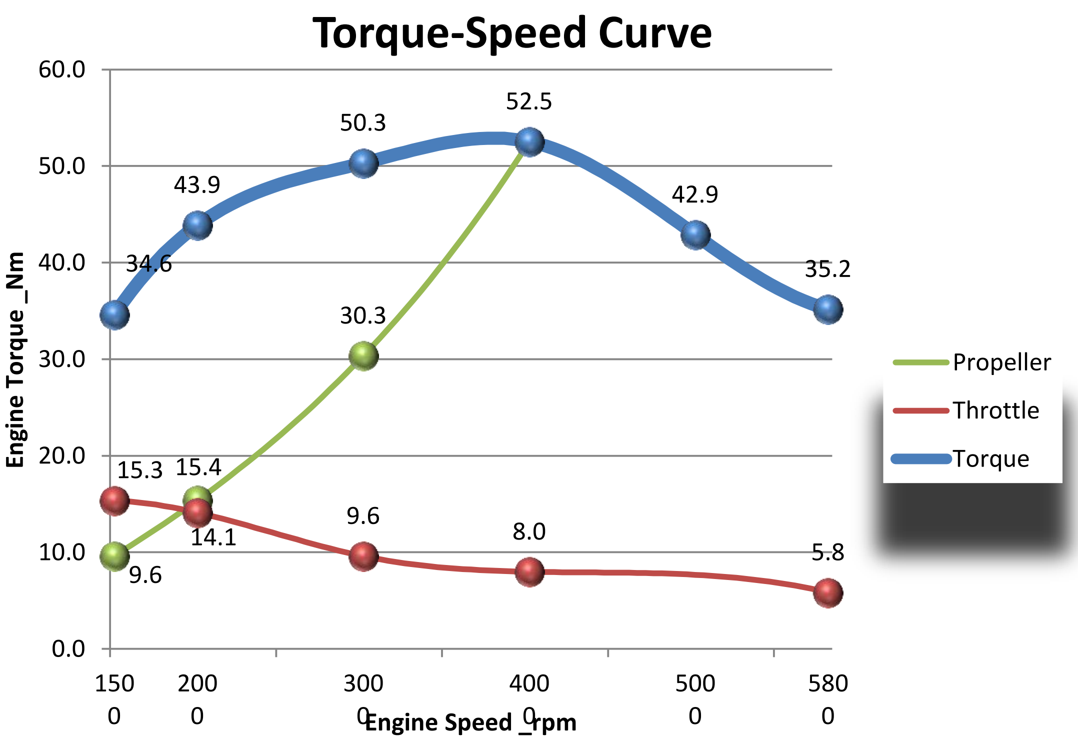


Figure 6-1 Torque-Speed curve

The max torque curve is represented in thick blue. The torque of each point is shown near the corresponding point, and they are shown in Nm. It can be seen that the speed of 900 rpm was removed from the planned engine map. During the measuring it was noticed that the engine was so unstable at this speed that it was hard to set the precise load in the screen, making the reading not very trustable. Even though points of 0 Nm, 5 Nm and 10 Nm @900 rpm were measured, they will not be used.

It also can be seen how the peak torque is at 4000 rpm. The propeller curve was modified as to match it with this point, adding one additional point to the measuring map. As it has been described in chapter 5.3, the constant C1 is now found to be 3.988x10-5.

As it was established in chapter 5, the red curve stands for the point at which the throttle starts to open. Below that curve the engine works as a diesel engine, in the sense that the increment in speed and/or load is achieved by the injection timing and mass of fuel injected.

The rest of measuring points are shown in Figure 6-2.

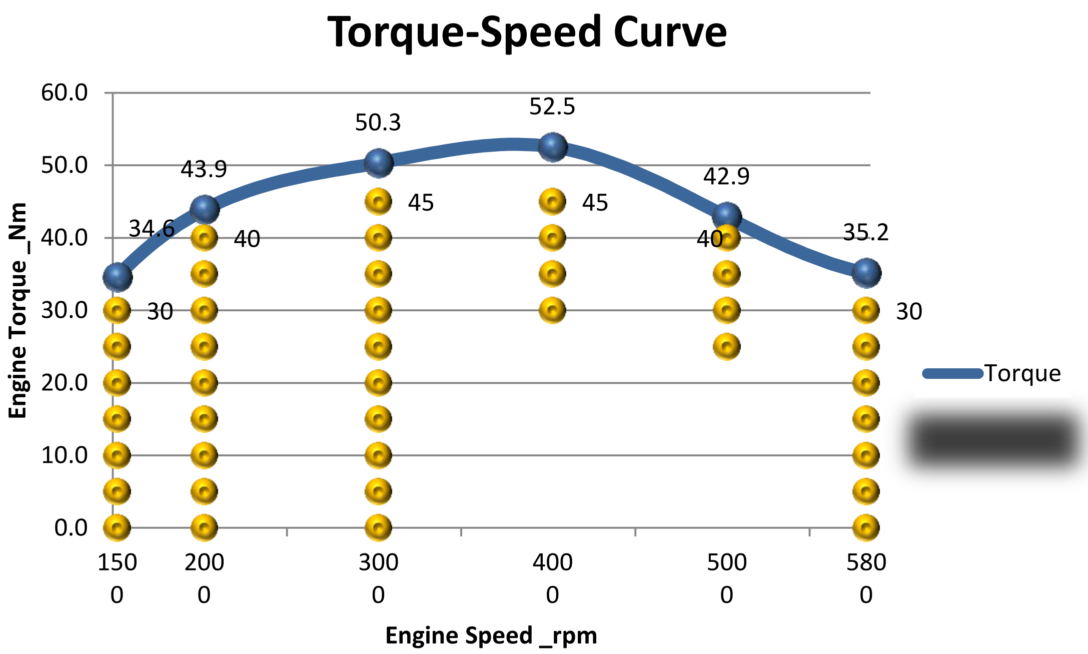


Figure 6-2 Real set of measurement points

Comparing this figure to Figure 5-8, it can be seen that some points have been added. In order to have a complete set of measurements for the peak torque and peak power speeds, measurements were taken to all the loads for 3000 rpm and 5800 rpm. After the measurements were finished, it was found that the peak torque point is at 4000 rpm. Due to temperature problems in the gear box at high speeds, it was decided not to perform the missing measurements at 4000 rpm.

The description of the complete set of points measured is shown in Table 4.

Table 4 Naming of measured points also used in Matlab

	Measurement points												
	load												
speed	0	5	10	15	20	25	30	35	40	45	wot	Throttle op	Propeller
900	1	2	3										45
1500	6	7	8	9	10	11	58				12	13	46
2000	14	15	16	17	18	19	20	21	22		23	24	47
3000	49	50	51	52	53	54	25	26	27	59	28	29	48
4000							30	31	32	60	33	34	63
5000						35	36	37	61		38		
5800	55	56	57	40	41	42	62				43		

A total of 59 points were measured, as points 4, 5, 39 and 44 were decided not to be done. The order of the numbering of the points is not completely in order because it was more flexible in that way during the measurement process. This was useful to face unexpected situations very fast, as it was sometimes needed due to the problems in the temperature of the gearbox, the temperature of the pressure sensor and some others. The first step during the processing of the data in Matlab was to order all the information according to the structure described in chapter number 5.4, and Table 4 is used to identify any point needed.

The calculation of the power is simply done by multiplying the speed and the torque. The curve obtained is shown in Figure 6-3.

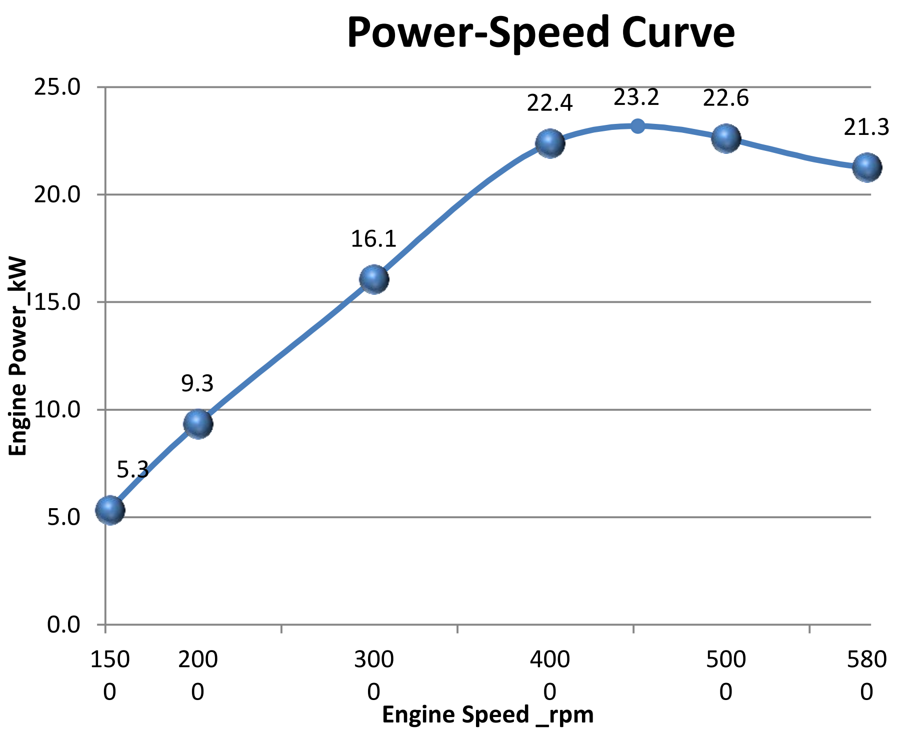


Figure 6-3 Power-Speed curve

As it has been seen that all the torque values where higher than expected, it is easy to think that the power curve will hold higher values also. Of course, the max power

@5800 rpm is higher than the reported in the technical specifications; from 18 kW to 21.3 kW. But also it is seen that the maximum power is somewhere between 4000 rpm and 5000 rpm as the spline interpolation is showing.

6.2 Injector parameters

One of the main objectives of the present work is to measure the electrical parameters of the injector in order to find out how to control it. It wasn't needed to measure both, current and voltage thru all the points in the engine's map. The objective was to identify the changes in the current and in the voltage when the engine is increasing speed at a constant load, and when the engine is developing more torque at a constant speed.

Figure 6-4 and Figure 6-5 show the measurements of the driving voltage and current on the injector when the engine is developing 5 Nm and 46 Nm at 3000 rpm respectively. In them it is possible to see that the driving voltage is constant at 40 V during the time desired for the injection event. The current increases from cero to a maximum value. As the injector is a coil, it takes time for it to fully charge of energy. When that happens, the current shows a constant value as well. This event happens at around 1.5 mSec.

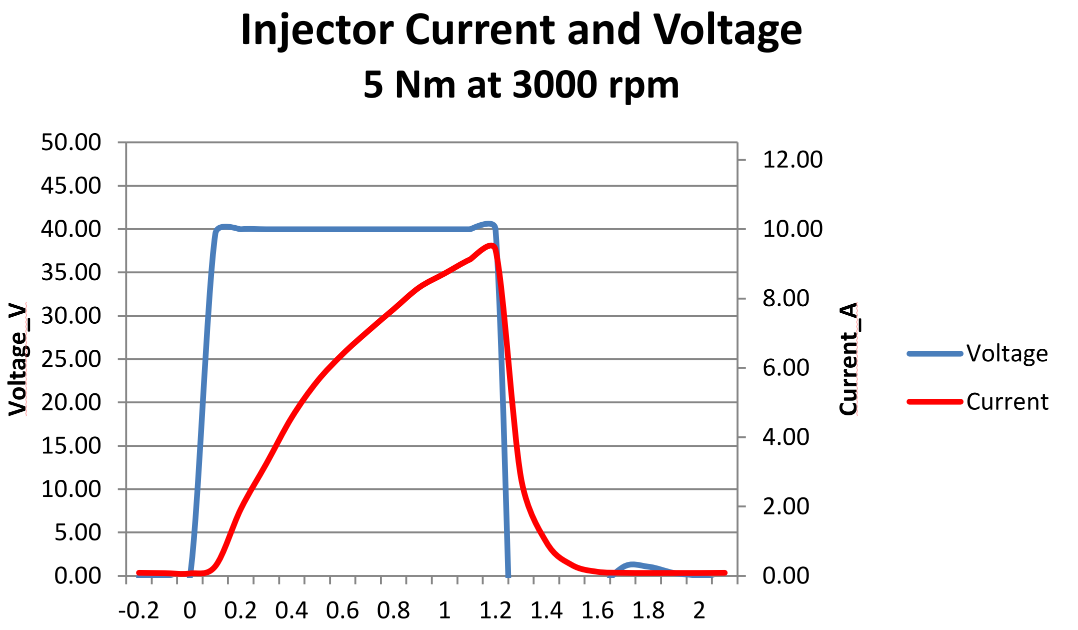


Figure 6-4 Electrical measurements at 5 Nm and 3000 rpm

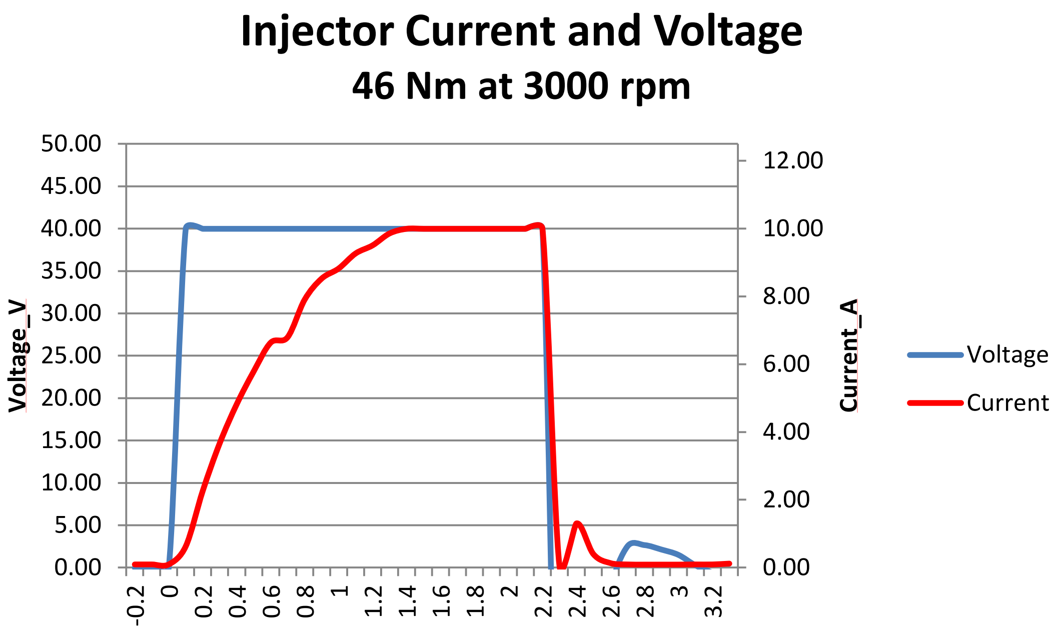
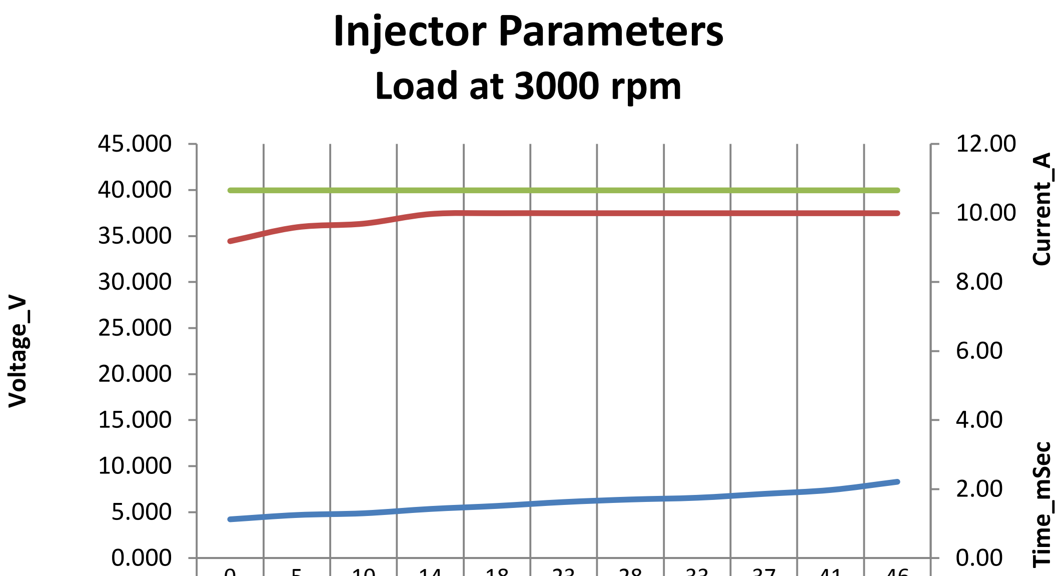


Figure 6-5 Electrical measurements at 46Nm and 3000 rpm

The injector is basically an electro magnet driving a piston which, when pushing the fluid, has to overcome the force of a spring which maintain the needle closed (see chapter 4). This means that the needle will open at the same pressure. Doesn't matter how much more voltage is fed in to the coil, the pressure of the fluid going through the needle is always going to be the same. For this reason there were suspicion that the voltage fed in to the injector had always the same magnitude, and that the current would follow the function of the coil when charging energy. Being this right, it would then mean that the only parameter which has to change in order to operate the injector is the time during which the voltage is being fed.

Figure 6-6 shows the measurements of the injector parameters along different loads at a constant speed of 3000 rpm.



