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Modelling and CNG distribution study of a Natural Gas-Diesel Dual Fuel Engine

David Sancho Casarrubios

Thesis for the degree of Master of Science in Engineering

Division of Combustion Engines
Department of Energy Sciences
Faculty of Engineering | Lund University

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David Sancho Casarrubios

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This degree project for the degree of Master of Science in Engineering has been conducted at the Division of Combustion Engines, Department of Energy Sciences, Faculty of Engineering, Lund University

Supervisor at the Division of Combustion Engines was Professor Per Tunestål
Examiner at Lund University was Professor Martin Tunér

The project was carried out in cooperation with Pablo García

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Division of Combustion Engines

Department of Energy Sciences

Faculty of Engineering, Lund University

Box 118, 221 00 Lund

Sweden

www.energy.lth.se

NOMENCLATURE

Φ	Equivalence ratio
AFR	Air to Fuel Ratio
BDC	Bottom Dead Centre
CAD	Crank Angle Degree
CH_4	Methane
CI	Compression Ignition
CNG	Compressed Natural Gas
CO	Carbon Monoxide
CO_2	Carbon Dioxide
DDF	Diesel Dual Fuel
DOC	Diesel Oxidation Catalyst
DPF	Diesel Particulate Filter
EGR	Exhaust Gas Recirculation
EVC	Exhaust Valve Closing
EVO	Exhaust Valve Opening
IVC	Intake Valve Closing
IVO	Intake Valve Opening
LNT	Lean NO_x trap
NO_x	Nitrogen Oxides
PINGD	Pilot Ignited Natural Gas Diesel
PM	Particulate Matter
ROHR	Rate of Heat Release
SCR	Selective Catalytic Reduction
SI	Spark Ignition
TDC	Top Dead Centre
TWC	Three-Way Catalyst

UHC	Unburned Hydrocarbons
UNFCCC	United Nations Framework Convention on Climate Change

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ABSTRACT

Nowadays, environmental concerns have created the need of evolution in combustion engines for trying to give a solution to the growing problem of pollution produced by the combustion of fossil fuels. This master thesis project has been motivated by the developing of dual fuel engines with natural gas as main energy source for being an alternative to traditional combustion engines. Dual fuel is still a new technology that needs of research for a complete understanding of its operation, such as advantages and drawbacks, in order to improve it.

For that reason, a CNG-diesel dual fuel engine has been modelled through computer software GT-POWER with the main objective of studying CNG distribution in the intake manifold and how this distribution affects to cylinder-to-cylinder variations.

1. - INTRODUCTION

During the last decade, the automotive industry has been forced to reduce the harmful and pollutant gases emitted in the design of combustion engines: concern for greenhouse gases emissions and their effect on the climate change has posed an additional challenge to the development of engines.

Diesel engines are traditionally preferred due to their lower running cost and better torque characteristics. However, due to stricter emissions standards for NO_x and soot, emission trade-off in diesel engine is a concern for engine developers. The reason is that the properties of the diesel fuel itself have mandated the use of several advanced combustion and after-treatment technologies, which in turn have made the diesel engine power train expensive and complex.

Consequently, many countries are looking for alternative fuels with rich resources and low cost for vehicles, alternative fuels. In this context, natural gas is an excellent fuel. Its combustion and emission characteristics are superior to any other realistic competing fuel: it not only helps to reduce emissions but also aids in optimizing the engine at numerous operating points for best fuel efficiency along with a decent torque and power output. The CNG fuelled engine is one of those which could help in achieving the desired targets without major compromises. On the other hand, CNG has also the advantage of being cheaper than diesel or gasoline in most countries.

As an option, diesel-CNG dual fuel combustion is considered as a promising solution for highly efficient internal combustion engines. This concept offers the possibility to combine a diesel pilot injection as a high energy combustion initiation event with an indirect injection of methane as main energy source. The dual fuel combustion strategy dramatically lowers operational costs, and extends maintenance intervals and engine life, NO_x and soot emissions reduce.

1.1. – Engine modelling

With the objective of a deeper study and understanding of how it works under different conditions, an engine simulation of a diesel-CNG dual fuel engine with a GT-Power 1D model is carried out in this master thesis project. These simulations are usually fast and multiple parameters can be monitored and analysed. The advantages of engine modelling are many, from safety to economic reasons.

GT-POWER is the market leading engine simulation software, used by every major engine manufacturer for the design and development of their engines. It is applicable to all sizes and types of engines, and its installed base includes a highly diverse group of car, truck, motorcycle, motor sport, marine, locomotive, power generation, mining and construction, agricultural, and lawn and garden equipment manufacturers. It is a part of GT-SUITE, software of Gamma Technologies. GT-POWER counts on two main powerful tools called GT-ISE (Integrated Simulation Environment) that builds, executes, and manages the simulation process and, GT-POST, a post-processing tool that provides access to all the plot data generated by the simulation. [1]

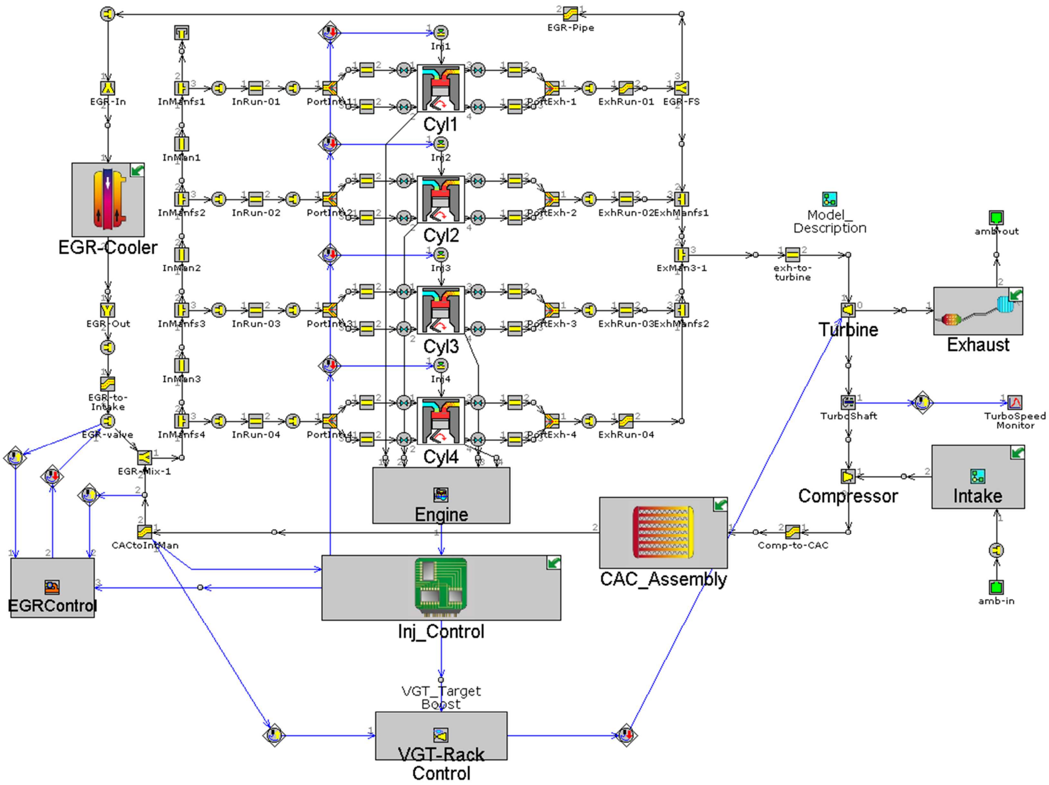

Figure 1.1: Example of a four cylinder diesel engine model with EGR and Turbo in GT-ISE, GT-POWER.

The GT-Power software simulates different engine components including pipes and flow splits that are used to build up the engine geometry. More specialized parts such as cylinders, turbochargers, etc. use built-in models to calculate the performance parameters such as cylinder pressure, heat release, and thermal efficiency. At its core, the GT-Power solver is based on the 1D solution of the fully unsteady, nonlinear Navier-Stokes equations. Beyond this software core is the thermodynamic and phenomenological model solvers to capture the effects such as combustion, heat transfer or droplet evaporation.

2. - LITERATURE REVIEW

2.1. – Climate change

Climate change is the effect in climate, i.e. regional temperature, precipitation, extreme weather, etc., caused by the increase in the greenhouse effect. The greenhouse effect is the process wherein greenhouse gases, such as water vapour, CO_2, CH_4, etc., in the atmosphere absorb and re-emit heat being radiated from the earth, trapping warmth. The greenhouse gases concept refers to gases that contribute to the greenhouse effect by absorbing infrared radiation, heat.

Energy-related emissions are the majority of global greenhouse gas emissions. The fight against climate change has become a defining feature in energy policy-making, but the implications are daunting. Meeting the emission goals pledged by countries under the United Nations Framework Convention on Climate Change, UNFCCC, would still leave the world 13.7 billion tones of CO_2, or 60%, above the level needed to remain on track for just 2ºC warming by 2035. [2]

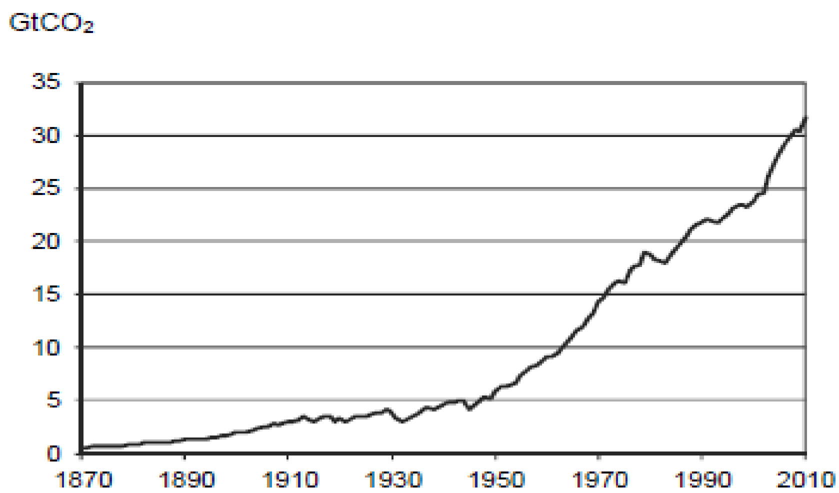

Figure 2.1: Trends in CO_2 emissions from fossil fuel combustion. [2]
Original source: Carbon Dioxide Information Analysis Centre, Oak Ridge National Laboratory, US Department of Energy, Oak Ridge, Tenn., United States.

2.2. - Emissions in combustion engines

The undesired gases and particles which are released into the air or produced by various sources are known as emissions. Their amount and the type differ from industrial activity, technology, and a number of other factors, such as air pollution regulations and emissions controls. In combustion engines, undesirable emissions are of major concern due to their negative impact on environment and human health. For this reason, legislators regularly introduce progressively stricter emission standards to reduce the harmful emissions from road transports. For example, it is used the Euro 6 standards (or Euro 5 in some cases) in the European Union, or the Federal Test Procedure, FTP-75, in the United States.

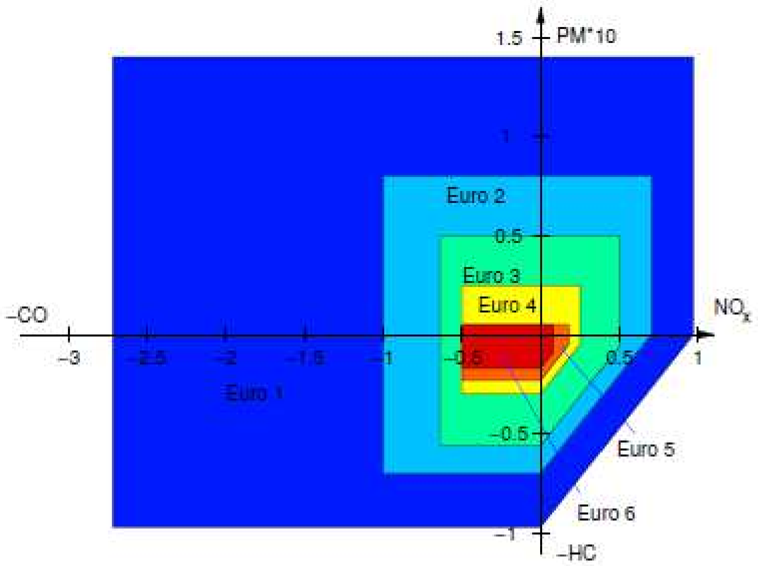

Figure 2.1: Historical, present and future diesel passenger car emission standards in the European Union. Emission limits are given in g/km over the entire NEDC drive cycle. [3]

Internal combustion engines produce air pollution emissions due to combustion of carbonaceous fuel. These emissions can be divided into two subcategories: local emissions, which negatively affect the environment close to the emitter; and global emissions, which affect the entire planet equally, producing the greenhouse effect. Currently legislation and certification of vehicles take only local emissions into account, and therefore these receive the most attention from engine developers. Global emissions, the main one being CO_2, are generally analogous to poor fuel economy and are therefore minimized in order to have a competitive product.

2.2.1. - Local emissions

2.2.1.1. – Carbon monoxide CO

Carbon monoxide, CO, is formed as an intermediate step in combustion of hydrocarbons. The lack of oxygen or/and low temperatures in the cylinder can produce a failure in the oxidization process of CO into CO_2. Then the emissions of carbon monoxides are mainly controlled by the equivalence ratio of the mixture in the combustion chamber and mixture temperature. They are promoted by a rich combustion, tending to increase constantly with an increasing equivalence ratio. Diesel engines operate on lean mixtures, so carbon monoxide emissions are commonly low enough to be insignificant. Therefore, SI engines are more prone to carbon monoxide emission. [4]

CO is toxic to humans because it prevents blood from transporting oxygen, which can cause symptoms ranging from light headaches to death. CO emissions are effectively handled by an oxidation catalyst and are therefore not as problematic as emissions of HC or NO_x.

2.2.1.2. – Unburned Hydrocarbons UHC

Unburned hydrocarbon emissions, UHC, are a result of incomplete combustion of the hydrocarbon fuel. Hydrocarbons react in the presence of nitrogen oxides and sunlight to form ground-level ozone, a major component of smog. Ozone irritates the eyes, damages the lungs, and aggravates respiratory problems. It is our most widespread and intractable urban air pollution problem. A number of exhaust hydrocarbons are also toxic, with the potential to cause cancer. [5]

In diesel engines there are two major sources of hydrocarbon emissions: At first, during the combustion delay period the mixture is too lean, and thereby, outside the flammability boundaries of the fuel. Late in the combustion process fuel leaves the injector with a low velocity. This results in under mixing of the fuel, consequently resulting in unburned hydrocarbons.

In SI engines the origin of hydrocarbon emissions is primarily due to fuel-air mixture trapped in crevices during combustion (approximately 80% of the emitted unburned hydrocarbons). The largest crevices are mainly located between cylinder walls, piston and piston rings. During the compression stroke unburned mixture is forced into crevices, and cooled due to heat transfer from the cylinder walls. During combustion more mixture is forced into the crevices. Then the flame propagates into the crevices region, and partially burns the mixture, or the flame quenches at the crevice entry region. Furthermore, absorption of hydrocarbons by the oil layers, incomplete combustion and flame quenching at the cylinder walls play a secondary role in the emission of unburned hydrocarbons [4].

2.2.1.3. – Nitrogen Oxides NO_x

Nitrogen oxides, NO_x, like hydrocarbons, are precursors to the formation of ozone, but also contribute to the formation of acid rain. [5]

The formation of NO_x is propagated by high charge temperatures and excess oxygen concentrations. Therefore, the nitrogen oxides emissions are highest in zones close to stoichiometric, slightly lean mixture. The combustion temperature and access to oxygen is strongly affected by variations in equivalence ratio ϕ, ignition timing and exhaust gas recirculation, EGR. However, in CI engines, combustion is heterogeneous and generally takes place at an ϕ quite different from that measured in the exhaust pipe. Even at low loads, where mixture is very lean, the mixing controlled combustion always occurs at a local ϕ close to one (stoichiometric). For this reason, the global ϕ will only have a limited effect on the formation of NO_x in CI engines. Moreover, fuels with higher adiabatic flame temperatures compared to diesel tend to produce higher NO_x emissions.

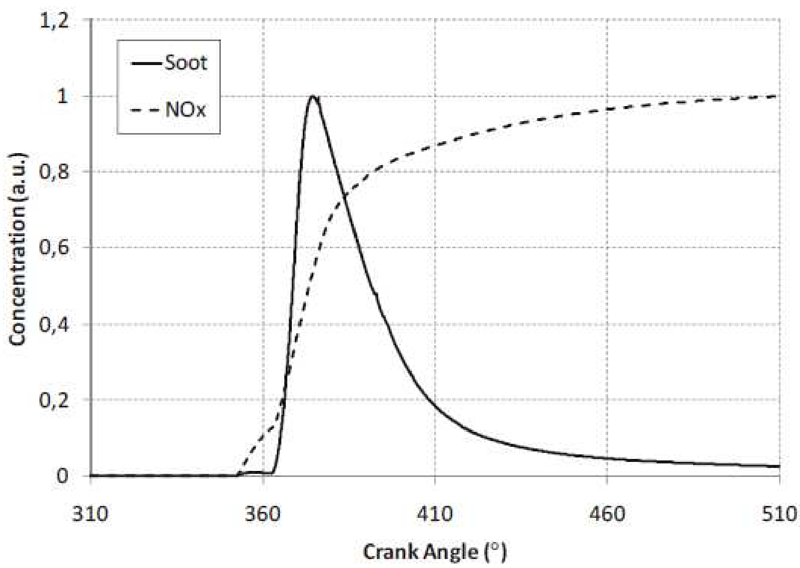

Figure 2.2: Typical soot (solid line) and NO_x (dashed) histories in the cylinder of a light-duty CI engine. [3]

2.2.1.4. – Particulate matter PM (Soot)

Particulate matter, PM, is not a well-defined substance, but a complex mixture of non-gaseous emissions. This emission is common in compression ignition engines (diesel engines). The solid part of it consists of soot and ash compounds from lubrication oil additives and engine wear. It is connected with serious health effects like lung cancer as well as respiratory and cardiovascular diseases. [3]

Particularly, soot is theorized to be the second largest cause of global warming. It consists of carbonaceous material formed during the combustion process and is generally the dominating part of the PM emissions. The soot formation rate is strongly dependent on temperature and the local mixture equivalence ratio ϕ. Generally, ϕ must exceed a value of roughly two for significant soot formation to occur.

The soot mass emitted from the engine is the result of two processes: formation and oxidation. The typical soot history is characterized by rapid formation at the beginning of combustion, followed by a slower, but larger, reduction due to oxidation during the late cycle [4]. In a good combustion engine, soot emission is on the order of 1% of the soot formed in the cylinder.

2.2.2. - Global emissions

2.2.2.1. – Carbon dioxide CO_2

Carbon dioxide, CO_2, is a product of all hydrocarbon combustion. Even though it is not harmful for living beings, it is a greenhouse gas. An increase in CO_2concentration in the atmosphere is a cause of global warming, changing the climate.

There are three main methods of reducing CO_2 emissions. At first, they are reduced if the fuel consumption is reduced. Another method is reducing them by using a biofuel, i.e. a fuel derived of biological materials. Finally, it is possible to reduce them by using a fuel containing less carbon, such as methane CH_4, which is the main component of natural gas.

2.2.2.2. – Methane CH_4

Methane, CH_4, is not directly toxic, but is more difficult to break down in a catalytic converter, so in effect a "non-methane hydrocarbon" regulation can be considered easier to meet. Since methane is a greenhouse gas, interest is rising in how to eliminate emissions of it. Its lifetime in the atmosphere is much shorter than CO_2, but CH_4 is more efficient at trapping radiation than CO_2. Pound for pound, the comparative impact of CH_4 on climate change is over 20 times greater than CO_2 over a 100-year period. [6]

2.2.3. – Use of catalytic converters

The task of a catalytic converter for exhaust after-treatment is to oxidize CO and HC and/or to reduce NO_x. Catalytic converters rely on surface oxidation and/or reduction where the catalyst itself is most often a precious metal but could also be a non-precious metal. Palladium is used for reduction and rhodium is used for oxidation. Platinum is also widely used for both oxidation and reduction. An automotive catalytic converter most often consists of a ceramic core with a honeycomb structure in order to maximize the surface area for the catalytic material. The core is then covered with a wash coat containing the catalytic material.

