Politecnico di Torino

College of Mechanical, Aerospace, Automotive and Production Engineering

M.Sc. in Automotive Engineering

Master Thesis

Development of a methodology for the numerical simulation of a high performance 2 stroke engine by means of 1D-3D coupling

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Academic Year 2019-2020

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Ringraziamenti

Ringrazio innanzitutto Alessandro Gallo per il suo continuo e indispensabile supporto nello sviluppo della metodologia. Sempre presente nonostante i numerosi impegni. Grazie di cuore. E inoltre, un Piemontese che insegna a fare la pizza ad un Campano e' piu' unico che raro.

I would like to thank Matthias Boeger for the technical and motivational support during the year spent in Porsche Engineering, but particularly for letting me steal all his food at work. Remember: the bread dough for pizza has to rest 6 hours, not 30 minutes.

Al Maestro Vincenzo Bevilacqua che mi ha arricchito professionalmente, e non solo, con le sue lezioni di ingegneria e di vita. Te ne saro' sempre grato.

Special thanks to Luca, Michal and David for having made fun of me while capsizing in the Vltava River. I will never forget that experience.

Menzione speciale va a Giovanni Corvaglia, mia prima Guida all'ingresso del mondo del lavoro. Dopo aver condiviso tanta acqua e gioia alla variante della Roggia, mi assegna un due tempi bicilindrico.. e poi se ne scappa a lavorare su V12 aspirati. Non c'e' miglior definizione per Pezzo di.. pane.

Ringrazio Emanuele Magliano per la stima che ha riposto nei miei confronti. Nun succede.. ma se succede non ti delutero'.

A Papa', Mamma e mio Fratellone, per avermi offerto sempre il loro supporto fisico e morale anche quando non ero stato io a chiederlo.

Abstract

The 2 stroke engines have wide variety of applications, especially in the field of recreational vehicles, handled products and small two wheelers. This is due to the 3 advantages that characterize these kind of engines: the high power density, the low weight and the low production costs. The main research activities in this field are focused on the optimization of these advantages while minimizing the well known disadvantages: high emissions and fuel consumption. The aim of this work is the development of a simulation methodology able to predict the performance of the engine under development with special attention to the scavenging process, mixture preparation and combustion. Up to now, in the field of simulation, two state of art approaches exist. The first one characterized by the detailed description of the gas exchange and thermodynamic processes by means of 3D-CFD software tools able to model and analyse complex geometries. But unfortunately this kind of approach requires reliable initial conditions and, in addition, the model build-up and computational times are very high. So, during the development process of a two stroke engine, the 3D simulation results are not achieved before the prototype phase begins. The second approach, on the other hand, is performed by means of 1D-CFD software tools. It is less time consuming compared to the 3D approach and requires a lower demand on initial conditions since an higher number of iterations can be calculated to get a converged solution. But on the negative side it has to be considered that it cannot provide predictive models for what concern crucial aspects of two stroke engines, such as the scavenging and the mixture preparation. Since a detailed understandings of the thermodynamic processes and a predictive simulation strategy performed in a reasonable amount of time is needed, the combination of the two approaches is necessary. The focus of this thesis is the evaluation of the feasibility of the 1D-3D methodology and the accuracy of the results for the simulation of a real world high performance two stroke engine.

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Chapter 1

Introduction

1.1 Past, Present and Future of 2 stroke engines

Although the legislative pressure of many countries is going toward more and more strict limits on the pollutant emissions and is forcing the development of alternative propulsion systems, nowadays motorcycles, scooters and mopeds equipped with two stroke engine are still largely produced for the general transport and for recreational purpose [1]. If the two stroke engine will be still in the future the main mass produced engine for motorcycle application will depend also on the results of the research and development conducted by the two stroke engine supplier.

There exist many and different applications of this kind of engine due to peculiarities that characterize two stroke engines such as lightness, high specific power and construction simplicity.

Anyway the most efficient application of the two stroke engine is that of the marine diesel propulsion unit. An example can be the Harland and Wolff uniflow-scavenged two-stroke diesel ship propulsion engine of 21,000 bhp, with a 900 mm bore and a 1800 mm stroke, working at 60-100 rpm and producing about 4000 bhp per cylinder.These marine engines have thermal efficiencies above 50%, making them one of the most efficient engines ever build. Same efficiency level can be reached by 1.6-liter turbo V6 F1 engines, but they are characterized by a much more complex architecture and of course it is a way more expansive solution.

However even if marine propulsion unit are a lot different in terms of size from the other two stroke engine, they are not in terms of design.

Two stroke engine are also used as lightweight power units for portable tools like

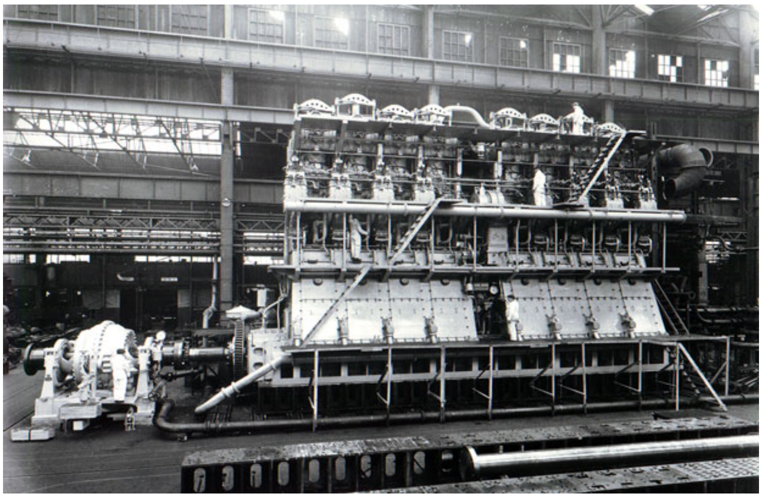


Figure 1.1: H&W uniflow-scavenged two-stroke diesel ship propulsion engine

chainsaws or brushcutters. In this kind of handheld application the lightness and the high specific power is a must.

They are used also for the outboard engine that, although it is mainly recognize as applied for sporting or recreational purpose, today actually it represents a widespread solution employed for the commercial fishing and for the water transport in many countries of the world.

Then recently some new recreational products that recently appear on the market are the skidoos and snowmobiles, and the engine that most of the time is used for these purposes is the two stroke one. The use of a two stroke on a snowmobile is particularly indicated since the lubrication system of such engine is perfectly suited for subzero temperature condition. Although snowmobile is seen mainly as a recreational vehicle, actually it is a really practical means of transportation for many people in an Arctic environment [2].

Chapter 2

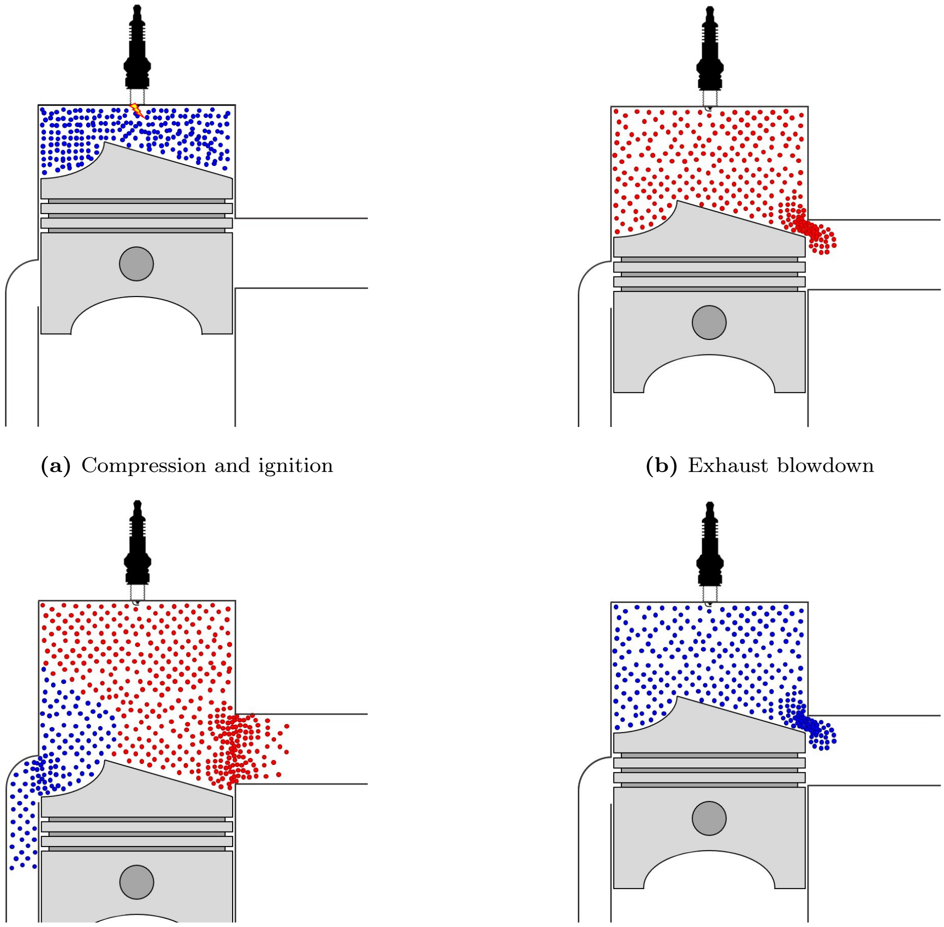
Two-Stroke Engine Fundamentals

2.1 Operating mode of a two-stroke engine

The simplicity of the engine is obvious, with all of the processes controlled by the upper and lower edge of the piston. In the Fig. 2.1 (a), above the piston, the air trapped and the fuel charge are ignited by the spark plug, producing a step rise of the pressure and of the temperature that push the piston down towards the bottom dead center BDC. Below the piston, the opening inlet port induces the air to move from the external environment to the crankcase due to the rise of the crankcase volume that reduces the pressure below the atmospheric level. The crankcase is sealed around the crankshaft in order to guarantee the maximum depressure inside it.

To introduce the fuel in the engine, various options exist: a carburetour, a port injection or a direct injection. If the two stroke engine is diesel, the only option is the last one, with a glow plug, replacing the spark plug, that aid the cold engine start.

Since the area of the exhaust port increases with the crank angle, and the pressure is decreasing in the cylinder, it is clear that the pressure profile in the exhaust duct increases up to a certain point and then it falls. Such pressure profile is known as blowdonwn. This flow process is described as insteady gas flow process and such pressure impulse can be reflected by all the area variations of the pipe or by the end of the pipe to the atmosphere. Such pressure wave reflections strongly affect the performance of the engine, as it will be deeply described in the chapter 4. Below the piston the compression of the new charge is occurring. Pressure and temperature



(a) Compression and ignition

(b) Exhaust blowdown

(c) Scavenging

(d) Stuffing of fresh charge

Figure 2.1: Two-stroke engine phases

reached will be proportional function of the crankcase volume reduction, or better of the Crankcase compression ratio.

In Fig. 2.1 (c), above the piston, the initial part of the exhaust phase, known as blowdonwn, is almost completed and, with the piston that uncovers the intake ports, it connects the cylinder directly to the crankcase by means of the transfer ports. If the crankcase pressure is higher than the one inside the cylinder, the fresh charge enters the cylinder: a process known as scavenging. Clearly, if the transfer ports are not well designed, the new fresh charge can directly exit through the exhaust port and be completely lost from the cylinder. This process, known as "shortcircuit", increasing the amount of exhaust gas residuals, can compromise the filling of the cylinder and the combustion itself. Even worse, in a two stroke engine equipped with carburetour, all the fuel would be lost at the exhaust with a subsequently huge increase of unburned hydrocarbons emission and fuel consumption. Therefore, the fresh charge directioning should be realized, with a proper design of the ports, in such a way to maximize the cylinder filling.

The following chapter will deeply focus on the scavenging process and on all the aspects that contribute to maximize it. It should be pointed out that such a process can not occur perfectly, since part of the fresh charge could find a way direct to the exhaust port. In the same way, the exhaust process can not ensure the complete emptying of the cylinder from the burned gases.

In a monocylider race engine, it is possible to double the mass of fresh air trapped by means of a tuned tailpipe, and that means double the engine power. After that the exhaust port is finally closed, the compression starts up to when the spark plug ignites the air fuel mixture and the combustion takes place. The compression ratio of a two stroke engine is different from the one of a four stroke engine. In the first case as reference volume is considered the one of the cylinder at exhaust port closure and it is defined as *trapped compression ratio*, that has to be distinguished from the *geometric compression ratio* that considers the complete cylinder volume as reference one.