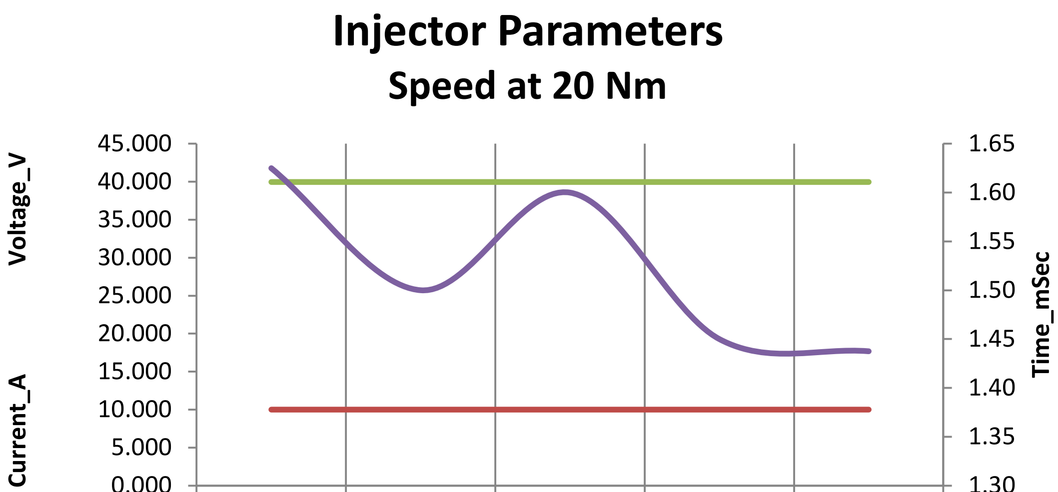
Load_Nm	0	5	10	14	18	23	28	33	37	41	46
Voltage_V	39.98	39.98	39.98	39.98	39.98	39.98	39.98	39.98	39.98	39.98	39.98
Time_mSec	1.13	1.25	1.30	1.43	1.51	1.63	1.70	1.75	1.86	1.98	2.21
Current_A	9.185	9.589	9.694	9.971	9.995	9.995	9.995	9.995	9.995	9.995	9.995

Load_Nm

Figure 6-6 Injector electrical measurements at 3000 rpm

The plot clearly shows that the voltage is constant through all the load increment at a certain speed. The numbers in the table at the bottom of the plot confirm that there is no difference among them. The red curve, which represent the amount of current flowing in to the injector's coil, reveals the time that it takes for the coil to complete charge energy. The blue curve, representing the time of injection, shows the steady increment of mass of fuel injected.

Figure 6-7 shows the measurement of injector parameters performed at constant load.



	1500	2000	3000	4000	5000
Voltage_V	39.980	39.980	39.980	39.980	39.980
Current_A	9.995	9.985	9.995	9.995	9.987
Time_mSec	1.63	1.50	1.60	1.45	1.44

Figure 6-7 Injector electrical measurements at 20 Nm

This plot reveals again that the voltage has the same magnitude. Now the injection time is erratic, even it tends to be reduced as the speed increases. It looks strange since the higher the power in the engine, the higher the demand for energy from the fuel. But we have to keep in mind that the timings for the injector and for the spark plug are changing. As for the current, the value at 1.5 mSec seems not to make completely sense with the others. The other values completely make sense, even if they are crossed with the values of Figure 6-6. From there, a conclusion can be drawn that the injector's coil charges completely at 1.45 msec. Also, from Figure 6-5 it can be said that the span of injection time is 1.1 mSec.

6.3 Injection Timing

Going further in the injector analysis, attention now has to be put in the injection and spark timings, along with the injection time. Figure 6-8 shows the progression of the injection and ignition along different loads at 3000 rpm. This figure expands the contents of Figure 6-6, which shows the same information but for the injector current and voltage. Figure 6-9 shows the time between the injection event and the ignition event in mSec, which reflects the mixing time that the fuel has with the air in the combustion chamber. Also, Figure 6-10 and Figure 6-11 show the time plots of injection in mSec and CAD.

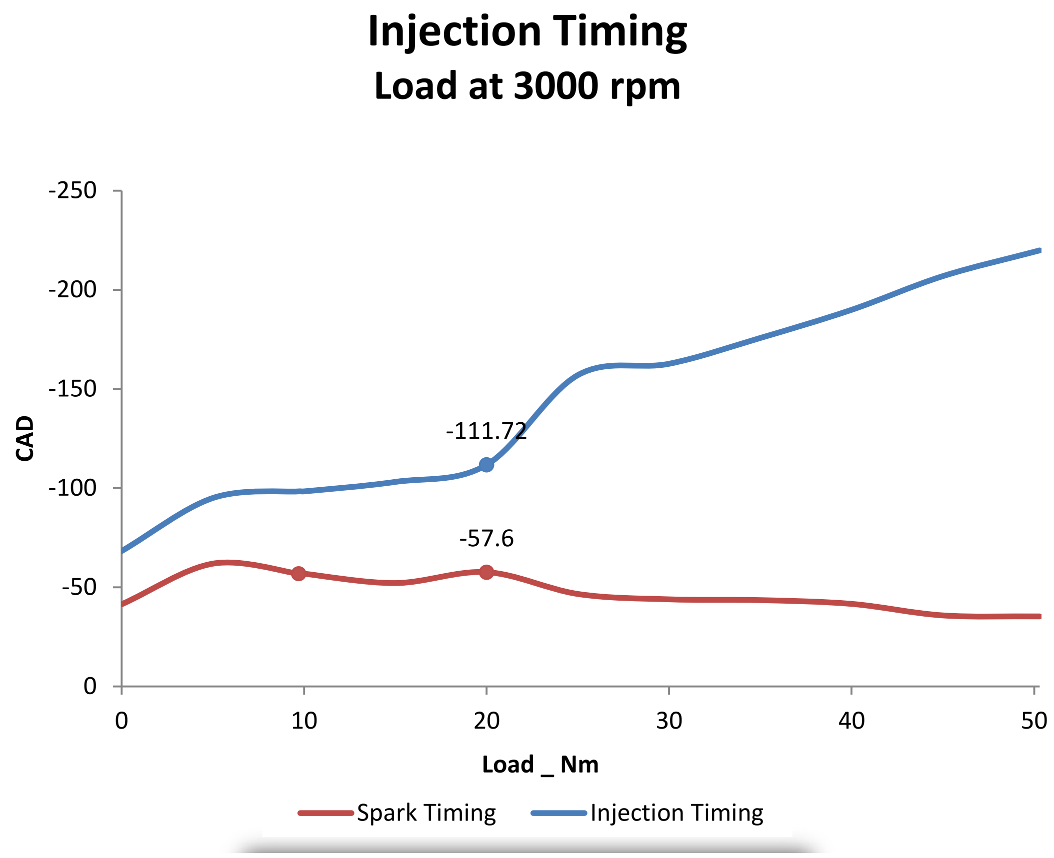


Figure 6-8 Ignition and Injection timing at 3000 rpm

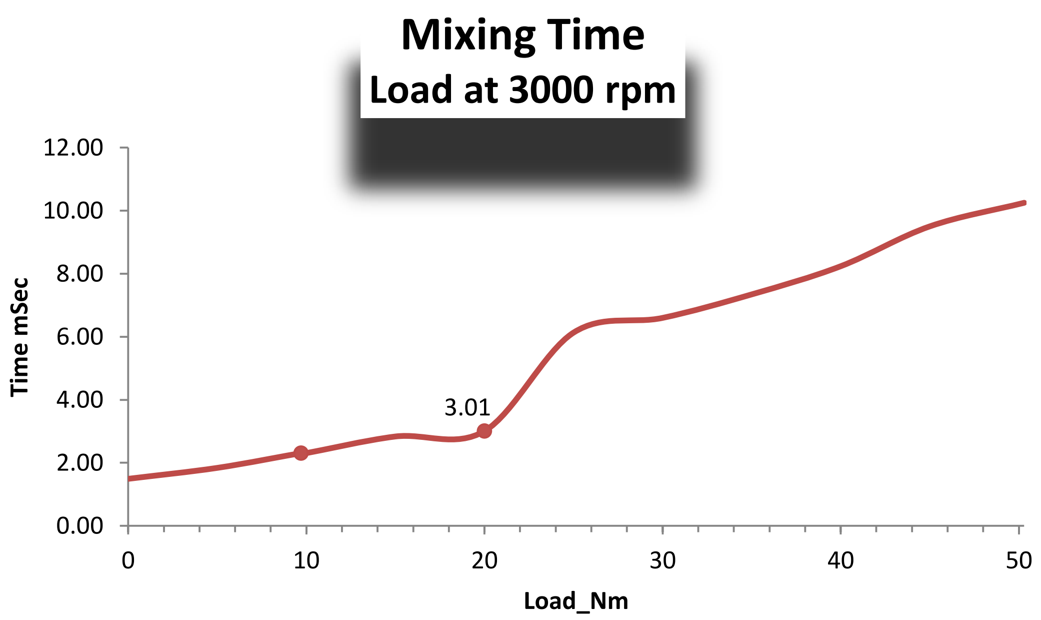


Figure 6-9 Time between injection and ignition at a constant speed of 3000 rpm

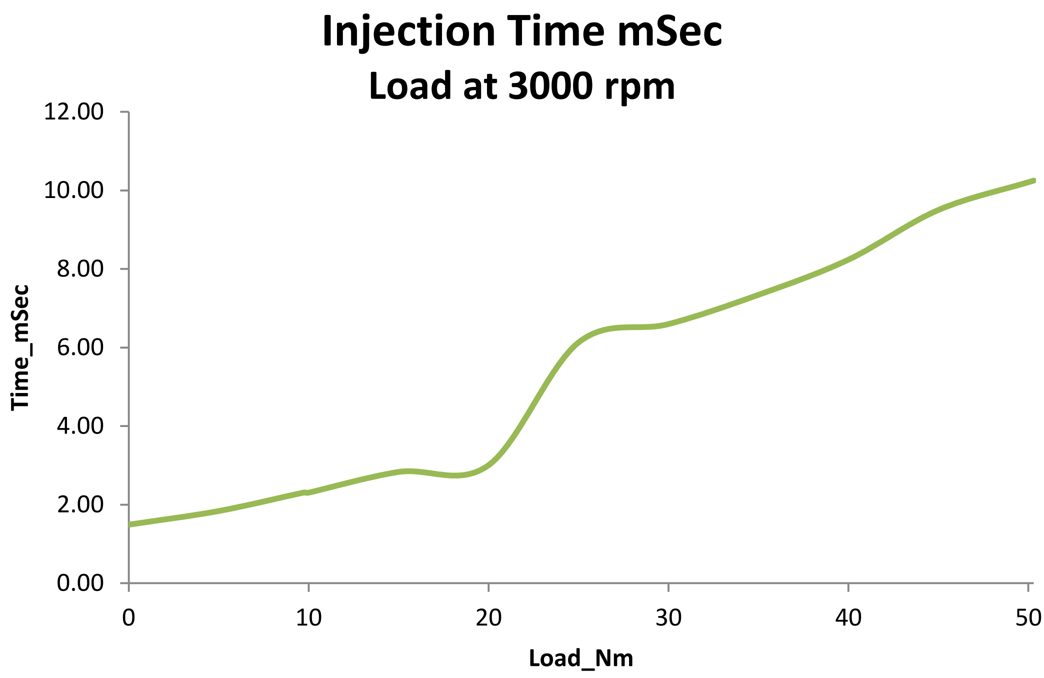


Figure 6-10 Injection time in mSec at 3000 rpm

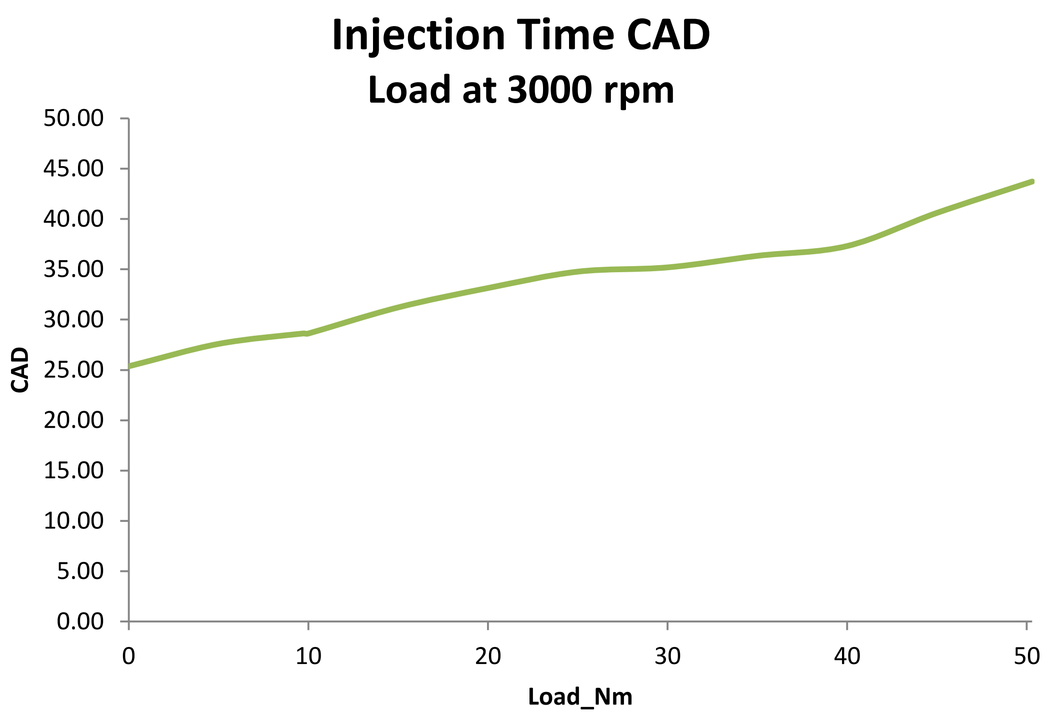


Figure 6-11 Injection time in CAD at 3000 rpm

The red curve in Figure 6-8 reflects the spark plug timing, which moves between -20 and -60 CAD, as it is possible to see in the axis to the left. The blue curve reflects the injector timing. It is clearly visible that the injector time increases while the injection timing advances more and more. The points on the curves show the moment at which the throttle starts opening.

In Figure 6-7 it was shown that the injection time reduce while the speed increase at a constant load of 20 Nm. Figure 6-15 represents the injection time for a constant load of 30 Nm and confirms the observation. Also Figure 6-12 reveals that at low speed the injector changes its timing, but at high 4000 rpm it stays in the same value.