The bulk of the wash coat consists of silicon dioxide, titanium dioxide or aluminium oxide and the catalytic material is suspended in the wash coat. The purpose of the wash coat is to provide a rough surface which further increases the surface area over which the engine exhaust can react with the catalyst material.

In SI engines, the three way catalyst, TWC, is the most common catalytic converter for exhaust after-treatment. The TWC is capable of both reducing NO_x and oxidizing HC and CO. In order for the oxidation of HC and CO to be effective there must be some oxygen in the exhaust, i.e. lean operation. In order for the reduction of NO_x to be effective, however, the oxygen concentration has to be very low, i.e. rich operation. It turns out that in a very narrow air to fuel ratio, AFR, interval around stoichiometric the TWC can both reduce NO_x and oxidize HC and CO. This means that it is essential to control the AFR very accurately in order to meet emission targets. [3]

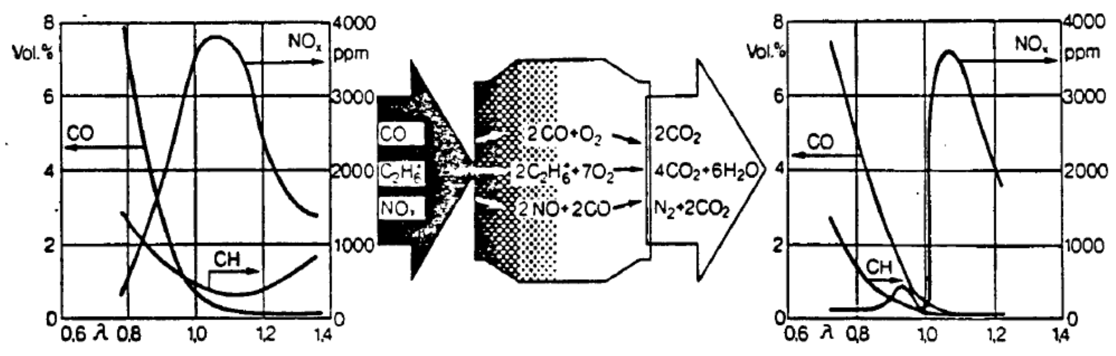

Figure 2.3: CO, HC and NO_x emissions before and after TWC as a function of λ.

2.3. - Overview of compression ignition engines

Compression ignition (CI) engines are also called diesel engines after their inventor, Rudolf Diesel. In contrast to conventional spark ignition (SI) engines, where combustion is premixed, the mixing of fuel and air take place during combustion in CI engines. The CI engine require the fuel to auto ignite for the combustion to start at all. Long carbons chains are easily broken thus large heavy hydrocarbons are very suitable. A good diesel has a high cetane number, i.e. short ignition delay. [3]

Since the mixing rate limits the rate of combustion, the process is called mixing limited combustion or, sometimes, diffusion combustion. CI engines are traditionally associated with higher tailpipe emissions of NO_x and PM than SI engines. On the other hand, they boast significantly lower fuel consumption. This is partly due to diesel fuel containing about 10% more energy per unit volume than gasoline, but also to the higher efficiency of the CI combustion process. A CI engine typically reduces the fuel consumption by 30-40% as compared with an SI engine, estimated for a combined US city/highway test cycle. Similar reductions in CO_2 emissions over a European drive cycle have been reported. Taking the energy densities of the fuels into account, a diesel car is about 17% more energy efficient than its gasoline counterpart, based on official fuel consumption numbers for European cars in 2008.

Several factors contribute to the higher efficiency of the CI engine with respect to the SI engine. First, CI engines have considerably lower pumping losses than conventional SI engines. This is because load control is achieved by regulating the amount of fuel injected, rather than by throttling the intake airflow. Second, since the fuel is injected near TDC, compression ratios are not limited by knock and are typically much higher than in SI engines. Third, the overall fuel-lean conditions in CI engines keep both global average temperatures and concentrations of CO_2 and H_2O in the produced gases low. This means that the post-combustion gases are characterized by a high specific heat ratio, allowing a greater amount of work to be extracted during expansion. Finally, fuel is injected into a cavity in the piston and is thereby kept from entering engine crevices. This leads to combustion efficiencies that are typically greater than 98%. In combination, these factors result in an efficiency advantage as compared to conventional (port-injected) SI engines.

CI engines have traditionally been used in applications where fuel efficiency is of central importance, for example in long haul trucks where fuel cost is a significant part of the total cost of ownership. With the increasing focus on energy efficiency, CI engines have become more popular in passenger cars as well.

In the European Union, the share of diesel-powered cars has exceeded 50% for a number of years and the share is increasing in other markets as well. This is due both to the superior fuel economy and to technological advances that have steadily decreased the harmful emissions from diesel vehicles.

2.3.1. – Exhaust after-treatment

Exhaust after-treatment for diesel engines has a much shorter history than for is very different of SI engines. The main reason is that diesel engines always run lean which means that a TWC cannot reduce NO_x, and this is a big drawback of exhaust after-treatment for CI engines. However, it is possible to oxidize HC and CO and thus diesel oxidation catalysts, DOC, are widely used on diesel engines. The DOC does not convert soot but it does remove a large part of the soluble organic fraction of the exhaust. [3]

In-cylinder measures, like increased injection pressure, cannot remove the soot emissions completely. A great portion of the remaining soot can be removed from the exhaust stream using a diesel particulate filter, DPF. DPF consists of a porous material with channels in it. The channels are plugged in alternate ends of the filter, forcing the exhaust gases that flow into them to pass through the porous wall where the PM is filtered out. To prevent the filter from clogging up the trapped particles are removed from the filter surface by increasing the temperature until the soot particles are combusted, which normally require a temperature of around 600ºC, a temperature normally reached only at high load conditions. By using a catalyst this temperature can be reduced by at least 150ºC, making filter regeneration possible in a greater part of the engine's operating range.

At low loads, however, regeneration must be forced by somehow increasing the exhaust gas temperature. This is often achieved by injecting fuel very late during the expansion stroke, a procedure associated with a fuel consumption penalty. The catalyst is sometimes fuel-borne, meaning that it is mixed into the fuel and incorporated into the soot particles. Fuel-borne catalysts are often cerium or iron based. It is also common to use catalytically coated filters, which are often palladium or platinum based. The advantage of coated filters is reduced complexity, since no system for adding catalyst to the fuel is needed. With a fuel-borne catalyst, on the other hand, the DPF can be regenerated more rapidly since all particles are exposed to the catalyst, not only those closest to the filter surface. A potential drawback in fuel-borne systems is that the catalyst may be emitted to the atmosphere during regeneration.

Removing NO_x from diesel exhaust is more difficult than from SI engine exhaust. A technology that has become industry standard is selective catalytic reduction, SCR. It removes NO_x from the exhausts by reducing it into nitrogen N_2 and water H_2O over a catalytic surface. A reductant must be added to the exhausts upstream of the catalyst. Ammonia, NH_3, is used as a reductant in some applications but, since it is a hazardous chemical, urea is typically used in vehicles. Modern SCR catalysts typically need about 200 ºC to operate. This, together with the fact that the water solution of urea can freeze makes SCR difficult to use in cold winter conditions.

Typically the urea dosage is controlled based on an engine exhaust model combined with a catalyst model. The control system can also be complemented with one or two NO_x sensors up- and downstream the SCR catalyst. Since the urea is converted to ammonia in the high-temperature exhaust, some ammonia may escape unreacted from the SCR catalyst. For this reason, an SCR system is usually complemented with an ammonia slip catalyst as well.

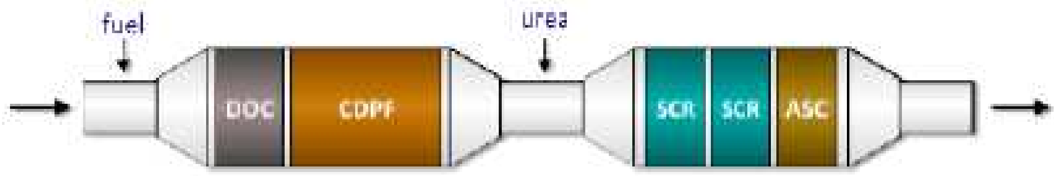

Figure 2.4: Schematic of a typical US 2010 diesel exhaust after-treatment system with exhaust fuel injection, DOC, catalytic DPF (CDPF), urea injection, SCR catalyst and ammonia slip catalyst (ASC). [Source: BASF][1]

An alternative technology is the so-called NO_x trap, or lean NO_x trap (LNT). This system does not require an extra reductant. On the other hand, using fuel only to purge the trap obviously increases the fuel consumption. [3]

2.4. – Compressed Natural Gas CNG as fuel

Compressed Natural Gas, CNG, is made by compressing natural gas, which is mainly composed of methane, CH_4, to less than 1 percent of the volume it occupies at standard atmospheric pressure. It is mainly methane stored at high pressure. It can be used in place of gasoline, diesel or propane. Although it is a fossil fuel, CNG offers a 20% reduction in greenhouse gas emissions compared to gasoline or diesel because of the greater H to C ratio of methane. Natural gas is relatively abundant and will be available at competitive prices for the upcoming decades.

CNG has many properties making it a suitable automotive fuel. Since it is gaseous, exceptional mixing can be achieved and soot emissions are for this reason unlikely. It is a very simple and stable molecule. Because of this stability, it has a high auto-ignition temperature and good resistance to knock, enabling high compression ratios and resulting benefits in thermal efficiency. However, the stability of the methane molecule also presents a challenge for the after-treatment system since methane requires a high light-off temperature and the conversion efficiency of methane does not reach 100% for the catalyst, even at high temperatures, as shown in figure 2.6. For this reason, minimizing engine-out emissions of unburned methane is of great importance. [7]

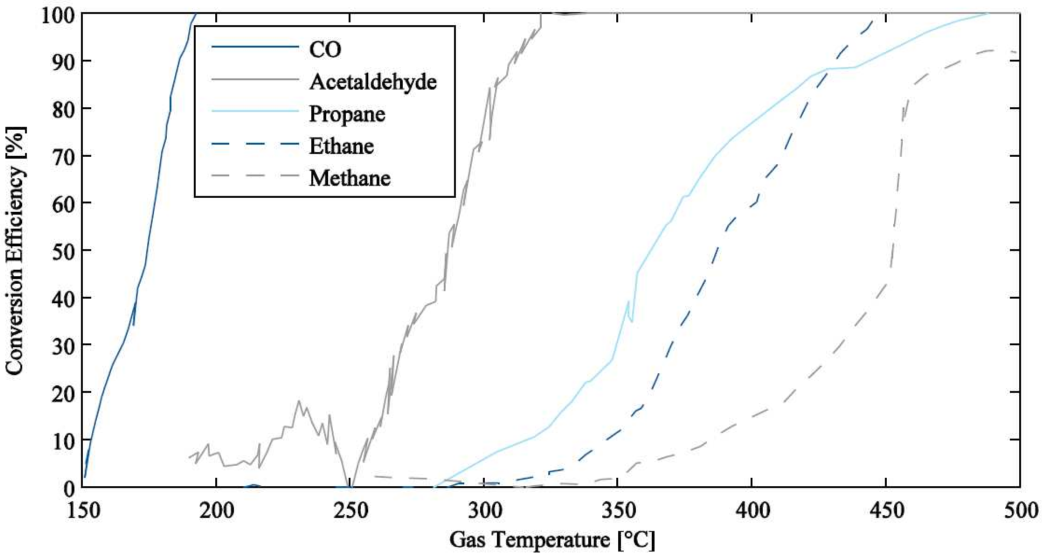

Figure 2.6: Conversion efficiency of an aged oxidation catalyst versus gas temperature for CO and different hydrocarbons [8]

Due to these characteristics, CH_4 is primarily used in SI engines. In light duty applications, these engines are usually bi-fuel engines; they must be able to operate on gasoline fuel as well. This limits the compression ratio, and effectively limits the efficiency of the engine as well. For medium and heavy duty applications, SI gas engines are typically mono-fuel, meaning that they can be optimized for methane operation, providing better efficiency. These engines still suffer from the known Achilles heel of the Otto engines, pumping losses and poor part load efficiency. The pumping losses can be addressed to some extent, by running the engine lean and by adding EGR. Lean operation of an SI engine is the source of many complications since the three-way catalyst can no longer be used to reduce NO_x and a lot of strain is put on the ignition system.

2.4.1. – Real applications of natural gas in engines

Nowadays, it is possible to find applications of CNG as fuel in different fields:

- Cars: Use of CNG as fuel in cars is increasing around the world. Several manufacturers, as Volvo, Ford, Honda, Mercedes-Benz and Volkswagen, among others, are already producing models that run on CNG. An example is the Volvo V60, shown in figure 2.7, which has a bi-fuel SI engine, able to run on either CNG or gasoline. [9] On the other hand, there is a growing number of cars that have been modified in order to use CNG as fuel of their engine.

Figure 2.7: Volvo V60 Bi-Fuel on CNG-Gasoline

- City buses: Use of CNG as fuel in city buses is growing all around the world due to their significant environmental and economic advantages. [10] They are not only in big cities, if not in smallest cities as well. Some manufacturers of CNG buses are MAN, Volvo or Mercedes-Benz. In addition, some manufactures now offers CNG hybrid buses.

Figure 2.8: A CNG powered Mercedes-Benz O530 Citaro bus, operated by Bergkvarabuss in Lund, Sweden.

- Transportation trucks: Nowadays there are progressively more trucks and vans used in transportation operated by CNG engines and Bi-fuel engines.

Figure 2.9: Freightliner Cascadia Natural Gas truck. [11]

- Marine applications: An example is Wärtsilä dual-fuel engines, which are already well-known in the marine market, and the technology they are based on is a daily reality for hundreds of owners all around the world. [12] One of these engines is the Wärtsilä 50DF [13], which has had a big success in the carrier market in the last years.

Figure 2.10: The LNG Carrier Gaselys was the first vessel to be powered by the Wärtsilä 50DF dual-fuel engine. [14]

- Stationary engines: A usual application of natural gas as fuel in stationary engines is in power plants for electricity generation. An example of these engines is the MAN 51/60DF engine shown below.

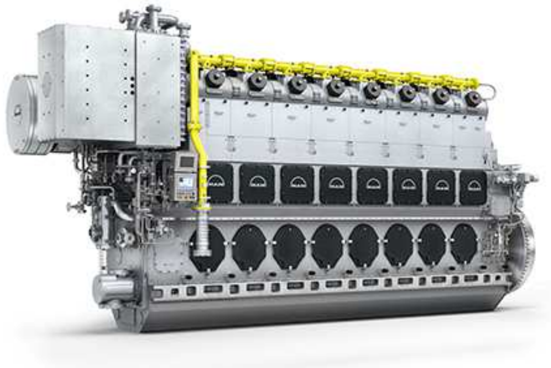

Figure 2.11: MAN 51/60DF stationary engine. [15]

2.5. – The Diesel-CNG Dual Fuel Engine

One solution to using methane in a CI engine is to introduce a pilot fuel with higher cetane number, diesel in this case, which initiates combustion and ignites the methane, thus giving rise to the diesel dual fuel, DDF, engine. For the application of dual-fuel operations a combination of CI and SI combustion concepts is applied. To be precise the primary fuel, CNG, is injected in the intake manifold and the mixture of fuel and air is compressed during the compression stroke. Since CNG is a fuel with a high octane number, it is resistant against auto ignition, and therefore, not likely to ignite during the compression stroke. At the end of it, close to TDC, diesel is injected, which spontaneously ignites due to its high cetane number, which makes it prone to auto ignition. The ignited diesel fuel actually acts as an ignition source, and ignites the available mixture of CNG and air. [16]

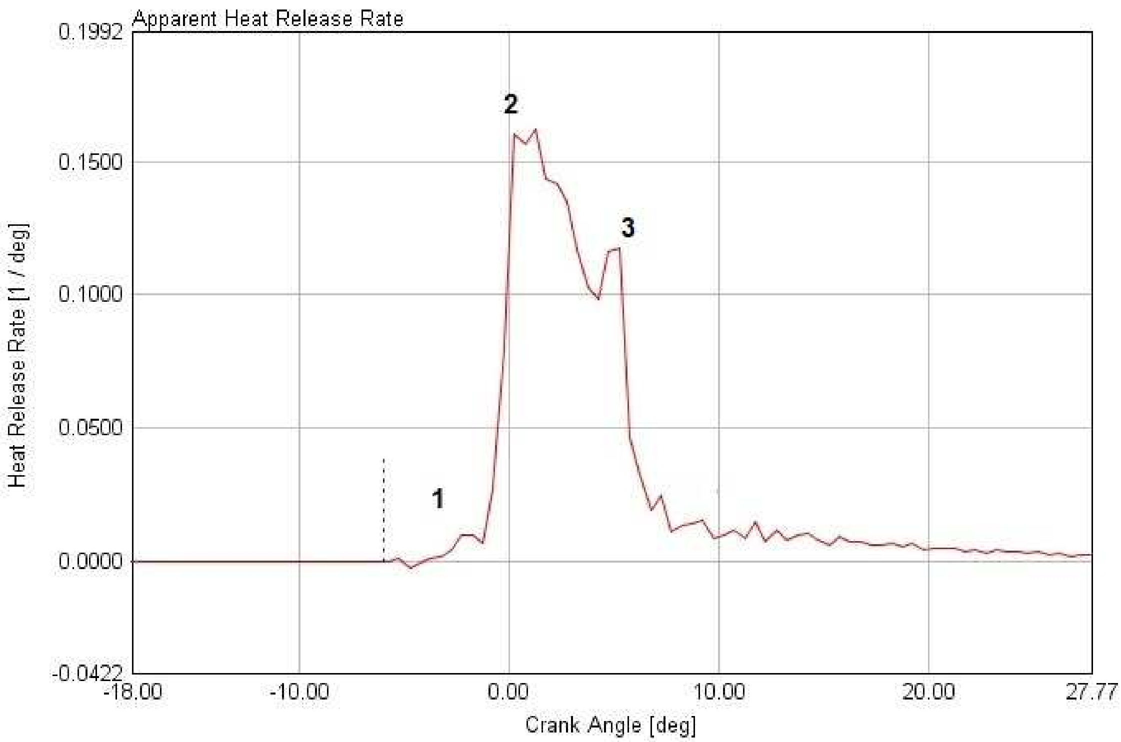

Figure 2.11: Rate of Heat Release, ROHR, for dual fuel combustion as a function of CAD. [15]

Analysis of the DDF combustion process divides the rate of heat release into three phases as is depicted in figure 2.11.

1. First the premixed combustion of the direct injected diesel fuel and a minor part of the CNG entrained in the diesel spray. Heat release rate in this point is usually very small.
2. The second phase is characterized by the premixed combustion of the major part of CNG and small amounts of diesel. This is the main combustion, where most part of heat is released.
3. Finally, the third phase of the combustion belongs to the flame propagation, where the rest of both fuels is burned.

It seems trivial that the importance of phase one and two is characterized by the amount of diesel substitution by CNG. However, this is not the case. The peak of the ROHR during the first phase seems to depend on the amount of pilot fuel that can be burned during this phase. Only when the amount of pilot fuel is decreased below this certain limit the importance of this phase decreases. The importance of the second phase on the other hand is determined by the amount of diesel substitution.

The advantage of this concept is that it makes use of the difference in flammability of the used fuels. When leaving out the CNG the engine operates as a normal diesel engine. However, since diesel is necessary for ignition it is not possible to run only on CNG.

Several strategies for operating a compression ignition engine under dual fuel mode has been developed and evaluated. They are distinguished by the way the gaseous fuel is introduced in the cylinder and the amount of diesel fuel that is displaced by it. The two most important strategies are presented below.