Finally, it can be said that a two stroke engine is a double acting device. Above the piston the processes of compression and combustion take place, while below the piston, in the crankcase, the fresh charge is inducted, usually by means of reed valves, from the external environment and prepared for the transfer in the cylinder [4].

2.2 Engine performance

The mass of the fresh charge that has been supplied to the crankcase from the atmosphere is indicated as $\mathrm{m_{as}}$. Measuring the atmospheric conditions, environmental pressure and temperature, $\mathrm{p_{at}}$ and $\mathrm{T_{at}}$, the density of the air can be calculated from the thermodynamic state equation, where $\mathrm{R_a}$ is the gas constant for the air:

$$\rho_{\mathrm{at}} = \frac{p_{\mathrm{at}}}{R_{\mathrm{a}} T_{\mathrm{at}}} \tag{2.1}$$

The delivery ratio, DR, of the engine defines the ratio between the air mass supplied to the cylinder during the scavenging process and a reference mass, $\mathrm{m_{ref}}$, that is the mass required to fill the swept volume at atmospheric condition, i.e.:

$$\mathrm{m}_{\mathrm{dref}} = \rho_{\mathrm{at}} V_{\mathrm{sv}} \tag{2.2}$$

$$\mathrm{DR} = \frac{m_{\mathrm{as}}}{m_{\mathrm{dref}}} \tag{2.3}$$

The scavenging ratio, SR, of a naturally aspirated engine defines the ratio between the air mass supplied to the cylinder during the scavenging process and a reference mass, $\mathrm{m}_{\mathrm{sref}}$, that is the mass required to fill the entire cylinder at environmental condition, i.e.:

$$\mathrm{m}_{\mathrm{sref}} = \rho_{\mathrm{at}}(V_{\mathrm{sv}} + V_{\mathrm{cv}}) \tag{2.4}$$

$$\mathrm{SR} = \frac{m_{\mathrm{as}}}{m_{\mathrm{sref}}} \tag{2.5}$$

If the engine is supercharged or turbocharged, the new reference mass, $\mathrm{m}_{\mathrm{sref}}$, is calculated at the pressure and temperature conditions of the scavenge air supply, p_{s} and T_{s}.

$$\rho_{\mathrm{s}} = \frac{p_{\mathrm{s}}}{R_{\mathrm{a}} T_{\mathrm{s}}} \tag{2.6}$$

$$\mathrm{SR} = \frac{m_{\mathrm{as}}}{\rho_{\mathrm{s}}(V_{\mathrm{sv}} + V_{\mathrm{cv}})} \tag{2.7}$$

The abovementioned theory has been treated in terms of a mass flow referred to the swept volume of a single cylinder engine. Anyway, if the engine is a multicylinder device, the volume considered is total volume swept by the cylinders.

2.2.1 Scavenging efficiency and purity

The best that a two stroke engine could be get is that the scavenge efficiency, SE, should be equal to the scavenge ratio, SR. The scavenge efficiency is defined as the ratio between the air mass delivered in the cylinder, $\mathrm{m}_{\mathrm{tas}}$, and the air mass retained at exhaust port closure, m_{tr}. The charge trapped is made up of only fresh charge, $\mathrm{m}_{\mathrm{tas}}$ and exhaust gases, m_{ex} and any other fuel air mixture unburned from the previous cycle, m_{ar}, where:

$$\mathrm{m}_{\mathrm{tr}} = m_{\mathrm{tas}} + m_{\mathrm{ex}} + m_{\mathrm{ar}} \tag{2.8}$$

So, the scavenge efficiency, SE, defines the effectiveness of the scavenge process, like it can be seen from the following equation:

$$\mathrm{SE} = \frac{m_{\mathrm{tas}}}{m_{\mathrm{tr}}} = \frac{m_{\mathrm{tas}}}{m_{\mathrm{tas}} + m_{\mathrm{ex}} + m_{\mathrm{ar}}} \tag{2.9}$$

However, after the gas exchange process, combustion takes place. The air trapped in the cylinder reacts with the fuel and it is of utterly importance defining the purity of the air fuel mixture. The purity of the trapped charge, Π , is defined as the ratio between the air trapped in the cylinder before the combustion, m_{ta}, and the total charge mass of the cylinder, where:

$$\mathrm{m}_{\mathrm{ta}} = m_{\mathrm{tas}} + m_{\mathrm{ar}} \tag{2.10}$$

$$\Pi = \frac{m_{\mathrm{ta}}}{m_{\mathrm{tr}}} \tag{2.11}$$

In many technical papers and books on two stroke engines, *scavenging efficiency* and *purity* are somehow interchanged, wrongly assuming that the value of m_{ar} is zero, that is generally true for most of spark ignition engine and in particular when combustion occurs with a rich air-to-fuel ratio, but this would not be true for two stroke diesel engine where the air is never entirely involved in the combustion process, it would not be true even for spark ignition engine with a stratified charge combustion process.

2.2.2 Trapping efficiency

The trapping efficiency, TE, is the ratio between the air mass trapped into the cylinder, $\mathrm{m}_{\mathrm{tas}}$, and the one supplied, m_{as}:

$$\mathrm{TE} = \frac{m_{\mathrm{tas}}}{m_{\mathrm{as}}} \tag{2.12}$$

Or it can be written also:

$$\mathrm{TE} = \frac{m_{\mathrm{tr}} SE}{m_{\mathrm{sref}} SR} \tag{2.13}$$

It can be seen that, in ideal condition, m_{tr} can be considered equal to $\mathrm{m}_{\mathrm{sref}}$ and that the equation (2.13) can simplify as follow:

$$\mathrm{TE} = \frac{SE}{SR} \tag{2.14}$$

2.2.3 Charging efficiency

The charging efficiency, CE, defines the ratio between the charge air filling of the cylinder, and the ideal charge air filling of the cylinder at the beginning of the compression stroke. After all, the optimal design of a two stroke engine for what consists the gas exchange process is based on maximizing the trapped air mass in the cylinder and so maximize the quantity of fuel that can burn with that amount air. Therefore, the charging efficiency, is defined as:

$$\mathrm{CE} = \frac{m_{\mathrm{tas}}}{m_{\mathrm{sref}}} \tag{2.15}$$

It can be written also as the product between the trapping efficiency and the scavenge ratio, as shown below:

$$\mathrm{CE} = \frac{m_{\mathrm{tas}}}{m_{\mathrm{as}}} \cdot \frac{m_{\mathrm{aas}}}{m_{\mathrm{sref}}} = TE \cdot SR \tag{2.16}$$

2.2.4 Air-to-fuel ratio

It is important to know that there exist different limits for what concern the acceptability of the combustion of air and fuel, like gasoline or diesel. In the case of gasoline, that can be approximated to the octane, C_8H_{18}, burns "perfectly" with the air considering a balanced equation called stoichiometric reaction. Air is made up of approximately, in terms of volume percentages, 21 parts of oxygen and 79 parts of nitrogen. So, the chemical equation for the complete combustion becomes:

$$2\mathrm{C_8H_{18}} + 25[\mathrm{O_2} + \frac{79}{21}\mathrm{N_2}] = 16\mathrm{CO_2} + 18\mathrm{H_2O} + 25\frac{79}{21}\mathrm{N_2} \tag{2.17}$$

From this equation it can be evaluated the ideal air-fuel stochiometric ratio, AFR or also α, so that for approximately two parts of octane, 25 parts of air are needed. Since it is usual to deal with masses, knowing the molecular weight of O_2, H_2, N_2 and C that are respectively 32, 2 e 28 and 12, so:

$$\mathrm{AFR} = \frac{25 \times 32 + 25 \times 28 \times \frac{79}{21}}{2(8 \times 12 + 18 \times 1)} \tag{2.18}$$

Since the equation is balanced, with the exact quantity of oxygen needed to burn all the fuel in carbon dioxide and water vapour, such combustion process does not produce carbon monoxide, CO and unburned hydrocarbons but since the combustion reaction produce also to a step increase of temperature, this leads to a non negligible emission of NO_x, the different nitrogen oxides. In fact, at high temperature, nitrogen and oxygen react and combine each other in different ways forming NO, N_2O, NO_2 etc.[6].

A normalized parameter λ is defined as the ratio between the actual AFR and the AFR_{st}.

$$\lambda = \frac{AFR}{AFR_{st}} \tag{2.19}$$

So it can be stated that:

$$\begin{cases} \lambda < 1 & \text{rich mixture} \\ \lambda = 1 & \text{stoichiometric mixture} \\ \lambda > 1 & \text{lean mixture} \end{cases}$$

For what concern combustion limit, it can be considered as limit for the rich combustion a λ value equal to 0.6, while the meximum power can be reached with a slightly rich mixture, λ 0.95 (the fuel in excess evaporates and increase the density of the charge inside the cylinder), while the maximum thermal efficiency (or minimum fuel consumption) with a slightly lean mixture 1.05, and finally the limit for a lean combustion around 1.6 [3]. The values of air-fuel ratios just indicated refers to an homogeneous charge at the moment of the spark ignition, and they are defined as trapped air-fuel ratios, AFR_t. The air-fuel ratio derived from the equation (2.18) is, more properly, the trapped air-fuel ratio, AFR_t, needed for the stoichiometric combustion.

2.2.5 Cylinder trapping conditions

All that has been explained in the previous paragraph intend to highlight that the effect of the scavenging process of the cylinder is to fill it with a mass of air, m_{ta}, inside a total mass of charge, m_{tr}, at trapped condition. This total mass highly depend on the trapping pressure, as shown in the equation below:

$$m_{tr} = \frac{p_{tr}V_{tr}}{R_{tr}T_{tr}} \tag{2.20}$$

where

$$V_{tr} = V_{ts} + V_{cv} \tag{2.21}$$

Anyway, the trapping volume, V_{tr}, is a constant, as it is also the gas constant, R_{tr}, for the gas composition at the trapping point. The gas constant for the exhaust gases, R_{ex}, is almost equal to the one of the air, R_a. Since the gas composition of the cylinder is mainly air, considering R_{tar} equal to R_a induces to negligible errors. For many different kinds of scavenging process, the trapped temperature changes, at most, of 5%. Therefore, the most significant variable, is the trapped pressure, p_{tr}. Like it has been said in the previous sections, the value of the trapped pressure is directly controlled by the dynamic of the pressure waves of the exhaust system, both for a monocylinder engine with or without a tuned exhaust pipe or for a multicylinder engine with a branched exhaust pipe. The value of the mass of fuel trapped, m_{tf}, can be determined as:

$$\mathrm{m_{tf}} = \frac{m_{\mathrm{ta}}}{AFR_{\mathrm{t}}} \tag{2.22}$$

2.2.6 Heat released during the burning process

The value of the total heat released by the combustion process from the fuel is QR:

$$\mathrm{Q_r} = \eta_{\mathrm{c}} m_{\mathrm{tf}} H_{\mathrm{i}} \tag{2.23}$$

Where η_c is the combustion efficiency and H_i is the lower heating value of the fuel.