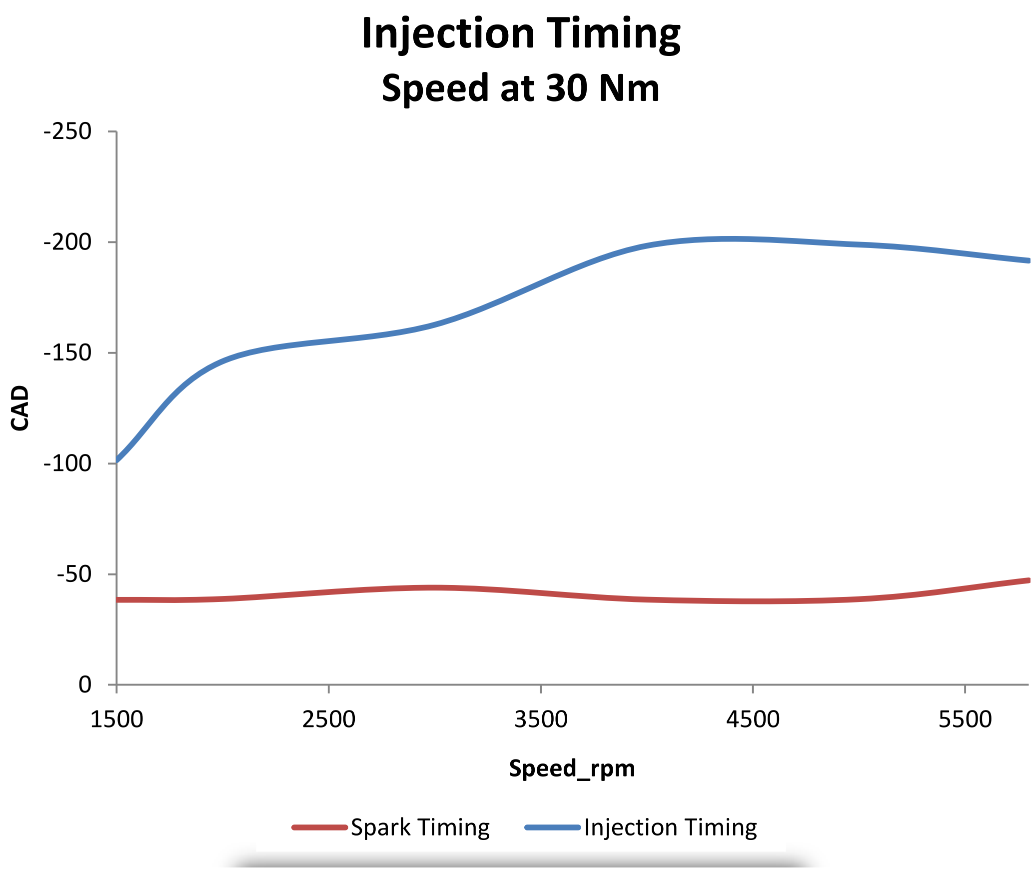


Figure 6-12 Ignition and Injection timing at 30 Nm

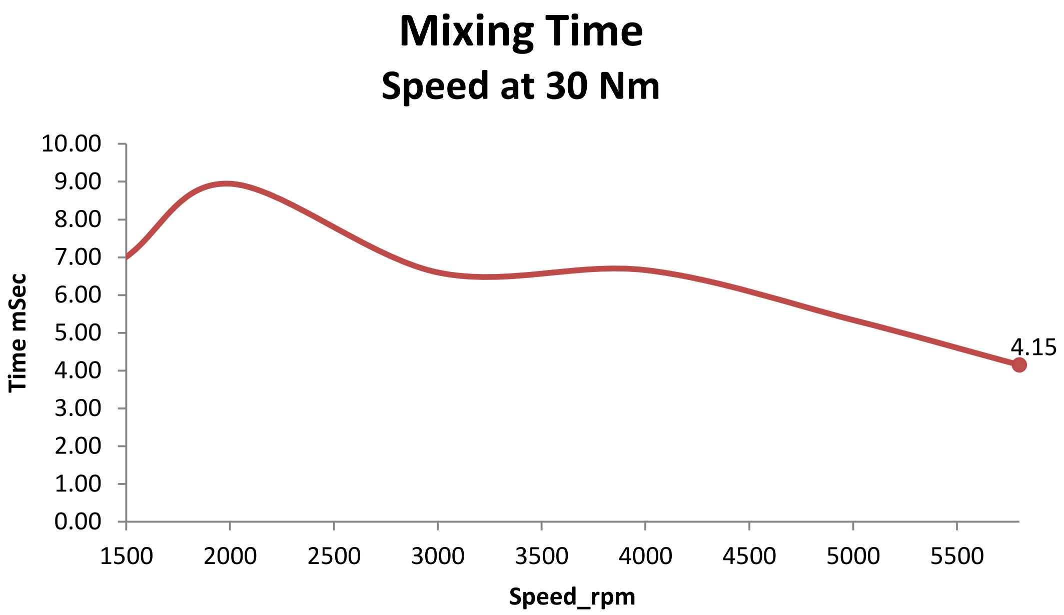


Figure 6-13 Time between injection and ignition at a constant load of 30 Nm

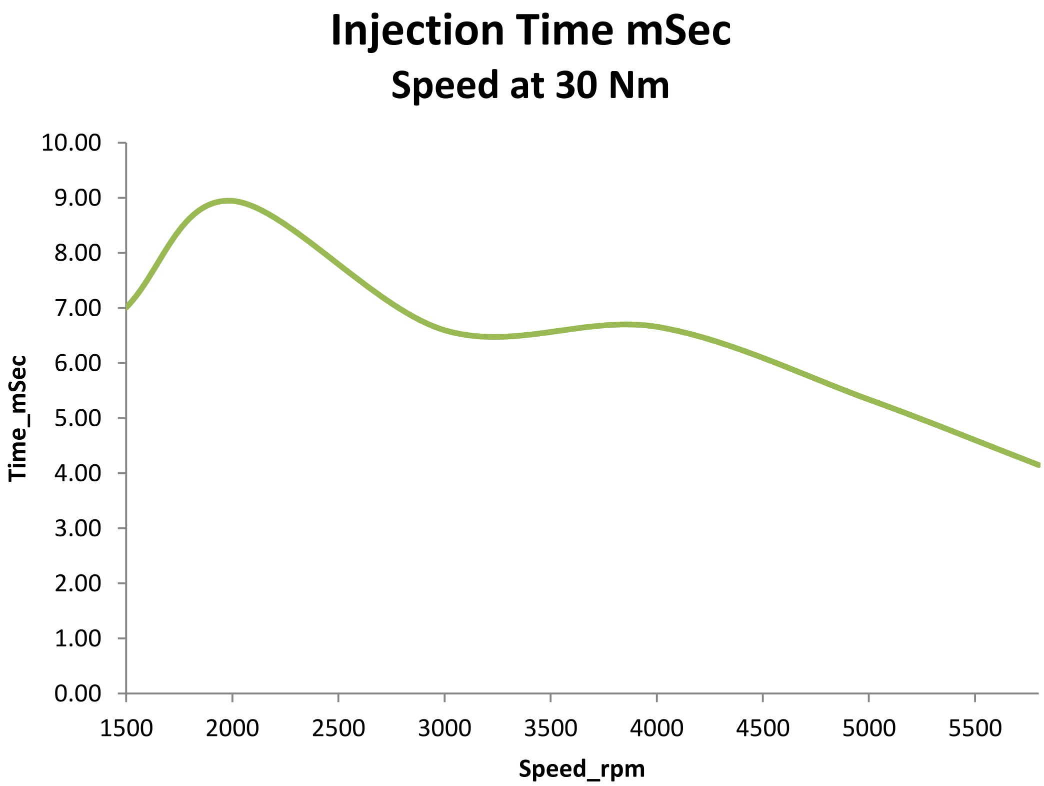


Figure 6-14 Injection time in mSec at 30 Nm

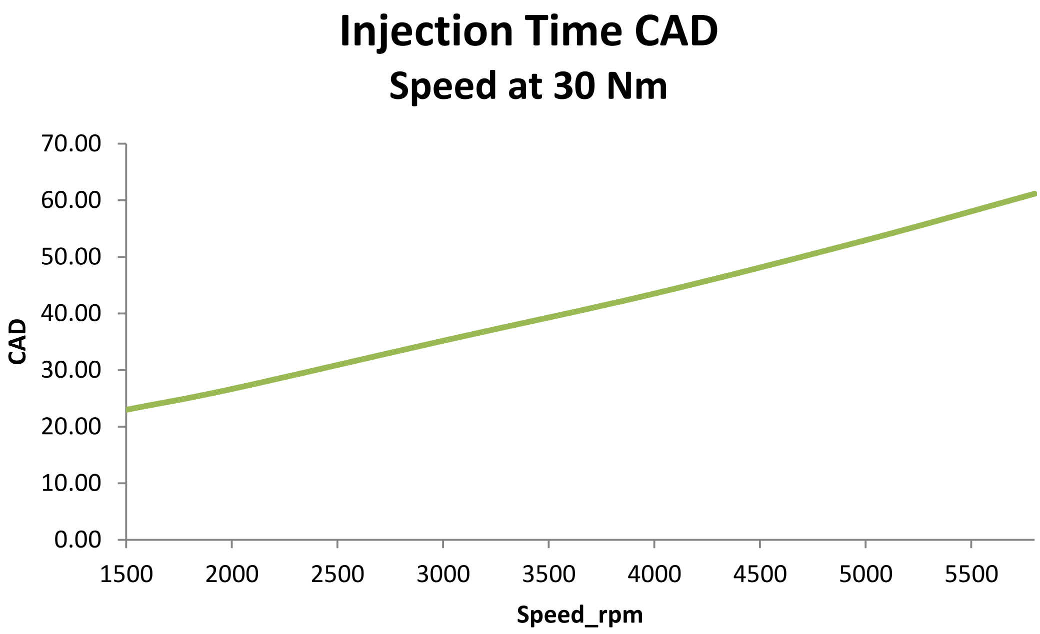


Figure 6-15 Injection time in CAD at 30 Nm

Figure 6-16 shows the injector timing data for the propeller curve. In this case the engine is increasing both, speed and load, and as a consequence the plot seems to be more complex.

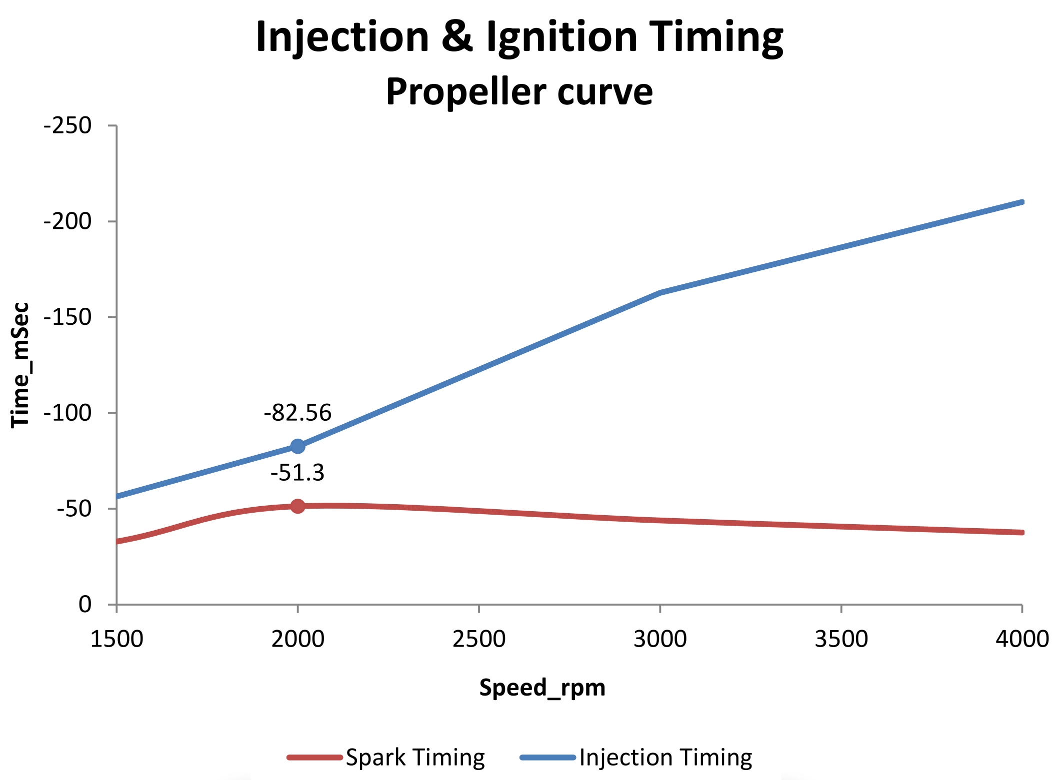


Figure 6-16 Ignition and Injection timing along propeller curve

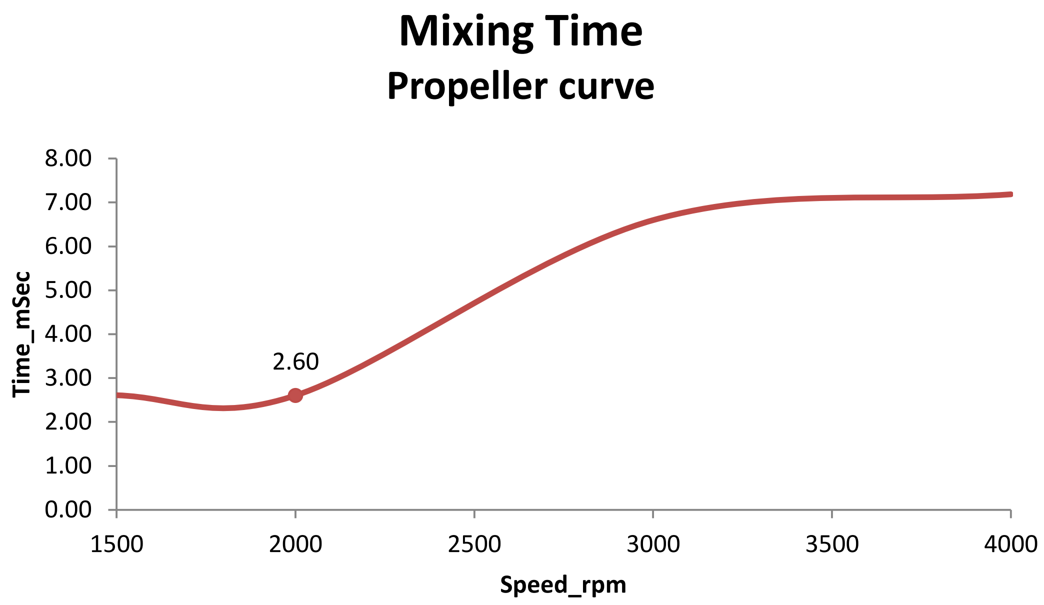


Figure 6-17 Time between injection and ignition along the propeller curve

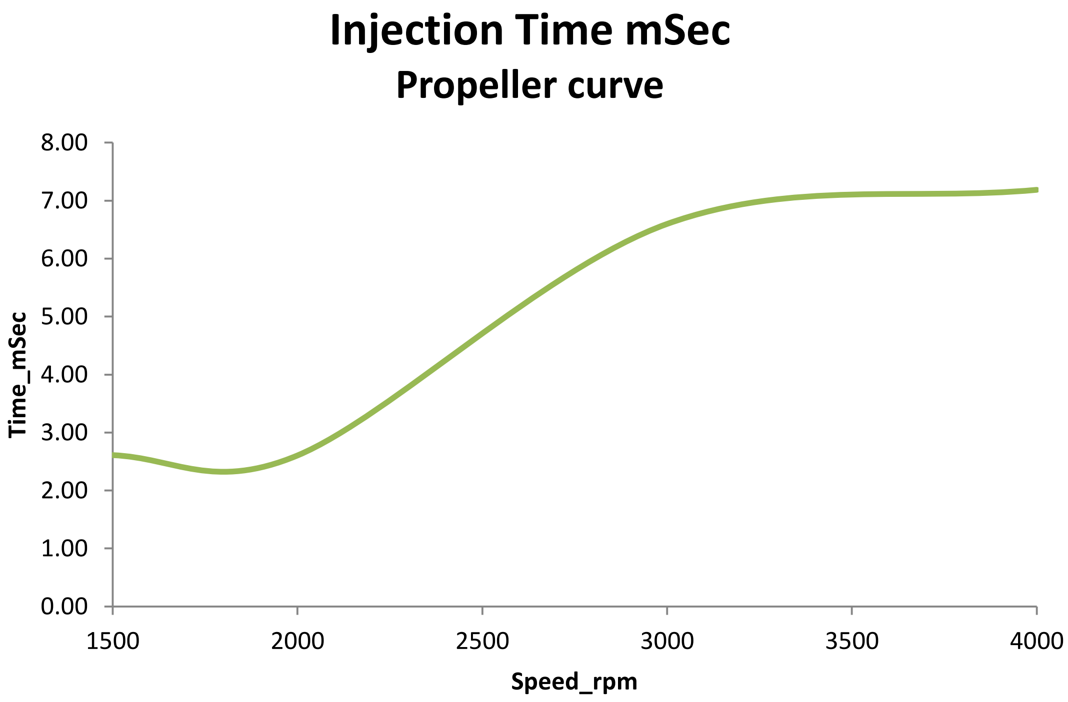


Figure 6-18 Injection time in mSec along propeller curve

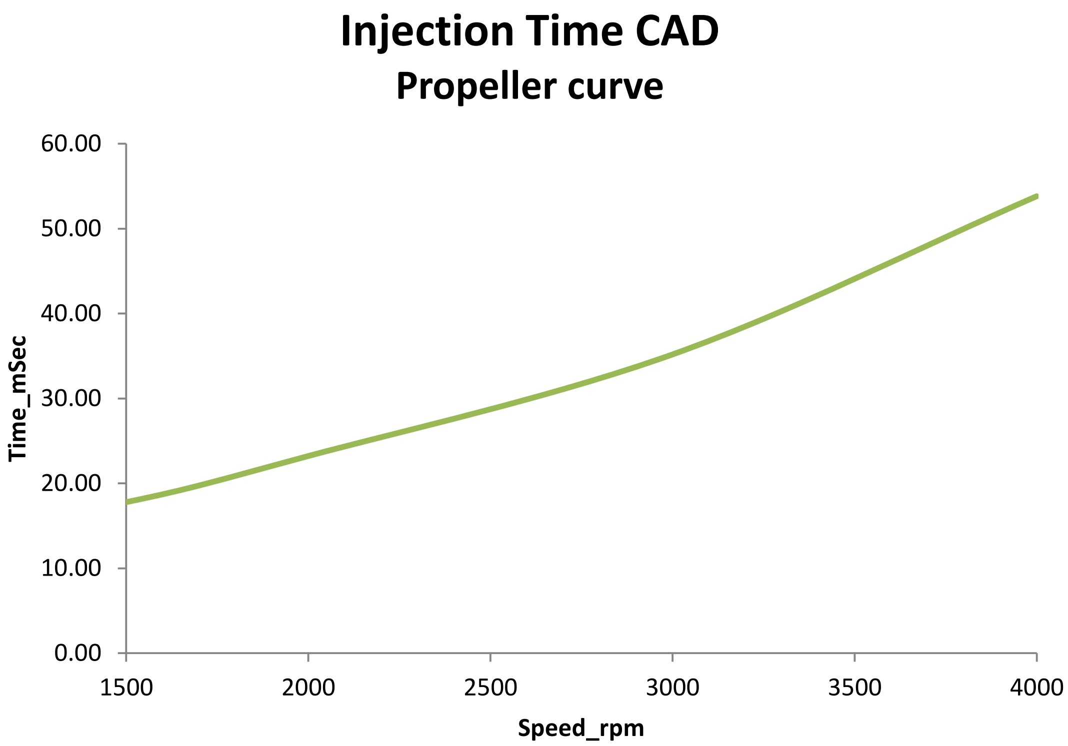


Figure 6-19 Injection time in CAD along propeller curve

One of the most important features that this engine has is the flexibility (being direct injected) of changing the time of injection in a very wide area, and making the engine to run over different injection modes. Figure 6-20 shows a contour plot over the bmep curve showing the variations in injection timing. It can be observed that the latest injections are in the low load - low speed areas, and the earliest ones are mainly where the highest energy is being produced. The curve at which the throttle starts opening is shown in bright green. Also the zone for which no measurements were taken, or shadow, is shown in gray line.

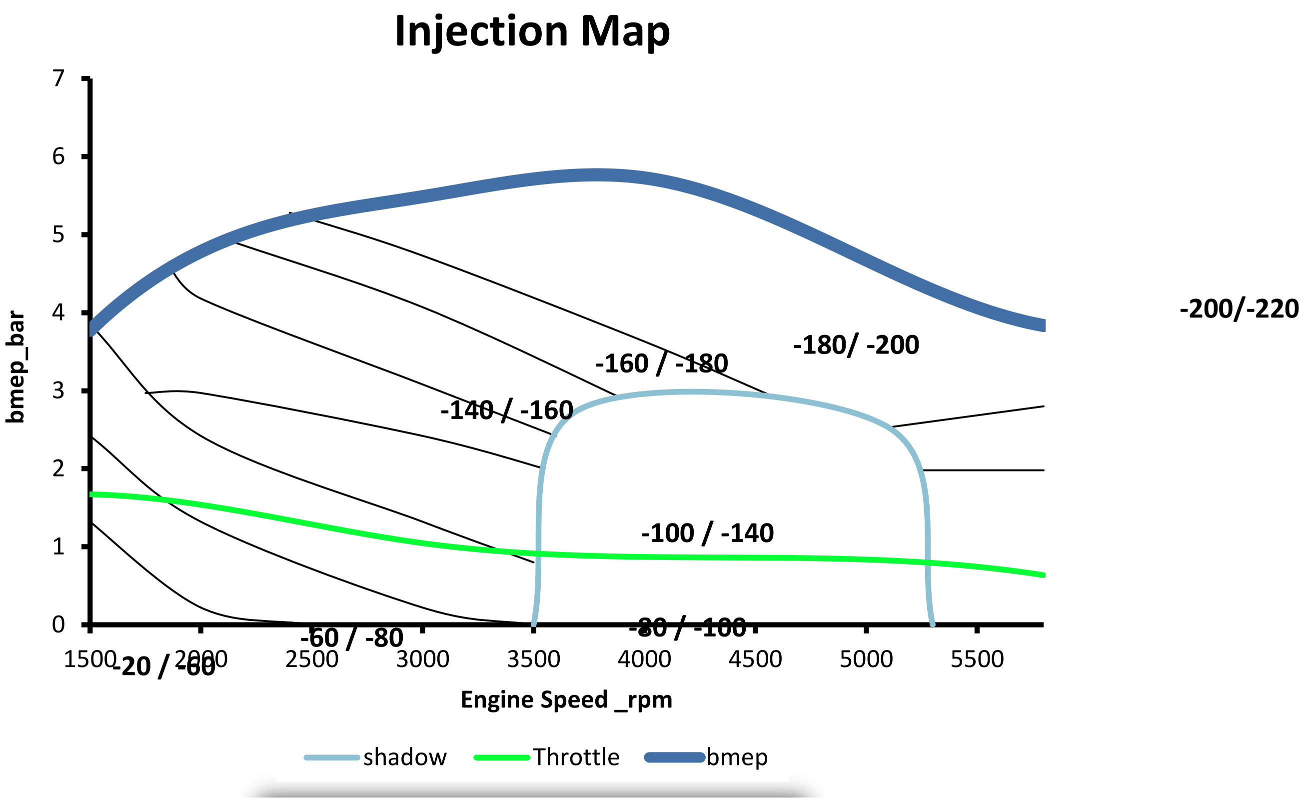


Figure 6-20 Injection timing map. The values in each area of the contour plot show the range in CAD the injector is injecting at when the engine is operating inside such area.