- <u>Conventional Dual Fuel Strategy:</u> CNG is fumigated in the intake manifold and the mixture of gas and air is compressed in the compression stroke which is then ignited due to auto ignition of the diesel fuel spray. The quantity of diesel in this case can vary from 100% to 20% and this strategy is preferable on an engine which does not have electronically controlled diesel injection. During starting and full load conditions, the engine runs with 100% diesel as a safety measure to avoid knocking, whereas under part load conditions it runs in DDF mode.

- <u>Pilot Ignited Natural Gas Diesel, PINGD, Strategy:</u> In this case, CNG is used as the primary fuel and the pilot diesel injection serves as the ignition source only. CNG addition can be by manifold of in-cylinder direct injection. In later cases, high-pressure gas is injected after charged air is compressed as in normal diesel engines. This is also called the "late cycle gas injection" strategy. Due to the poor self-ignition quality of the gaseous fuel, a small amount of diesel pilot fuel is used to ignite the injected gas fuel. The gaseous fuel burns mostly in the premixed manner as in dedicated natural gas engine. The quantity of diesel is usually limited to 20% and hence this operating mode has been found to further reduce soot and NO_x as compared to the Conventional Dual Fuel operation, but the power drop is higher and hence running cost is high.

Comparing the two dual fuel strategies, the conventional duel fuel operation offers benefits in terms of economy and lower CO and HC, whereas the PINGD offers further reduction in $\mathrm{NO_x}$ and soot but with a compromise in terms of lower power output, higher specific fuel consumption and higher CO and HC emissions. [17]

3. – METHOD

In this chapter the complete work procedure to follow of this project is described. It may be divided in four main steps:

1. Modelling of the engine in GT-POWER.
2. Design of the experiment
3. Data setup and final calibration of engine model
4. Collection of necessary results

All this steps are detailed below.

3.1. – Modelling of the engine

3.1.1. – Engine

The model is based on a four-stroke Volvo D4 lab engine, shown in the figure 3.1, available at the Faculty of Engineering, LTH, at Lund University, in Sweden. It is a Diesel-CNG dual fuel engine, able to operate in a range of diesel substitution ratio from 0, i.e. only diesel operation, to values close to 1. It has four cylinders, with a total displacement of 2 litters. It is equipped with a high pressure common rail system for diesel injectors and CNG gas injectors placed in the intake runners. The engine has also a cooled EGR system, which will be included in the model, but will not be studied in this project.

Figure 3.1: Volvo lab engine. Diesel-CNG Dual Fuel Engine

Some of the featuring engine specifications are shown in table 3.1.

Feature	Value
Number of cylinders	4
Displacement volume (cm^3)	1969
Bore (mm)	82
Stroke (mm)	93.2
Connection rod length (mm)	147
Peak firing pressure (bar)	190
Number of valves/ cylinder	4
Operating compression ratio	15.8

Table 3.1: Engine specifications

The engine is equipped with multiple measurement and control systems, which are connected to a computer. These systems are controlled via LabVIEW based software developed at the Division of Combustion Engines at Lund University. High resolution data (every 0.2 CAD) is sampled via a NI PXI-FPGA 7854R. Slow signals are logged via a NI PXI-6221 multifunction data acquisition card while temperatures are measured with a NI PXI-4353 card. Diesel direct injectors, high pressure diesel pump and diesel common rail dump valve are controlled using NI 9751 Direct Injector Driver Module kits. CNG injectors are controlled via an "in-house built" peak and hold driver. [18]

It allows to control different operating parameters as injection, speed, common rail pressure or EGR ratio, and at the same time collecting measurements of this operation, as pressure traces, fuels and air flows and conditions, engine load or exhaust emissions. Thanks to it, it is possible to carry out the necessary experiments for getting the data for the model.

3.1.2. – 1-D Model building

The first step of engine modelling is to create the elemental structure basing on the real engine. In this case, the engine model is composed of the following elements:

- Intake and exhaust boundaries
- Intake manifold
- Exhaust manifold
- Valve ports
- Engine crank train
- Cylinders
- CNG port injectors
- Diesel direct injectors
- EGR system
- Pipes attached to intake and exhaust manifolds

The most of these elements are not created as a single element, but a combination of different elements. The GT-POWER created model is shown, in the figure 3.2, where elements are marked for a better understanding.

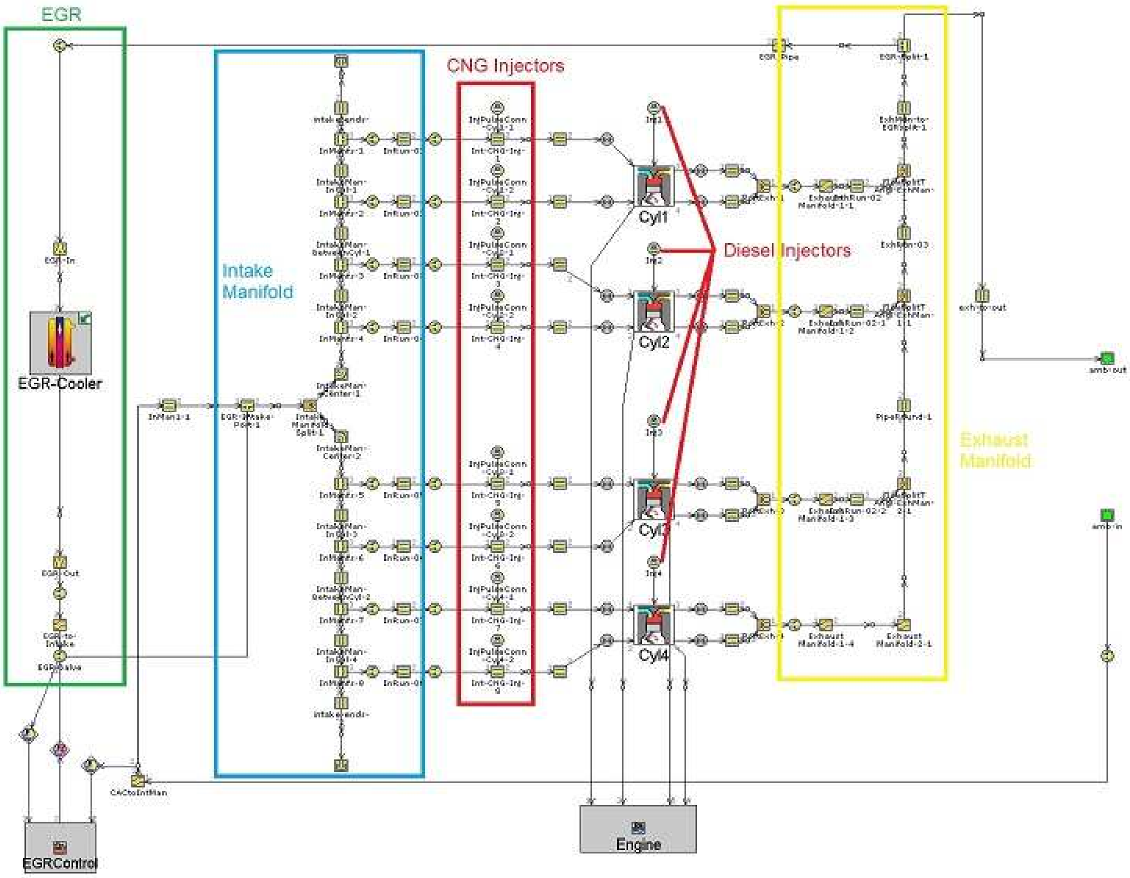

Figure 3.3: Map of the model

3.1.3. – General input data

Once the model is created, the next step is to set all the parameters that can be obtained without operating the engine. There is no need to measure for getting some of them, as engine or injectors specifications, since they are available. However there is need to measure and model other parameters:

- Intake and exhaust manifolds, EGR and attached pipes: Diameters, lengths and angle and radius of the bends have to be measured. In some cases, it´s necessary to calculate areas or volumes for a good setup, e.g. in flow split volumes.
- Valve ports specifications: Several parameters are necessary for well-modelled valve ports: diameters and positions in the cylinder, which can be easily measured, and intake and exhaust valve lifts, whose measurement is more complex. Valve lifts curves and opening and closing timings are shown in figure 3.3 and table 3.2.

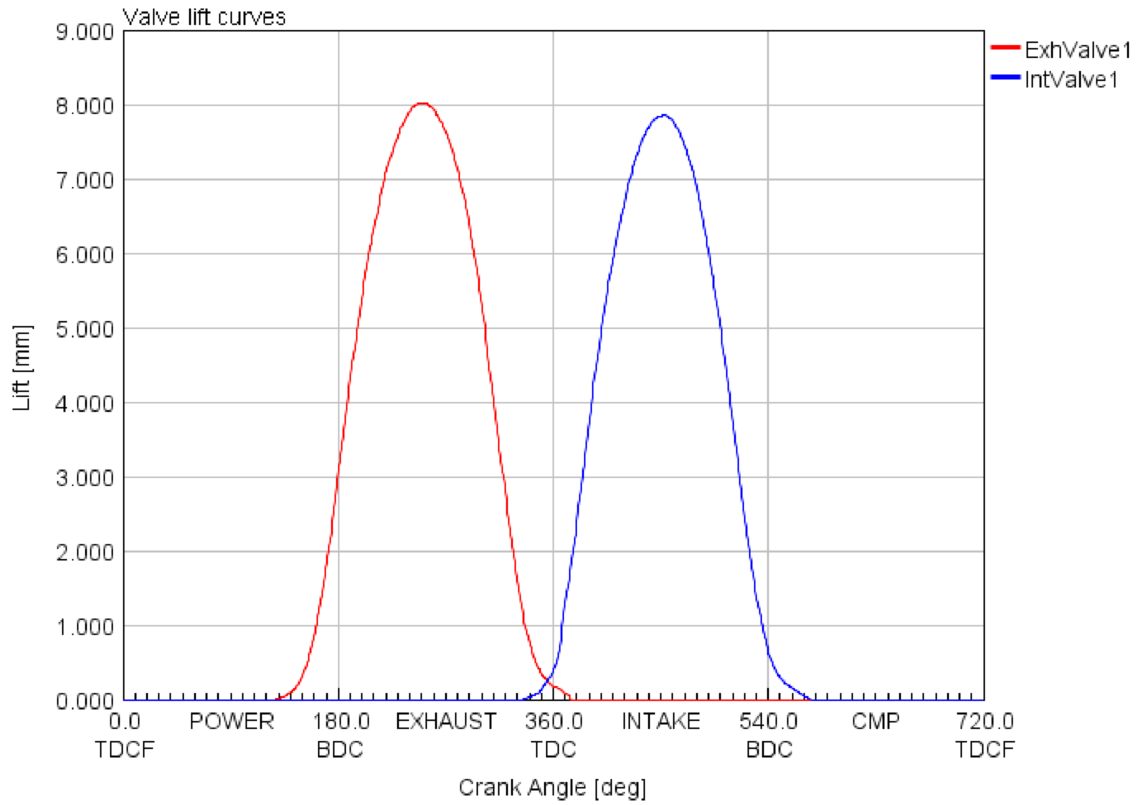

Figure 3.4: Valve lifts

Feature	Value
EVO (CAD ATDC)	128
EVC (CAD ATDC)	375
IVO (CAD ATDC)	335
IVC (CAD ATDC)	-144

Table 3.2: Valve lifts opening and closing timings

3.1.4. – Fuel modelling

Modelling fuel characteristics is very difficult since there is not an exact composition for each fuel, and then chemical and thermodynamic properties cannot be easily modelled. For this reason, it was decided to use the default diesel properties that GT-POWER provides, shown in table 3.3.

Property	Value
Carbon atoms per molecule	13.5
Hydrogen atoms per molecule	23.6
Lower Heating Value Q_{LHV} (MJ/kg)	43.25

Table 3.3: Diesel properties used in model [1]

However, GT-POWER does not provide a default model of CNG, or even natural gas. It has been modelled following the properties of CNG distributed in Sweden, provided by gas company Swedegas [19], with the objective of the best possible precision with respect to the experiments. As seen in the chapter, CNG is mainly composed of methane, but also ethane, propane and other gases in less proportion. CNG composition is shown in table 3.4, and its properties calculated from this composition and used in the model are shown in table 3.5.

Component	Molar fraction (%)
Methane CH_4	89.26%
Ethane C_2H_6	5.9%
Propane C_3H_8	2.37%
I-Butane C_4H_{10}	0.55%
Nitrogen N_2	0.35%
Carbon dioxide CO_2	0.93%
Other gases	0.64%

Table 3.4: CNG composition provided by Swedegas [18]

Property	Value
Carbon atoms per molecule	1.14
Hydrogen atoms per molecule	4.24
Oxygen atoms per molecule	0.019
Nitrogen atoms per molecule	0.007
Lower Heating Value Q_{LHV} (MJ/kg)	50.3

Table 3.5: CNG properties calculated for model

3.1.5. – Combustion modelling

Once fuels are modelled, it is time for doing the same about combustion parameters. As stated before, these parameters can only be obtained through running the engine. However, the objective of this run is to get data that does not have any effect in the following simulations. For this reason, it was chosen a random experiment that would have been carried out previously. The next step was set combustion parameters up in the model for using it as a calibration model, although most of them will change in following simulations.

Some of the experimental data used are:

- Start of combustion
- Pressure and heat release rate traces of each cylinder
- Friction data
- Temperatures and pressures in the intake and exhaust
- Fuel flows and their injection parameters: timing, injected mass, delivery rate, temperature, etc.
- Coolant properties
- Levels of exhaust emissions

As said before, this data can be directly measured from the engine. In this case, the combustion modelling was made through a Three Pressure Analysis, TPA. TPA is carried out directly by GT-POWER. It requires three measured pressures: intake, cylinder and exhaust, hence the name. The main purpose of this type of simulation is to analyse the measurements in order to obtain a single combustion burn rate for each operating condition. [1]

3.1.6. – Run simulation and calibration

When all the parameters are set up in the model, the last steps are to run the simulation, analyse the results and finally calibrate the parameters needed for a better approach to experimental data, i.e. engine performance.

Some parameters that were changed for calibration were compression ratio or intake and exhaust conditions. As example, measured and simulated pressure traces before and after calibration of one of the cylinders are shown in figure 3.4.

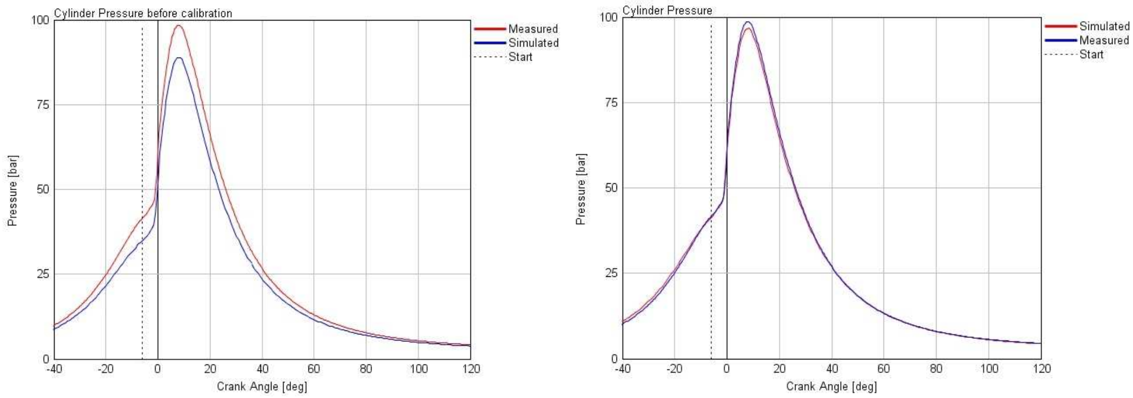

Figure 3.4: Differences between simulated and measured before (left) and after (right) calibration

3.2. – Design of the experiment

In this part, the planning of the research is made.

3.2.1. – Objective

The main objective of the experiment is to study how CNG distribution is affected by DDF operation in different working conditions. Especially, the objective is to research cylinder-to-cylinder variation produced by CNG distribution.

3.2.2. - Experimental design

Experiment is expected to be affected by several factors. If so, it is interesting to find out which factor has the strongest effect on the response, but another important question is if effects are separate or if the way in which the factors are combined is important. These both questions can be answered by full factorial experiments. [20] Then, this is the most suitable experimental design for this research.

It is also important to define the number of factors, i.e. number of different variables, and the level of the design, i.e. the number of different values for each factor. In this case, it has been carried out a two-level full factorial design with 3 factors, so the experiment is composed of $2^3 = 8$ cases. This type of design is represented in table 3.6.

A	B	C
-	-	-
+	-	-
-	+	-
+	+	-
-	-	+
+	-	+
-	+	+
+	+	+

Table 3.6: Two-level full factorial design with three factors

3.2.3. – Parameters definition

At first, factors have to be chosen looking for which ones are the most suitable for our objective. Looking at it, they were decided to be speed, CNG load and CNG injection timing. As CNG injectors have a fixed pressure, the factor will be CNG injection duration instead of load, which is easier to control and is going to represent the mass injected in the same way.

After this, it is time to define the levels, i.e. values, of these factors in the experiment:

- CNG injection duration: As a car engine, it is supposed to usually work at low and medium load. These conditions can be explained in terms of injection duration due to a constant injection pressure. Then, chosen values for low and medium load are 3250 and 8200 µs respectively.
- CNG injection timing: The main factor for defining the suitable values is the intake valve lift, in particular the opening timing. Looking at IVO, maximum lift and IVC timings, 375 CAD, 270 CAD and 144 CAD BTDC respectively, it was decided to use 2 symmetric values respect to maximum lift which can be considered early and late injection timings, but occurring while intake valve is still open. For this reason the

chosen values are 330 CAD and 210 CAD BTDC, corresponding to 60 CAD before and after maximum valve lift respectively.

- Speed: The usual operation of the engine is at low and medium speeds. So, the chosen values are 1250 and 2250 rpm.

The defined factors and their levels are shown in table 3.7.

Factor	A – CNG injection duration (µs)	B – CNG Injection timing (CAD)	C – Speed (rpm)
Level -	3200	330 BTDC	1250
Level +	8250	210 BTDC	2250

Table 3.7: Factors and levels of the experiment

3.2.4. Collection of experimental data

Finally the last step is to carry the experiment out in the real engine for collecting the experimental data that is needed for the simulation run, i.e. all new different parameters that we cannot simply set up without running the engine, e.g. pressure traces. It is desired that diesel does not affect cylinder variations. Therefore, before start of experiment, diesel injection timings and durations are calibrated in order to get no cylinder-to-cylinder variations in pressure traces. Moreover, in this case, there are two especial conditions for a good data collection, explained below.

- The first condition is that the experiment had to be made twice and in different days. The reason is that consecutives measurements tend to be more similar to each other than measurements made at different occasions, so they underestimate the effects of external noise.
- The second condition is about the order in which we are going to perform the cases. The reason of this condition is that some background results will remain unknown, and engine runs should not be affected by them. So the condition is that the order of the measurements has to be randomized and different for each occasion.

According to these conditions, experimental data is collected from the engine.

3.3. – Data setup in model and final calibration of the model

This step is a repetition of the last part of the engine modelling, but in this case the data used and the experimental cases are different. Once all setup is done, the simulation is run. Finally a last calibration is made for getting the desired results and finishing this step. Calibration result is shown through a comparison of measured and simulated pressure traces before and after the calibration in figure 3.6 as made in chapter 3.1.6. The most important parameters that have been calibrated are compression ratio, which has been reduced from 15.7 to 14.5 in all cases, and inlet and exhaust pressures, which values have varied on the scale of 0.01 bar with respect to experimental results and are different for each case, with values around 1.15 bar for inlet pressures and 1.3 bar for exhaust pressures.