2.2.7 The thermodynamic cycle for the two-stroke engine

The thermodynamic cycle of a two stroke engine is usually referred as a derivative of the Otto cycle. A representation of such cycle is shown in the logp-logV and pV diagrams respectively in Fig. 2.2 and Fig. 2.3, relative to the experimental results of the two stroke engine that will be later deeply analysed in this thesis. A comparison of the experimental results with the theoretical cycle at 8250 RPM is shown in Fig. 2.2. As it can be clearly seen, the following hypothesis have been made for the theoretical cycle:

(a) Compression begins at the trapping point

(b) The combustion of all the fuel occurs at TDC (Isochoric combustion)

(c) the exhaust is considered as a heat rejected process

(d) Compression and expansion processes are isentropic, with the air as working fluid, and so these processes are calculated like:

$$\mathrm{p}V^{\gamma} = constant \tag{2.24}$$

where γ is a constant. For the air, the ratio between the specific heats, γ, has a value of 1.4. It can be proved that such kind of ideal cycle shows a thermal efficiency equal to:

$$\eta_{\mathrm{t}} = 1 - \frac{1}{CR_{\mathrm{t}}^{\gamma-1}} \tag{2.25}$$

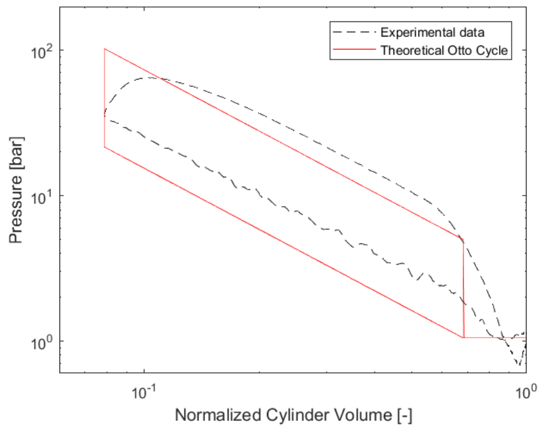


Figure 2.2: Logarithmic plot of pressure and volume

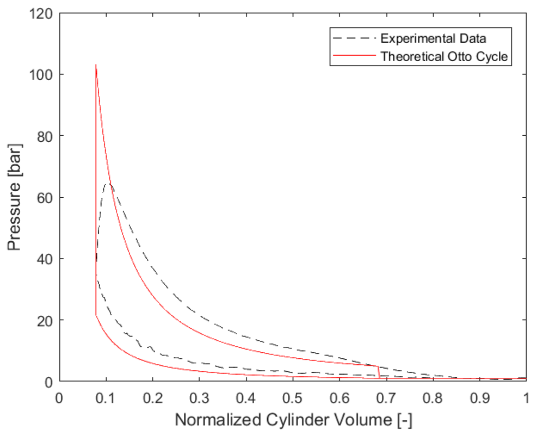


Figure 2.3: Ideal pressure cycle versus experimental data

The thermal efficiency is defined as:

$$\eta_{\mathrm{t}} = \frac{work\,produced\,per\,cycle}{heat\,available\,as\,input\,per\,cycle} \tag{2.26}$$

Since the engine under investigation has a trapped compression ratio of 5.66, the ideal thermal efficiency can be easily evaluated from the formula (2.24) and it is 0.50. It can be clearly seen that the the ideal value of the efficiency of the cycle is quite different from the one calculated experimentally, equal to 21%. Almost the half.

With a deeper investigation of the two cycles, it has to be pointed out that, even if the experimental and theoretical pressure trends in the cylinder are somehow similar, however, the ratios of the specific heats of the actual fluid are equal to 1.33 for the expansion and 1.17 for the compression, that are quite different of the ideal value of 1.4 for the air.

On the other hand, the actual compression stroke starts clearly before the trapping point (at the closure of the exhaust port), and this is also true for an engine without the tuned exhaust pipe. The assumptions of the isochoric combustion and exhaust process for the ideal cycle are quite far from the experimental data as it can be seen from the pressure trace. In fact, the peak pressure of the ideal cycle is 102 bar while the one measured at the test bench is 65, a quite remarkable difference.

The work on the piston along the cycle is ideally the work delivered at the crankshaft by means of the connecting rod. From the thermodynamic point of view, with the word "ideal" it is intended that friction or other losses, like the blow by, are not considered. Therefore, the ideal work produced each cycle (See equation (2.26) is the one carried out by the Force F, generated by the gas pressure p, on the piston. So the Work is given by the product of the Force and the displacement, x, or also it can be written like:

$$\text{Work produced per cycle} = \int F dx = \int pA dx = \int p dV \tag{2.27}$$

where A is the piston surface area.

Therefore, the work produced for any engine cycle, in the case of a two stroke engine for one rotation of the crankshaft, is the cyclic integral of the pressure-volume diagram. In the same way, the pumping work required is equal to the cyclic integral of the crankcase pressure-volume diagram. In both cases, the work is equal to the area enclosed in the pressure-volume diagram, be it a theoretical cycle or an actual one.

2.2.8 The concept of mean effective pressure

As said in the previous paragraph, the area enclosed in the pV diagram is the work produced in the cylinder, in the ideal and real cycle. In the Fig 2.4 another shaded rectangular area is shown, equal to the one enclosed by the pressure trace line in the p-V diagram. This rectangle has an height equal to a value called IMEP and a width equal to V_{sv}, where IMEP is the *indicated mean effective pressure* and V_{sv} is the swept volume. The word "indicated" has an historic origin. It derives from the fact that the pressure transducers for the engines were called "indicators" and the pV diagram was called "indicator card".

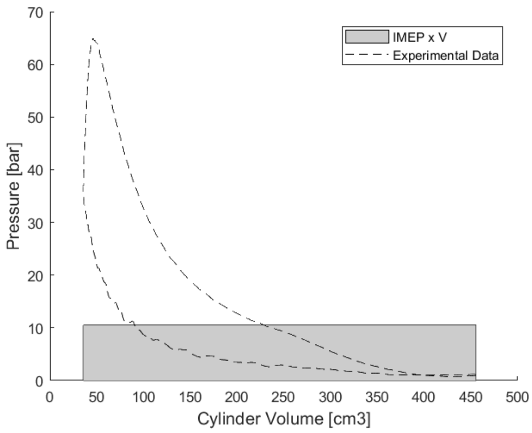


Figure 2.4: IMEP representation on a pV diagram

The mean effective pressure is an extremely useful parameter used by engineers to quantify the engine's capacity to do work that is independent of engine displacement. There are several types of MEP. These MEPs are defined by the location measurement and method of calculation. In general, the mean effective pressure is the theoretical constant pressure that, if it acted on the piston during the power stroke, would produce the same net work as actually developed in one complete cycle. The MEP can be defined as:

For example, the net indicated mean effective pressure, known as $IMEP_n$ is equal to the mean effective pressure calculated from in-cylinder pressure over the complete engine cycle (720° for a four-stroke engine and 360° for a two-stroke engine).

For two engines of equal displacement volume, the one with a higher MEP would produce the greater net work and, if the engines run at the same speed, greater power.

We can also use mean effective pressure, to define friction losses. The *brake mean effective pressure* (BMEP) is what we have at the crankshaft. BMEP is the difference between *indicated mean effective pressure* and *friction mean effective pressure* (FMEP).

$$\mathrm{bmep} = \mathrm{imep} - \mathrm{fmep} \tag{2.28}$$

where the friction mean effective pressure is the sum of *pumping mean effective pressure* (PMEP), *mechanical rubbing mean effective pressure* (RMEP) and *auxiliary mean effective pressure* (AMEP). It is often very difficult to segregate the separate contributions of frictions in measurements taken in a laboratory except by recording crankcase pressure diagrams and by measuring friction power using a motoring methodology that eliminates all pumping action at the same time [8]. So is quite common to refers directly to the friction mean effective pressure:

$$\mathrm{fmep} = \mathrm{pmep} + \mathrm{rmep} + \mathrm{amep} \tag{2.29}$$

These kind of phenomena deteriorates the performance of an engine, and their effects are taken into account with a parameter called mechanical efficiency, η_{m}. The mechanical efficiency of the engine is the ratio between the *brake mean effective pressure* and the *indicated mean effective pressure*:

$$\eta_{\mathrm{m}} = \frac{\mathrm{bmep}}{\mathrm{imep}} \tag{2.30}$$

Replacing equation (2.38) in (2.40) gives:

$$\eta_{\mathrm{m}} = \frac{\mathrm{imep} - \mathrm{fmep}}{\mathrm{imep}} = 1 - \frac{\mathrm{fmep}}{\mathrm{imep}} \tag{2.31}$$

From equation (2.41) it can bee seen that, the lower is the *friction mean effective pressure*, the higher is the mechanical efficiency of the internal combustion engine. So, reducing the friction losses for a given engine, will reduce the fuel consumption and improve the power output [7].

2.2.9 Power, torque and fuel consumption

The Power is defined as the rate of doing work. If the engine speed is rps, rotations per second, and since the two stroke engine has a working cycle each revolution, then the power delivered to the piston by the gas trapped in the cylinder is defined indicated power, $\dot{W}_{\mathrm{i}}$, where:

$$\begin{aligned} \dot{\mathrm{W}}_{\mathrm{i}} &= imep \times V_{\mathrm{sv}} \times (work\, cycles\, per\, second) \\ &= imep \times V_{\mathrm{sv}} \times rps \quad for\, a\, two\, stroke\, engine \end{aligned} \tag{2.32}$$

For a four stroke engine, that has a working cycle each two revolutions of the crankshaft, the speed in the formula (2.27) has to be divided by two. In other words, a four stroke engine with same power output and speed has the double of the imep of a two stroke engine. The indicated torque, $\mathrm{Z_j}$, is the turning moment of the crankshaft and it is related to the indicated power output by the following equation:

$$\dot{\mathrm{W}}_{\mathrm{i}} = 2\pi Z_{\mathrm{i}} rps \tag{2.33}$$

If the engine consumes is running with a known fuel mass flow whose lower heating value is equal to $\mathrm{H_i}$, then the thermal indicated efficiency, η_{i}, of the engine can be calcuated with an equation derived from the Eq. (2.25):

$$\eta_{\mathrm{i}} = \frac{power\ output}{rate\, of\ heat\ input} = \frac{\dot{W}_{\mathrm{i}}}{\dot{m}_{\mathrm{f}} H_{\mathrm{i}}} \tag{2.34}$$

Of high interest and common use in the engineering practice is the concept of specific fuel consumption, the rate of fuel consumption per unit of output power. Then, the indicated specific fuel consumption, isfc, is given by:

$$\mathrm{isfc} = \frac{fuel\, consumption\, rate}{power\ output} = \frac{\dot{m}_{\mathrm{f}}}{\dot{W}_{\mathrm{i}}} \tag{2.35}$$

Comparing the equations (2.29) and (2.30) it can be noticed that thermal efficiency and specific fuel consumption are mutually correlated. Since most of the petrol based fuel has similar lower heating value, instead of comparing the thermal efficiency, it is common use, and more practical for engineers, to compare the specific fuel consumption of two engines.

2.2.10 Brake Torque

The engine power is measured by means of an engine dynamometer, or simply dyno. The reaction torque on the external case of the dyno, that is exactly equal to the engine torque, is measured on a lever of lenght L, from the central line of the dyno, on which acts a force F. The lever avoids the external case to rotate so that the dyno is capable of absorbing the torque and the power of the engine under examination. As consequence, the reaction torque measured is the brake torque, M_b, and it is calculated as:

$$\mathrm{Z_b} = F \times L \tag{2.36}$$

Therefore, the work per revolution of the engine id the distance "traveled" by the force F on a circumference of radius L:

$$\text{Work per revolution} = 2\pi FL = 2\pi Z_b \tag{2.37}$$

The power measured, the brake power, $\dot{W}_b$, is the capability of the engine to supply work, and it is calculated as:

$$\begin{aligned} \dot{\mathrm{W}}_b &= (Work\,per\,rev) \times (rev/s) \\ &= 2\pi Z_b rps = \pi Z_b \frac{rpm}{30} \end{aligned} \tag{2.38}$$

The brake thermal efficiency, η_b, is then given by the following equation:

$$\eta_b = \frac{power\,output}{rate\,of\,heat\,input} = \frac{\dot{W}_b}{\dot{m}_f H_i} \tag{2.39}$$

The same logic can be applied for the brake specific fuel consumption, bsfc:

$$\mathrm{bsfc} = \frac{fuel\,consumption\,rate}{power\,output} = \frac{\dot{m}_f}{\dot{W}_b} \tag{2.40}$$

However, it is also possible to measure the mean effective pressure relative to the brake power measured. This is called brake mean effective pressure, bmep, and it can be derived from the equation (2.27):

$$\mathrm{bmep} = \frac{\dot{W}_b}{V_{sv} \times rps} \tag{2.41}$$

It is clear that the brake power and brake mean effective pressure are only part of the corresponding indicated power and indicated mean effective pressure, since the engine has lost a certain amount of power as consequence of the internal friction and other dissipative effects.

The lost mechanical work is regarded as "friction" work. It has the generic name "friction" but it contains all the engine losses [2].

The "friction" losses work is made up from:

- Pumping work W_p (intake and exhaust)
- Rubbing friction work W_r (piston assembly, connecting rod, crankshaft, balance shaft, valve train system)
- Auxiliary devices work W_a (oil pump, fuel pump, water pump, alternator, AC compressor, etc.)

$$W_f = W_p + W_r + W_a \tag{2.42}$$

Chapter 3

Pollutant emissions

3.1 Formation mechanisms of CO, HC, NOx and PM

3.1.1 Carbon monoxide - CO

The emission of carbon monoxide for a spark ignition engine, derives from the incomplete oxidation of such molecule in carbon dioxide. This phenomenon occurs mainly for two reasons: in rich mixture, for the lack of oxygen and during the expansion due to the freezing in composition of the intermediate combustion product. The former situation occurs mainly during cold start, acceleration transients and at high load-high speed where the fuel enrichment is usually adopted. But also for an overall stoichiometric mixture, the air fuel ratio imbalances among the cylinders cause a significant CO emissions. In the latter case, it has to be highlighted that the oxidation reaction of CO to CO2 is quite slow compared to the partial oxidation of hydrocarbon in CO. For this reason also for stoichiometric mixture the concentration of the carbon monoxide is relevant. Furthermore, due to the high temperature reached during combustion (above 2850 k) the following equation

$$CO2 \rightleftharpoons CO+O \tag{3.1}$$

reaches the equilibrium and the increasing of the reverse reaction rate leads to the dissociation of the CO2 in CO and O. Then, the step drop in temperature of the gases due to the expansion stroke of the piston, freezes the chemical composition.