Figure 6-21 shows a contour plot of the ignition timing against a bmep curve. The most advanced times are in the lower center of the plot. It shows no significant change across the speeds which can explain the increase of speed with a reduction of injected mass when in constant load.

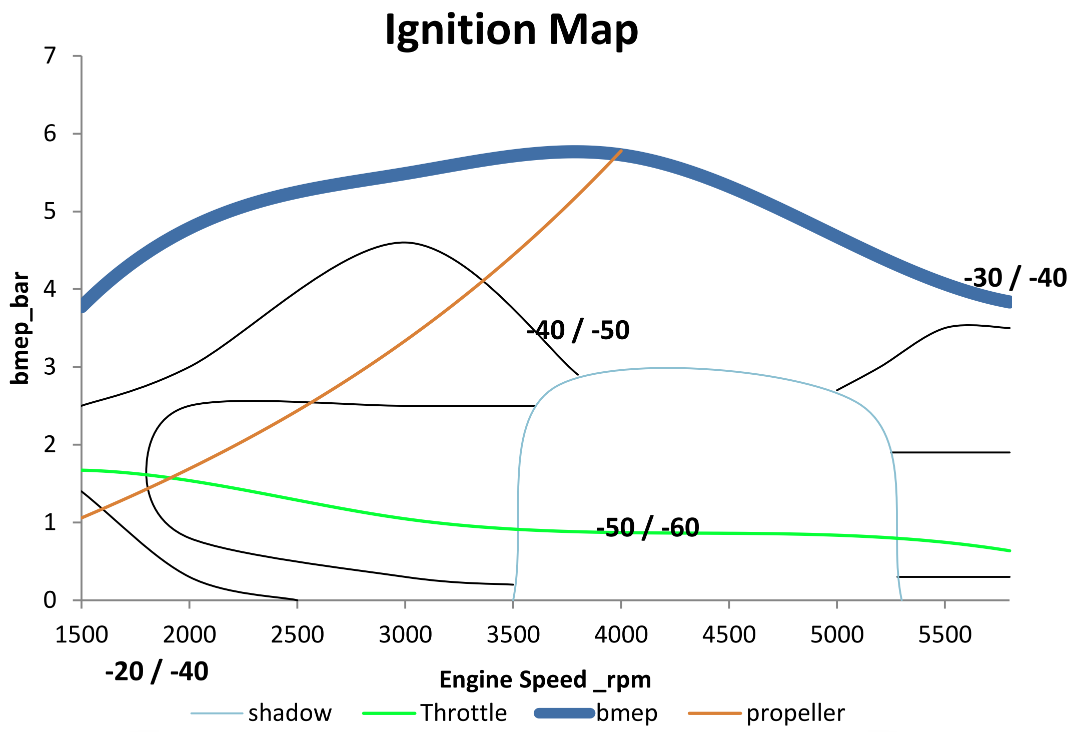


Figure 6-21 Ignition timing map. The values in each area of the contour plot show the range in CAD the spark plug is igniting at when the engine is operating inside such area.

The throttle angle may be also interesting when trying to figure out the injection mode of the engine. The contour plot for the throttle angle is shown in Figure 6-22.

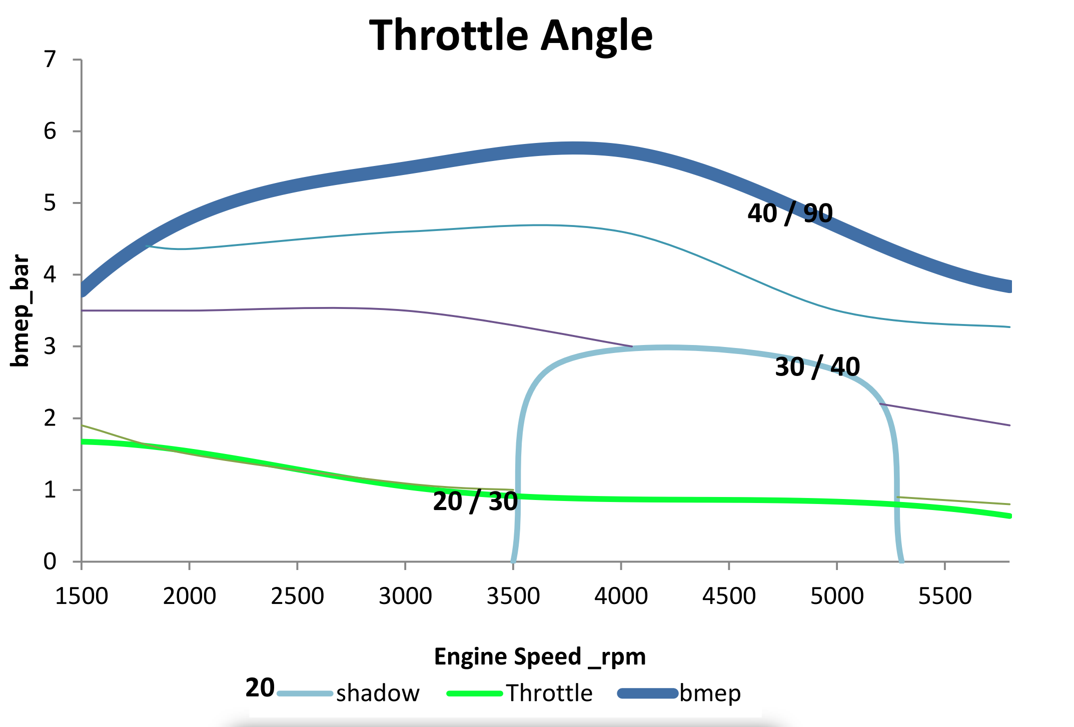


Figure 6-22 Throttle angle map. The values in each area of the contour plot show the range in degrees the throttle is positioned at when the engine is operating inside such area.

6.4 Specific Fuel Consumption and Emissions

Figure 6-23 shows a superimposed plot of the bmep, propeller and throttle curves over a colored plot of the specific fuel consumption of each measured point. The colour of the background in each cell indicates the value of the sfc, being red the higher and green the lowest.

What is interesting to see in this figure is that the optimum fuel consumption is not in the low speed high load region as in automotive engines is. In this case the optimum point is near the center of the bmep curve in the high loads. When compared to the propeller curve, it can be seen that this curve is near to that area, which seems logical as this engine is projected to be running along the propeller curve, and the optimum point being at the highest speed along this curve. SFC values in the plot are given in g/kWh.

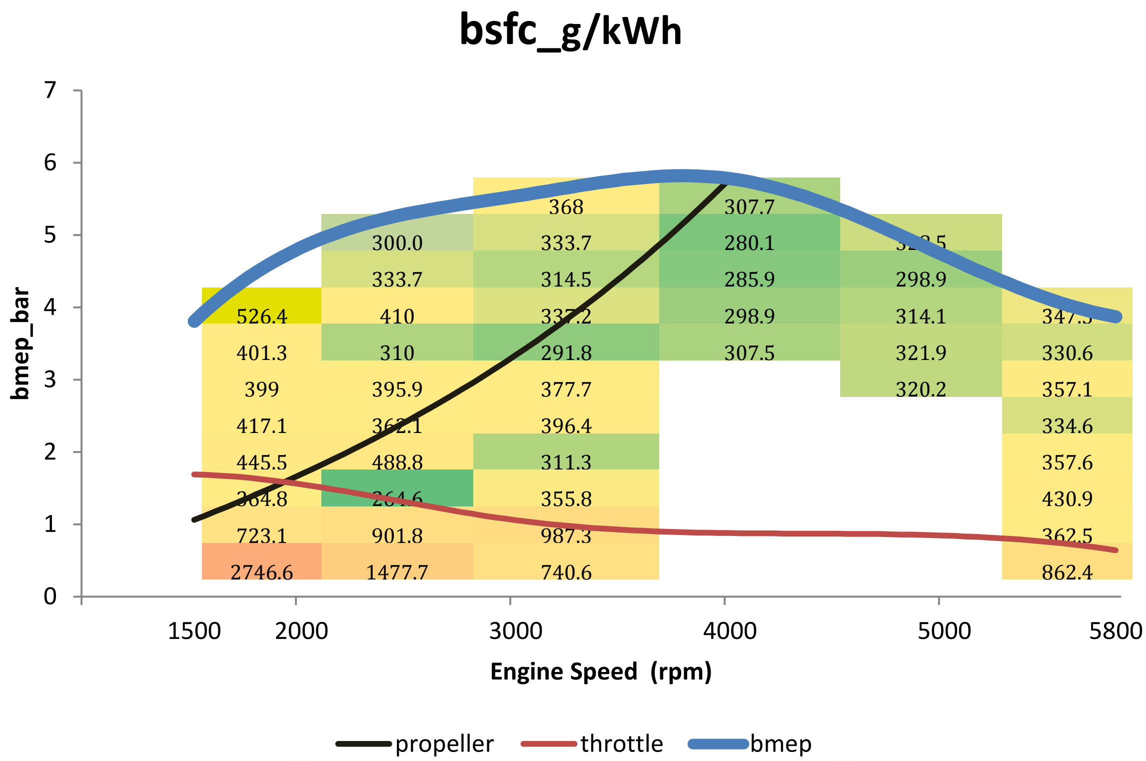


Figure 6-23 bsfc map

The superimposed plots for emissions are shown in the following figures.