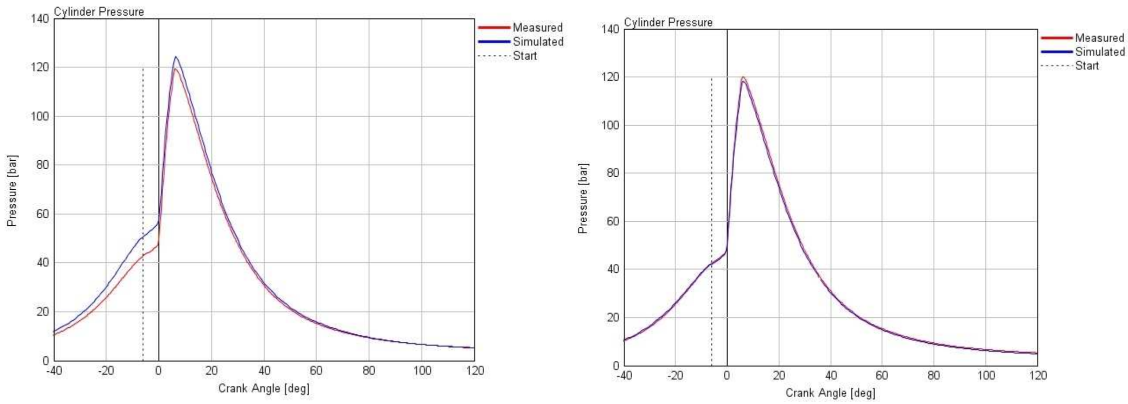

Figure 3.5: Differences between simulated and measured before (left) and after (right) final calibration in case 2

3.4. – Collection of final results

The famous physicist Lord Kelvin said that it is important for scientists to be able to measure what they are speaking of because “when you cannot express it in numbers, your knowledge is of a meagre kind”. The point of this is that, to provide an unambiguous answer to a research question, observations must be quantified at some level. Research requires a high degree of inter-observer correlation, meaning that other researchers must be able to repeat your experiment and come to the same result. [20] For this reason, an appropriate collection of final results is very important, and it requires data to be chosen and, in some cases, post-processed for a better interpretation.

3.4.1. – Choice of results

For a good choice of results, it is important to know which the research objective is. So, as a reminder of chapter 3.2.1, the research objective is to study the effects produced by CNG distribution. Then the chosen results should be able to answer to that question.

According to this, it has been decided that the results that should be analysed are, for each cylinder:

- Pressure trace and especially maximum in-cylinder pressure
- CNG in-cylinder flow
- Air in-cylinder flow
- Injection event
- Fuel flows in the intake manifold

In addition to cylinder parameters, engine parameters are analysed too.

3.4.2. – Post-processing of results

All the direct results are chosen. However, most of these values cannot give enough information by themselves and they need to be processed. As cylinder-to-cylinder variations are being studied, there is need to calculate a suitable value to quantify that variations. So the used value is the coefficient of variation, COV, defined by the equation shown below:

$$COV\ X = \frac{S}{\overline{X}}$$

Equation 3.1 Formula of coefficient of variation, COV

Where S is the standard deviation, defined in equation 3.2, and $\overline{X}$ is the mean.

$$S = \sqrt[2]{\frac{1}{N-1}\sum_{i=1}^{N}(X_i - \overline{X})^2}$$

Equation 3.2 Formula standard deviation S

Where

- N is the number of different values in each case, i.e. number of cylinders (4 cylinders)
- X_i is the defined value of cylinder i.

Lastly, as described in chapter 3.2.2, it is interesting to find out which factor has the strongest effect on the response, and if effects are separate or if the way in which the factors are combined is important. Therefore, a full factorial design is used, supplemented with columns for interactions, as shown in table 3.8.

	A	B	C	AB	AC	BC	ABC	Response
$\boldsymbol{M_{1_F}}$:	-1	-1	-1	1	1	1	-1	Y_1
$\boldsymbol{M_{2_F}}$:	1	-1	-1	-1	-1	1	1	Y_2
$\boldsymbol{M_{3_F}}$:	-1	1	-1	-1	1	-1	1	Y_3
$\boldsymbol{M_{4_F}}$:	1	1	-1	1	-1	-1	-1	Y_4
$\boldsymbol{M_{5_F}}$:	-1	-1	1	1	-1	-1	1	Y_5
$\boldsymbol{M_{6_F}}$:	1	-1	1	-1	1	-1	-1	Y_6
$\boldsymbol{M_{7_F}}$:	-1	1	1	-1	-1	1	-1	Y_7
$\boldsymbol{M_{8_F}}$:	1	1	1	1	1	1	1	Y_8
Divisor:	4	4	4	4	4	4	4	
Effect:	E_A	E_B	E_C	E_{AB}	E_{AC}	E_{BC}	E_{ABC}	

Table 3.8: The full factorial design supplemented with columns for interactions. [20]

The values in the table are:

- M_j is the response multiplier of case j for calculation of effect of factor or combination of factors F and is referred to factor level
- Y_j is the response of case j that is analysed
- Divisor is the number of measurements of each factor level, i.e. 4 measurements per level
- E_F is the effect of each factor or combination of factors F in the response, and it is calculated according to equation 3.3 defined below

$$E_F = \frac{1}{4} \sum_{j=1}^{8} M_{j_F} \cdot Y_j$$

Equation 3.3 – Formula for calculation of effects

4. – RESULTS AND DISCUSSION

This chapter is divided in two parts. In the first part the results of each case are presented and discussed in a separate way, while in the second part every case are analysed together and compared for a global discussion. Experiment cases are summarized in the following table:

Case	A – CNG injection duration (µs)	B – CNG Injection timing (CAD BTDC)	C – Speed (rpm)
1	3200	330 BTDC	1250
2	8250	330 BTDC	1250
3	3200	210 BTDC	1250
4	8250	210 BTDC	1250
5	3200	330 BTDC	2250
6	8250	330 BTDC	2250
7	3200	210 BTDC	2250
8	8250	210 BTDC	2250

Table 4.1: Summary of cases

4.1. - Case 1: Inj. Duration = 3200 µs; Inj. timing = -330 CAD; Speed = 1250 rpm

Case 1 shows engine operation with low CNG injection duration and early CNG injection at low speed. Pressure traces of the four cylinders are shown in figure 4.1.

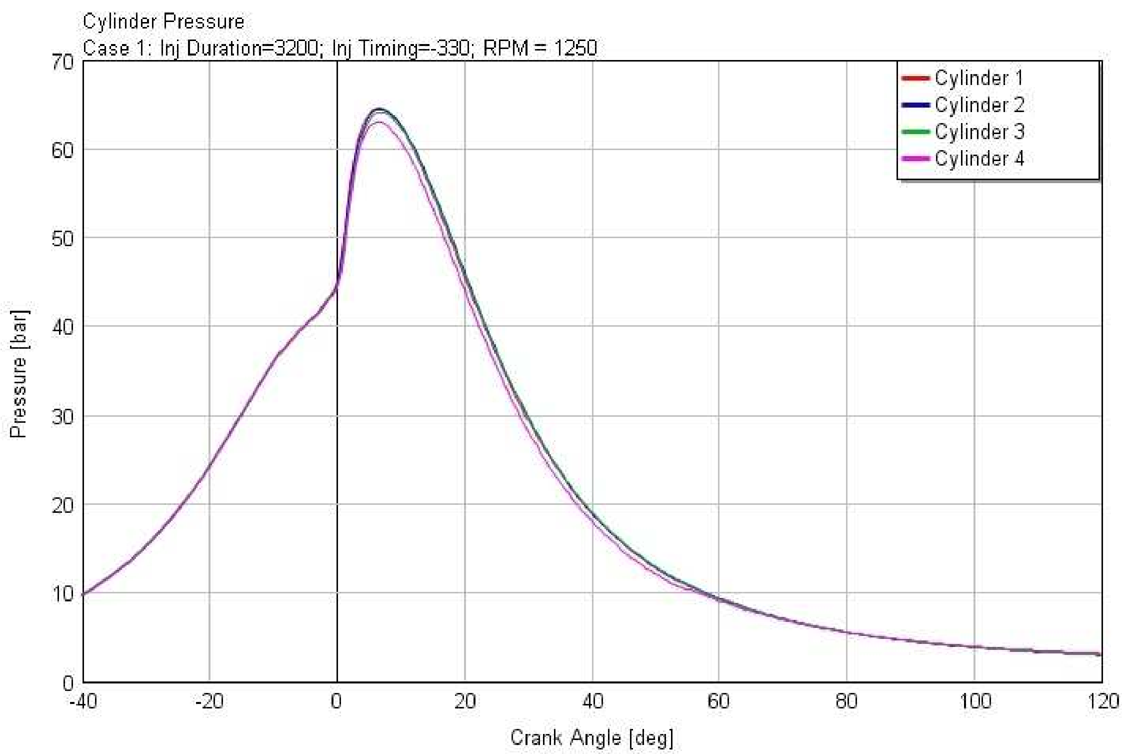

Figure 4.1: Cylinder pressure traces in case 1

Looking at the pressure traces there is no proof that there are high cylinder-to-cylinder variations in this case. For more information, mass flow rates of one of the intake valves of Cylinder 1 and its closer CNG injector are plotted in function of crank angle for a better understanding of CNG flow in figure 4.2.

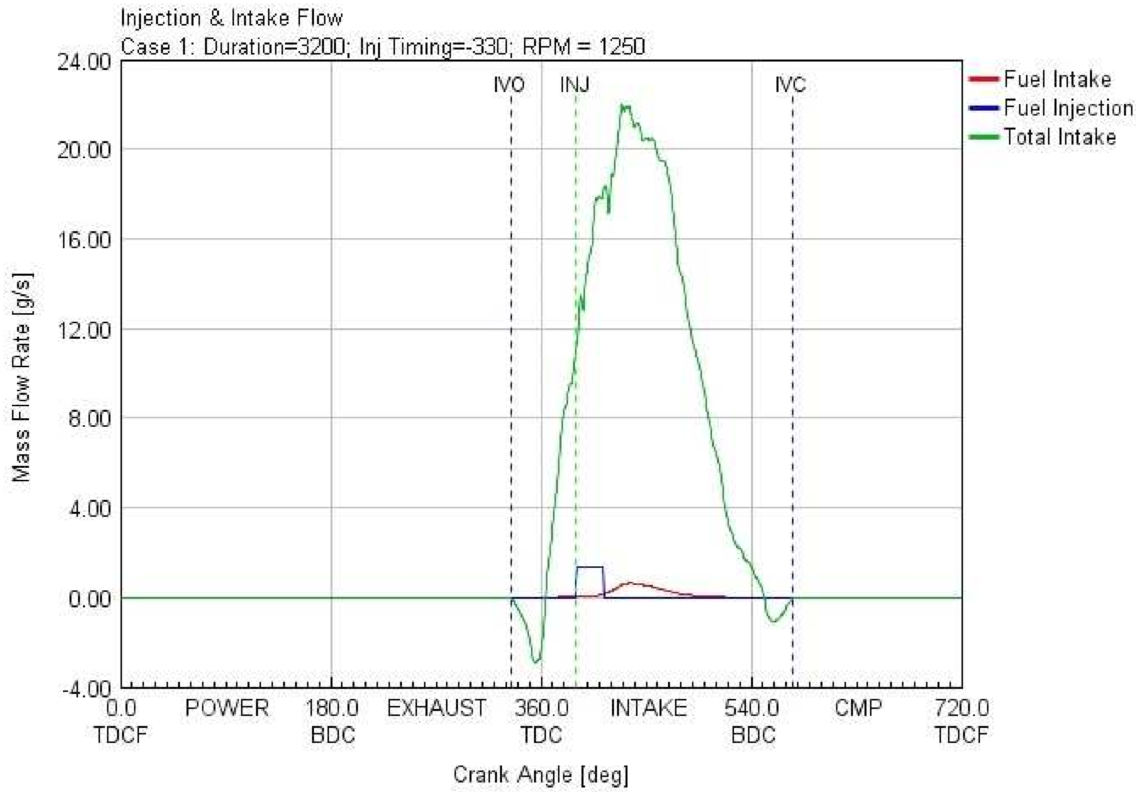

Figure 4.2: CNG injection and intake valve mass flow rates in Cylinder 1, case 1

According to the plot, it seems that all fuel injected is trapped by its own cylinder, and no fuel mass stays in the runners. In table 4.2 some cylinder information is shown.

Feature	Cylinder 1	Cylinder 2	Cylinder 3	Cylinder 4	COV
Maximum pressure (bar)	64,6	64,5	64,2	62,9	1,20%
Trapped mass at IVC (mg)	617	616	616	616	0,54%
Trapped air mass at IVC (mg)	568	567	567	568	0,55%
Trapped fuel mass at IVC (mg)	8,74	8,75	8,74	8,74	0,05%
AFR at IVC	64,9	64,9	64,9	65,0	0,54%

Table 4.2: Cylinder data in case 1

As expected, there are no significant differences in the trapped fuel mass of each cylinder, as quantified by COV, so there are no variations between cylinders.

4.2. - Case 2: Inj. Duration = 8250 μs; Inj. timing = -330 CAD; Speed = 1250 rpm

Case 2 shows engine operation with long CNG injection duration and early CNG injection at low speed. Pressure traces of the four cylinders are shown in figure 4.3.

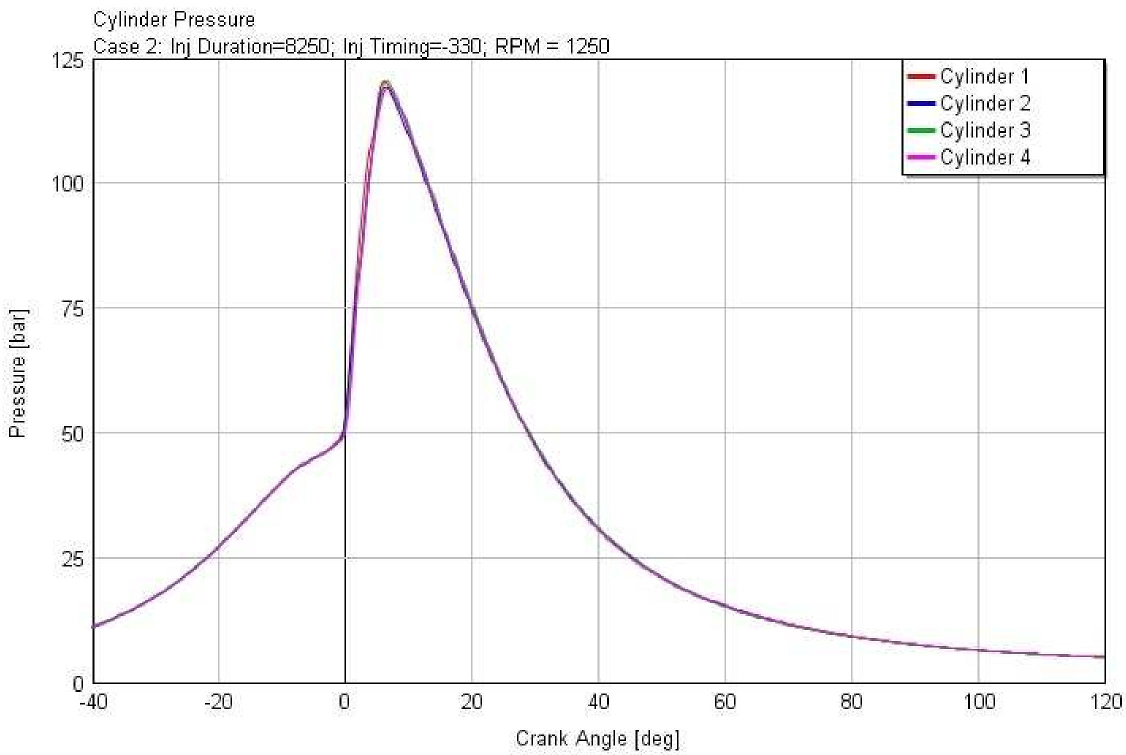

Figure 4.3: Cylinder pressure traces in case 2

According to the pressure traces, all of them are almost the same. In this case, no cylinder-to-cylinder variations can be expected. As done in case 1, mass flow rates of one of the intake valves of Cylinder 1 and its closer CNG injector are also plotted in figure 4.4.

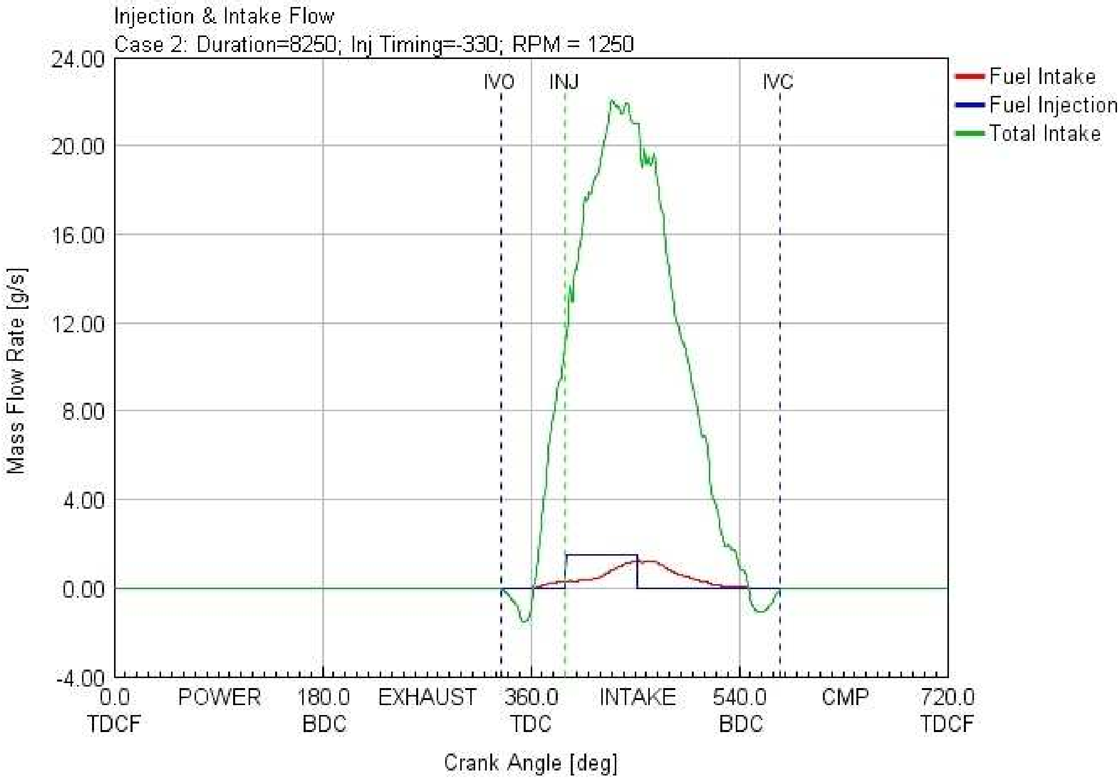

Figure 4.5: CNG injection and intake valve mass flow rates in Cylinder 1 case 2

In the figure it is able to see that all fuel is injected before the maximum instantaneous mass flow in intake valve, but there is a low fuel flow before start of injection. It is possible that some fuel is staying in the intake runners and going into the cylinder in the next cycle. For that reason, some cylinder information, shown in table 4.3, could be important to find out if there are differences in the cylinder contents.

Feature	Cylinder 1	Cylinder 2	Cylinder 3	Cylinder 4	COV
Maximum pressure (bar)	119	118	118	117	0,45%
Trapped mass at IVC (mg)	611	611	611	613	0,11%
Trapped air mass at IVC (mg)	559	559	559	561	0,16%
Trapped fuel mass at IVC (mg)	25,2	25,1	25,1	25,2	0,10%
AFR at IVC	22,2	22,3	22,3	22,3	0,13%

Table 4.3: Cylinder data in case 2

Analysing trapped mass, it that a low injected fuel mass could not be inducted into the cylinder, but it is in the next cycle, so there are no variations in the cylinders.

4.3. - Case 3: Inj. Duration = 3200 μs; Inj. timing = -210 CAD; Speed = 1250 rpm

Case 3 shows engine operation with low CNG injection duration and late CNG injection at low speed. Pressure traces of the four cylinders are shown in figure 4.5.