In fact, the CO concentration at the tailpipe are found to be reasonably higher than what it could be expected from the chemical equilibrium at exhaust temperatures.

3.1.2 Nitrogen oxides - NOx

During the combustion, the high temperatures reached at the flame front dissociate the molecular bond of O2 and N2 and then atoms react leading to the formation of NOx behind the flame front. The reaction rate of the NO formation mechanism increase esponentially with the temperature. The value of the reaction constant can be calculated from the Arrhenius equation:

$$\mathrm{k} = Ae^{-\frac{E_\mathrm{a}}{RT}} \tag{3.2}$$

Where:
A = frequency factor.
Ea = activation energy.
R = Gas constant.
T = Temperature.

The formation mechanism just described is called *thermal mechanism* (or also *Zeldovich mechanism*). The condition by which there is the maximum production of Thermal NOx is for slightly lean mixture (λ 1.1), due to the high concentration of Oxygen and the high peak temperature reached during combustion. As for the carbon monoxide, also here due to the step drop in temperature during the expansion stroke, the chemical composition freezes and the concentration of NOx at the exhaust are higher than expected from the chemical equilibrium at exhaust temperatures. Another NOx formation reaction is called *prompt mechanism*. In this case, the mechanism extremely fast and less sensitive to temperature. It takes place inside the flame front where the CH radicals lead to the formation of HCN and N, that can both react with oxygen to form NO. The last formation mechanism, of marginal importance, involve the so called *fuel NO* where nitrogen oxides are formed from nitrogen bound in the fuel which is oxidized in fuel-lean regions to NO.

3.1.3 Unburned hydrocarbons - HC

Most of the HC emissions at tailpipe are made of partially unburned hydrocarbons, particularly reactive for photochemical smog reaction. The main mechanisms of formation of HCs are the followings:

Flame quenching
The presence of a thermal boundary layer, whose thickness is inversely proportional to the in-cylinder pressure, quenches the flame front near the walls due to the heat transfer which reduces the temperature too much for the oxidation reactions.

Fuel trapped into the oil layer
When the fuel partial pressure is high, part of the fuel vapor is absorbed by the oil layer and does not take part to the combustion. Further, the solubility of the oil depends on its thermodynamic condition, especially the lower the temperature the higher the solubility.

Mixture trapped into crevices
During the compression, part of the fuel is pushed into crevices where the flame front cannot propagate, such as in the ring pack volume. This effect is enhanced at cold start when the clearances are larger.

Bulk quenching
When the mixture is particularly lean or the amount of exhaust gas residuals overcomes the misfire limit, the flame front cannot propagate properly and a large amount of the fuel does not burn

Fuel trapped by deposits
They can absorb or release fuel during transient operations leading to an enleanment or enrichment of the mixture that increases the HC emissions

Mixture enrichment
During transients and full load-high speed operations where the excess of fuel is needed to reduce the exhaust gas temperature for component protection.

Another source of formation of HC is, especially for 2 stroke engines, the fuel that is shortcircuited directly to the exhaust port during the scavenging process of the cylinder.

All these phenomena leads to a decrease of the engine efficiency. It is worth to mention that part of the HC can oxidize during the exhaust stroke if the temperature level is high enough and if some oxygen is still available. For this reason, the condition by which there is the minimum emission of HC is for slightly lean mixture λ 1.15.

3.1.4 Particulate matter - PM

Particulate matter, usually called PM, is a complex mixture of extremely small particles and liquid droplets. The PM is composed of elemental carbon particles (soot) which agglomerate and adsorb other species, including nitrates, sulfates, organic chemicals, metals, and soil or dust particles, to form complex structures from the physical-chemical point of view. The size of particles is directly linked to their potential for causing health problems. The smaller the size of the particle, the deeper they can travel through our respiratory system and the worse are the health effects. Traditionally SI engines were never affected by a reasonable amount of particulate matter emissions. But more recently, innovative injection methods have been implemented in modern SI engines (such as direct injection, stratified-charge injection, etc.), that can originate extremely fuel rich zones ($\lambda < 0.5$) and so leading to PM formation. For this reason the more recent European emission standards limit also the number and the amount of PM emitted for SI engine.

3.2 Pollutant emissions regulation

The emission regulations for two stroke engine are defined mainly by the European emission standards for what concern the on-road vehicle and the US EPA standards (Environmental protection agency) for non-road mobile machinery.

3.2.1 Emissions standard for the on-road two and three wheelers

Euro emission standards are implemented by the European commission and, among all the different categories of road transport emissions regulated, it includes also the so called L-category that comprises the on-road two and three wheelers. At first the regulation focused on the reduction on the carbon monoxide (CO), unburned hydrocarbons (HC) and nitrogen oxides. Then, from Euro 4, an evaporative emission test was introduced (SHED Sealed Housing for Evaporative Determination) together with the on board diagnosis (OBD) in order to self-diagnose and to ensure emission control is maintained [10].

Now, with Euro 5, from 2020 for new motorcycles and for all L-category vehicles by 2021, along with a further tightening of CO, HC and NOx limits, also a limit of 0.045 g/km for particulate matter (PM) has been introduced. Another important parameter to consider is the durability of such emission requirements, it means for how long the vehicle has to comply the emission limits. The Table 2.1 briefly summarize all the aspects discussed above.

Table 3.1: L-category Euro emissions limits

	Euro 1	**Euro 2**	**Euro 3**	**Euro 4**	**Euro 5**
CO [g/km]	13.0	5.5	2.0	1.14	1.0
HC [g/km]	3.0	1.0	0.3	0.17	0.1
NOx [g/km]	0.3	0.3	0.15	0.09	0.06
PM [g/km]	-	-	-	-	0.0045
SHED test	-	-	-	Yes	Yes
OBD	-	-	-	OBD1	OBD2
Durability	-	-	-	20000 km	Lifetime

The emissions measurement is performed during the so called World Motorcycle Test Cycle (WMTC). Since 2017 this procedure is adopted from all European manufacturers, Brazil, Chile, Indonesia, Singapore, Thailand, Vietnam, India and Japan while in the USA, the WMTC is accepted as an optional regime to the local FTP (Federal Test Procedure).

Up to now, there is no regulations aiming to the reduction of particulate number PN and CO2, probably because compared to other sources of pollution, the contribution from motorcycles is very small [9].

3.2.2 Emissions standard for the non-road mobile machinery

For what concern the non-road mobile machinery (NRMM) European standards harmonize with the US EPA standards. NRMM includes control emissions of engines that are not used primarily on public roadways (All-Terrain-Vehicles ATVs, snowmobiles, skidoos, etc.).

The emissions measurement is performed during the so called Non-road Transient Cycle (NRTC) developed by the US EPA in cooperation with the authorities in the European Union (EU).The cycle is a transient driving schedule and it is performed on an engine dynamometer. It lasts 20 min. ca.. In Fig. 3.1 the normalized engine speed and torque during the NRTC test are shown.

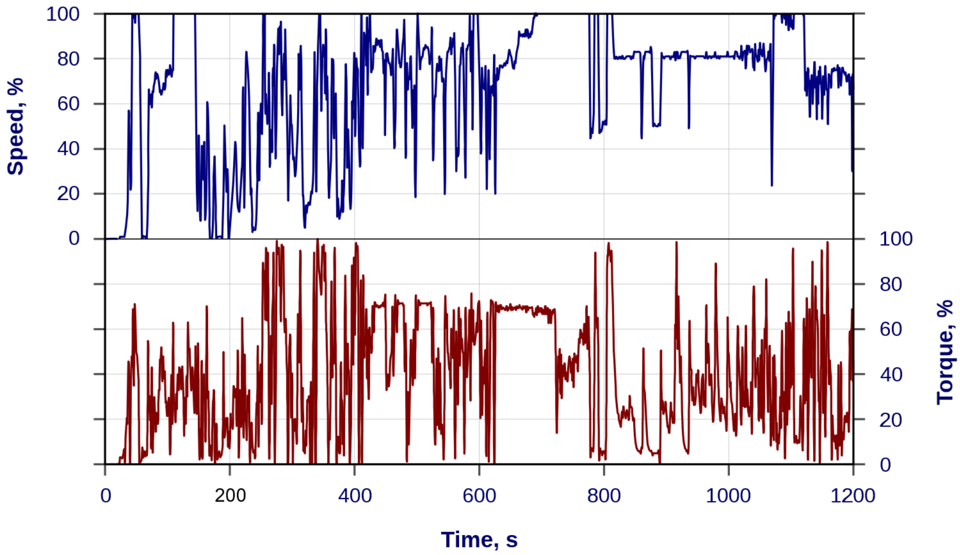


Figure 3.1: Normalized speed and torque over NRTC cycle

The NRTC is run twice, with a cold and a hot start, with a waiting period of 20 minutes between the 2 runs [11]. The emission limits for the NRMM are briefly summarized in the Table 2.2 [12].

Table 3.2: Recreational Vehicle Exhaust Emission Standards

Vehicle	**Model Year**	**Emission standards**	
		HC [g/kWh]	**CO [g/kWh]**
Snowmobile	2006	100	275
	2007 - 2009	100	275
	2010	75	275
	2012	75	200
Off road Motorcycle	2006	2.0	25.0
	2007 - ongoing	2.0	25.0
ATV	2006	1.5	35.0
	2007 - ongoing	1.5	35.0

Chapter 4

Scavenging

The process of emptying the cylinder of burned gases and filling it with a fresh mixture (or air) - the combined exhaust and intake process - is called scavenging.

The two stroke engine, in order to supply power for each rotation of the crankshaft, it has to rely on the action of an auxiliary device, known as scavenging pump, to remove the exhausted gases from the cylinder and filling it with a new charge. The scavenging pump can be physically separated from the cylinder or the combined action of cylinder-crankcase-piston can work as a scavenging pump itself.

The ideal goal of the scavenging process is to substitute completely combustion products in the cylinder with a fresh charge supplied from the intake ports, without losing any amount of the fresh mixture through the exhaust port. In this way, the engine can deliver the maximum theoretical power. In practice, to obtain this ideal scavenging process is not possible, so that the quality of such process can be evaluated comparing the mass of fresh air trapped into the cylinder with respect to the mass of fresh air supplied from the scavenging pump [12].

4.1 Method of the scavenging the cylinder

Scavenging arrangements are classified into: (a) cross-scavenged, (b) loop-scavenged, and (c) uniflow-scavenged configurations. The location and orientation of the scavenging ports control the scavenging process, and the most common arrangements are indicated. Cross- and loop- scavenging systems use exhaust and inlet ports in the cylinder wall, uncovered by the piston as it approaches BDC. The uniflow system may use inlet ports with exhaust valves in the cylinder head, or inlet and exhaust ports with opposed pistons.

4.1.1 Crossflow scavenging

In a cross flow scavenged engine, intake and exhaust ports are placed on opposite sides of the cylinder, and a deflector on the top of the piston directs the fresh charge on the upper part of the cylinder, pushing the exhaust gases on the other side of the deflector and out through the exhaust port. the deflector increase the piston weight and the exposed surface, affecting the cooling of the piston and making difficult to design an efficient shape of the combustion chamber.