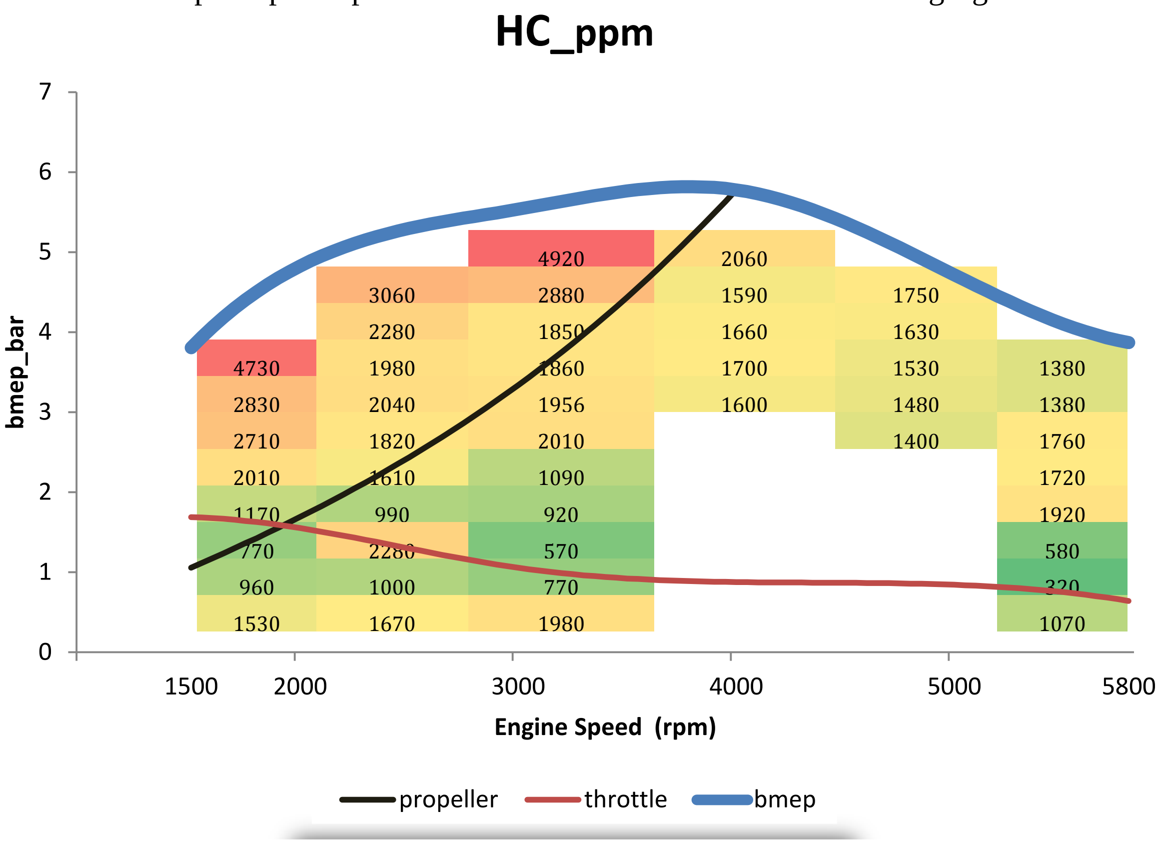


Figure 6-24 HC map

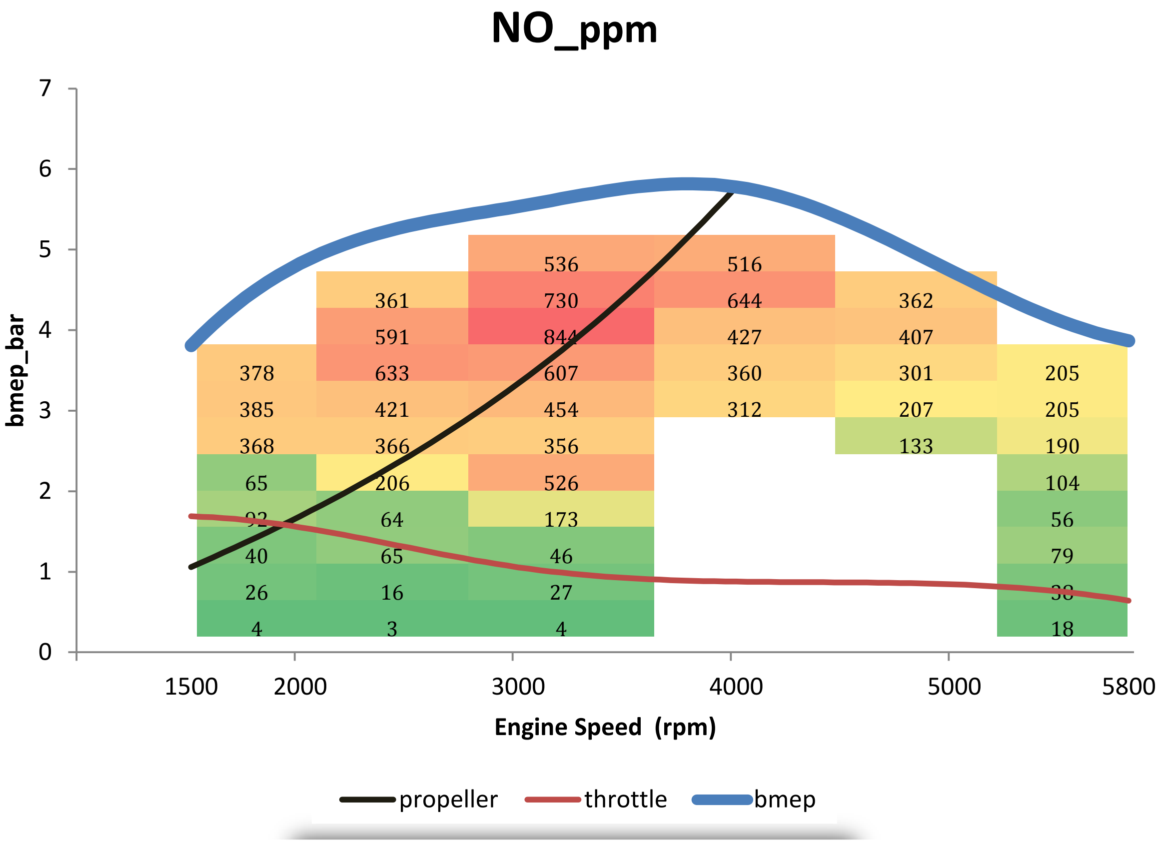


Figure 6-25 NO map

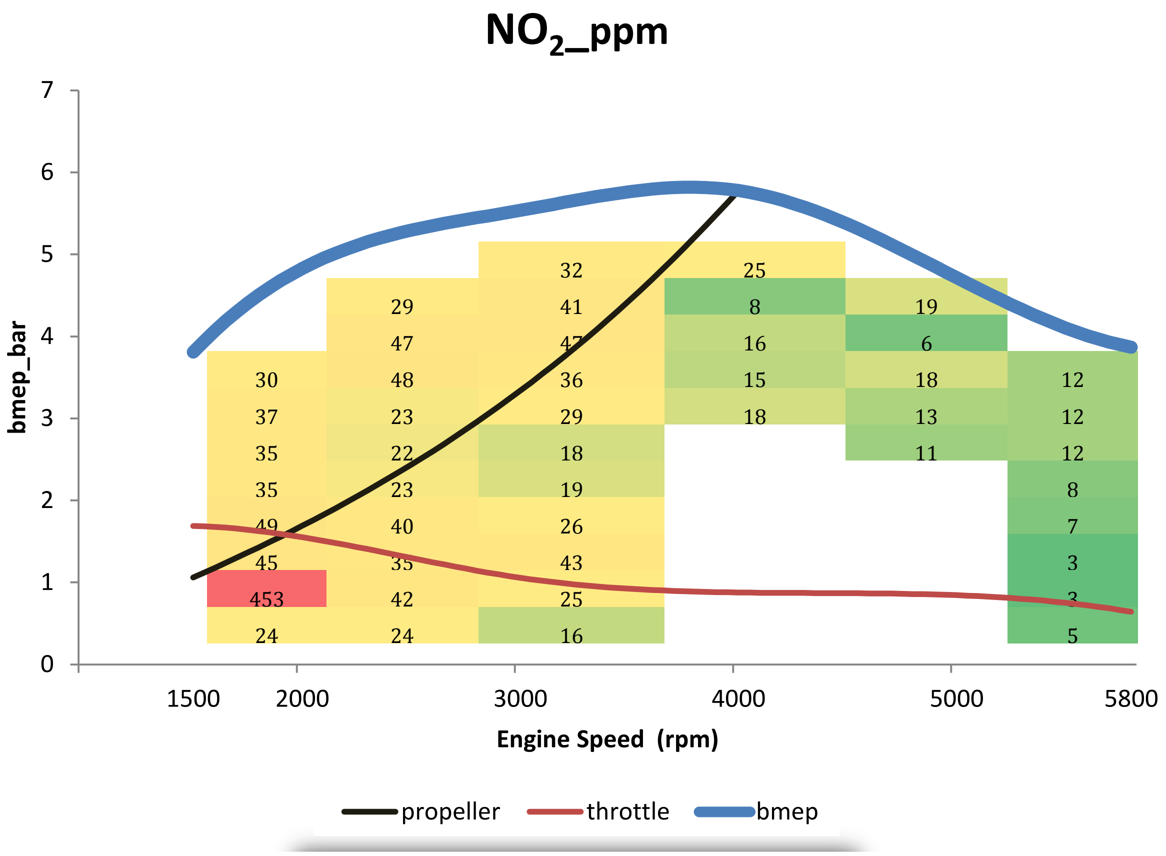


Figure 6-26 NO2 map

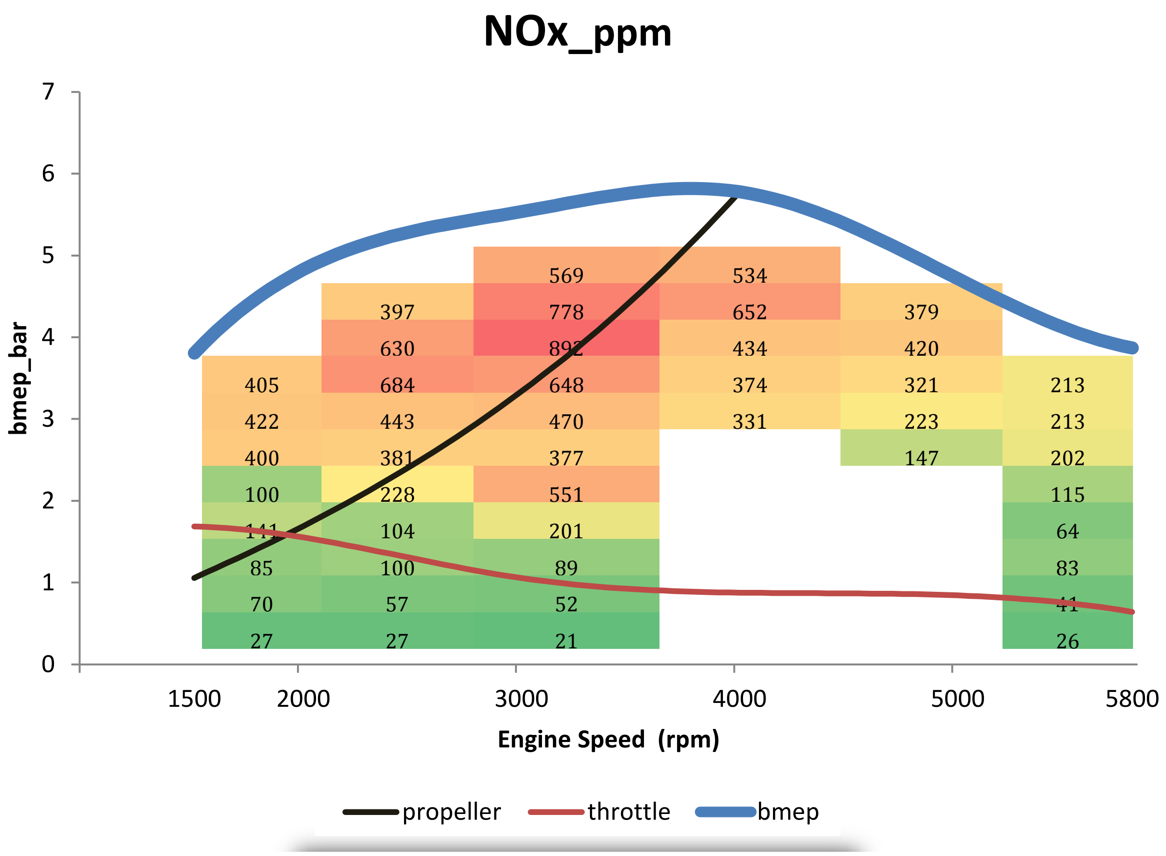


Figure 6-27 NOx map

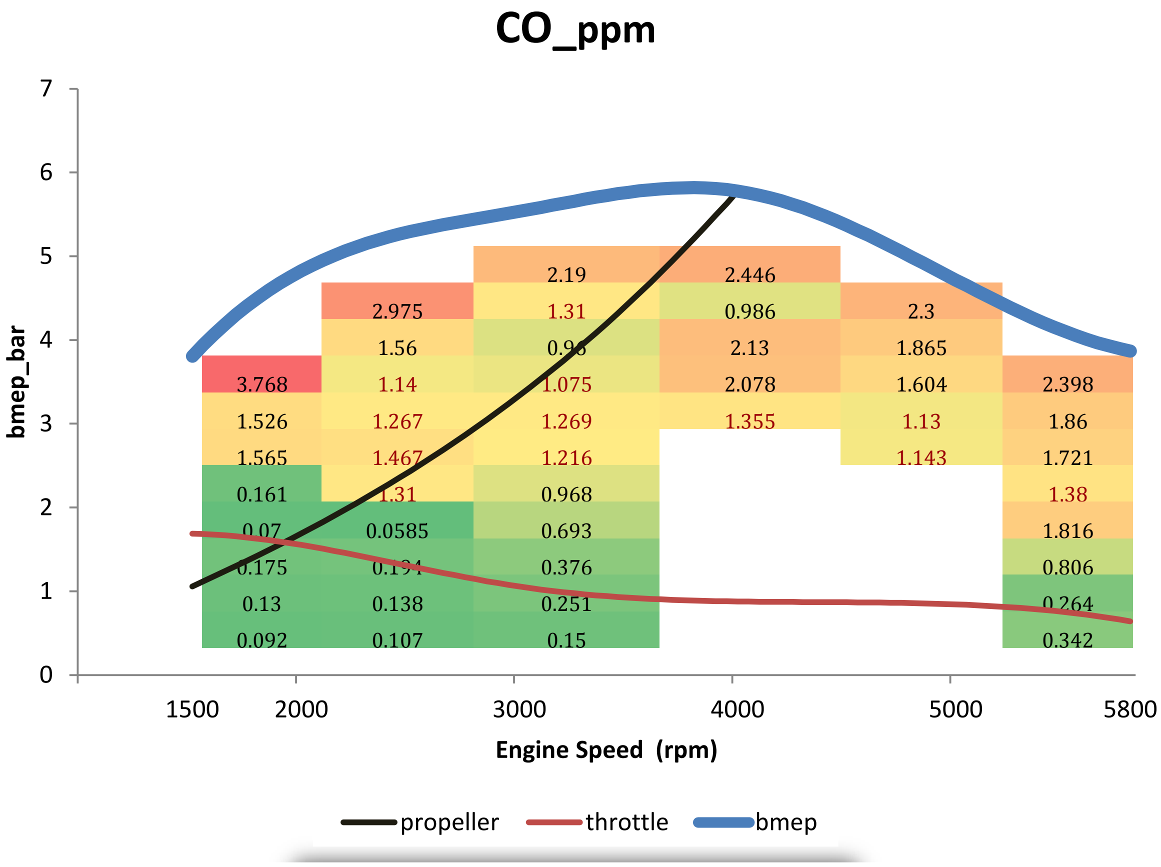


Figure 6-28 CO map

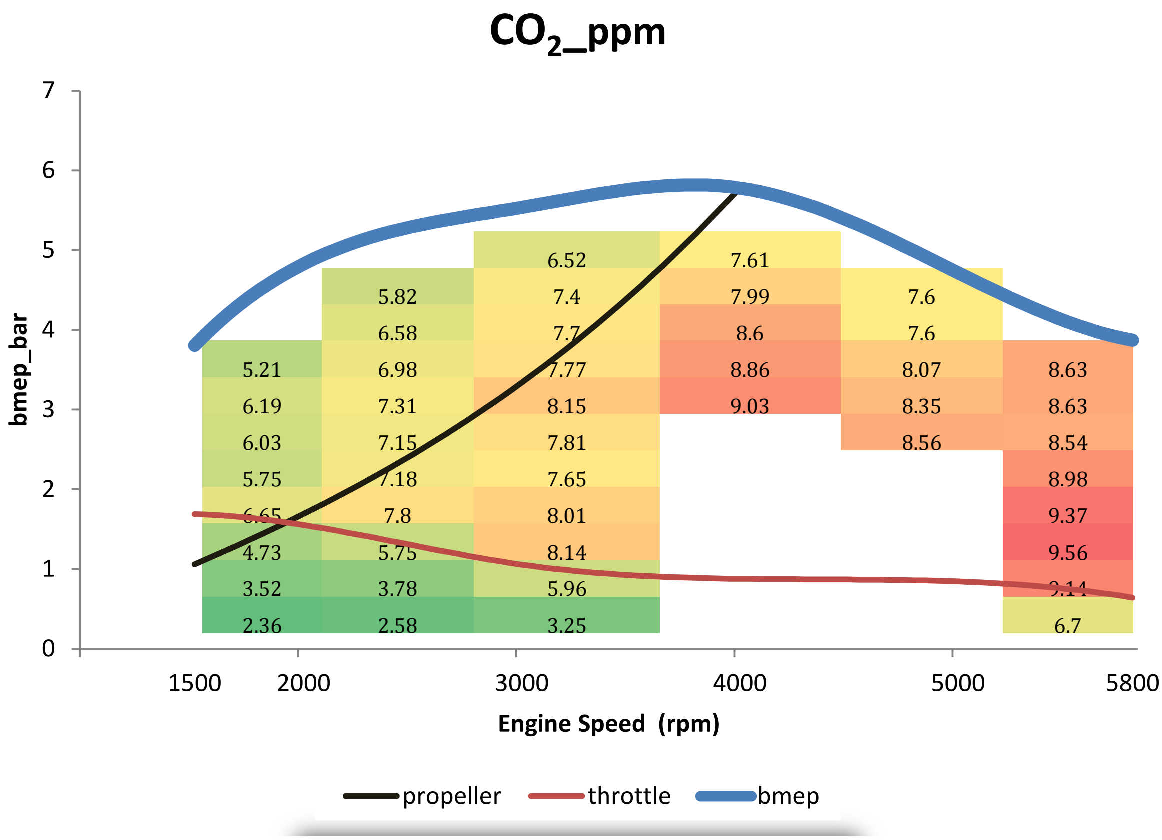


Figure 6-29 CO2 map

6.5 Spray Characterization

As part of the project a spray characterization of the Evinrude injector was done. Some results were obtained, as more time is needed to do a complete set of spray characterization, being subject of a different project.

Figure 6-30 shows a measurement in millimetres of the injector's spray at .5 bar back pressure and 2.3 mSec of injection time.

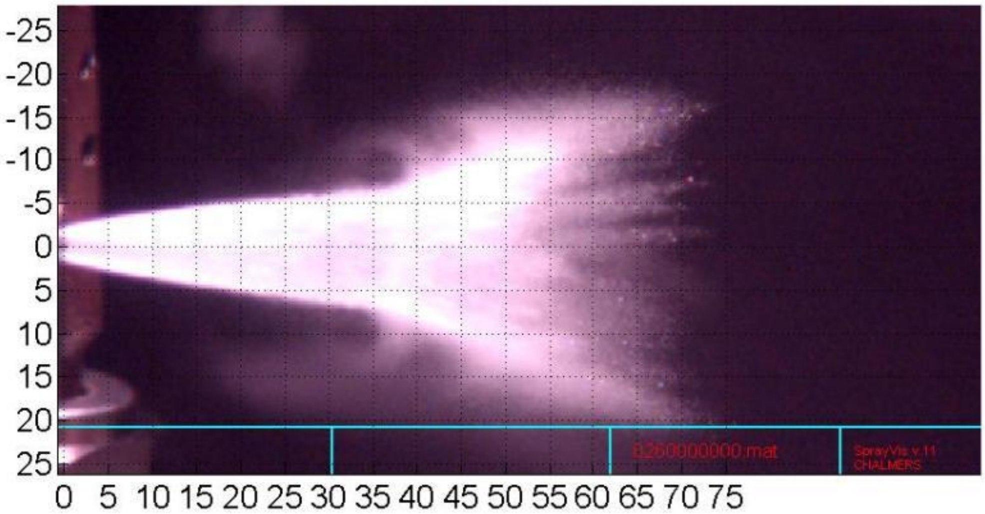


Figure 6-30 Cone diameter and spray penetration

From this figure it can be seen that the penetration of the injection reaches up to 60 mm. The diameter of the hollow cone is around 12 mm.

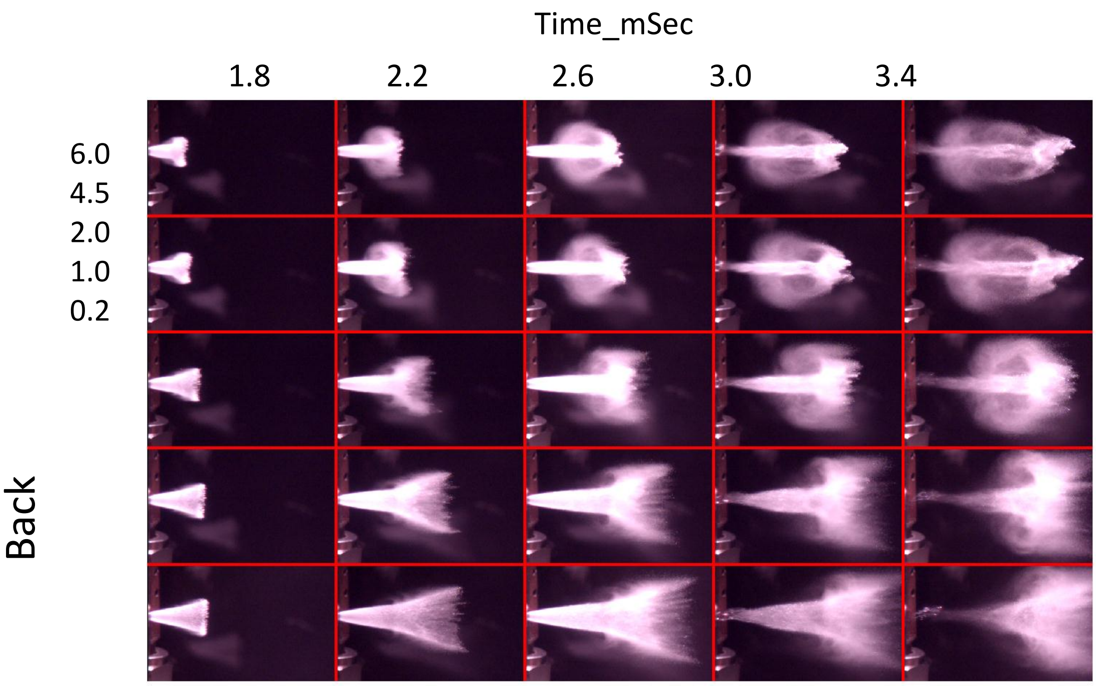


Figure 6-31 Comparison of sprays at different back pressures

In this image is possible to see how the pressure affects the shape of the spray. In chapter 7.2 it was presented the way that the mass of injected fuel changes with different back pressures along the propeller's curve.

Table 5 and Table 6 show in detail the response of different events of the injector along different back pressures. There is possible to see in detail how the injection time (considering the time during which fuel is going through the needle, instead of the time while the injector is being feed with voltage which is the one used in previous chapters) changes with the back pressure, and also the lag of the injector between the time when it is feed with voltage and when it shows a response.

Table 5 Injector events at different back pressures

Back Pressures	Needle Open	Start of Injection	End of Injection	Injection Time
6	1.490	1.520	3.200	1.71
	1.490	1.520	3.180	1.69
	1.480	1.520	3.180	1.70
	1.480	1.510	3.190	1.71
	1.490	1.520	3.200	1.71
	1.490	1.520	3.190	1.70
	1.487	1.518	3.190	1.70
4.5	1.480	1.510	3.160	1.68
	1.470	1.510	3.190	1.72
	1.480	1.500	3.180	1.70
	1.470	1.500	3.190	1.72
	1.470	1.510	3.190	1.72
	1.480	1.510	3.190	1.71
	1.475	1.507	3.183	1.71
3	1.480	1.500	3.180	1.70
	1.470	1.500	3.180	1.71
	1.480	1.510	3.200	1.72
	1.480	1.510	3.180	1.70
	1.480	1.510	3.180	1.70
	1.480	1.510	3.190	1.71
	1.478	1.507	3.185	1.71
2.5	1.470	1.500	3.190	1.72
	1.470	1.510	3.210	1.74
	1.470	1.500	3.210	1.74
	1.480	1.510	3.200	1.72
	1.470	1.500	3.190	1.72
	1.480	1.510	3.190	1.71
	1.473	1.505	3.198	1.73
2	1.460	1.500	3.220	1.76
	1.470	1.510	3.190	1.72
	1.472	1.506	3.198	1.73
1.5	1.460	1.500	3.220	1.76
	1.470	1.500	3.210	1.74

	1.468	1.503	3.206	1.74

Table 6 Injector events at different back pressures (continued)

Back Pressures	Needle Open	Start of Injection	End of Injection	Injection Time
1	1.490	1.490	3.210	1.72
	1.490	1.490	3.250	1.76
	1.475	1.498	3.216	1.74
0.8	1.460	1.490	3.250	1.79
	1.470	1.490	3.200	1.73
	1.475	1.494	3.222	1.75
0.5	1.460	1.480	3.230	1.77
	1.460	1.500	3.240	1.78
	1.467	1.492	3.226	1.76
0.3	1.460	1.480	3.240	1.78
	1.460	1.480	3.240	1.78
	1.464	1.488	3.233	1.77
0.2	1.450	1.480	3.270	1.82
	1.450	1.490	3.260	1.81
	1.458	1.485	3.245	1.79
0.1	1.450	1.480	3.240	1.79
	1.460	1.490	3.240	1.78
	1.455	1.485	3.248	1.79

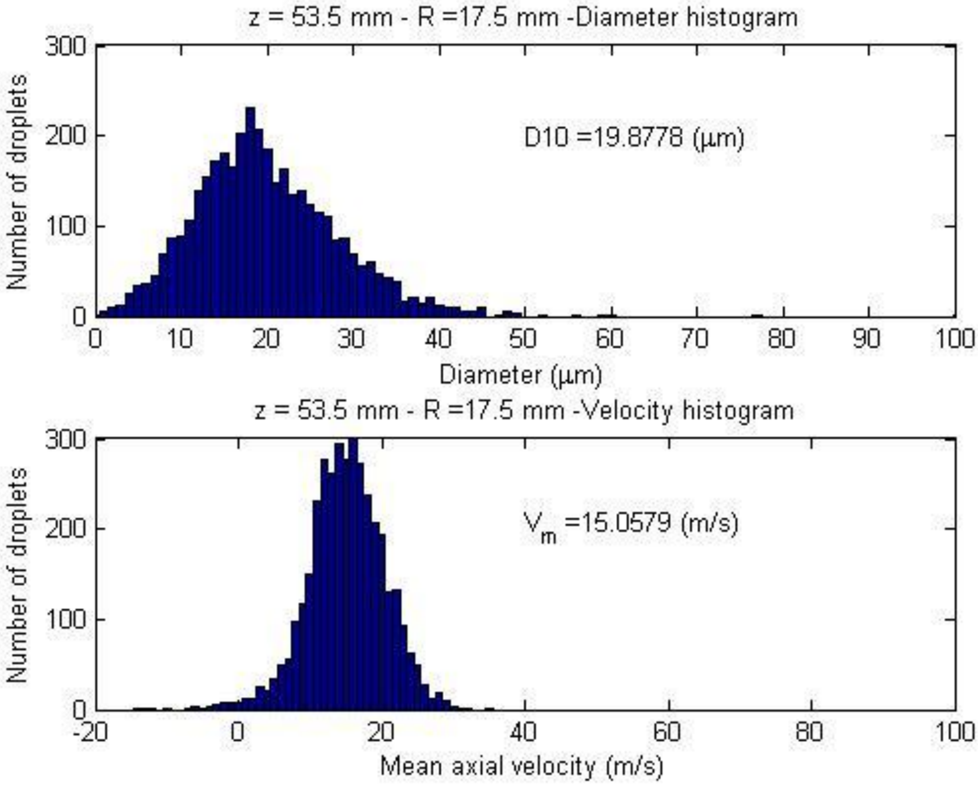


Figure 6-32 Drop speed and diameter histogram for .5 bar back pressure and 2.3 mSec injection

Figure 6-32 shows information about the speed and the diameter of the drops in the spray of the injector. From there it is seen that the average speed is at 15 m/sec, and the average diameter is around 19.87 µm.