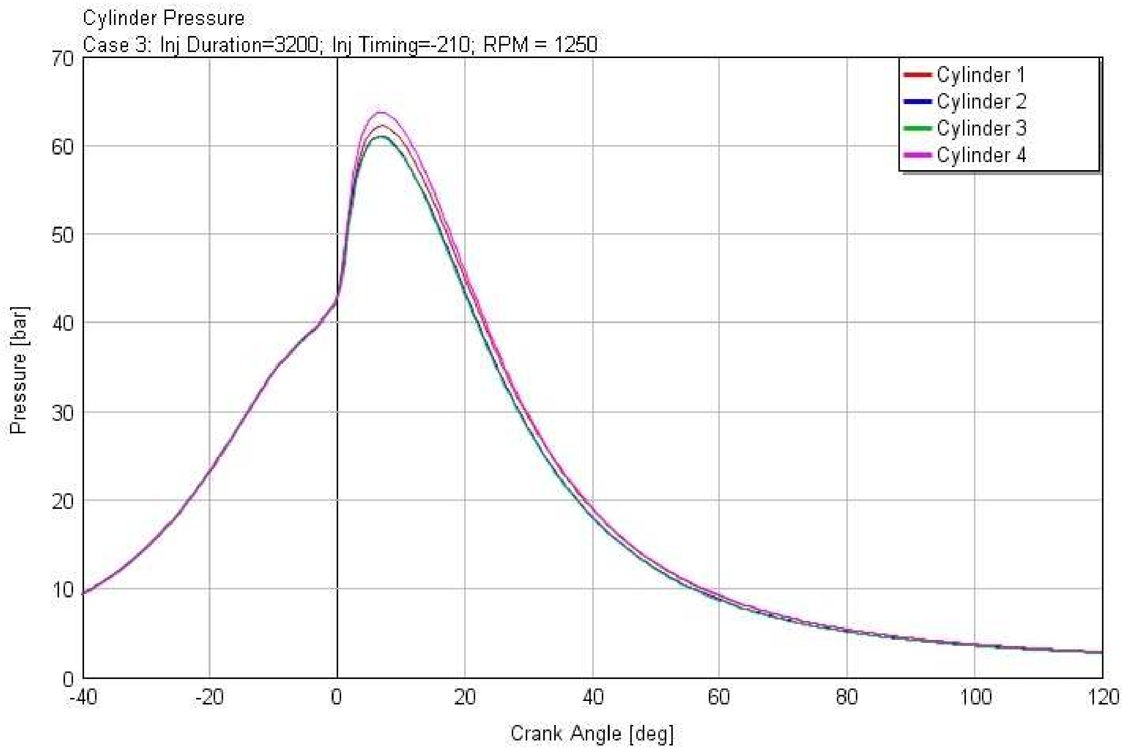

Figure 4.6: Cylinder pressure traces in case 3

In this case, pressure traces show a little variation as far as combustion is referred. Then it is expected that differences in trapped mass by each cylinder. Figure 4.6 represents mass flows in one of the intake valves of cylinder 1, and fuel injection in its attached pipe.

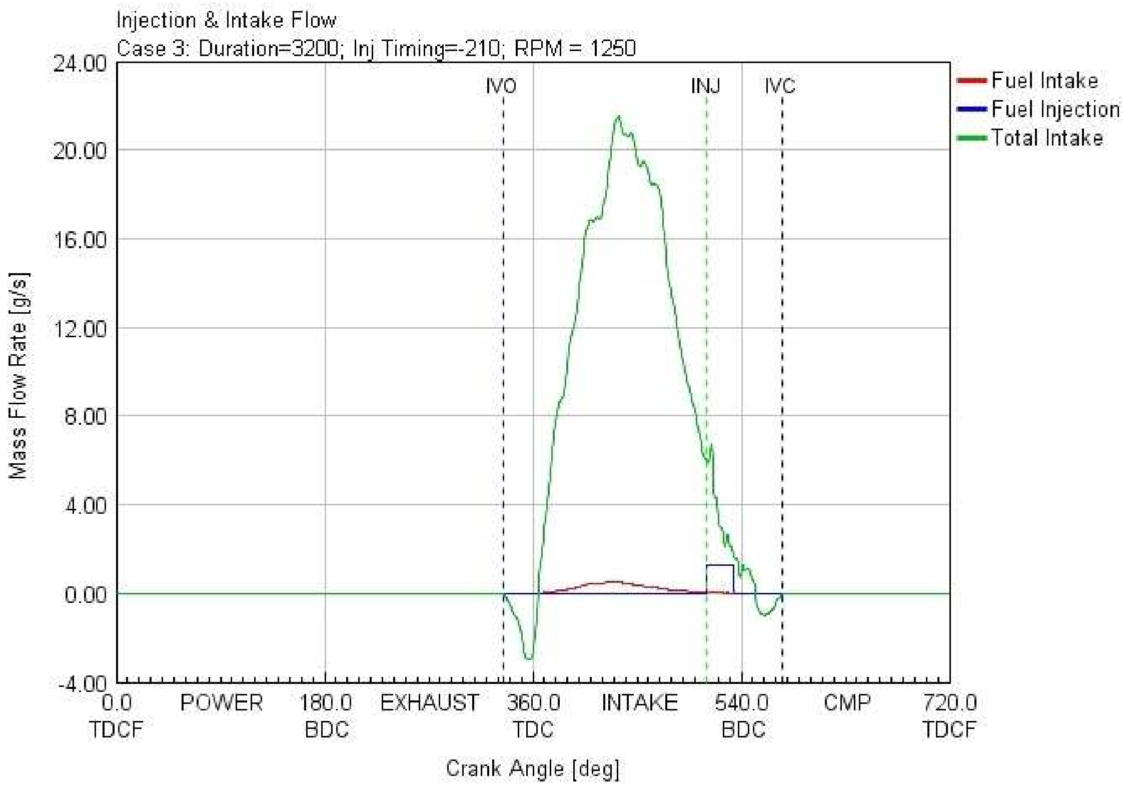

Figure 4.7: CNG injection and intake valve mass flow rates in Cylinder 1, case3

As exposed before, figure 4.6 shows that fuel mass inducted into the cylinder is not the mass injected in that cycle. Then it could explain differences seen in figure 4.5.

Feature	Cylinder 1	Cylinder 2	Cylinder 3	Cylinder 4	COV
Maximum pressure (bar)	62.2	61.1	61.0	63.8	2,08%
Trapped mass at IVC (mg)	587	591	592	585	0,54%
Trapped air mass at IVC (mg)	538	541	542	536	0,55%
Trapped fuel mass at IVC (mg)	9,06	7,78	7,77	9,08	8,86%
AFR at IVC	59,4	69,6	69,7	59,0	9,39%

Table 4.4: Cylinder data in case 3

As expected, cylinder contents shown in table 4.4 are different for each cylinder. Higher fuel mass and lower air mass, i.e. lower AFR, in cylinders 1 and 4 explain why pressures reached due to combustion are higher than in cylinders 2 and 3. There is an interaction between fuel quantities that should go into each cylinder that will be studied from a general point of view in chapter 4.10.

4.4. - Case 4: Inj. Duration = 8250 µs; Inj. timing = -210 CAD; Speed = 1250 rpm

Case 4 shows engine operation with long CNG injection duration and late CNG injection at low speed. After studying case 3, it may be expected that most of injected fuel mass in this case will also be too late, producing interactions in the intake manifold and, consequently, variation in the cylinders. At first, pressure traces of the four cylinders are shown in figure 4.7.

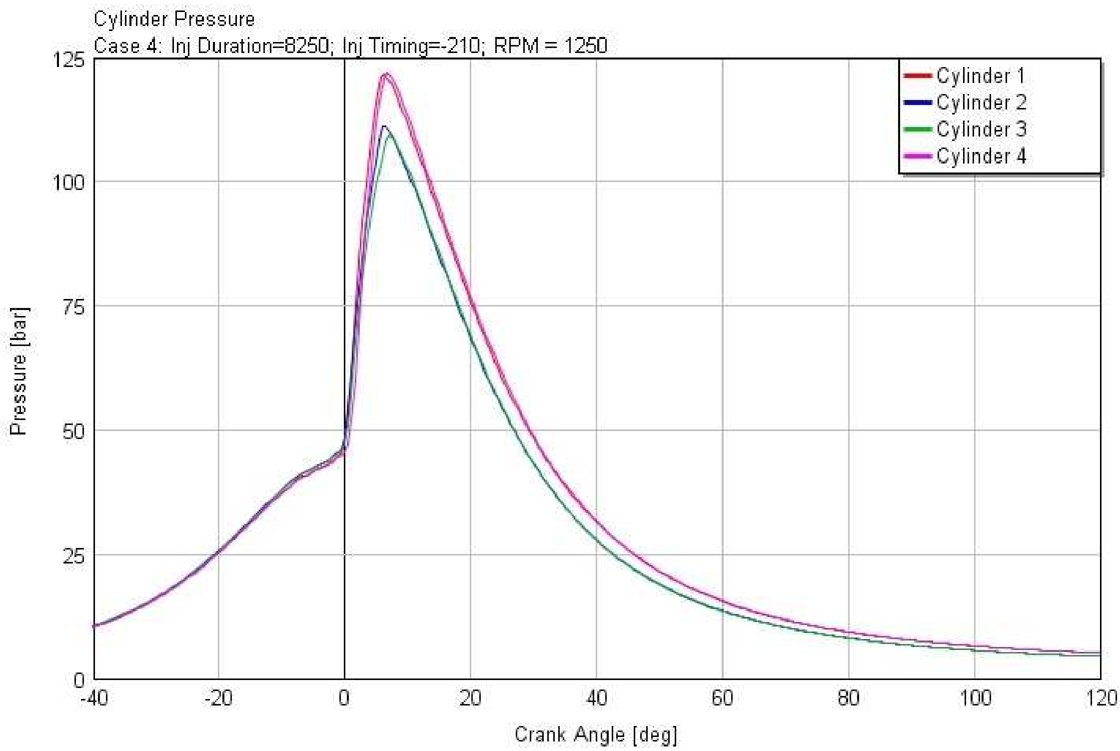

Figure 4.8: Cylinder pressure traces in case 4

This figure shows a significant difference between cylinder 1 and 4 and cylinders 2 and 3. These differences are similar to those seen in case 3, then it seems that both events could be explained in the same way.

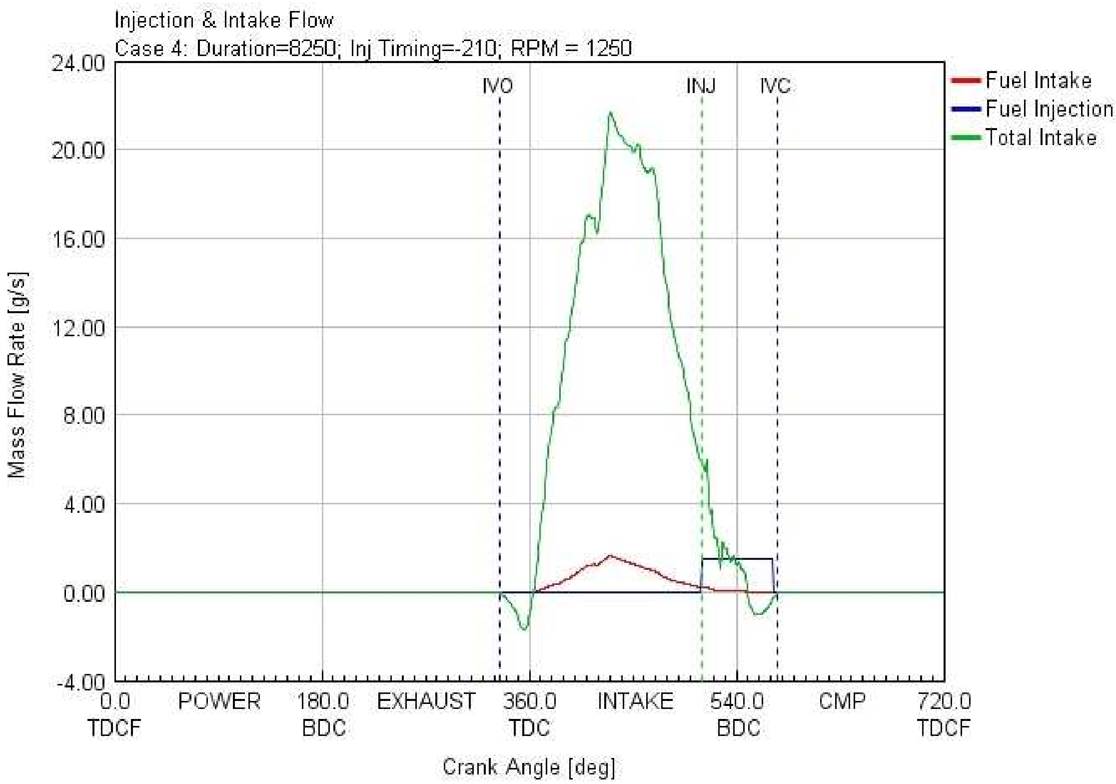

Figure 4.9: CNG injection and intake valve mass flow rates in Cylinder 1, case 4

Figure 4.8 shows that injected fuel mass is not inducted into the cylinder during its corresponding cycle, as in case 3, so it can stay in the intake runners or even be directed to other cylinders. As said, it will be studied in a general way in chapter 4.9. Cylinder information of this case is shown in table 4.5

Feature	Cylinder 1	Cylinder 2	Cylinder 3	Cylinder 4	COV
Maximum pressure (bar)	122	111	109	122	5,87%
Trapped mass at IVC (mg)	577	609	610	577	3,12%
Trapped air mass at IVC (mg)	523	557	558	523	3,70%
Trapped fuel mass at IVC (mg)	28,3	21,2	21,2	28,3	16,5%
AFR at IVC	18,5	26,2	26,3	18,5	20,1%

Table 4.5: Cylinder data in case 4

Trapped mass information shown in the table demonstrates that the assumptions are right: there are big differences in the mass and air mass flows that go into the cylinder, producing a richer combustion in cylinders 1 and 4.

4.5. - Case 5: Inj. Duration = 3200 μs; Inj. timing = -330 CAD; Speed = 2250 rpm

Last five cases show the same engine operation at higher speeds, then it can be assumed that similar events occur in each case, but with possible effects due to speed. Case 5 shows engine operation with low CNG injection duration and early CNG injection at high speed. Its pressure traces are plotted in figure 4.9.

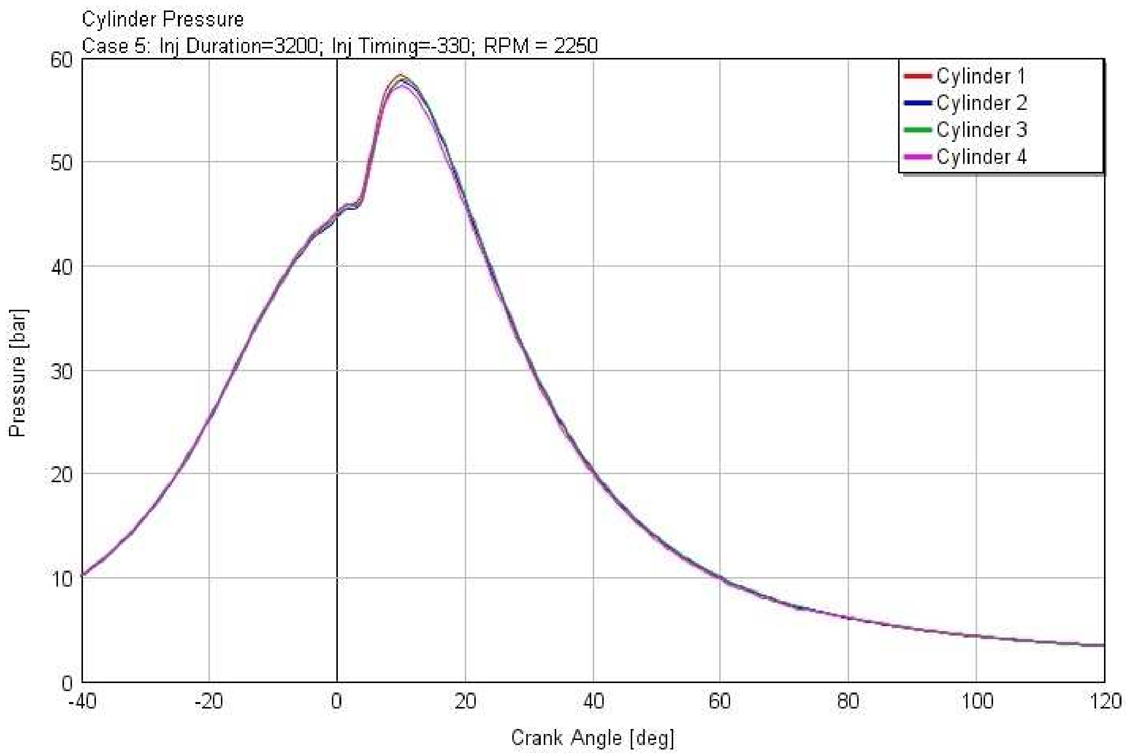

Figure 4.10: Cylinder pressure traces in case 5

In this figure, a little difference, but not necessarily significant, is seen between pressures during combustion. There is need for more information to prove if there are variations. In figure 4.10 injection and intake events are represented.

Figure 4.11: CNG injection and intake valve mass flow rates in Cylinder 1, case 5

According to the plot, events are similar to those seen in case 1: it seems that all fuel injected is trapped by its own cylinder, and no fuel mass stays in the runners, then no difference in cylinder contents is expected, as can be analysed from table 4.6. The table shows that there is no difference in trapped fuel mass by each cylinder, but there is a little difference in air fuel mass, which could explain the low variation in maximum pressures.

Feature	Cylinder 1	Cylinder 2	Cylinder 3	Cylinder 4	COV
Maximum pressure (bar)	58,4	57,8	57,9	57,3	0,75%
Trapped mass at IVC (mg)	636	631	631	635	0,37%
Trapped air mass at IVC (mg)	588	584	584	589	0,43%
Trapped fuel mass at IVC (mg)	8,46	8,43	8,44	8,46	0,16%
AFR at IVC	69,5	69,3	69,2	69,6	0,28%

Table 4.6: Cylinder data in case 5

4.6. - Case 6: Inj. Duration = 8250 µs; Inj. timing = -330 CAD; Speed = 2250 rpm

Case 6 shows engine operation with long CNG injection duration and early CNG injection at high speed, i.e. same operation than case 2 at high speed. According to this, big cylinder-to-cylinder variations are not expected, but speed could produce them in a lower scale. Pressure traces are shown in figure 4.11.

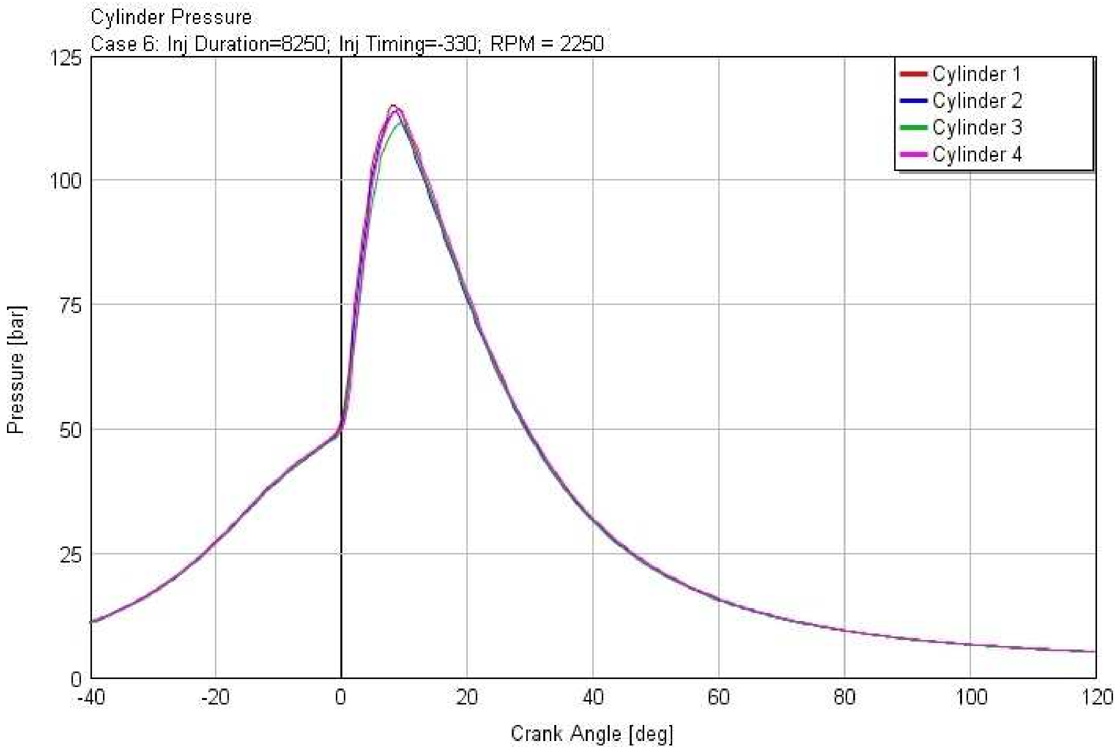

Figure 4.12: Cylinder pressure traces in case 6

According to this graphic, all of them are almost the same. As assumed, behaviour is similar to that seen in case 2. No cylinder-to-cylinder variations can be expected. Mass flow rates of injection and intake events of cylinder 1 are plotted in figure 4.12.