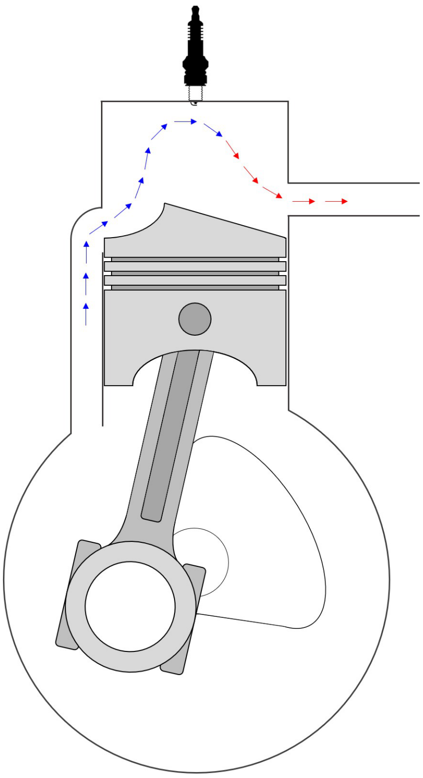


Figure 4.1: Crossflow scavenging

In addiction to that, with a non compact combustion chamber in which there is a protuberant deflector exposed, the engine has no significant improvements in terms of power and fuel consumption. The potential for a detonation or preignition, derived from the high surface/volume ratio of the combustion chamber and from the edges of the hot deflector, is quite high and so the compression ratio that can be adopted for this kind of engine tends to be a little smaller with respect to the correspondent loop scavenged engine. This kind of engine, on the other hand, present some advantages in terms of manufacturing and packaging over the loop scavenged engine. The cylinder-to-cylinder spacing for a multicylinder configuration is as small as possible (as far as a proper cooling of the cylinders is guaranteed). The equivalent situation for a loop scavenged engine, shown in Fig. 4.2, is limited by the presence of the transfer ports on the sides of the cylinder. Furthermore, it is possible to drill the intake and the exhaust ports with one operation, reducing, in this way, the production costs of the cross flow scavenged engine with respect to the equivalent loop or uniflow scavenged one.

4.1.2 Loop scavenging

The goal is to produce an optimal scavenging process in a cylinder with two or more ports (intake and exhaust ports) placed on the same side of the cylinder by means of a piston whose top is usually flat.

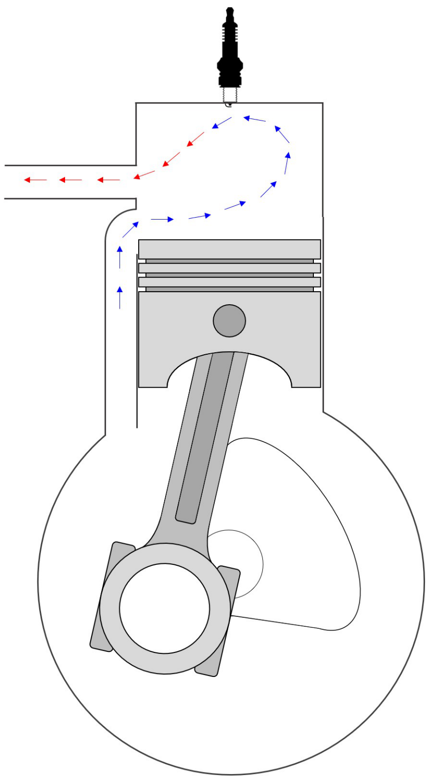


Figure 4.2: Loopflow scavenging

In the Fig. 4.2 a possible layout of loop scavenged engine can be seen. The common element for this layout is the sweep back angle of the "main" transfer ports, directing the flow motion far from the exhaust port. (main transfer port near to the exhaust port). An advantage of the loop scavenge design is the possibility to adopt a compact combustion chamber above the flat piston that allows a fast and efficient combustion process.

4.1.3 Uniflow scavenging

In an uniflow engine, the mixture, or the air for diesel or direct injection gasoline engine, enters from one end of the cylinder (usually an intake ports controlled by the piston motion) and exits, once the combustion occurs, on the opposite end (usually through a valve or an exhaust port). The scavenging gas flow moves in one direction only, and so the name uniflow.

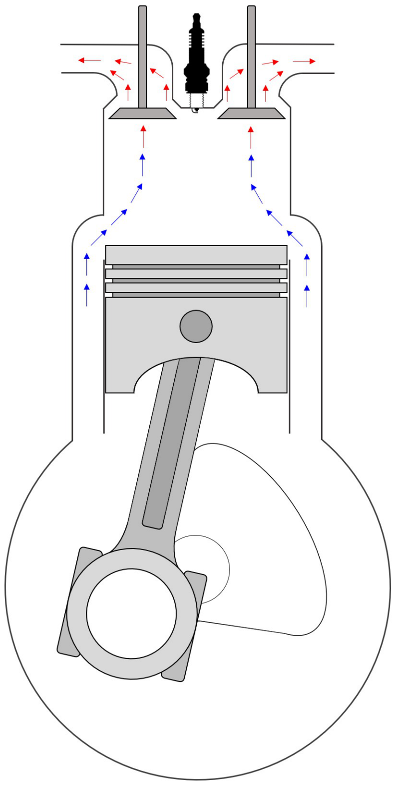


Figure 4.3: Uniflow scavenging

Usually the charge turbulence is enhanced by properly placing and orienting the ports or masking the poppet valves. The swirled motion of the charge is particularly suited to promoting an efficient combustion. The uniflow scavenging, while being good, is not significantly better than loop scavenging. For spark ignition engine, since the uniflow scavenging involves a reasonable higher mechanical complexity with respect to the other scavenging arrangements and since actually there is not a clear improvement in terms of performance, the increase of engine weight and cost is not justified by the small advantage in terms of power or of efficiency. For these reasons the uniflow scavenging arrangement is more rarely adopted.

4.2 Scavenging not employing the crankcase as an air pump

The essential characteristic of the simplicity of a two stroke engine is the use of the crankcase as air pumping device for the engine. Most of the two stroke engine designs are based on this kind of concept. The conventional lubrication method consists in mixing the lubricant with the fuel and supply it through a carburetor in ratio gasoline/lubricant that varies from 25:1 to 100:1, depending on the application, on the ability of the engine designer and on the characteristics of the bearings adopted for the crankshaft.

Since the lubrication is of the total-loss type, and about the 10-30% of the fuel charge is shortcircuited in the exhaust duct with the air, the exhaust manifold is rich of unburned hydrocarbons and lubricant, some of which partially burned and others completely unburned, and it is clearly visible like smoke. Nowadays this is ecologically unacceptable and so two stroke engine manufacturers have introduced some separate devices for pumping the lubricant oil in order to reduce its consumption and as a consequence to reduce the emissions of oil in the atmosphere.

Such systems can reduce the effective oil/gasoline ratio up to 200 or 300 and leading to an oil consumption comparable with four stroke engines. Even so, the visible exhaust smoke is still unacceptable and so, for future projects, like it is always been for marine and automotive two stroke diesel engine, it can be used a pressure fed lubrication system of the crankshaft based on plain bearings with dry or wet sump. By definition, it means that the crankcase cannot anymore be used as air pumping device and so an external pumping device is needed. This can be both a positive displacement blower of Roots type, or a centrifugal blower driven by the crankshaft. Clearly It would be thermodynamically more efficient a turbocompressor, where the exhaust gases expand in a turbine that supply the power to the compressor. A combined use of a blower and a turbocompressor is shown in the Fig. 3.4. The blower works mainly at low speed improving the low end torque, while the turbocompressor works at its best efficiency and performance at high speed and high load. To avoid the shortcircuit of the fuel to the exhaust, an injector should be employed in order to supply the fuel directly into the cylinder, ideally after the closure of the exhaust port. An engine of this type offers good results in terms of fuel consumption and unburned hydrocarbons, carbon monoxide an nitrogen oxides emissions compared to an equivalent four stroke engine.

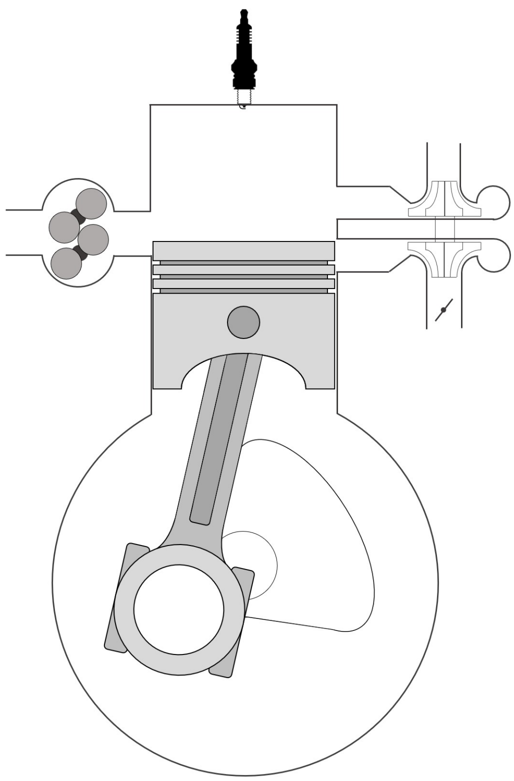


Figure 4.4: Turbocharged 2 stroke engine

However, it has to be pointed out that an engine with the crankcase-cylinder-piston as air pump and lubrication of the total-loss type allows a simple design, lightness and high specific power. For such characteristics no others solutions are an effective alternative, and these requirements are of utter importance for a wide range of applications: from chainsaws to recreational transportation (snowmobiles, ATVs, mopeds, etc.). Alternative to this architecture would be an heavy, bulky and expansive four stroke engine.

4.3 Pressure waves

Performance characteristic of a two stroke engine are significantly controlled by the unsteady gas motion and by pressure waves. This is true for all kind of two strokes engines, from the 2 hp outboard engine to the 150 Gran Prix one. If the tuned exhaust pipe is removed from a monocylinder engine while it is running at maximum power, the power could drop at least of 50% at that speed. The tuned exhaust pipe exploits the motion of the pressure wave generated at the exhaust port opening in order to trap a larger mass of air inside the cylinder. Without it, the engine can trap only half of the fuel air mixture in the cylinder. In order to design such exhaust systems, and improve engine performance, it is necessary to have a clear understanding of the pressure wave reflection mechanism, explained qualitatively in the following paragraphs.

4.3.1 Pressure wave reflection in a pipe open at both ends

Let's consider a positive pressure wave inside a pip open at both ends. It reaches one end of the pipe and its momentum carries it out at the open air, where it expands in all directions. Now, since it expands in all directions, its pressure drops down really fast up to nearly the atmospheric pressure. However, it has still momentum to moves away from from the pipe end. As consequence, it creates a small suction: the air that follows behind is sucked out from the pipe (like the air flow that is sucked behind a truck that is moving at high speed). Now the suction at the end of the pipe draws air further up the tube, that in turn draws air further up the tube and so on. The result is that a high pressure wave that moves along a pipe is reflected at its open end as a low pressure wave, with a phase variation of 180°. In an open-open pipe, such reflection occurs at both ends.

Summarizing, at the open end of a pipe the pressure wave is reflected with another pressure wave with opposite sign.

(a) incident compression wave

(b) reflected pressure wave

Figure 4.5: Pressure wave reflection in a pipe open at both ends

4.3.2 Pressure wave reflection in a pipe closed at one end

Let's now consider the pressure wave reflection in a pipe with a closed end. Here the explanation looks easier: the positive pressure waves pusher the air against the closed end, it is pushed back (third Newton's law) away from pipe closed end. So, a positive pressure wave is reflected as a positive pressure wave, with a phase change of zero in pressure. An additional thing that can be noted is that the oscillation period in the first example (open-open pipe) is two time the pipe length, while for the closed-open pipe it lasts four time the pipe length [13].

(a) incident compression wave

(b) reflected compression wave

Figure 4.6: Pressure wave reflection in a pipe closed at one end

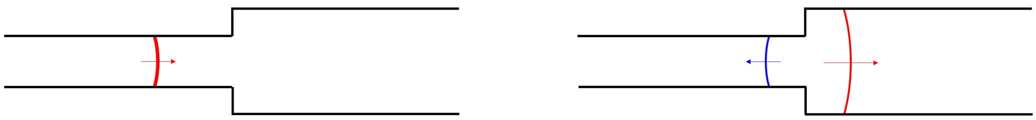
4.3.3 Reflection of pressure waves at a sudden area change

It's quite common to have a sudden area change in a pipe or in a duct connected to the engine. The main difference, from the dynamic point of view, is that, while in a pipe with a sudden area change the flow is always considered to be unidimensional, for a pipe connected to a plenum the flow is considered to be unidimensional in the pipe and tridimensional in the volume. One additional consideration is that the speed of the particle in a plenum is so low that can assumed equal to zero in any thermodynamic analysis. This leads to a change in amplitude of the transmitted impulse beyond the area change and also causes a reflection of the pressure wave.

Different kinds of sudden area change are reported in the following paragraphs, and the area can contract or expand at the junction. In all the examples below, the incident wave, travelling towards the sudden area change, moves from left to right.