Figure 6-33 shows the change in spray when the driving pressure (fuel pump) is changed. The specified working pressure is 2.5 bar (35 psi)[9]. The changes considered from 5 bar to 0.1 bar. No change was noticed. Even it was noticed that the injector can run if no pressure is given by the pump at all. The injector seems to have a reservoir of fuel when the main frame is full of fuel, and being small the amount of fuel injected in relation to the reservoir of fuel, it takes some time until this reservoir gets empty. And as Figure 6-33 is showing, the driving pressure makes no difference in the injector behaviour, then the only thing that the injector needs to operate is to have fuel in it reservoir.

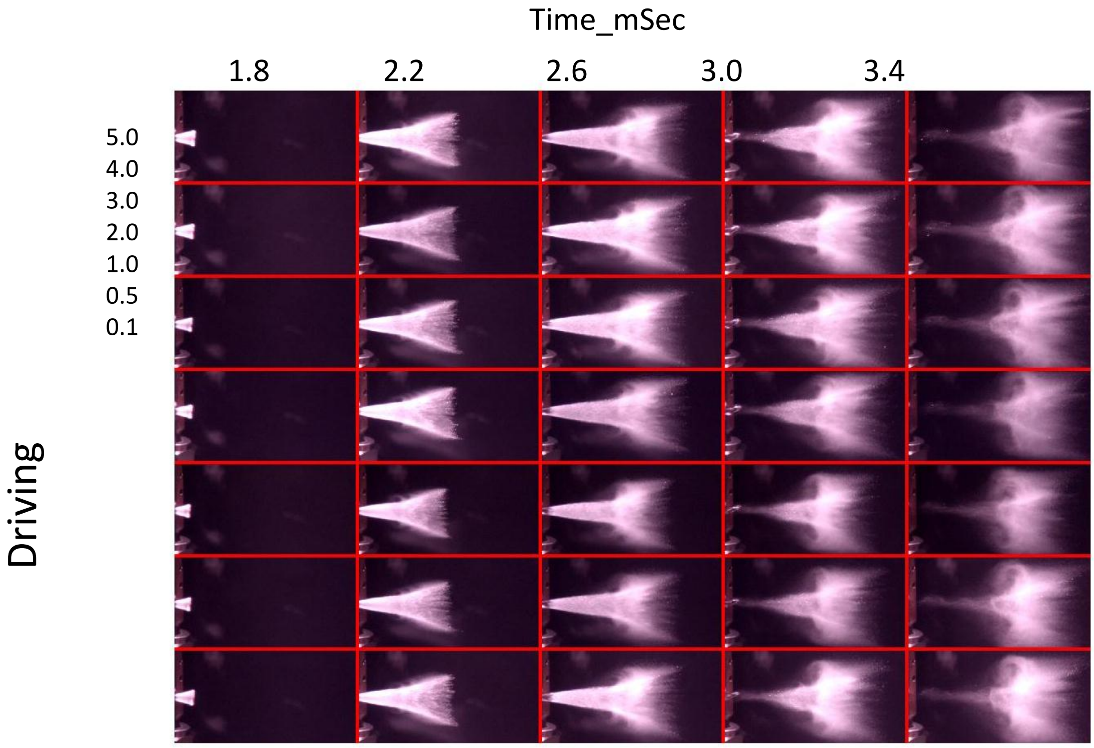


Figure 6-33 Comparison of sprays at different driving pressures

[9] Strauss Sebastian, Zeng Yangbing and Montgomery David. *Optimization of the E-TEC™ Combustion System for Direct-Injected Two-Stroke Engines Toward 3-Star Emissions*. SAE 2003-32-0007.

7 Discussions and Conclusions

Being a direct injected engine, there is the advantage of having the possibility of running over both, stratified and homogeneous modes. One of the biggest interests in this work is to find out in which points does the engine runs at each mode, and it can be done by using the results presented in the previous chapter. This discussion is the first section of this chapter.

The second section contains the discussion over the phenomena of the engine speeding up at a constant load while the injection time reduces.

7.1 Injection mode

The lambda value is related to the injection mode, in the sense that stratified injection runs with lean mixtures and homogeneous mode uses richer mixtures. It's important to mention that the lambda value being referred to is the lambda value of the exhaust gases. During the combustion, due to the non uniform spread of the fuel when fuel injecting on stratified mode, there are areas of the combustion chamber which are burning rich mixtures, even though the overall mixture is lean. This makes that the lambda value during the combustion is different than the lambda value from the combustion gases. The same could happen in homogeneous mode, but is less likely. A good signal that the engine is running whether at homogeneous or stratified is the exhaust lambda value, and a plot of the lambda values measured from the engine would be useful.

The problem is that the measured lambda values are not trustable. Since the engine being measured is a two stroke engine which doesn't have valves (uniflow) then some part (about 20%) of the fresh air coming in to the combustion chamber short circuits to the exhaust port, altering the readings of the lambda sensor.

An alternative to the lambda value is the CO value, which increases when the mixture becomes rich, and decreases when it becomes lean. Figure 7-1 illustrates this.

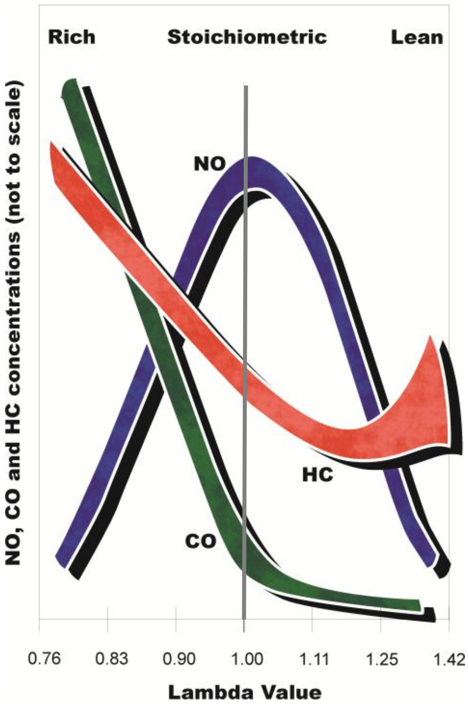


Figure 7-1 Lambda-Emissions plot

The grey line in the middle represents the stoichiometric point. On the right is the lean area, and on the left the rich one. It is shown how CO has a big change in the rich area and under the stoichiometric point. In the lean area the change is not very high. This means that deducting the lambda value from the CO information is practical only in the rich area, as the errors in the CO measurements could be bigger than the change of CO in the lean area.

Figure 6-28, shown in section 6.4, displays a color plot of the CO values measured in the engine. Figure 7-2, shown below, converts those CO values to lambda values.[10]

[10] This process of converting the CO and CO/CO_2 values to lambda was provided by the thesis sponsor, and is protected under secrecy agreement.

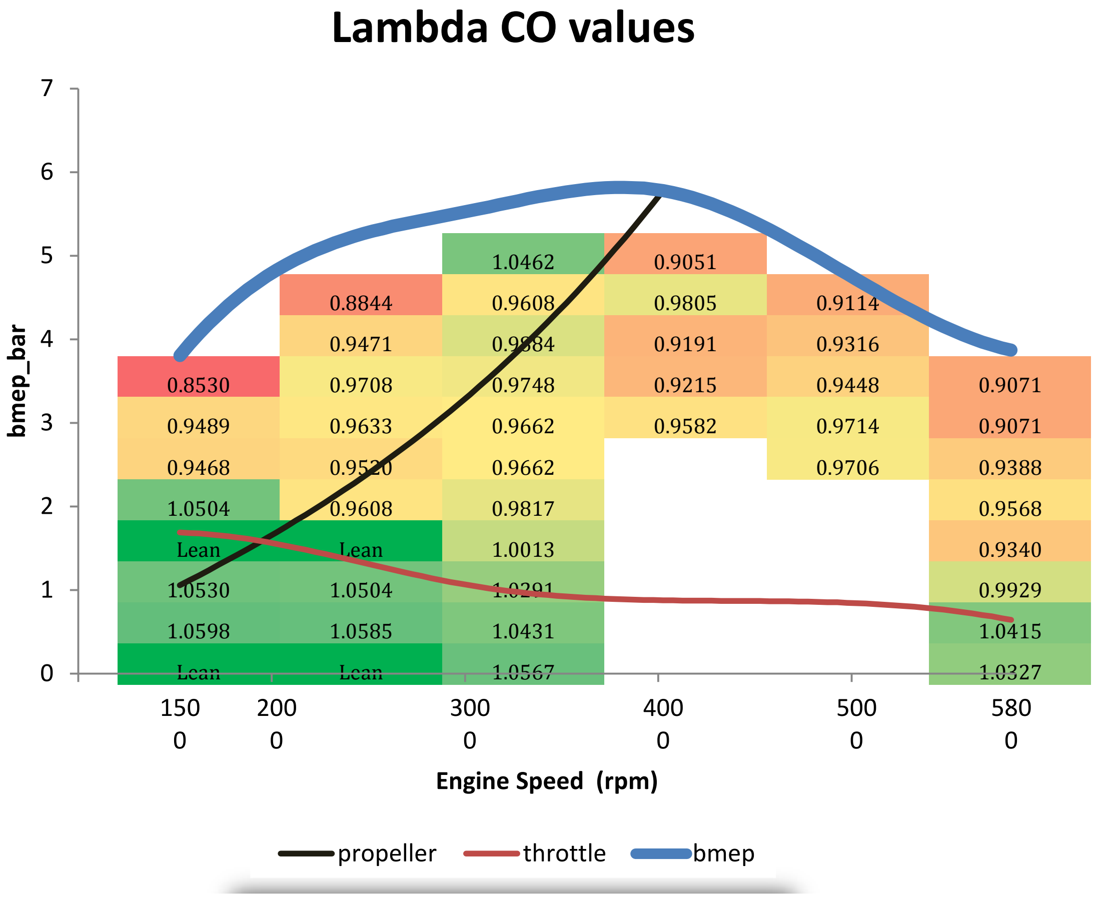


Figure 7-2 Lambda calculation from CO values

Values in green reading "Lean" are values which are beyond the frontier of the confidence of the calculation, so they are the leaner.

Another way to calculate lambda from exhaust emissions is by using the relation CO/CO2. The values of the conversion are presented in Figure 7-3.

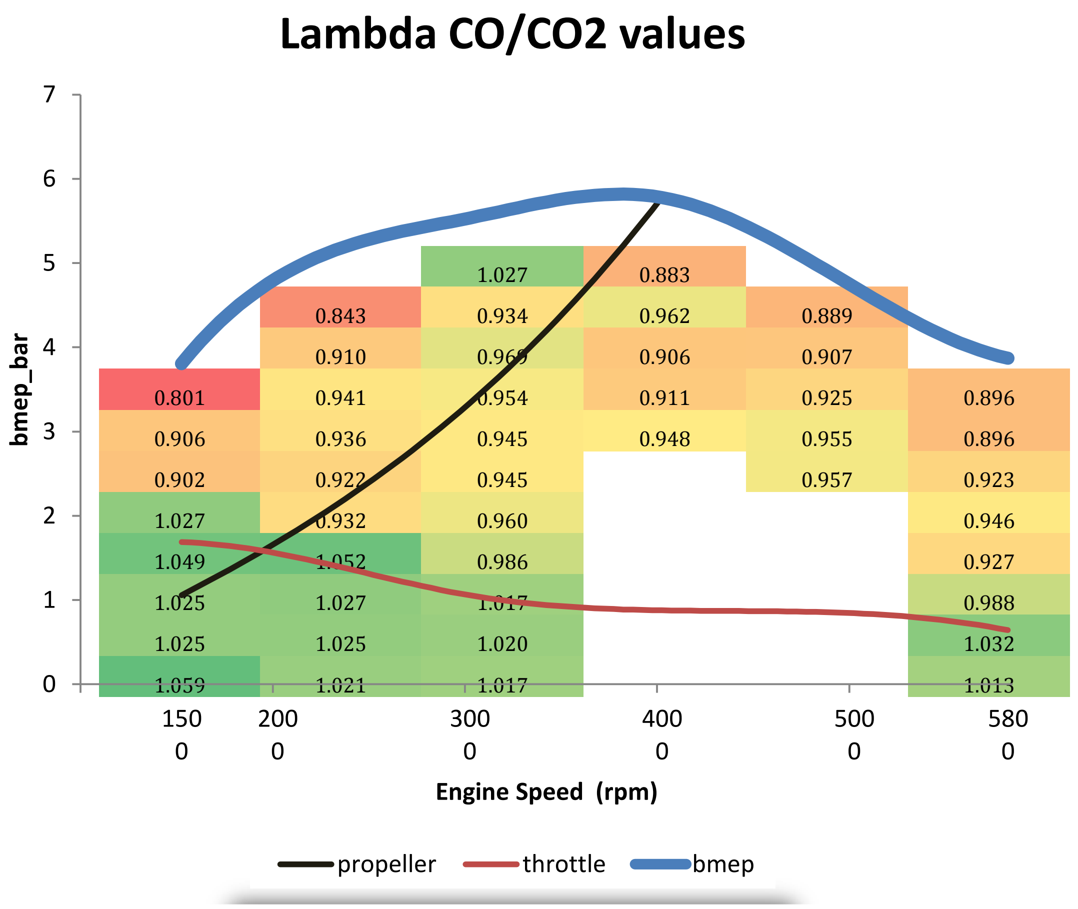


Figure 7-3 Lambda calculated from CO/CO2 ratio

Looking at both plots, it can be seen that they keep a good correlation between them. Even though the values obtained are not as lean as expected, having this correlation points out that the conversion procedure is trustable enough. The first glance over these graphs allows us to identify the green (lean) area in the low load and low speed area, and also in the low speed area at 5800 rpm. There is a lean point at the maximum bmep at 3000 rpm in both plots. As it is one of the points with highest bmep, the only possible reason is that there were some errors while taking that measurement.

To dig a little bit more in to this matter, Figure 7-4 shows a superimposition of the injector timing contour plot over Lambda CO.

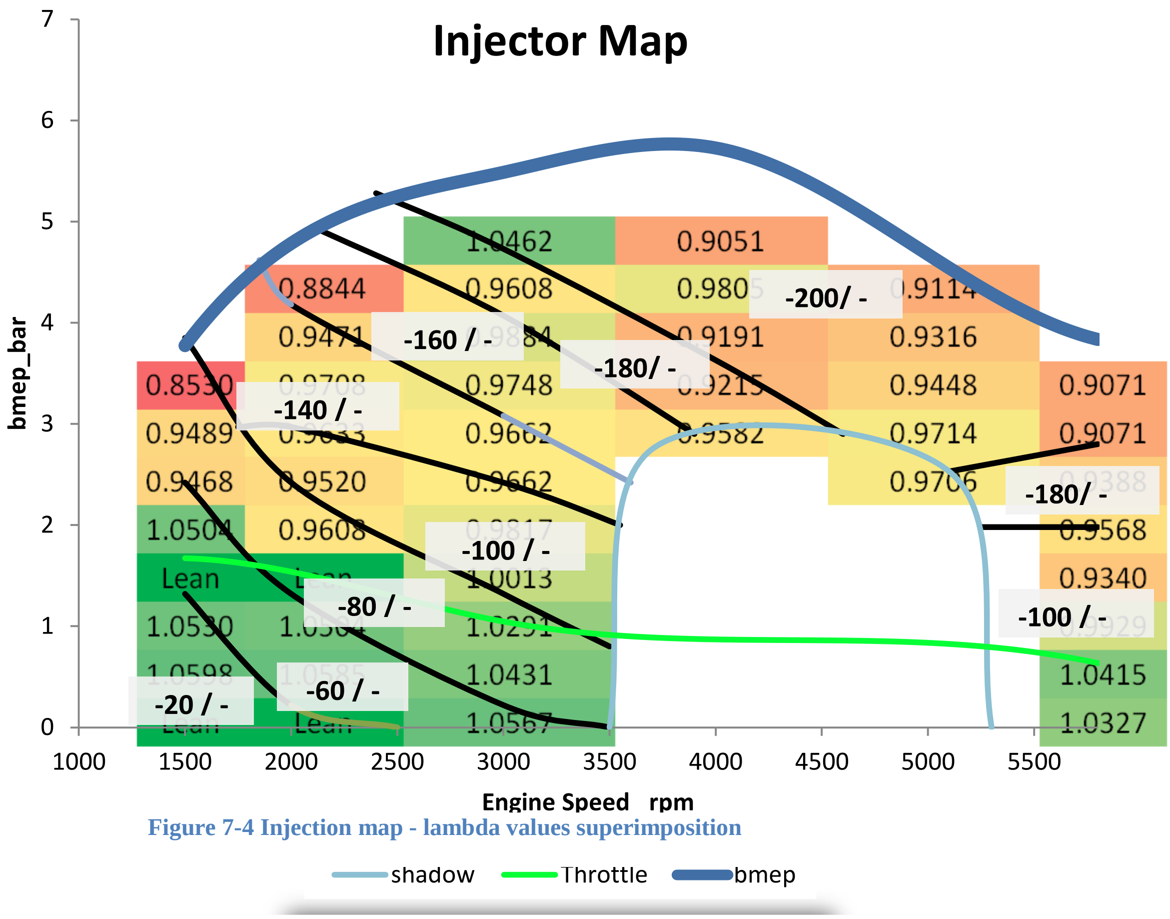


Figure 7-4 Injection map - lambda values superimposition

There is a strong change from green to yellow at 1500 and 2500 rpm in the color plot (lambda values). That change happens in the -80/-100 area. After that, at 3000 rpm, the color change is not so hard, and the transition part goes in to the -100/-140 area. At 5800, the change also happens in the -100/-140 area. In a stratified combustion, the ignition timing follows the injection timing, with enough time between them as to let the fuel to mix a little bit with the air, but not letting the cloud of fuel to spread too much, as to avoid having more than one flame in the combustion chamber.