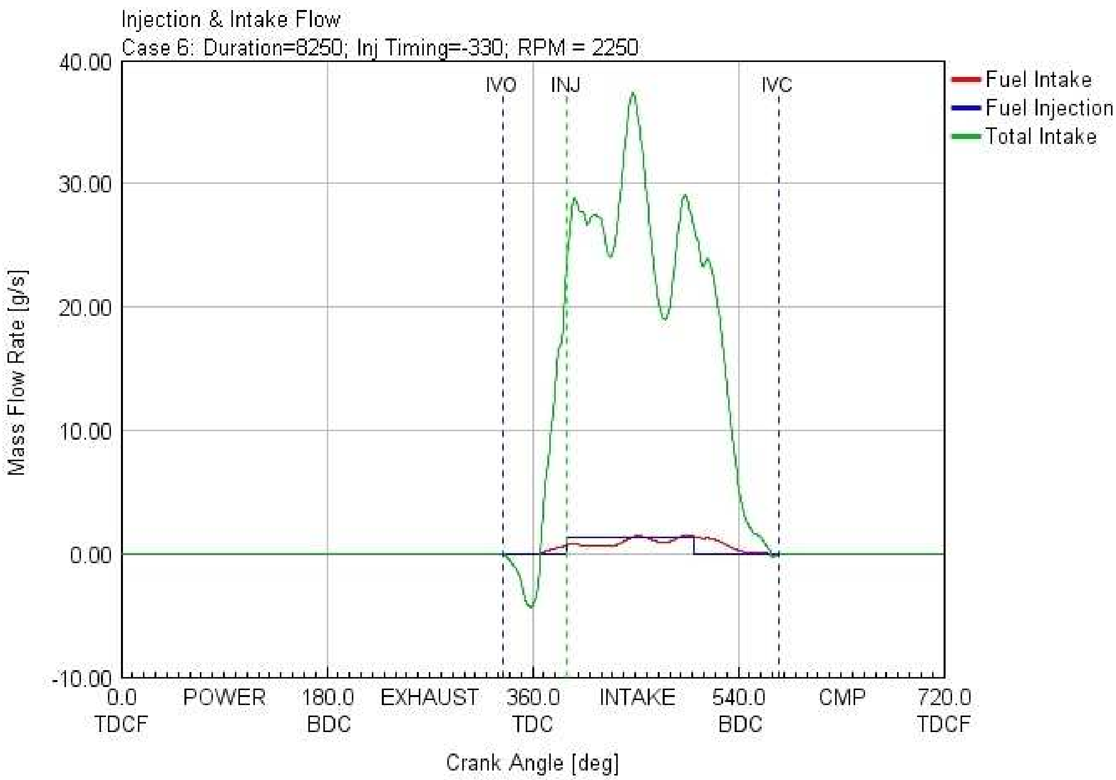

Figure 4.13: CNG injection and intake valve mass flow rates in Cylinder 1, case 6

According to the plot, and as case 2, there is some fuel mass flow before start of injection, but in this case it is able to see a second peak which shows that part of the injected mass is being introduced in the cylinder.

Feature	Cylinder 1	Cylinder 2	Cylinder 3	Cylinder 4	COV
Maximum pressure (bar)	115	114	111	114	1,41%
Trapped mass at IVC (mg)	626	622	623	625	0,27%
Trapped air mass at IVC (mg)	571	568	568	570	0,29%
Trapped fuel mass at IVC (mg)	22,7	22,1	22,1	22,7	1,71%
AFR at IVC	25,1	25,7	25,7	25,1	1,44%

Table 4.7: Cylinder data in case 6

Data collected in table 4.7 show that not all the injected mass is trapped by the closest cylinder to the injector, but it could be considered almost insignificant if cylinder pressures are analysed. However, it is an evidence of speed effect on CNG distribution as far as it is compared to same conditions at low speed, i.e. case 2.

4.7. - Case 7: Inj. Duration = 3200 µs; Inj. timing = -210 CAD; Speed = 2250 rpm

Case 7 shows engine operation with low CNG injection duration and early CNG injection at high speed. Then it presents same injection conditions than case 3 with higher speed, which could increase cylinder-to-cylinder variations. Pressure traces of this case are shown in figure 4.13.

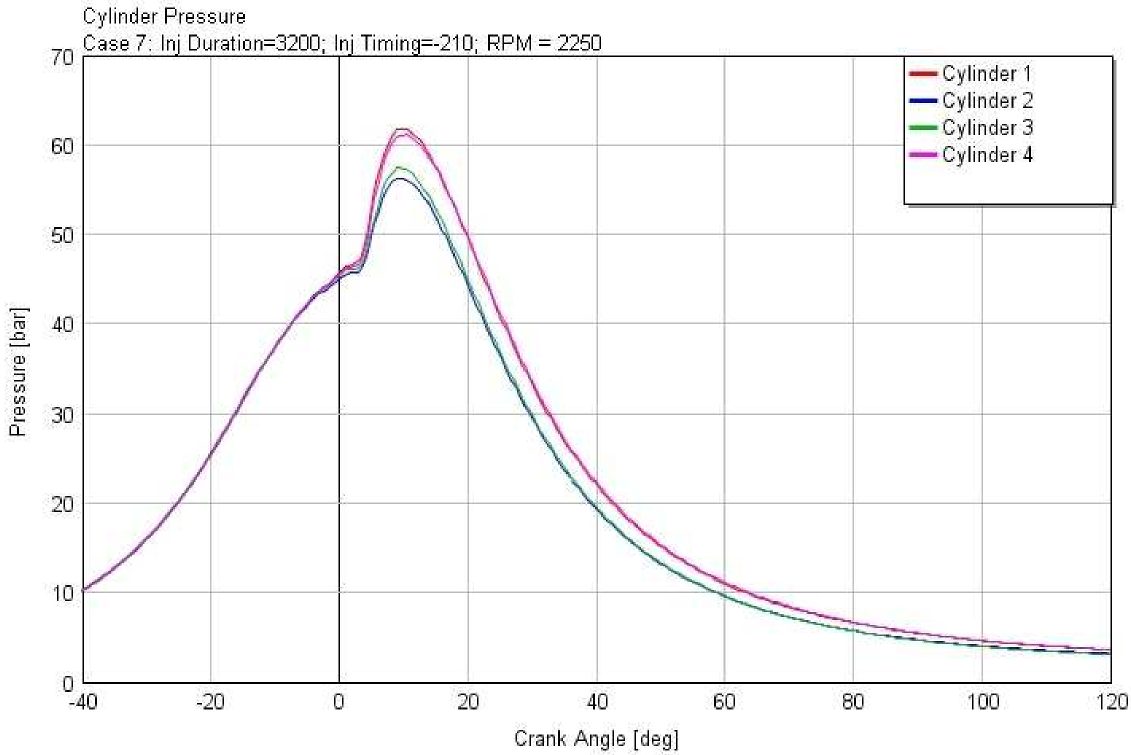

Figure 4.14: Cylinder pressure traces in case 7

In a first look, it can be seen that there is a big difference between pressure traces, where pressures in cylinders 1 and 4 are higher than pressures in cylinders 2 and 3. This difference seems to be bigger in relation to case 3.

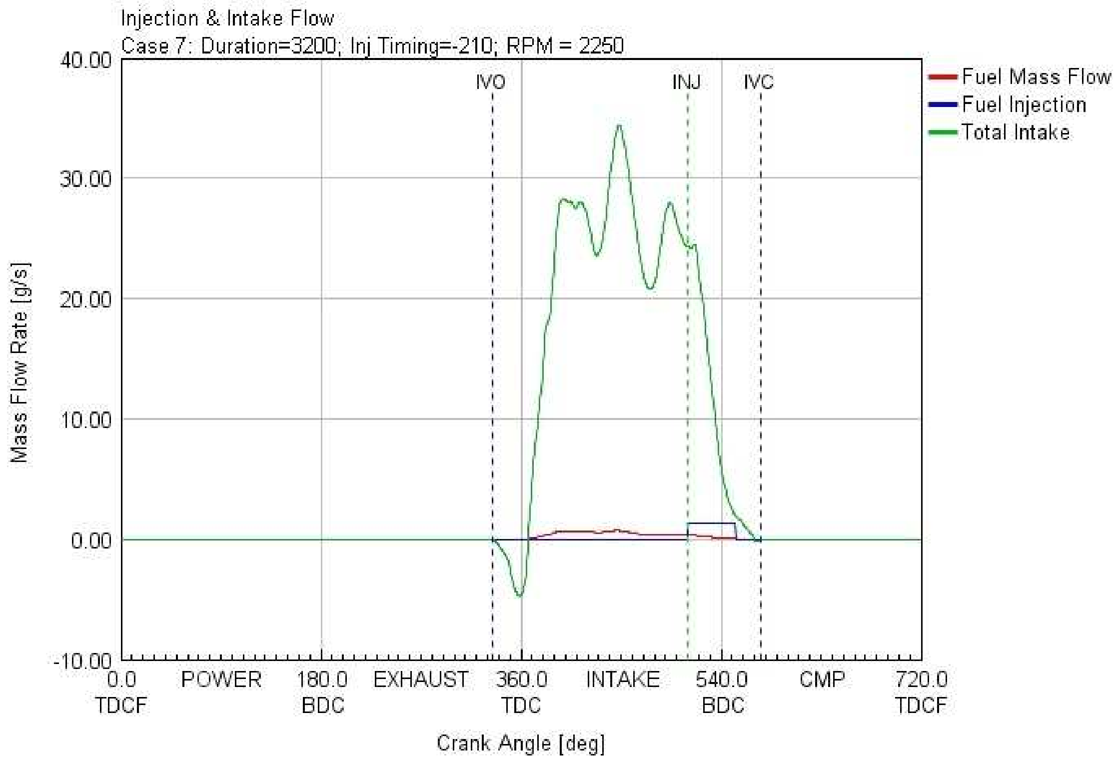

Figure 4.15: CNG injection and intake valve mass flow rates in Cylinder 1, case 7

On the other hand, figure 4.14 shows that most of fuel intake occurs even before start of injection. Then, as it happens in case 3, most of the fuel is not injected and inducted into the cylinder in the same cycle of the closest cylinder, which produces a fuel mass exchange between cylinders.

Feature	Cylinder 1	Cylinder 2	Cylinder 3	Cylinder 4	COV
Maximum pressure (bar)	61,8	56,3	57,5	61,1	4,56%
Trapped mass at IVC (mg)	632	639	639	632	0,63%
Trapped air mass at IVC (mg)	584	589	588	584	0,43%
Trapped fuel mass at IVC (mg)	9,96	6,94	6,95	9,94	20,5%
AFR at IVC	58,6	84,9	84,5	58,8	20,9%

Table 4.8: Cylinder data in case 7

Cylinder contents collected in table 4.8 show that these differences in fuel flow in the intake manifold are happening. Moreover, it is happening in a higher scale than in case 3, apparently due to effect of higher speed.

4.8. - Case 8: Inj. Duration = 8250 μs; Inj. timing = -210 CAD; Speed = 2250 rpm

Case 8 represents the most extreme case of this experiment: long CNG injection duration and late CNG injection at high speed. As a first analysis, previous cases, especially cases 4 and 7, can help to predict that cylinder-to-cylinder variations are going to be higher than in the other cases, especially because of a higher fuel mass in cylinders 1 and 4 than in cylinders 2 and 3. Pressure traces are plotted in figure 4.15.

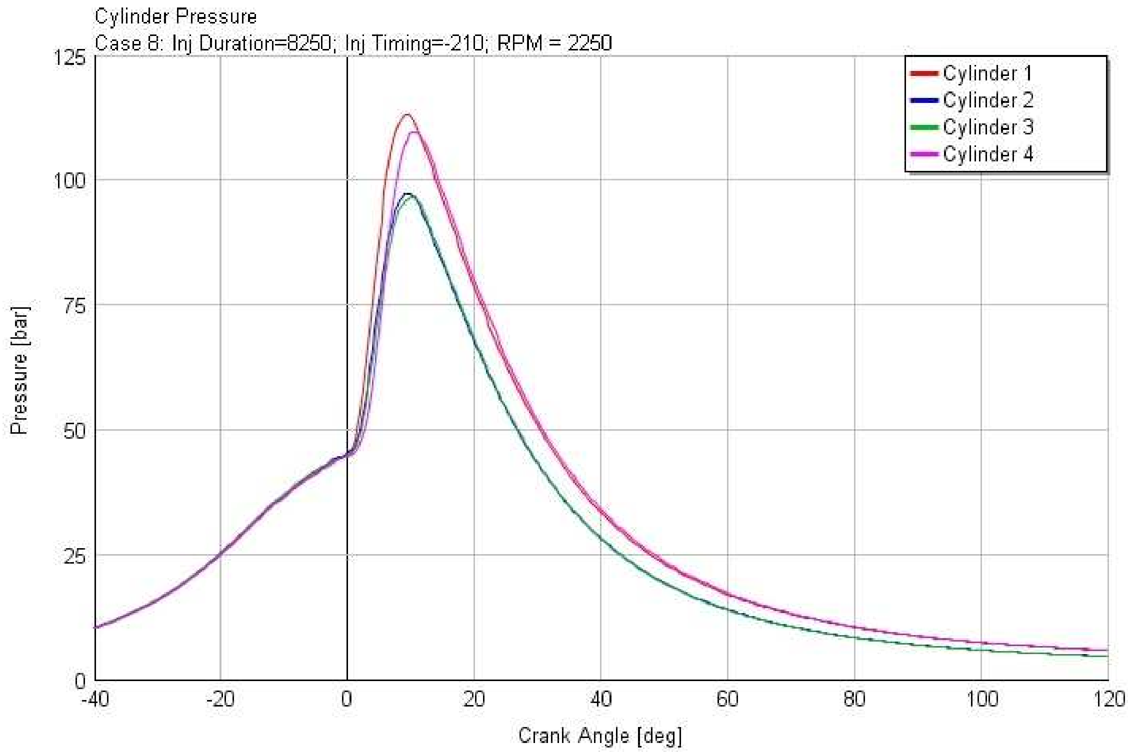

Figure 4.16: Cylinder pressure traces in case 8

As expected, there is a big difference between pressures in the external cylinders, 1 and 4, and internal cylinders, 2 and 3, such as happened in other cases which show cylinder-to-cylinder variations.

Feature	Cylinder 1	Cylinder 2	Cylinder 3	Cylinder 4	COV
Maximum pressure (bar)	113	97,4	96,6	110	8,1%
Trapped mass at IVC (mg)	567	596	596	567	2,9%
Trapped air mass at IVC (mg)	516	547	547	516	3,4%
Trapped fuel mass at IVC (mg)	27,1	17,8	17,8	27,1	23,8%
AFR at IVC	19,1	30,2	30,7	19,0	27,1%

Table 9: Cylinder data in case 8

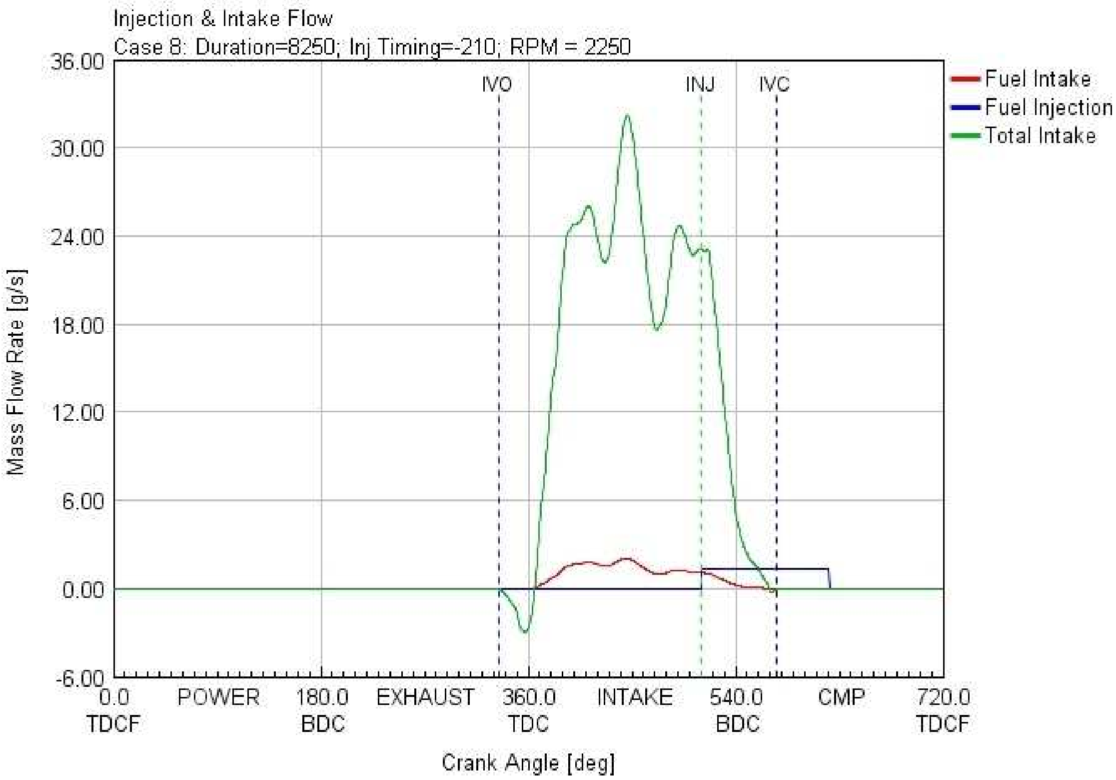

Figure 4.17: CNG injection and intake valve mass flow rates in Cylinder 1, case 8

Analysing data in table 4.9, variations in pressure traces can easily be explained: cylinder contents show that total trapped mass by internal cylinders is higher than trapped by external cylinder, but however external cylinders have a higher fuel mass, and so, lower AFR. For this reason, pressure differences make sense due to a richer combustion in external cylinders, despite of a lower total mass content. On the other hand, this difference in trapped fuel mass can be explained by figure 4.16. It shows that most of trapped fuel is inducted to the cylinder even before start of injection, as happened in previous cases, producing that fuel injected in the intake runner keeps out of cylinder and is directed toward the intake manifold. This event is studied in chapter 4.10, and this case seems to be perfect for doing it, since its variations are present in a higher scale.

4.9. Statistical study

Once all cases are presented, their results will be statistically studied together with the objective of a better understanding of how each factor affects to CNG distribution. As far as not only CNG cylinder content is important in variations, it was decided that the response used in analysis should be air-fuel ratio, AFR. In particular, its coefficient of variation, COV, is going to be used, because this value represents the variations in cylinders. Then it will be the response.