- **An enlargement for an incident compression wave**

The sudden enlargement of the area behaves like an "open end" of a pipe but a little less effective, where the reflected pressure wave is one of expansion, while the onward transmitted pressure wave in pipe 2 is also one of compression.



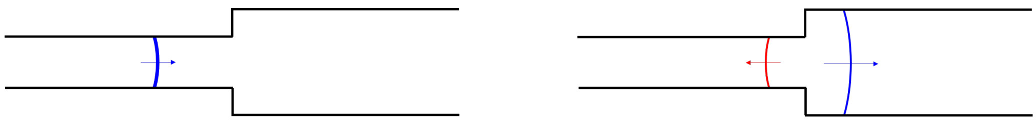
(a) incident compression wave

(b) reflected expansion wave and transmitted compression wave

Figure 4.7: Compressure wave reflection in a sudden area enlargement

- **An enlargement for an incident expansion wave**

As said above, the sudden enlargement of the area behaves like an open end but less effective, since it can be proved that in a case of a "perfect" bellmouth open end it produces a higher pressure ratio of the reflected pressure wave, instead of a weaker value in this case. The onward transmitted pressure wave in pipe 2 is an expansion wave, but with a lower pressure ratio.



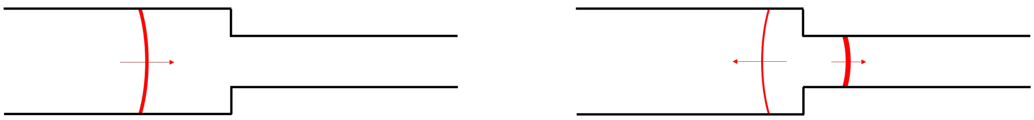
(a) incident expansion wave

(b) reflected compression wave and transmitted expansion wave

Figure 4.8: Expansion wave reflection in a sudden area enlargement

- **A contraction for an incident compression wave**

The sudden contraction behaves like a partially closed end, reflecting back a partial "echo" of the incident pulse. The onward transmitted pressure wave is also one of compression, but with a higher pressure ratio.

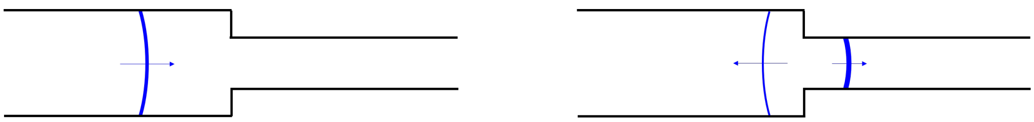


(a) incident compression wave **(b)** reflected and transmitted compression waves

Figure 4.9: Compression wave reflection in a sudden area contraction

- **A contraction for an incident expansion wave**

Like in (c), the sudden contraction of the area behaves like a partially closed end, reflecting back a partial "echo" of the incident pulse. Also the onward transmitted pressure wave is one of expansion, but it has to be noted that it has a higher expansion pressure ratio.



(a) incident expansion wave **(b)** reflected and transmitted expansion waves

Figure 4.10: Expansion wave reflection in a sudden area contraction

The theoretical presentation here is clearly too simple to be completely accurate in all circumstances. It is, however, intended to be a qualitative guide for the description of pressure wave reflection and transmission.

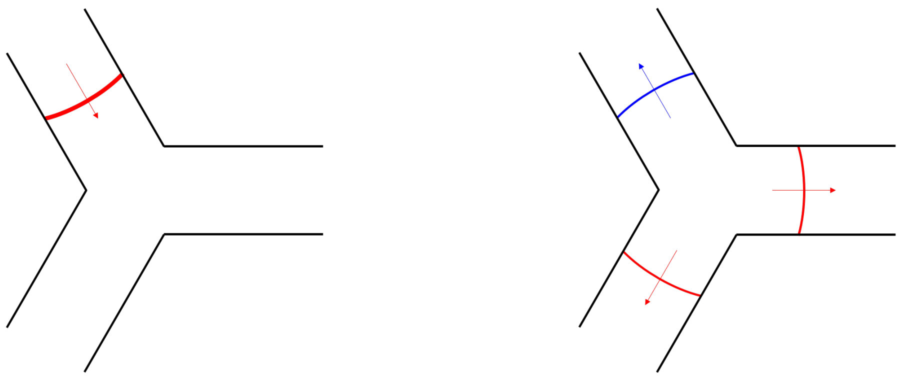
4.3.4 Reflections of pressure waves at branches in a pipe

An example of typical branch is shown in Fig. 4.11. The sign convention for all the branch theory here presented is that the propagation of a pressure wave toward the branch is considered as "positive".

Not surprisingly, the branched pipe can act both as a sudden area contraction or a sudden area enlargement for the gas flow. In different words, two pipe can supply gas flow to one pipe or a pipe supplies the other two. Let's consider these two cases in which all the pipes have the same area.

- **A compression wave coming down to the branch in pipe 1 through air and all other conditions in the other branches are "undisturbed"**

Pipe 1 supplies the other two, so the effect is an expansion. For what concern the pipe 1, the result is exactly the same as for a 2:1 sudden area enlargement discussed in the previous section. In the branch, the incident wave splits evenly between the other two pipes, transmitting onward a compression pressure wave and reflecting back a expansion pressure wave.

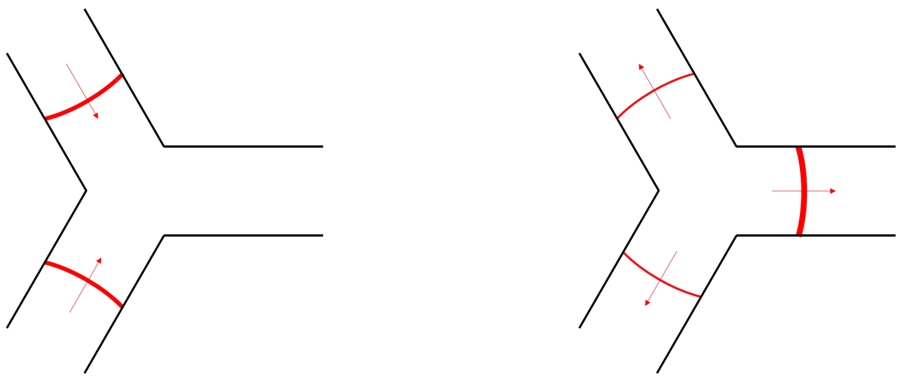


(a) incident compression wave **(b)** reflected and transmitted waves

Figure 4.11: Compression wave reflection at a branches in a pipe. Case 1.

- **Compression waves of pressure ratio arriving as incident pulses in pipes 1 and 2 leading up to the branch with pipe 3**

Pipes 1 and 2 supply the third pipe, so the effect is a compression. Pipe 3 has "undisturbed" condition. Now the branch behaves for the gas flow as a 2:1 area contraction. The effect of the contraction is highlighted by the reflection and the transmission of pressure waves.



(a) incident compression waves **(b)** reflected and transmitted waves

Figure 4.12: Compression wave reflection at a branches in a pipe. Case 2.

It is clear that also the angle between the branches affects in some way the amplitude of the transmitted and reflected pressure wave. When there are pipes with different area and a mix a compression and expansion wave interact in the branches, the qualitative description become a lot more complicated to have a clear understanding of what is happening [2].

At this point, with the aid of software tools the designer can focus more on the relevance of the results calculated by the software and less on the complex mathematical equations that rule these phenomena. One of this tool is GT-Power, used for the development of the methodology, that, by means of 1D modelling of the complete engine architecture, allows to study the dynamic effects and optimize the system.

4.4 Expansion exhaust manifold

It is true that the major contribution towards achieving actual high level of performance of the two stroke engine is given by the increasing understandings in the design of the exhaust systems. In origin, the exhaust tailpipe was designed to get the burnt gases out of the cylinder as fast as possible. Then, once engineers learnt more about pressure wave propagation and reflection, they tried to design the exhaust system in order to optimize the scavenging process.

As said at the beginning of this chapter, if the tuned exhaust pipe is removed from a two stroke engine while it is running at maximum power, the power could drop of 50% at that speed. Let's now analyse in detail how the tuned exhaust pipe exploits the motion of the pressure wave generated at the exhaust port opening in order to trap a larger mass of air inside the cylinder.

As the exhaust port opens, the burned gas, under still a considerably high pressure, are pushed in the exhaust duct, forming a positive pressure waves that moves at the speed of sound along the pipe.

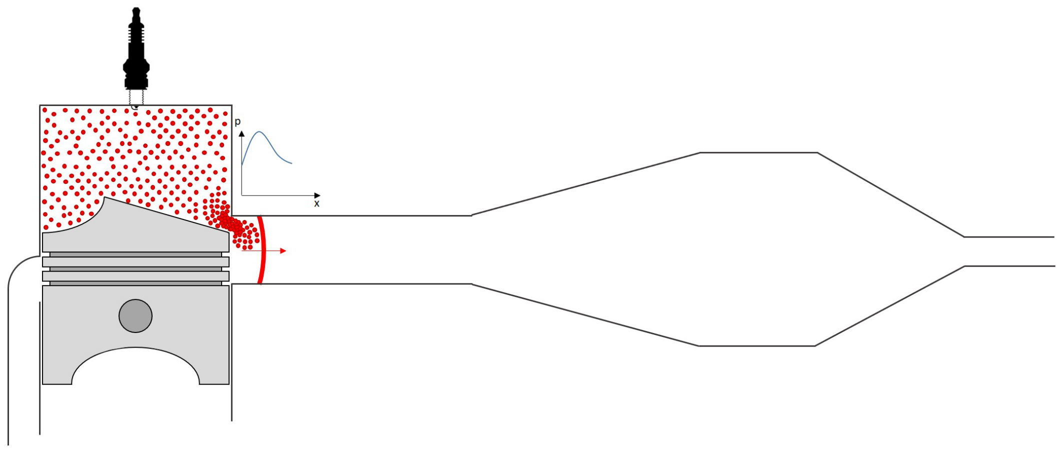


Figure 4.13: Blowdown pulse

After travelling for a relatively short distance, this pressure waves arrives at the first part of the expansion chamber, that is a diffuser (commonly called a megaphone). The walls of the diffuser diverge, and the wave reacts as it approaches a partially open end of a pipe, and as explained in this chapter, it is reflected back towards the the cylinder with opposite sign. In other words, the positive pressure wave is reflected back as a negative one. The biggest difference between the action of a diffuser and an open end is that the former gives back a stronger and more prolonged reflected pressure wave; It is a much more efficient converter (or inverter) of wave energy.

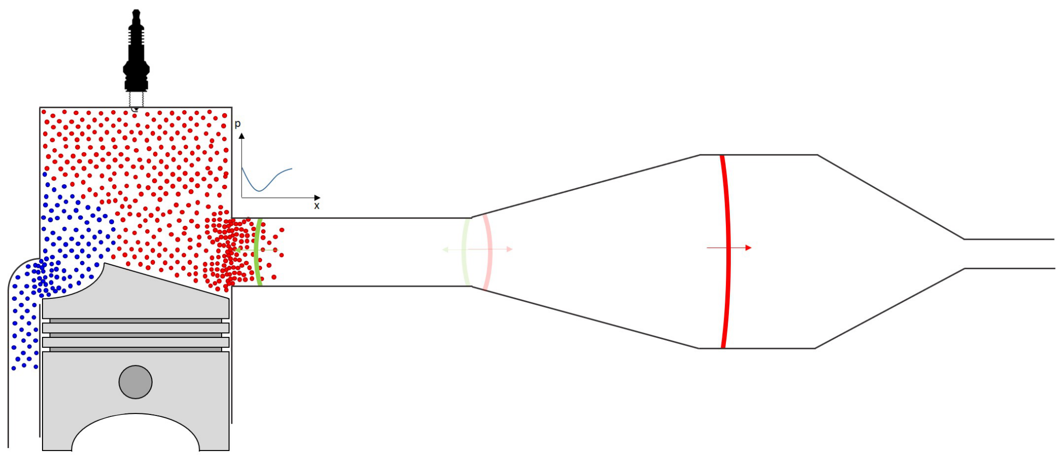


Figure 4.14: Suction pulse

While the initial wave is still moving downstream the diffuser, the process of inversion continues, giving back a negative pressure wave with a considerably high amplitude and duration. Further, overlaid on this effect there is the inertia of the fast moving exhaust gases, and the total effect is the creation of vacuum besides the exhaust port. This vacuum is stronger than one might suppose, reaching values of -7 psi. Adding this value to the one of the pressure of +7 psi inside the crankcase that works to push the fresh charge through the transfer ports inside the cylinder and it is clear how the scavenging process is that fast. Obviously, this pressure difference is particularly suitable to swept away also the residual exhaust gases coming from the previous power stroke. It is almost like having a compressor installed on the intake side but without all the mechanical complexity derived from it.