Figure 7-5 shows the superimposition of the ignition contour plot over the CO color map.

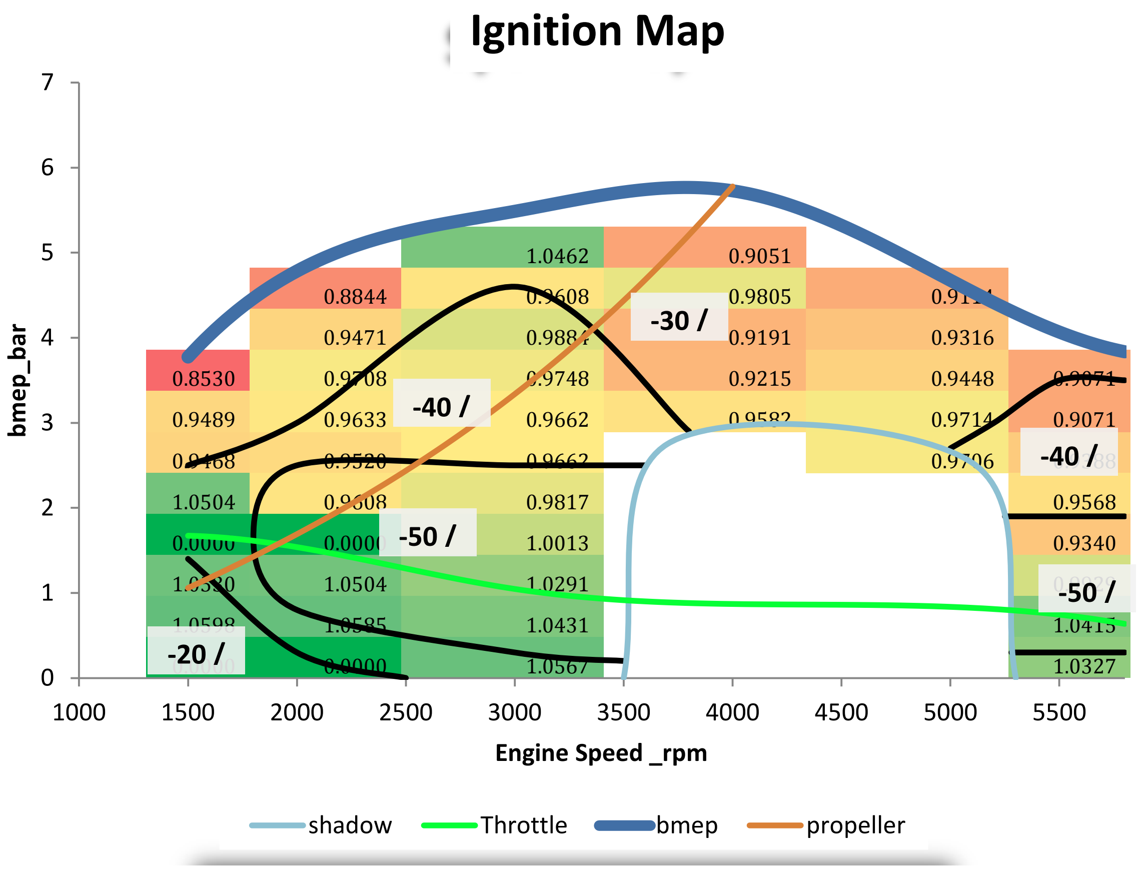


Figure 7-5 Ignition map - lambda values superimposition

In this plot it is possible to see that the ignition timing starts also very early in the leanest corner, and it keeps advancing until the change of color from green to yellow happens. From there, once the ignition angle is at -50/-60, it goes back again reducing the advance of ignition. Figure 7-4 states that the **injection** timing keeps advancing and advancing in the direction of the richest lambda values. Looking at Figure 7-5 again, we see that the **ignition** advances until the lambda color plot changes from green to yellow and it goes back again, letting the injector to keep advancing alone. This point where the ignition timing goes back indicates the area when the engine changes from stratified to homogeneous mode.

Attending to Figure 6-8 in section number 6.3, it can be easily seen how the ignition timing follows the injection timing when the engine increases its load at a constant speed of 3000 rpm. The CAD at which they completely split is near to -57.6 degrees for the ignition and -111.72 degrees for the injection when there is a difference of 54.12 degrees. This happens at 20 Nm, which corresponds to 2.1 bar. Looking back at Figure 7-5 it is possible to see that they match the areas where the change from stratified to homogeneous is suspected to happen.

Going to Figure 6-12, which represents the engine increasing its speed at a constant load of 30 Nm, it is not possible to see a clear relation between the ignition and injection timings. Also, the smallest difference among them is bigger than the difference they had when they split in Figure 6-8. Now, all the points at 30 Nm, which corresponds to 3.3 bar, if we look at the lambda color plot, are deep in to the suspected lean area, meaning that the combustion mode is homogeneous.

Finally, in Figure 6-16 the data for the propeller curve is shown. The point at which they seem to split is located at 2000 rpm. When locating this point along the propeller curve in figure Figure 7-5 it is seen that it is near to the frontier established between stratified and homogeneous modes.

7.1.1 Conclusion 1

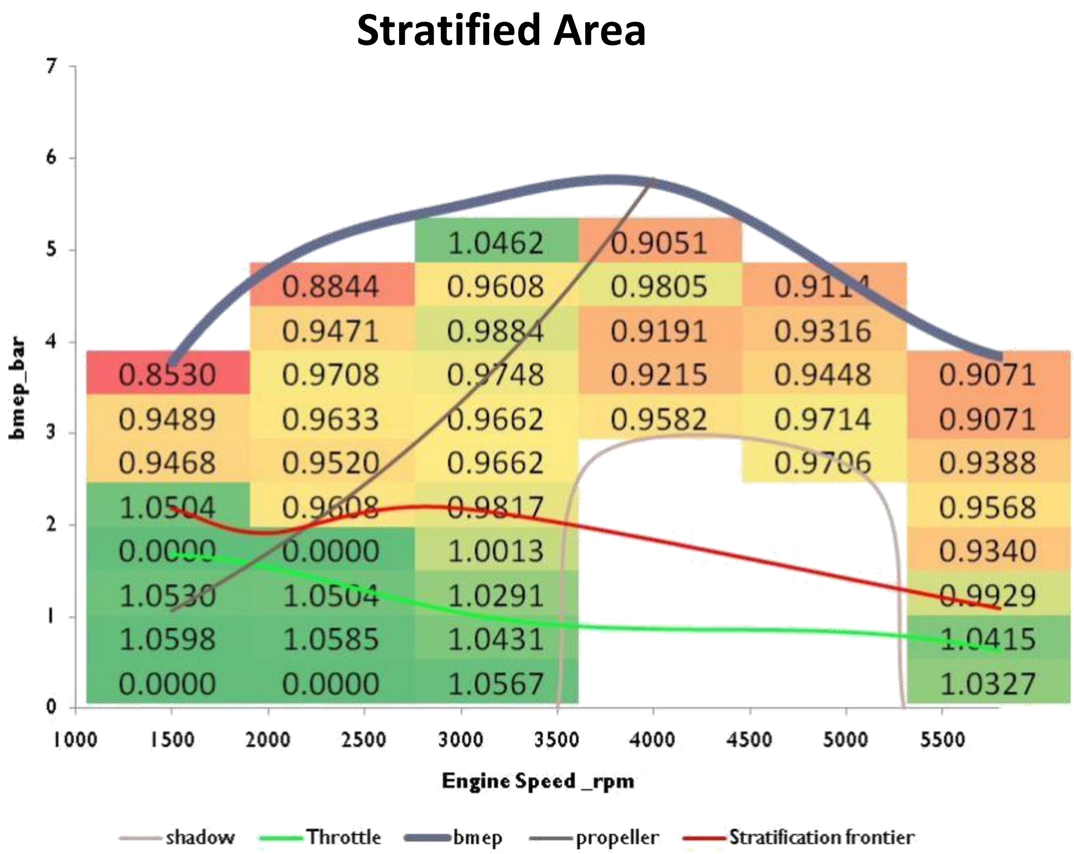


Figure 7-6 Stratified area delimitation

Figure 7-6 shows in red line the stratification frontier. Below it the engine is considered to operate in stratified mode; above it the engine is considered to operate at homogeneous mode.

7.2 Influence of Back Pressures

In chapter 6.3 Figure 6-18 was presented. It showed the injection time over the propeller curve, and it was observed that the injection time varied too little, and near to 1500 it even decreased while the speed increased.

At this point, back pressure is a parameter which becomes important. Back pressure is the pressure in the the pressure in the combustion chamber, which depends on the compression stroke. At the beginning of it,

the beginning of it, the pressure in the cylinder is relatively low, and at the end of the compression stroke the compression stroke the pressure is high. As the injection timing is changing, the back pressure is changing pressure is changing as well. This means that the amount of fuel injected will change if the back pressure if the back pressure changes even though the injection time remains the same, as the injector has to injector has to overcome more or less force to inject the fuel. This variation is clearly visible in visible in

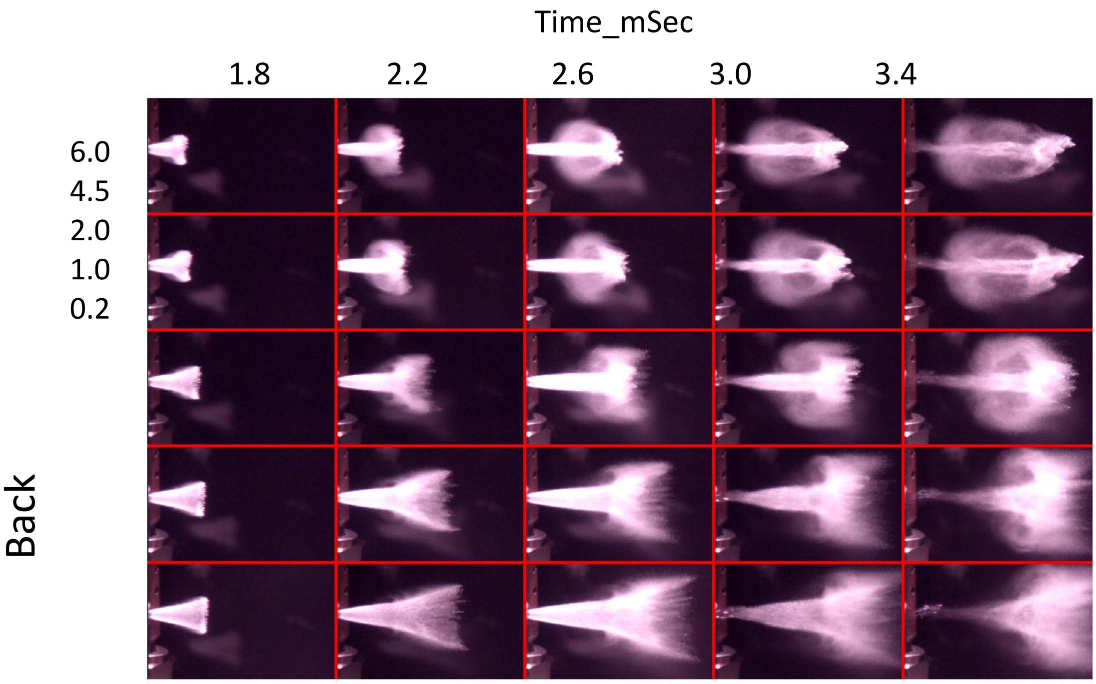


Figure 6-31.

7.2.1 Conclusion 2

Figure 7-7 shows the plot of the back pressure and the fuel injected in the combustion chamber when following the propeller's curve.

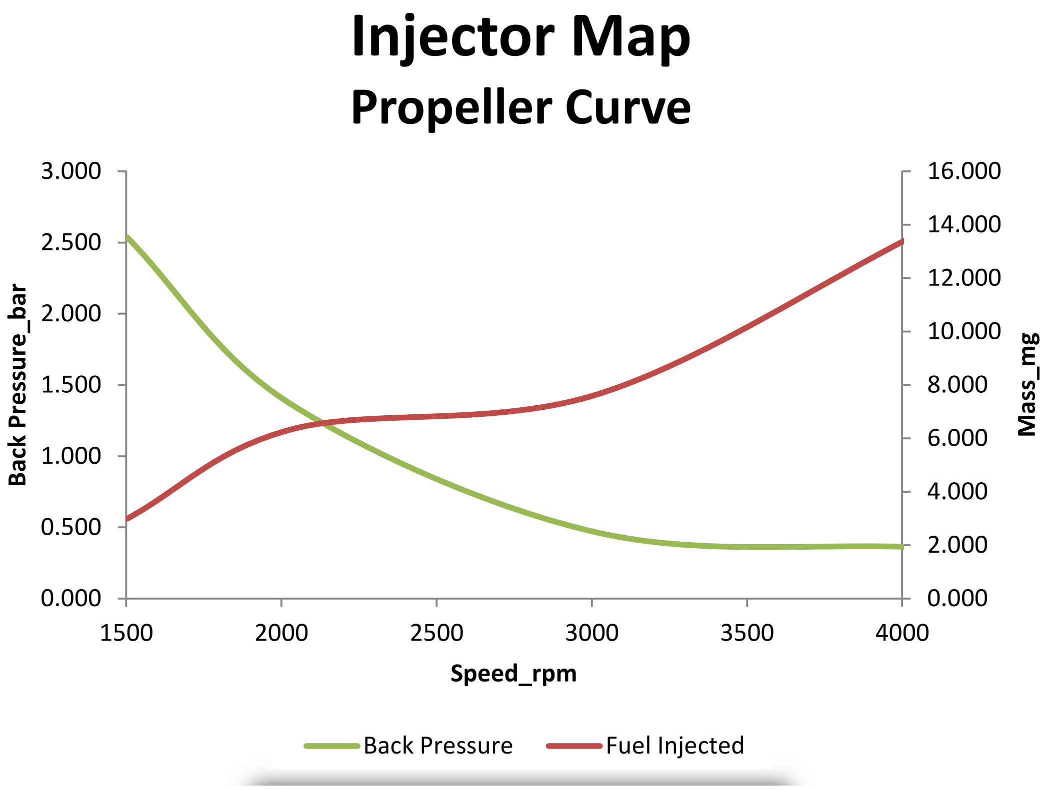


Figure 7-7 Fuel injected - back pressure comparison for propeller curve

In this plot is easy to see that the amount of fuel is increasing while the back pressure decreases, even though in Figure 6-18 it's seen that the injection time decreases or almost remains steady.

7.3 Influence of pressure pump in the injector's spray.

The injector produces its own pressure to inject fuel in the combustion chamber. But it was not sure if the pressure of the fuel pump feeding fuel in to the injector had any impact in the spray characteristics. It was even noticed that the injector was able to run with no pressure in the fuel pump at all. This information can be important when choosing a fuel pump for a different application.

7.3.1 Conclusion 3

It is clearly visible in Figure 6-33 that the spray characteristics of the injector do not change among different fuel pump pressures.

8 Appendixes

In this chapter figures which were not directly involved in the discussions, but still are interesting to know, are shown.