Factor	A – CNG injection duration (µs)	B – CNG Injection timing (CAD)	C – Speed (rpm)
Level -	3200	330 BTDC	1250
Level +	8250	210 BTDC	2250

Table 4.10: Factors and their levels

As explained in chapter 3.4.2, factor effects are calculated in table 4.11.

Case	A	B	C	AB	AC	BC	ABC	Response
1	-1	-1	-1	1	1	1	-1	0,54
2	1	-1	-1	-1	-1	1	1	0,13
3	-1	1	-1	-1	1	-1	1	9,39
4	1	1	-1	1	-1	-1	-1	20,1
5	-1	-1	1	1	-1	-1	1	0,28
6	1	-1	1	-1	1	-1	-1	1,44
7	-1	1	1	-1	-1	1	-1	20,9
8	1	1	1	1	1	1	1	27,1
Divisor:	4	4	4	4	4	4	4	
Effect:	4,39	18,8	4,90	4,02	-0,74	4,38	-1,52	

Table 4.11: Full factorial design results

Looking at the effects, it shows that the factor which has the strongest effect in variations is B, i.e. injection timing. This means that rise in cylinder to cylinder variations due to CNG distribution are consequence of a late CNG injection. This explains why cases 3, 4, 7 and 8, where CNG is injected late, show clearly a bigger variation than the other cases.

Other results show that long injection duration and high speed, by their own, also increase variations in cylinders. Moreover, they also have an increasing effect when they are combined with late injection duration. In other words, positive effect of AB interaction means that the response to a change in A, duration, is stronger when B is at the high setting, i.e. at late injection conditions, and vice versa, and same effect in interaction BC.

On the other hand, there are two combinations of factors with negative effect, which means that a rise in one factor has a weaker effect in variations when the other factors are at the high setting. They are the combinations of long injection duration and high speed, and the combination of all factors, i.e. long injection duration, late injection and high speed. However, this effect is not as strong as the other effects.

4.10. – CNG distribution

Variations produced by CNG distribution will be studied. For this reason, case 8 has been chosen as study object because it shows the strongest variations, and its results then should be representative enough of how CNG is being distributed before it is inducted into cylinders.

At first, most representative elements of the model have been chosen. Figure 4.17 shows a part of the model where these elements are highlighted and named.

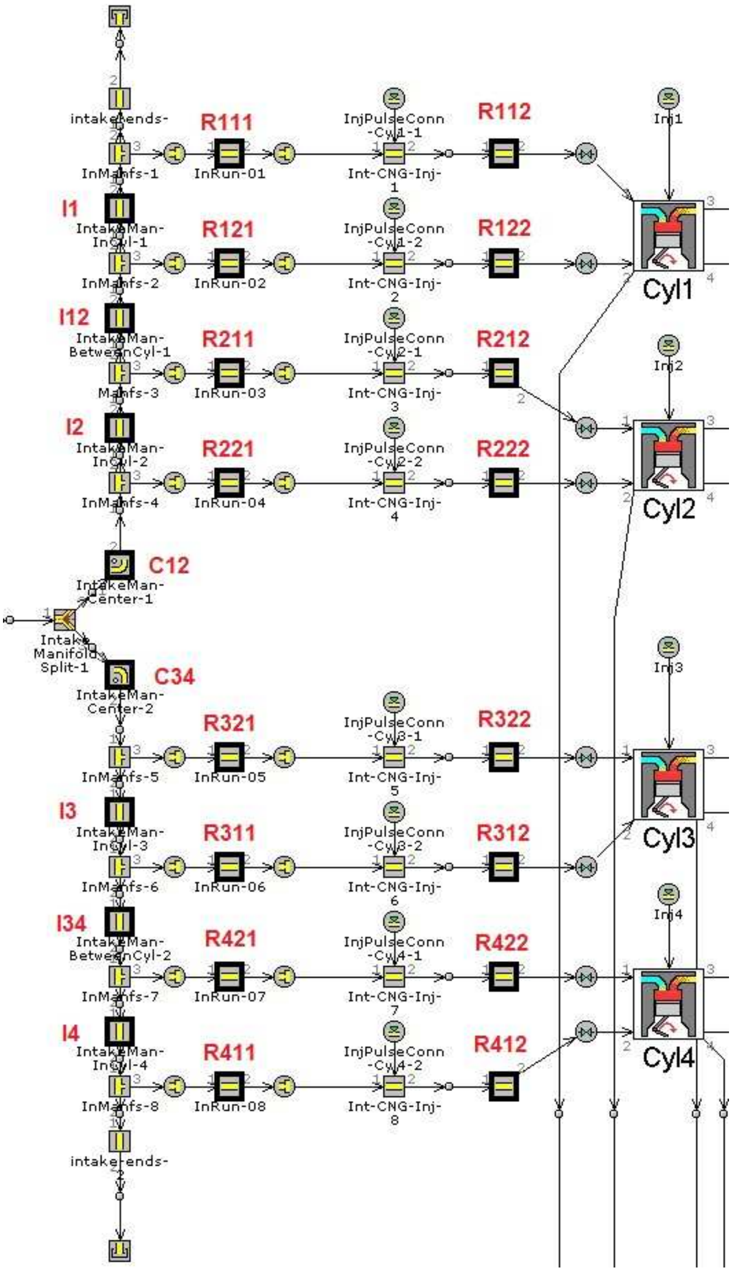

Figure 4.18: Map of intake assembly

Elements have been named according to its location for a clearer use in explanation and plots.

4.10.1. – Symmetry

As figure 4.17 shows, the intake manifold is geometrically symmetric with respect to the plane that divides it by half. According to this, it has been modelled in that way, such that each element has a symmetric element, except the flow split placed between C12 and C34. But it does not mean that CNG behaviour has to be symmetric in both halves of the manifold. Then it is going to be tested. Figures 4.18 and 4.19 show plots of calculated CNG mass flow of some elements and their symmetric counterparts together, as function of crank angle of cylinder 1. These CNG mass flows are not direct results of simulations, but calculated from multiplication of instantaneous total mass flow by instantaneous fuel fraction in each element, so results are not exact, but good enough for studying how CNG is moved in the intake manifold.

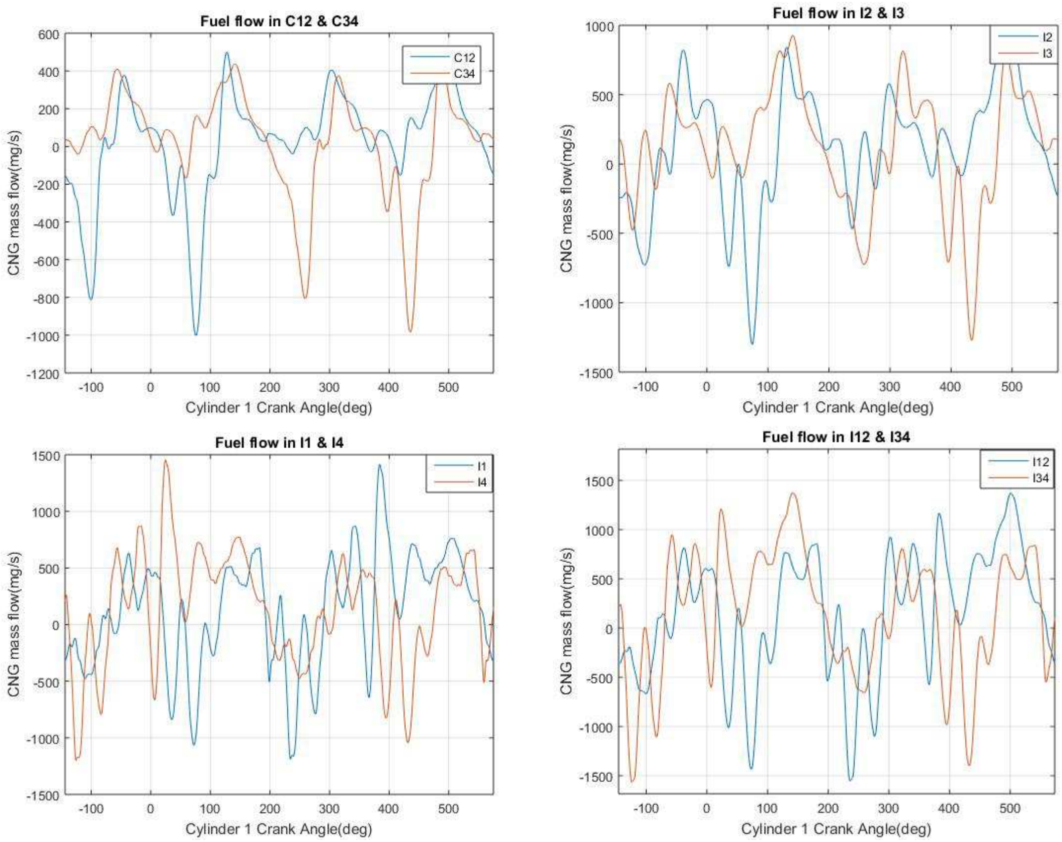

Figure 4.19: CNG mass flow of pairs of symmetric elements: C12-C34, I2-I3, I1-I4 and I12

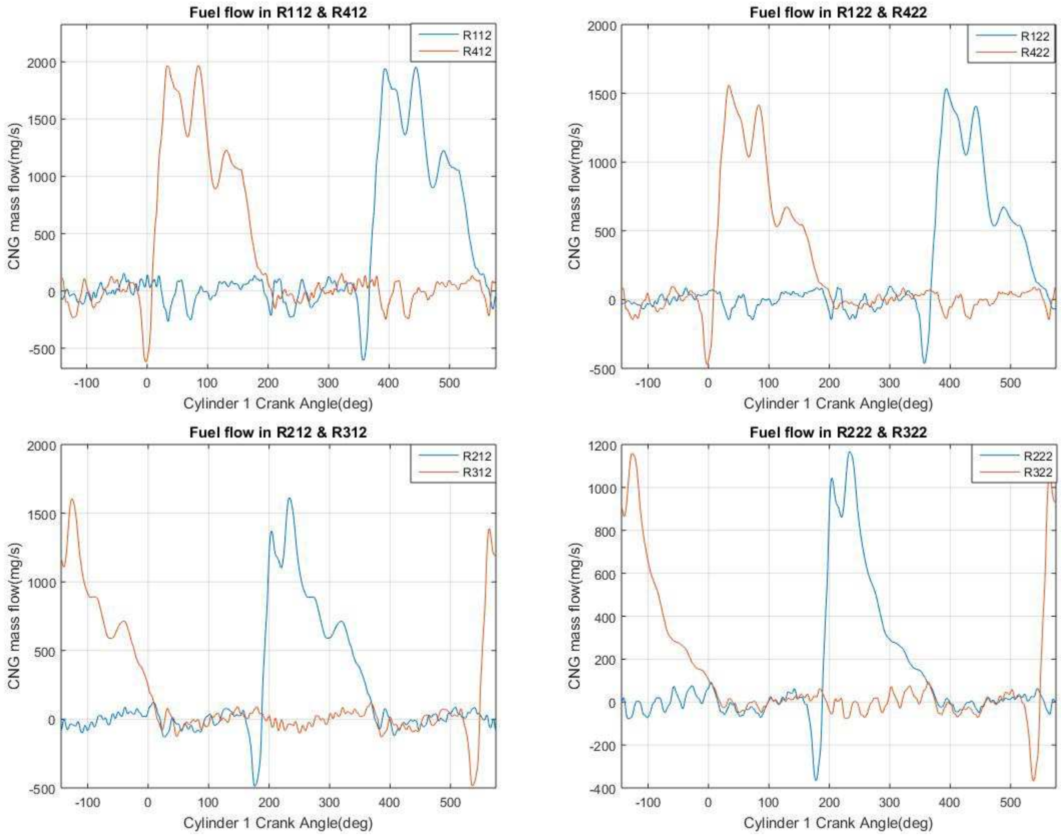

Figure 4.20: CNG mass flow of pairs of symmetric elements: R112-R412, R122-R422, R212-R312 and R222-R322

These figures show that CNG flow in each element is exactly the same as in its symmetric element, but with a phase difference of 360 CAD. This phase difference is produced by opening order of the valves, which is the same as the firing order of the cylinders: 1-3-4-2. Intake valve lifts are represented in figure 4.20 as function of crank angle of cylinder 1.

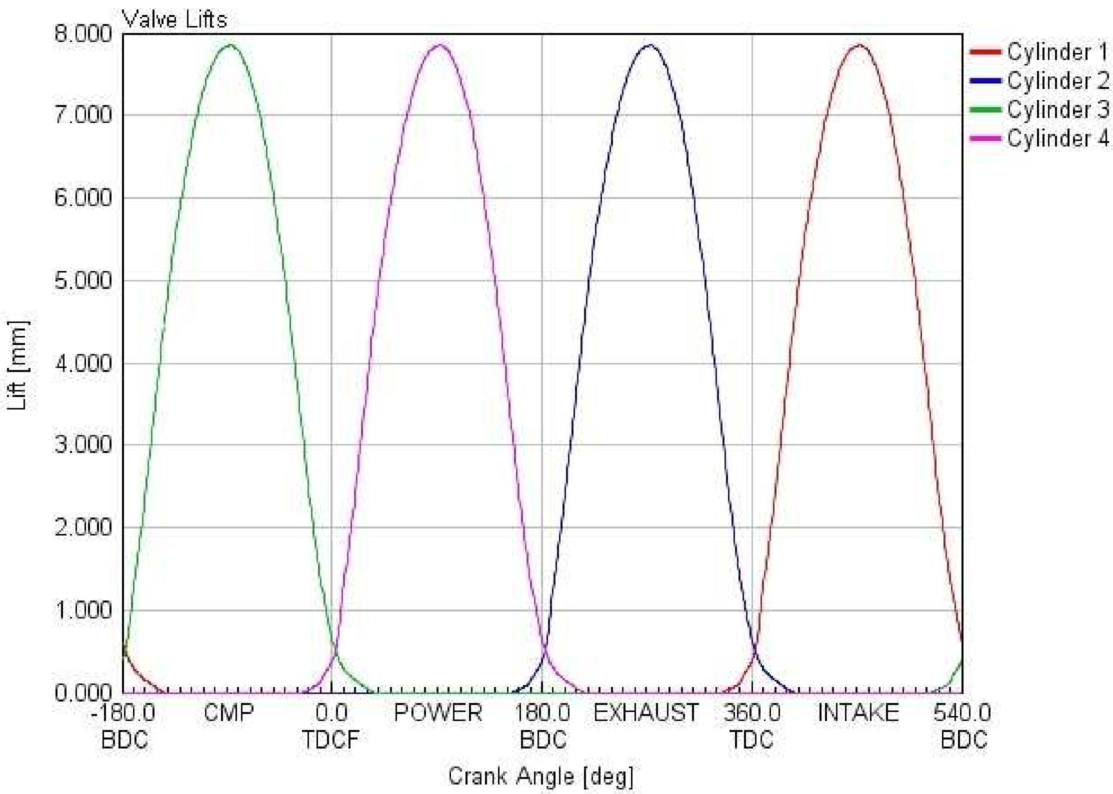

Figure 4.21: Intake valve lifts

In conclusion, it can be considered that CNG distributions in each symmetric half of the intake are identical. Therefore, next study will be focused exclusively in the half connected to cylinders 1 and 2, and will explain events that occur in both halves with a phase difference of 360 CAD.

4.10.2. – Interaction between symmetric parts

Before studying mass flows in the selected part, it is important that interaction between symmetric parts is studied. Therefore, accumulated mass flow of CNG in the middle element C12 is calculated and plotted against crank angle, as shown in figure 4.21.

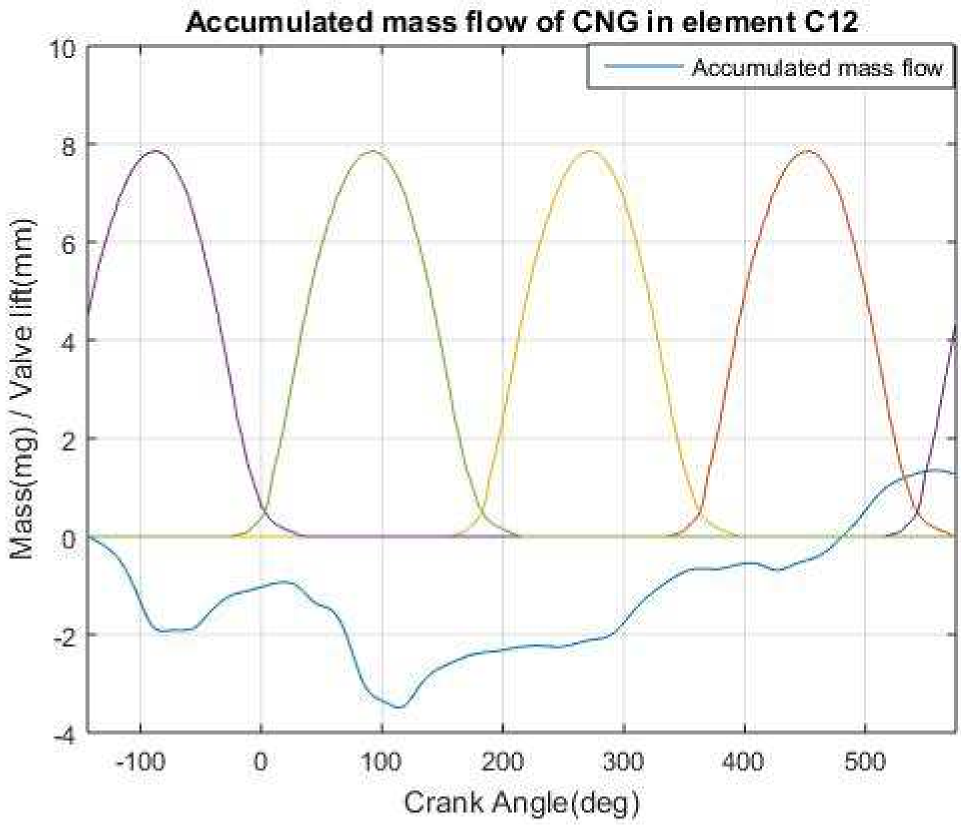

Figure 4.21: Accumulated mass flow of CNG in C12 and valve lifts

In this figure, it is easier to see how CNG flow is in element C12. It shows how induction flows produced by the opening of the different valves change the direction of CNG. Particularly, it shows that this half of the manifold is mainly receiving CNG when valves of cylinder 2 and 1 are open, i.e. when accumulated mass is increasing and expulsing CNG towards the other half when valves of cylinder 3 and 4 are open, i.e. when accumulated mass is decreasing.

However, there is an event that does not make sense in this part: there is a mass drop, represented in the crank angle limits of the plot, with positive value. In other conditions, this could mean that this half is receiving more mass of CNG each cycle, and it should be present in element C34 as a negative drop of accumulated mass. However, same event happens in this other element: there is a positive mass drop. This event means that CNG mass is being generated in a point between the elements, and this is not possible. So, as said before, it is considered an error of precision in the results due to calculation of CNG mass flows, and then a limitation of the simulation that has been carried out. For that reason, the conclusion of this study is that there is interaction between two symmetric parts, producing a mass exchange, but it cannot be quantified from obtained results, and it cannot be studied in a deeper way.

4.10.3. – CNG distribution study

As defined, this last study is focused in the half of the intake manifold that is connected to cylinders 1 and 2. This entire part is shown in figure 4.22. For a better understanding of CNG distribution, accumulated mass flows and mass factions of CNG of all elements of this part (except C12, which has already been analysed in the previous page) are plotted and shown in figures 4.23, 4.24 and 4.25.