The power is directly proportional to the amount of trapped air/fuel mass in the cylinder at the beginning of the compression stroke, so the vacuum effect has to stop in th right moment, otherwise the negative pressure wave at the exhaust would push outside the cylinder a reasonable quantity of the fresh charge. For this reason, after the diffuser, and sometime after a straight chamber, the pressure wave encounters a convergent cone that behaves like the closed end of a pipe. Part of the energy has been already supplied to the expansion wave generated at the diffuser, but still there is enough of its original energy to have a reflection also in this convergent end, and so the positive pressure wave is reflected back with the same sign. Hopefully, in the right moment, this positive pressure wave arrives at the exhaust port, blocking the outflow of the fresh charge. In fact, it will momentarily invert the flow in that point, stuffing what it would have been otherwise lost in the cylinder.

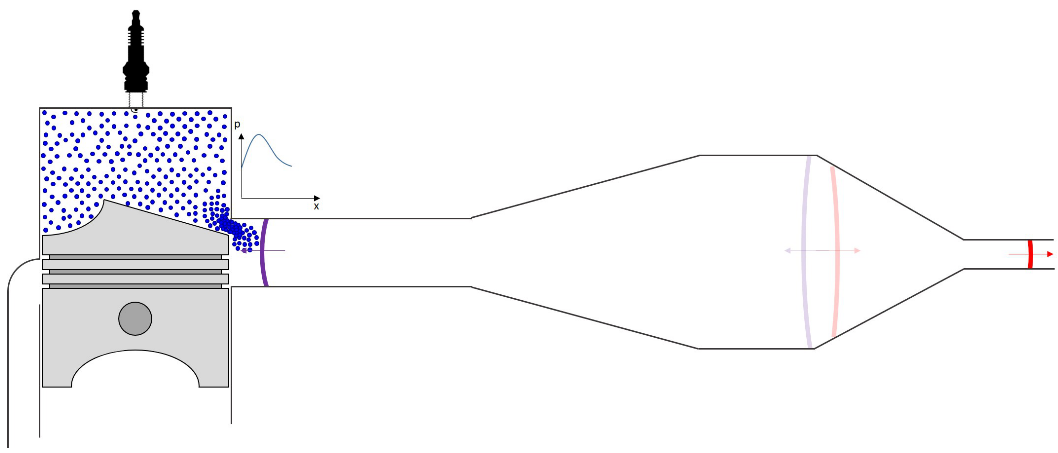


Figure 4.15: Stuffing pulse

The net result of these phenomena ruled by a perfectly tuned expansion exhaust chamber - once sucking out the exhaust gases and then stuffing the fresh charge in the cylinder - lead to a great boost in power.

The entire process can work magnificently good but it can also miserably fail if all the part of the expansion chamber are not properly dimensioned. All the pressure waves that suck and stuff the gas flow have to work in accordance with the engine's requirement. In this case, the propagation speed of the pressure waves is strictly related to the temperature of the exhaust gases and the time intervals between the initial wave departure and reflected wave components arrivals is function of speed of sound and the length of the system. So the task of designing an expansion chamber for a particular application is based on finding the right lengths, diameters and cone angles that will drive the pressure waves inside the exhaust systems for the benefit of the engine [5].

A CFD tool helps a lot the designer to find the right dimensions of the mechanical components without the need of any hardware, saving time and resources.

Chapter 5

Simulation Methodologies

To exploit all the advantages of the 2 stroke engine regarding the high power density and low production cost, advanced simulation methodologies has to be developed in order to optimize the scavenging process [14]. This means that the quality of the results has to be increased while the computational time reduced, while still getting a predictive simulation. So that the results are available early enough to be able to have an influence on the design of the engine. Of course, the thermodynamic simulations have to provide the engineers relevant information on the feasibility of some concepts with respect to others [17]. In such a way the understandings on the mechanisms that cause the scavenge losses, the mixture preparation and the AFR distribution in the combustion chamber are furtherly enhanced. It is worth reminding that these kind of informations can only be obtained from a simulation and not from test bench analysis. If the task is accomplished, this will allow the evaluation of different design alternatives and the optimization of them in the early stage of the 2 stroke engine development process [15]. The different engine concepts can include for example: variants of the ports, of the combustion chamber shapes, of injection technique, of the exhaust expansion chamber. Then, once the basic geometry parameters have been chosen, they are kept fixed and the optimization on the geometry is performed.

The use of numerical simulation software tools in the early stage of the engine development are already quite spread in engine simulation departments of the main automotive manufacturers. The main goal of the simulation analysis is to reduce the development time and increase the product quality without the needing of building a prototype engine as well as deepen all the physical processes behind a 2 stroke engine [16]. The loop-scavenged engine, like the one analysed in this thesis, is characterized by high scavenging losses due to the simultaneously opened exhaust and intake ports, and furthermore, It is not an easy task to measure at test bench the amount of residual gases trapped in the cylinder and the fresh charge short-circuited a the exhaust.

The possible simulation approaches that can be adopted in the analysis and optimization of a 2 stroke engine are described in the following sections

5.1 0D Model

In 0D-models all physical characteristics are only function of time (crank angle). Their calculation is based on the first law of thermodynamics. With this kind of model, of course, it is not solved local phenomena in the control volume but it is possible to assess very accurately the energy balance of the processes in a very short calculation time. The main assumption for 0D-models is is that the gas in the control volume is a mixture of ideal gases which are perfectly mixed. Physical parameters are evaluated by means of: Conservation of energy, conservation of mass and equation of thermodynamic state

5.2 1D Model

In 1D-models an additional geometric dimension is considered. So the physical parameters are function of time and position. Here the main assumptions are the geometrical dimensions reduced at 1D and the turbulence considered by means of friction parameters. In addition to the three equation solved in 0D models, here it is also solved the momentum conservation equation. 1D simulations are commonly used to calculate axial pressure pulses, flow rates and average temperatures in the cross-section of the fluid system. Main advantages of this approach are the simple model build-up and the fast numerical calculation. On the other hand, the well-known drawbacks are the inaccurate geometry simplification and the difficulties to describe accurately enough phenomena such as the scavenging process.

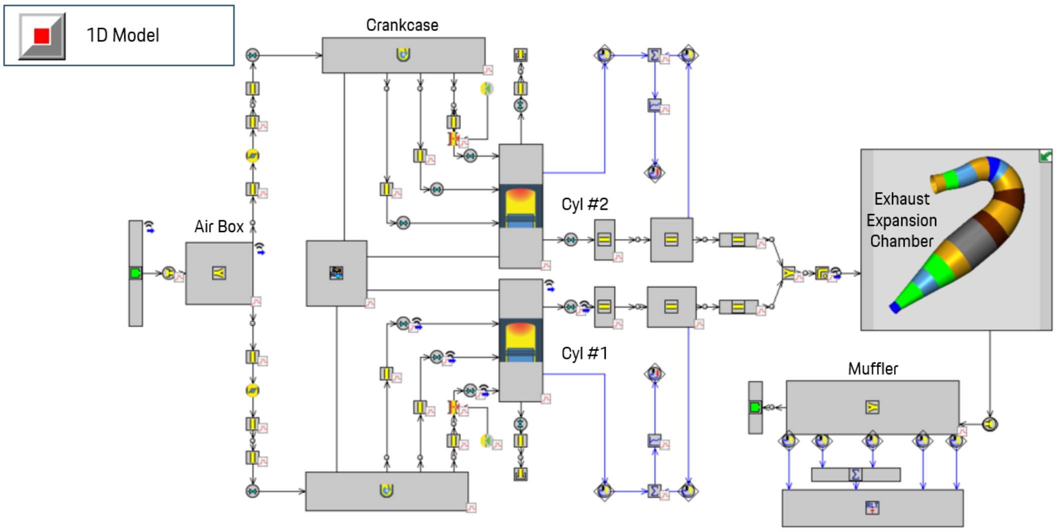


Figure 5.1: 1D engine model

5.2.1 Scavenging function

The tool that describes the gas exchange in GT-Power is called *Scavenging Function*. It defines the relation between two quantities: The cylinder Residual Ratio (CRR) that characterize the gas content inside the cylinder during the scavenging process. It is the ratio of the in-cylinder burned gas mass and the total in-cylinder mass. At the beginning of the exhaust port opening the CRR is equal to 1, supposing a complete combustion, and it goes to zero as all the burned gas is replaced by the fresh charge.

$$\text{Cylinder Residual Ratio} = CRR = \frac{m_{\text{brn}}}{m_{\text{tot}}} \tag{5.1}$$

Then, there is the so called Exhaust Residual Ratio (ERR) that describes the mass flow exiting the cylinder. It is the ratio of the burned gas mass flow at the exhaust port, with respect to the total mass flow. In an ideal 2 stroke engine the ERR is always equal to one, since for a ERR<1 means that part of the fresh charge is shortcircuited to the exhaust, leading to an increase of fuel consumption and pollutant emissions.

$$\text{Exhaust Residual Ratio} = ERR = \frac{\dot{m}_{\text{brn}}}{\dot{m}_{\text{tot}}} \tag{5.2}$$

The following picture qualitatively describes the quantity defined above.

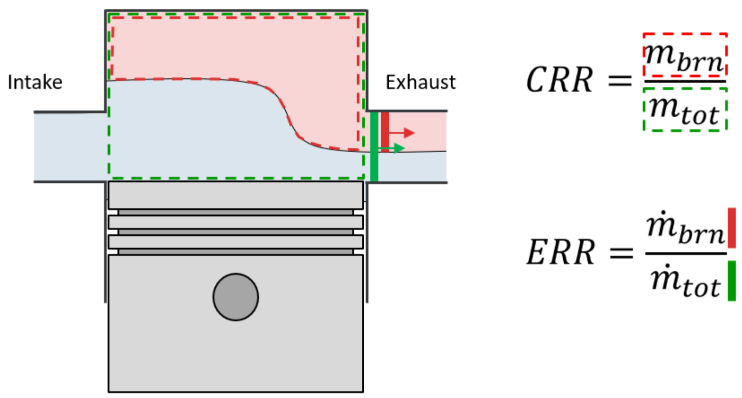


Figure 5.2: Cylinder and Exhaust Residual Ratio

A Perfect Scavenging and a Real Engine Scavenging are shown in terms of CRR and ERR in the picture below.

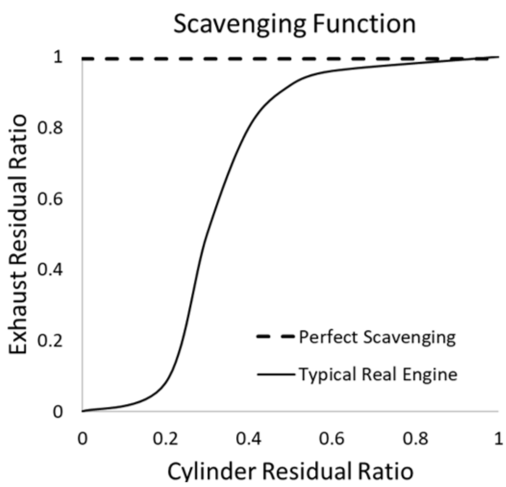


Figure 5.3: Scavenging function for an Ideal and Real 2 Stroke Engine

The limits for such approach are many. The scavenging function is given as input from 3D CFD analysis and it is not able to predict the effect of a change in the geometry of the exhaust expansion chamber (e.g.) on the gas exchange process. In addition to that, since it is a 1D modelling, it is not possible to get any information concerning mixture preparation, charge motion and combustion. For such kind of analysis a 3D approach is required.

5.3 3D Model

In 3D-models the complexity of intake, exhaust ducts, crankcase and cylinder are perfectly reproduced by means of 3D software tools. The computational time is strictly affected by the number of cells in the control volume and by the number of moving parts. The main advantages that, on the other hand, are possible to get from a 3D simulation are many: detailed analysis of the flame propagation, evaluation of locally different AFR in the cylinder, estimation of the exhaust residual gases trapped in the cylinder, etc. The negative side of such approach is that it requires much more effort for generating and for solving the simulation model than 0D or 1D simulations methods.