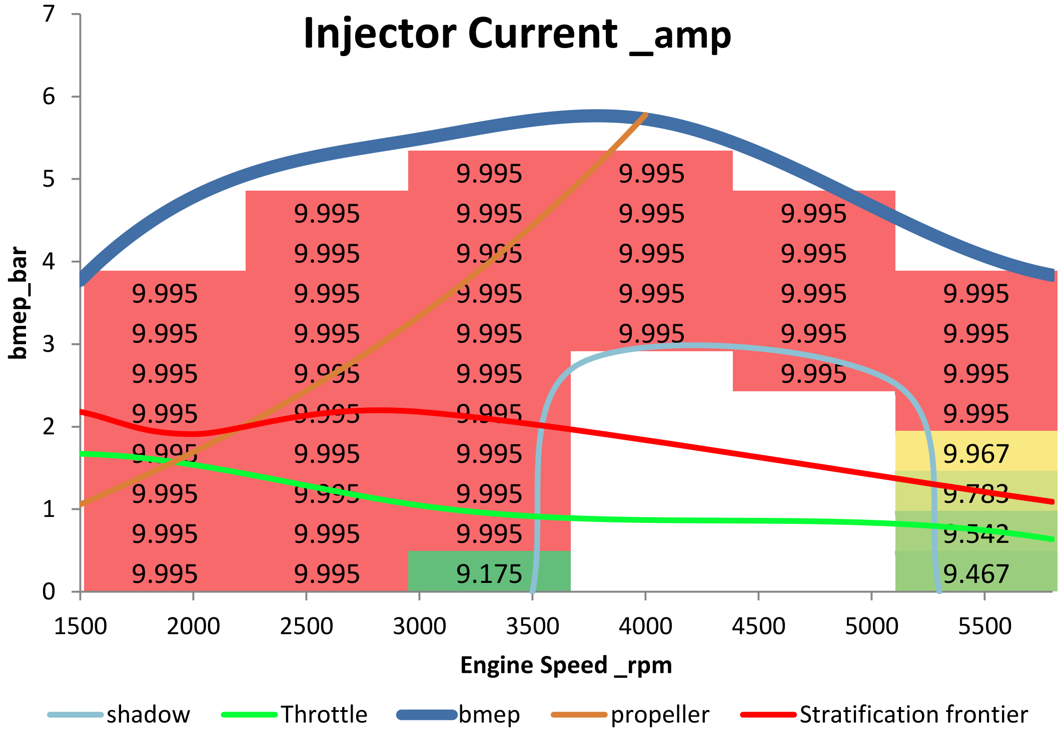


Figure 8-1 Injector Current

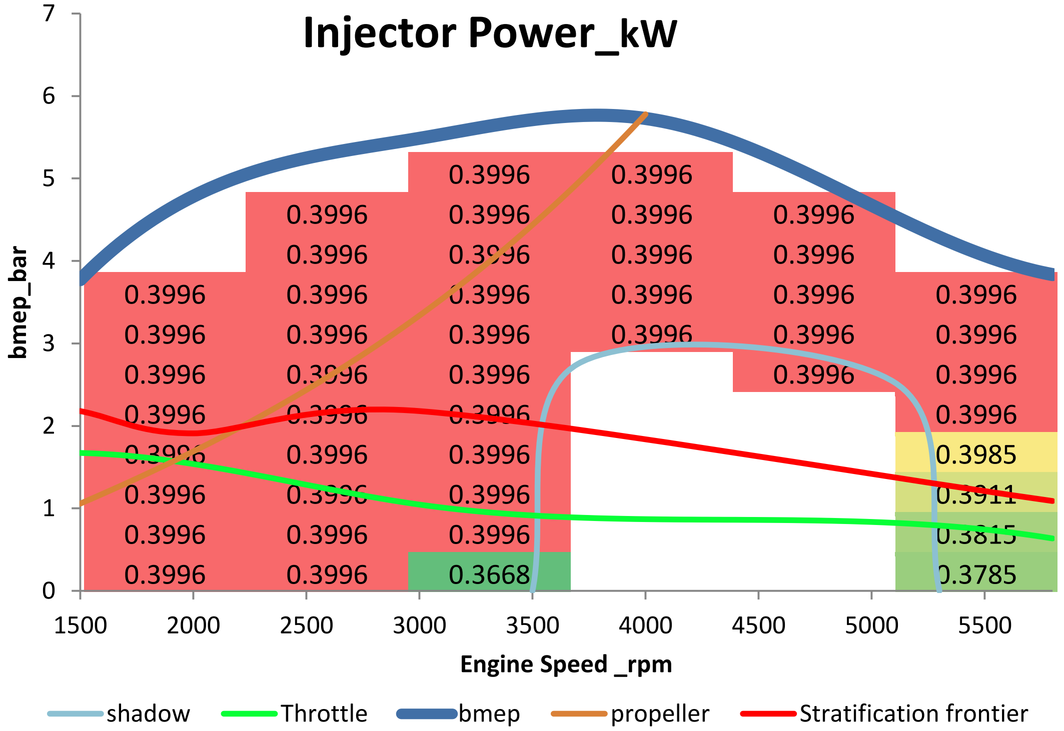


Figure 8-2 Injector Power

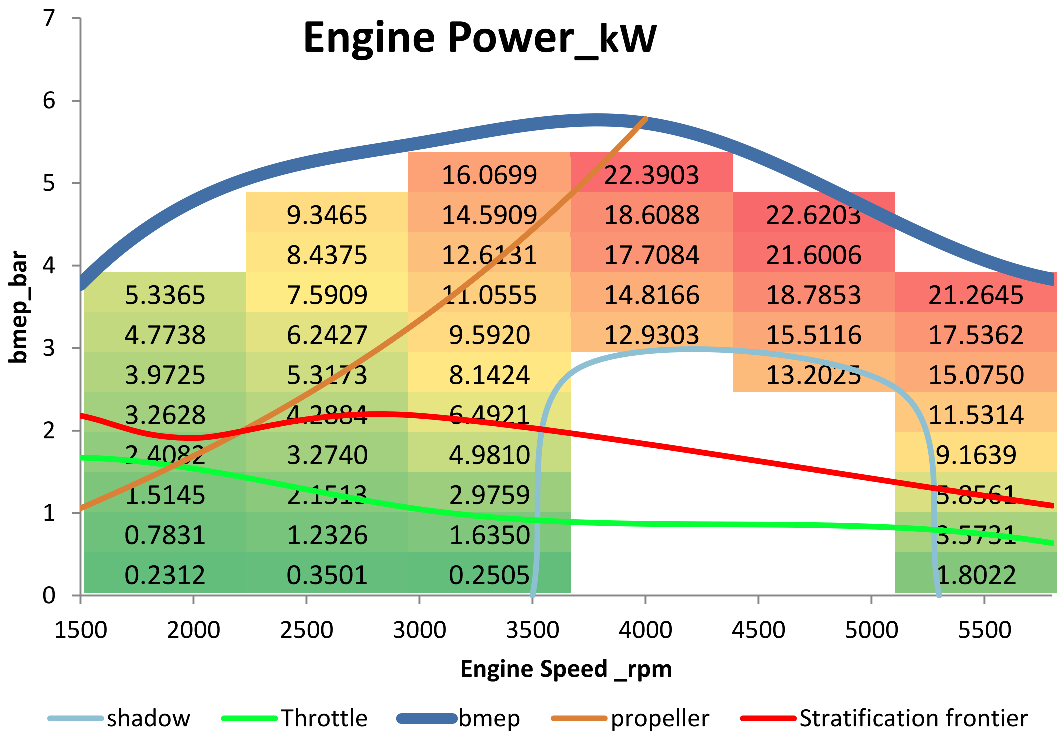


Figure 8-3 Engine Power

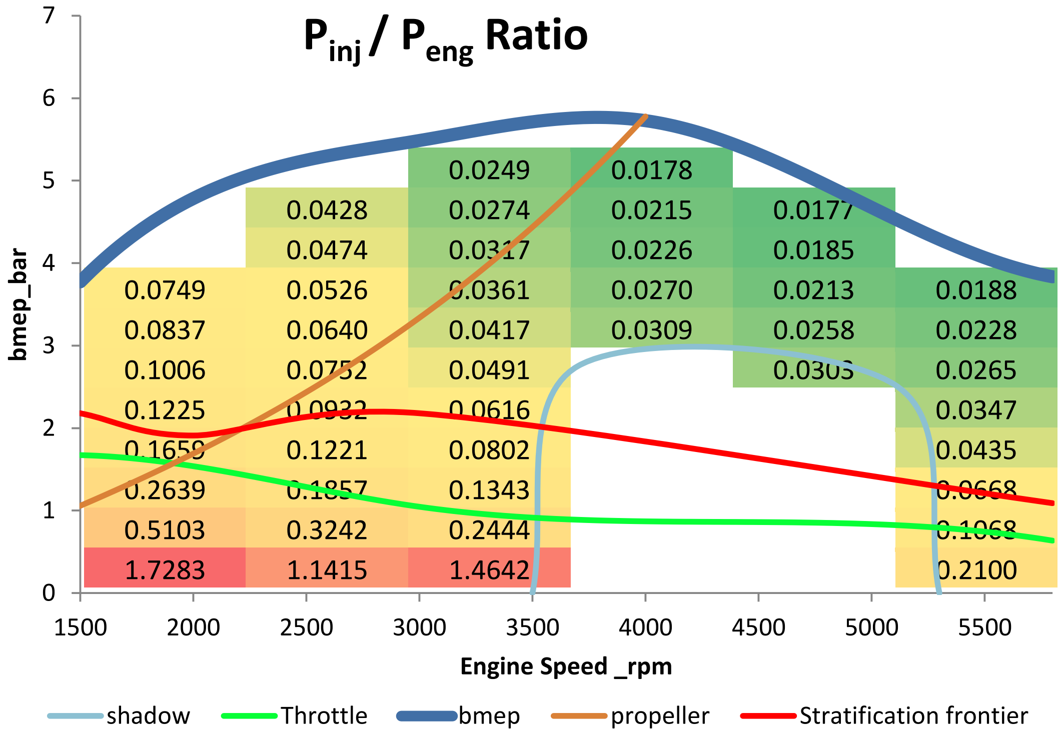


Figure 8-4 P_{inj} / P_{eng} Ratio

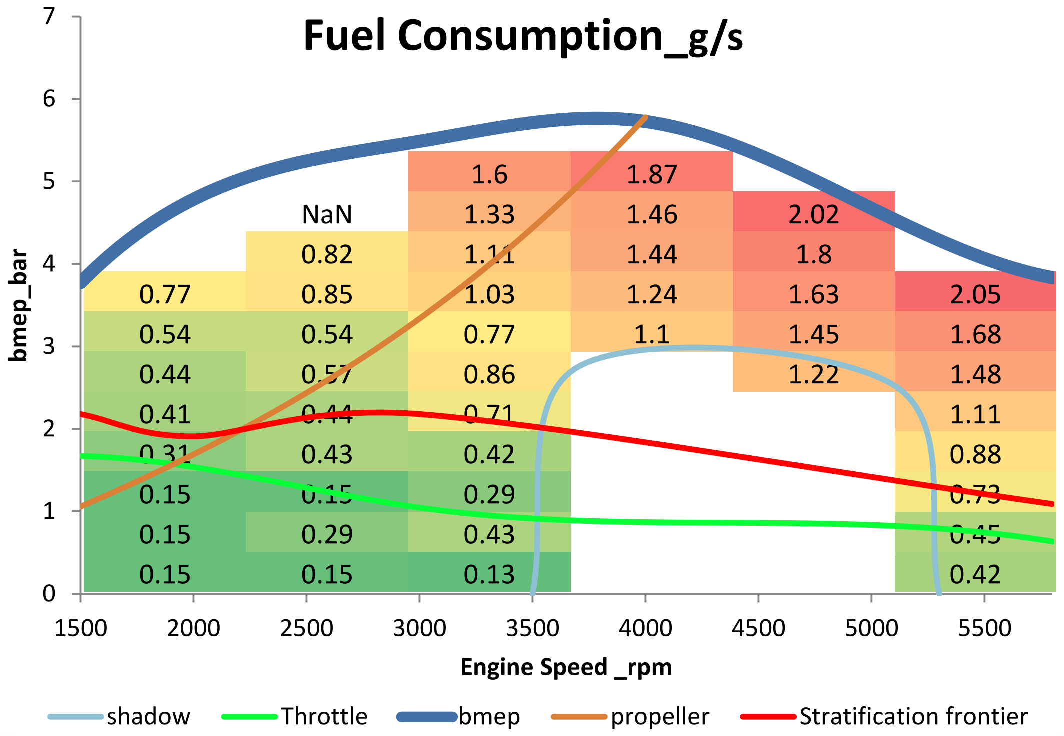


Figure 8-5 Fuel Consumption

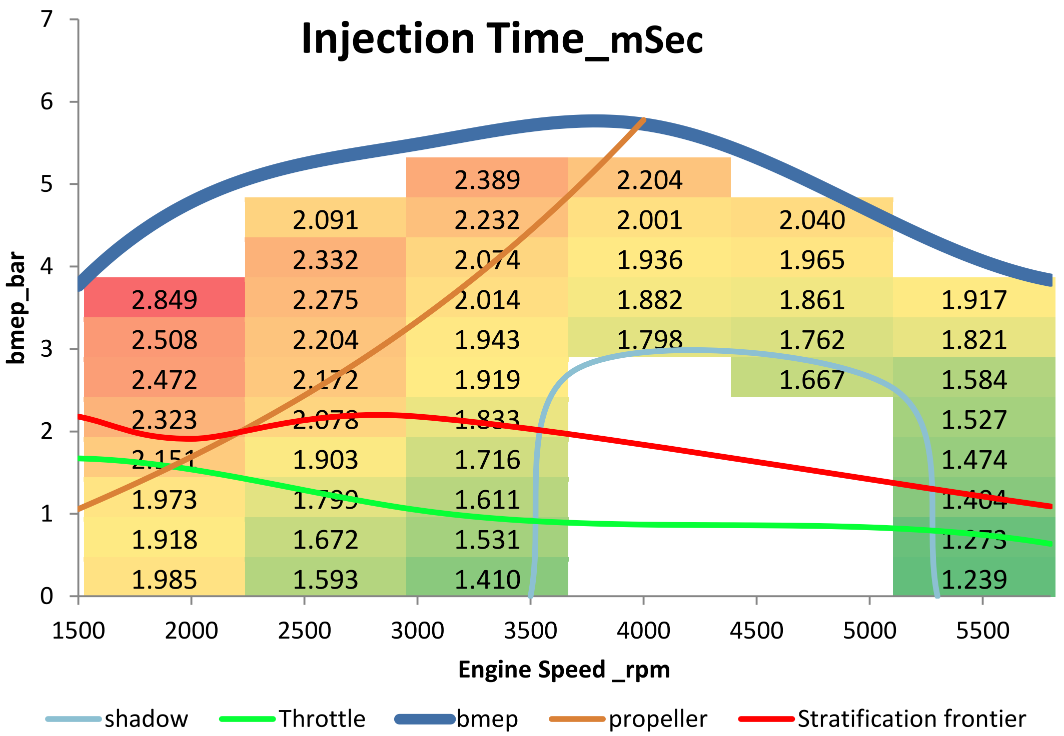


Figure 8-6 Injection Time

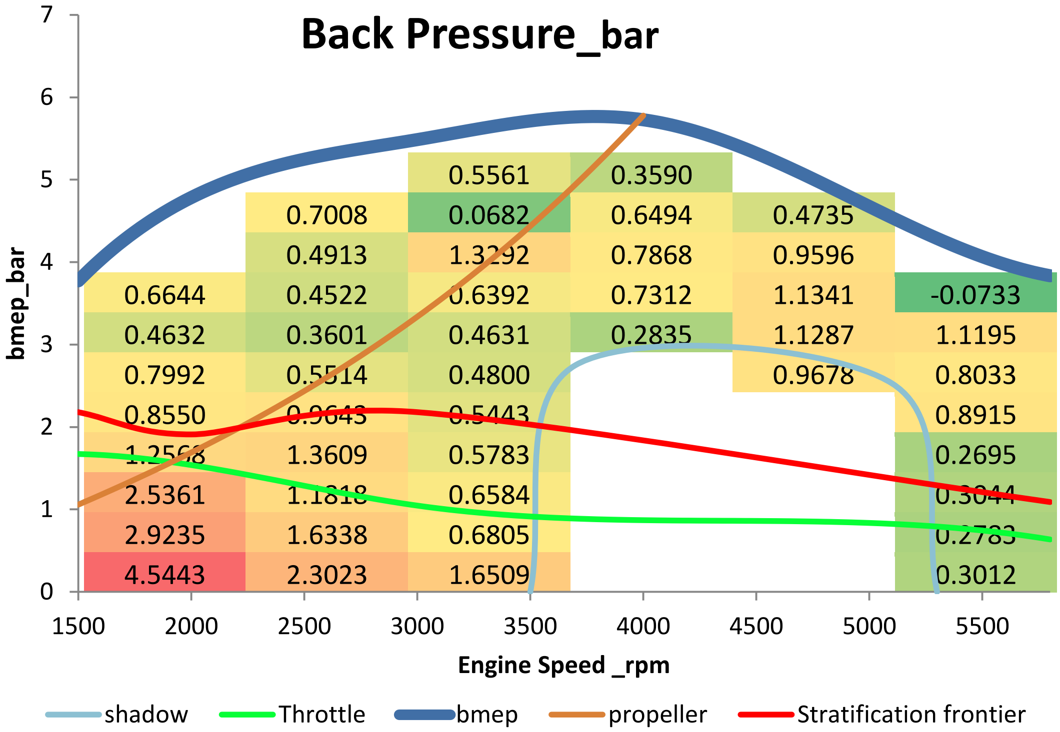


Figure 8-7 Back Pressure

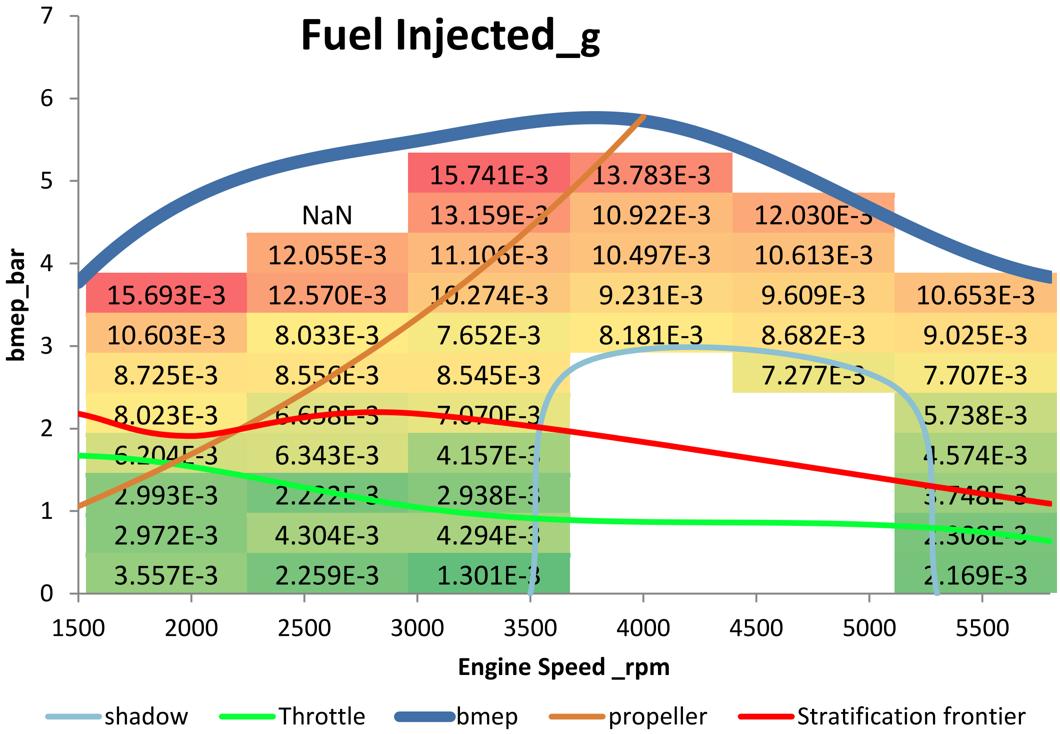


Figure 8-8 Fuel Injected

9 References

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