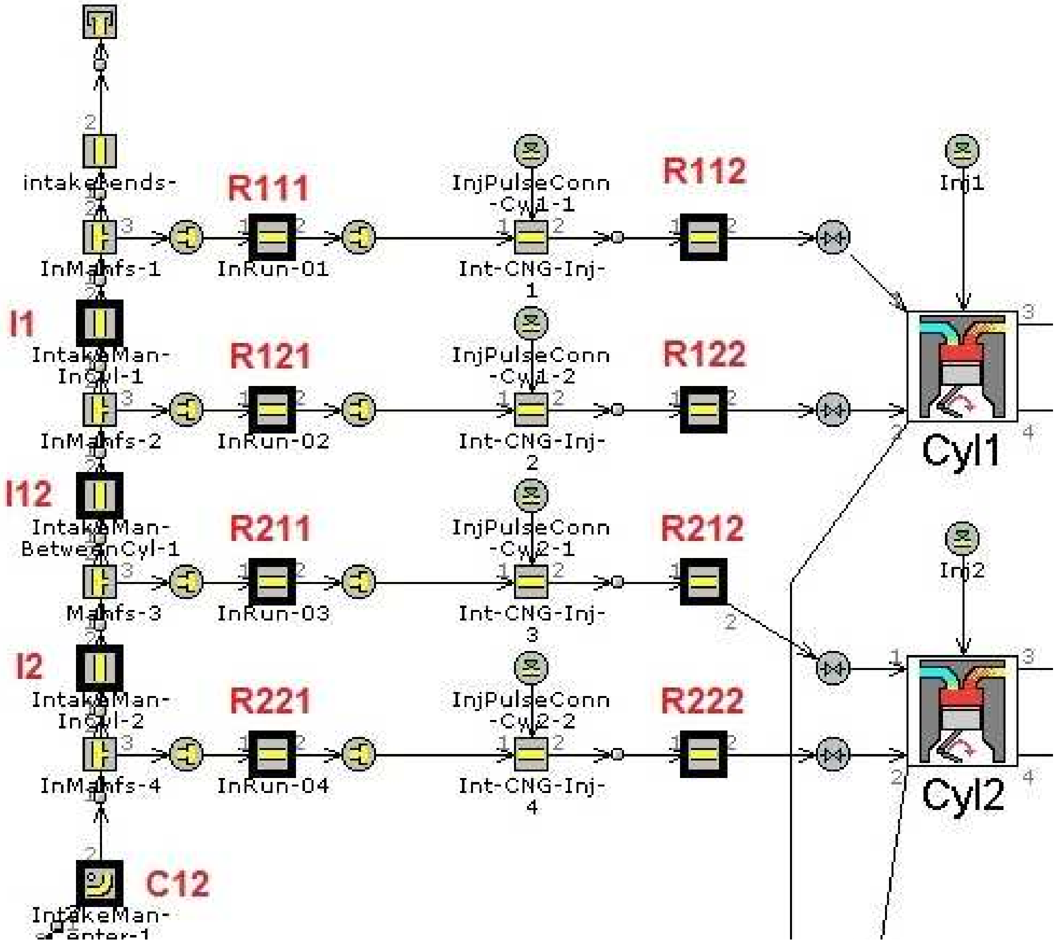

Figure 4.22: Map of studied part

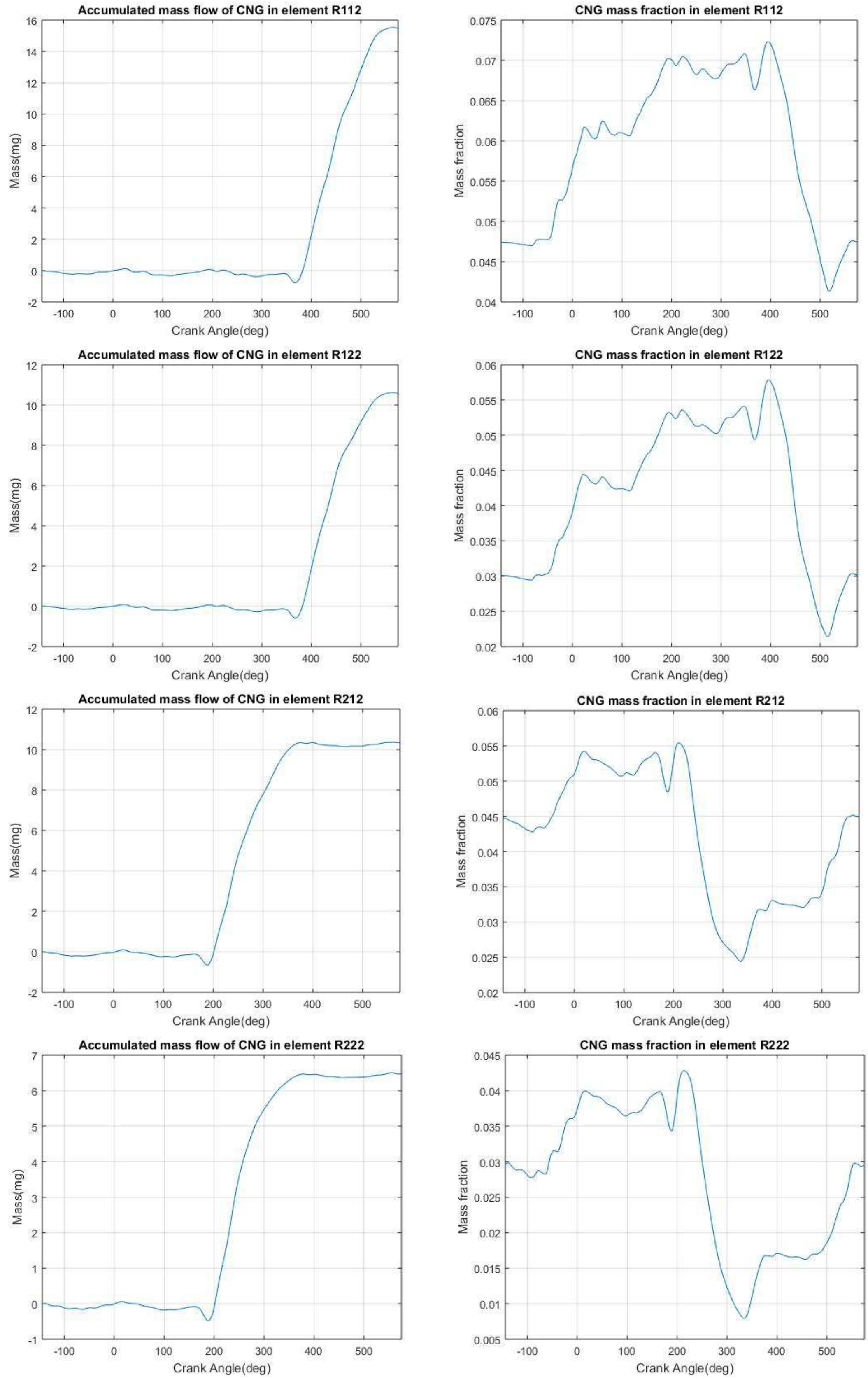

Figure 4.23: Accumulated mass flows and mass fractions of elements R112, R122, R212 and R222

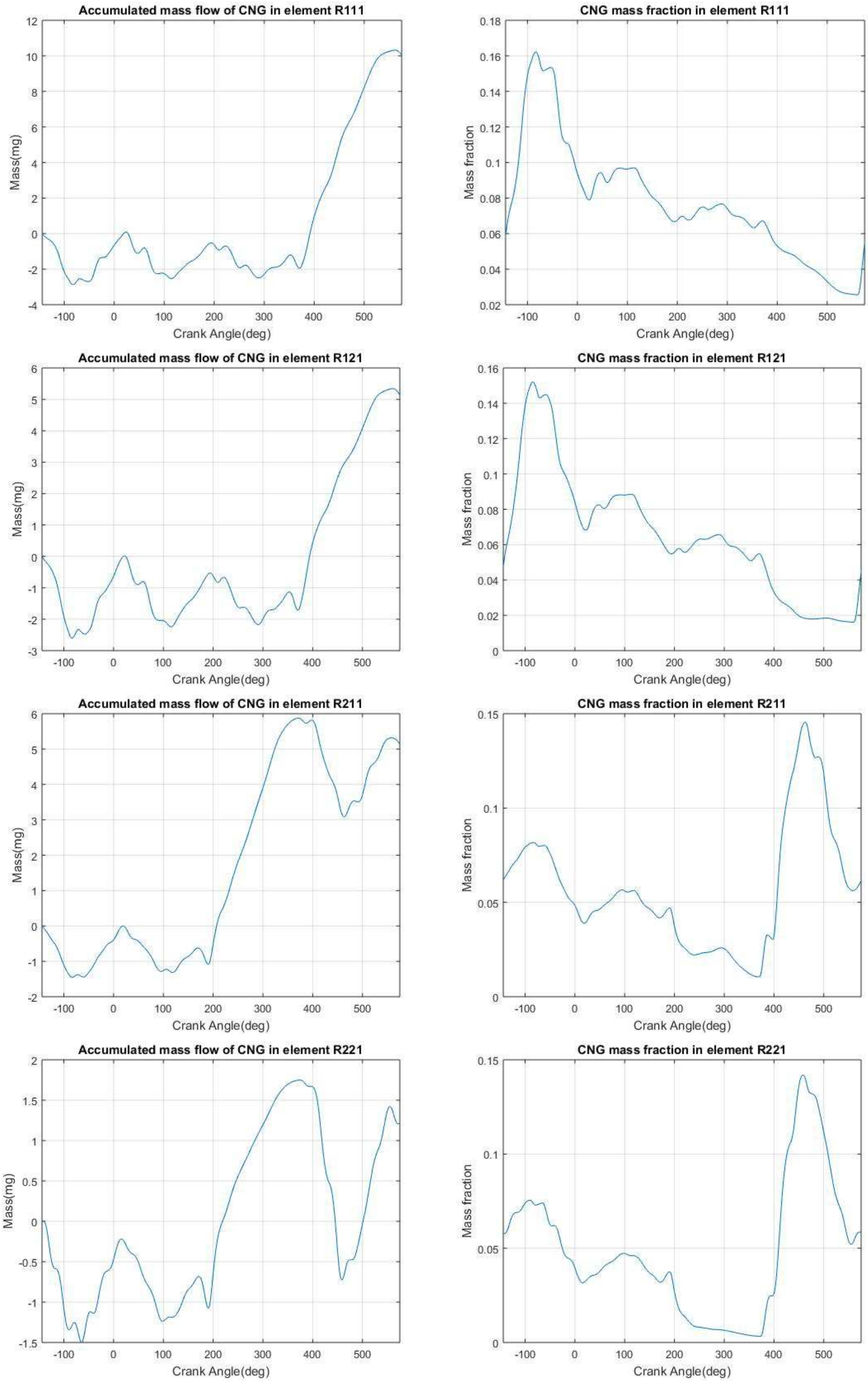

Figure 4.24: Accumulated mass flows and mass fractions of elements R111, R121, R211 and R221

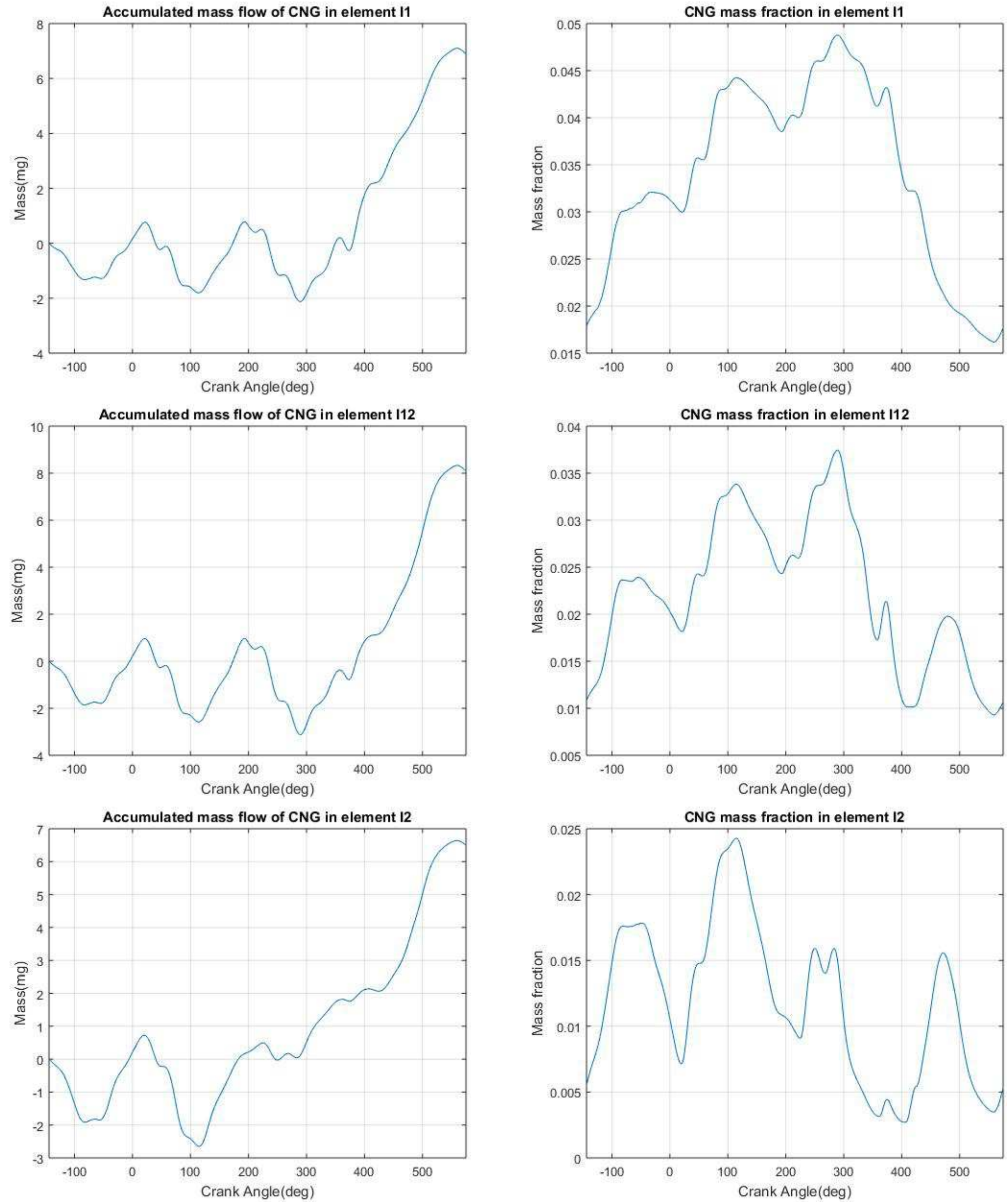

Figure 4.25: Accumulated mass flows and mass fractions of elements I1, I12 and I2

At first, figure 4.23 shows the plots belonging to elements directly connected to each valve. Therefore, accumulated mass flows may be understood as representative of mass flow through valves (when their connected valves are open), as far as these plots show the differences that exist in trapped mass by each valve. This shows that, while "internal" valves (valves connected to R122 and R212) are absorbing a very similar fuel mass quantity, the "external" valves have a significant in trapped mass. In mass fractions of figure 4.23, it can be seen that all these elements are accumulating fuel during each cycle until their connected valves are open, although each valve does it in a different scale with the same behaviour as the mass flows.

Figure 4.24 shows the plots belonging to elements that connect the common volume of the intake manifold to the element where CNG is injected. As in plots of figure 4.23, accumulated mass flows show that the scales are clearly higher in the runner connected to external intake valve of cylinder 1 when these valves are open. However, as in the study of interaction between parts, drops in accumulated mass should not be happening in the other three elements. Its reason is the fuel flow calculation, that as explained before, it is calculated by multiplication of total mass flow by fuel fraction. Due to this, mass fractions are used for a more accurate study. Mass fractions shows that in four elements similar mass fractions, around 0.155, is accumulated during next stroke in the elements once their valves are closed, but it is decreasing as each other valves are being open, until their valve is open and fuel mass fractions reach their minimum values. Analysing it together with data of elements R112, R122, R212 and R222, it can be concluded that while in element R111 fuel is being mostly moved toward to cylinder, in element R221 it is mostly being moving out of the runner. This can be produced by a higher induction pressure toward out of the runner produced by intake valve openings in cylinders 3 and 4 due to its higher proximity to their valves.

For completing this hypothesis, plots in figure 4.25 can perfectly define this behaviour. They show plots of elements I1, I12 and I2, which are the elements that simulate the parts of the volume between runners, and so can show fuel mass flows that are moving from a runner to other. Accumulated mass flows show that fuel is only being moving away from middle of the intake manifold, which explains why runner that are more distant of this point is absorbing more fuel mass. It is demonstrated by fuel mass fractions, which are visibly higher as their proximity to middle is lower. It can be seen that mass fraction is only decreasing during intake event of cylinder 2 in element I2, which is placed between the two runners of cylinder 2. On the other hand, mass fractions of elements I1 and I12 are mainly increasing, i.e. fuel is being accumulated, until reach their maximum value just before intake valves of cylinder 1 start to open.

In summary, it can be concluded that the difference in trapped CNG mass by each cylinder is consequence of mass flows that happen during the cycle. At first, these flows are expulsing part of fuel injected late out of the runner once their closest valves are closed, happening with a stronger effect in runners that are closer to the middle of the intake manifold, and accumulating the remaining part close to the valves. After this, fuel is mainly pushed away from the middle point of the intake manifold. This event is producing that once valves of each cylinder open, the cylinders that are more distant from this middle point, i.e. cylinders 1 and 4, are absorbing more fuel mass per cycle than the other cylinders.

4.10.4. – Weaknesses and limitations of results

For an engineer, an essential part of its work is the knowledge of the weaknesses and limitations of the work that is being carried out. As stated in previous points, in this project results of fuel mass flows, and consequently their accumulated values, have been approximated through multiplication of instantaneous total mass flow by instantaneous fuel fraction. As seen in the final results, this approximation represents at the same time a weakness and a limitation: weakness because it produces results that are not accurate, and limitation because this lack of accuracy has produced that results cannot be properly quantified.

Then, it can be considered that the source of the limitations of this study is the simulation of the model used, which still requires further improvement in order to calculate the fuel mass flows in an accurate way. However, modelling is an advantage above all, since it has made this study possible, but an improvement in software used would make possible to get the necessary results.

5. – CONCLUSIONS AND FUTURE WORK

The main conclusions of this research are organized in the following points:

- Engine modelling can offer a great advantage respect to real experiments. There are many aspects in terms of analysis of results that represent an improvement for research and developing of combustion engines. However, in this project, there have been some limitations in results that have affected the accuracy of needed results. Consequently, some results have been approached through others, such as fuel flows of the intake manifold.

- A statistical study concludes that the factor with the strongest effect on cylinder-to-cylinder variations in dual fuel engines is the injection timing of CNG. A later injection increases significantly cylinder-to-cylinder variations because it involves that part of injected CNG that is not inducted into the cylinder during its belonging cycle. Then, for a better balance of cylinder works, and so an improved engine performance, engine should operate with early timings of CNG or a different injection strategy.

- High engine speeds and long injections of CNG also increase cylinder-to-cylinder variations, but on a lower scale. Speed cannot be modified since either engine should be able to work in a determined interval. However, long injections could be reduced using higher CNG injection pressures.

- CNG distribution in the intake manifold when part of injected CNG is not trapped during it belonging cycle can explain why there are differences in the mass absorbed by each cylinder. In spite of lack of more accurate results for a better analysis, it is concluded that variations are mainly produced by fuel mass flows in each half of the intake manifold, and their behaviour can be considered symmetric with a phase difference of 360 CAD. According to the analysis, part of CNG injected in the closest runners to the intake manifold middle is pushed away to the most distant runners. This event involves that cylinders 1 and 4 absorb more CNG mass than predicted, and so cylinders 2 and 3 have a lower trapped fuel mass. This problem in CNG distribution may be resolved by using an injection strategy that supposes a higher fuel mass in runners of cylinders 2 and 3.

These conclusions from the project are summarized in some recommendations for future work. At first, the engine model should be improved in terms of combustion, since there were not enough data for a good combustion simulation, and consequently research its effects on engine performance. On the other hand, a deeper analysis through an improved model of fuel mass flows in the intake manifold should be done since results of this project have not been accurate enough, and so cannot be quantified and completely defined. Finally, it could be interesting to carry out a research of different CNG injection strategies in order to improve CNG distribution.

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