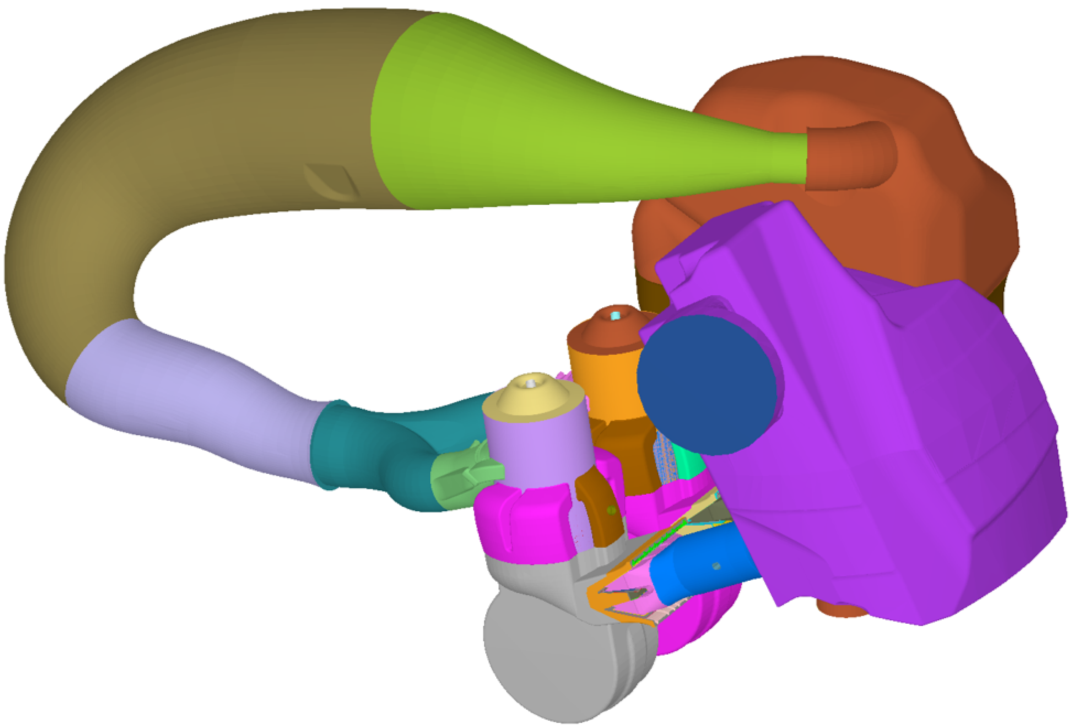


Figure 5.4: 3D engine model

5.4 1D-3D Model

The methodology that combines the short calculation duration and detailed results (where needed) is the 1D-3D CFD coupled approach. Crucial point of such method is the location of the coupling interface, since, at the 1D-3D coupling interface, the physical parameters (speed, pressure, temperature) has to be average out from the 3D-cells and given to the 1D software [18]. In the same way the 1D results have to be imposed to the cells in the 3D domain.

5.4.1 Requirements

In order to minimize the errors, the coupling interface has to be placed in a straight zone, otherwise, placing it near bends or in tapered pipes can cause problems due to to the radial components of the flow that can be ignored.

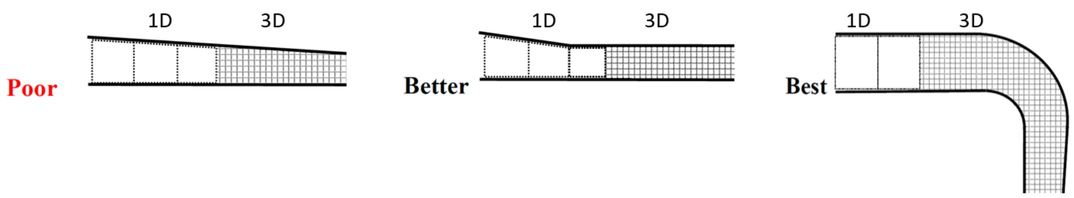


Figure 5.5: Recommended requirement for the positioning of the coupling interface

In addition to that, flow entering and leaving the 1D/3D boundary regions has to be fully one-dimensional and perpendicular to the 1D/3D interface plane otherwise oscillations, high frequency signals, or divergence may occur.

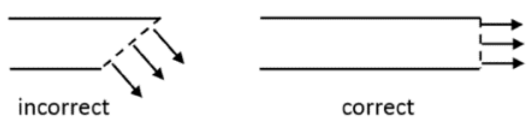


Figure 5.6: Correct and incorrect location of the coupling interface

5.4.2 Types of coupling

With the combined use of GT Power and CONVERGE the possible CFD interface types are the following:

1. Pressure

- GT provides static pressure to CONVERGE
- CONVERGE returns (a) velocity /(b) mass flow to GT

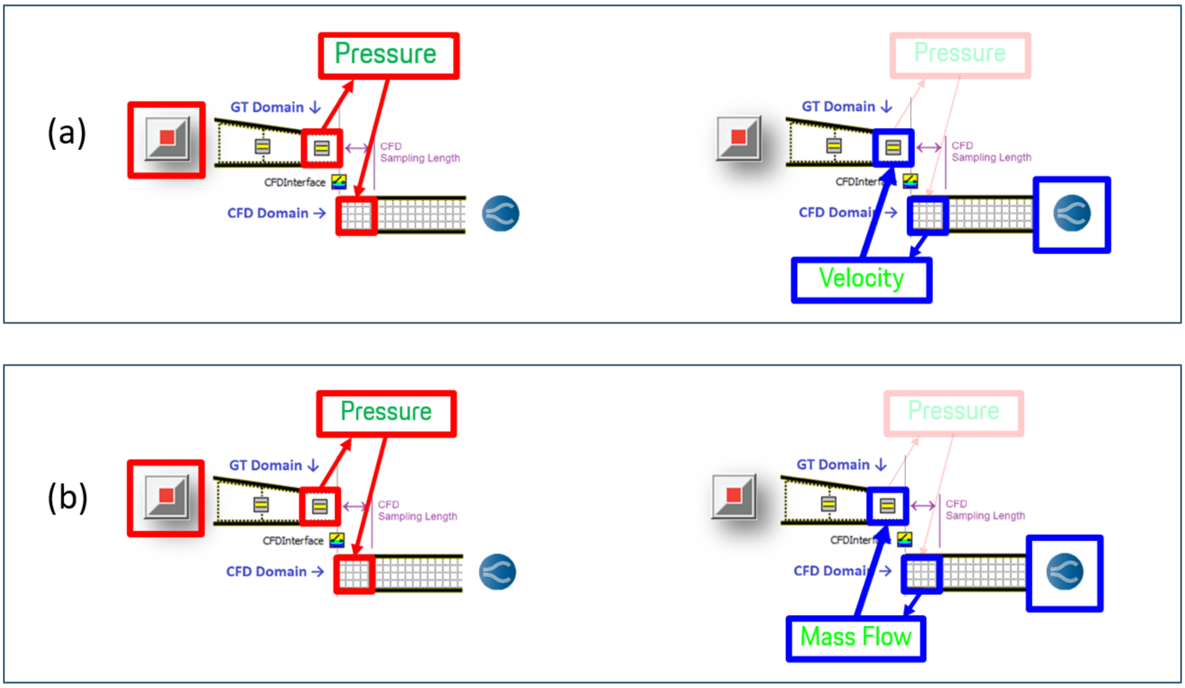


Figure 5.7: CFD interface type: Pressure

2. Velocity/mass flow

- GT provides (c) velocity/(d) mass flow to CONVERGE
- CONVERGE returns presure to GT

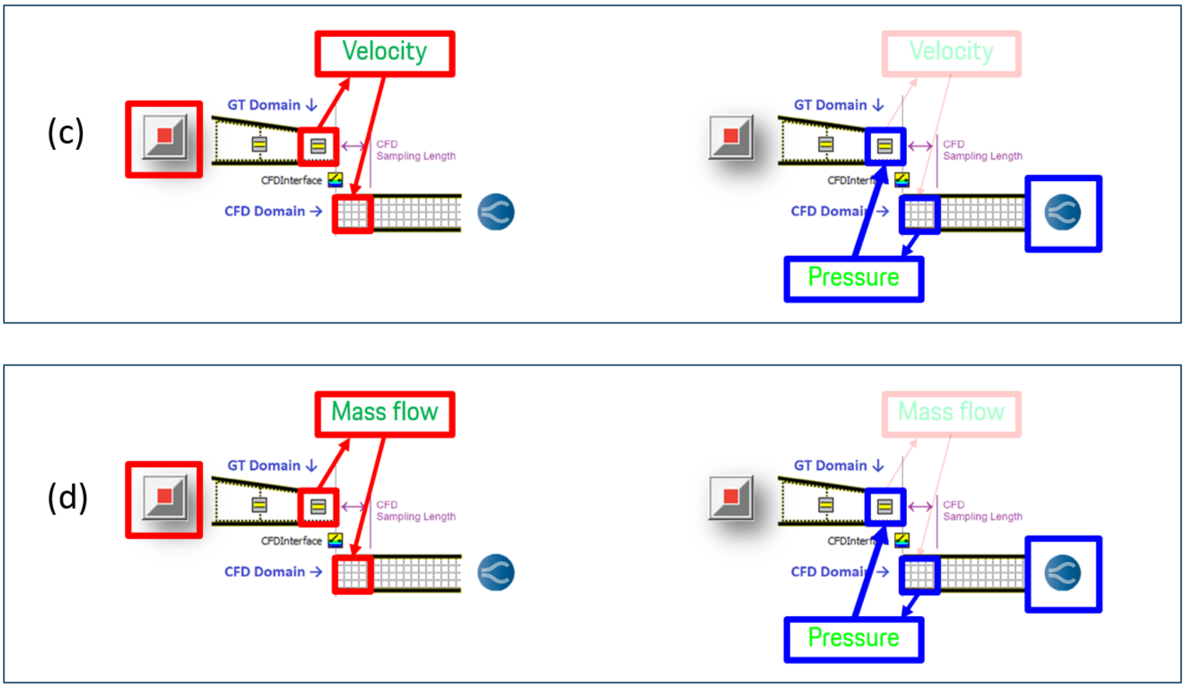


Figure 5.8: CFD interface type: Velocity/Mass flow

If velocity/mass flow is chosen for all coupling interfaces stability issues will occur.

5.4.3 Averaging methods

Averaging methods establish where the 1D results are imported in the 3D domain (and viceversa).

In the Converge Case set-up there is a parameter called gti-flag that define how the 1D results are averaged and imported in the 3D domain. For each number associated to the gti-flag corresponds an averaging method.

Table 5.1: Averaging methods for coupled simulations

Values of gti-flag			
Extrapolation	Averaged value at boundary	Average value in the subvolume defined by a virtual region	Averaged value in the subvolume defined by a region
1	2	3	4

In the following Figures, examples of the various Averaging methods are qualitatively described considering the coupled interface at the exhaust side of the 2 stroke engine under investigation.

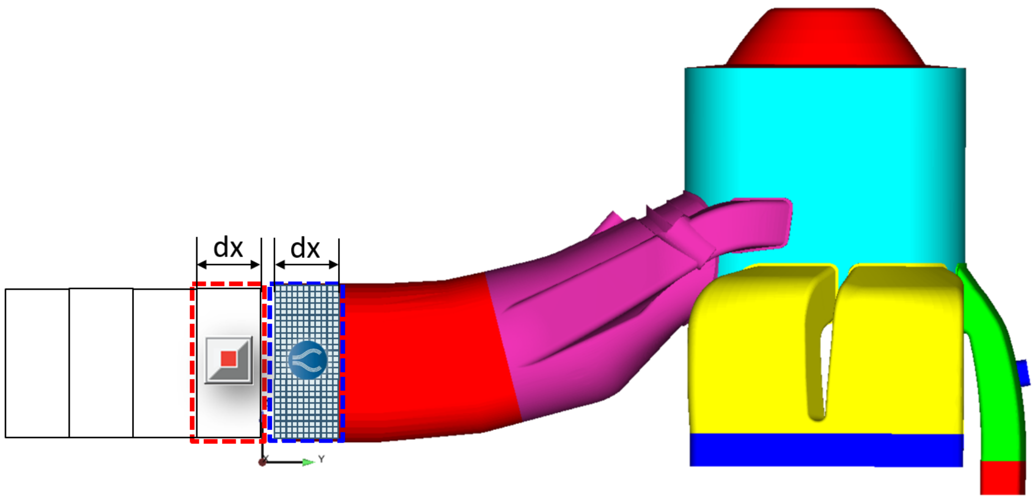


Figure 5.9: Averaging method 1: Extrapolation

With the averaging method number 1, 1D results calculated in the last pipe subsection (dx represents the discretization lenght) are imported in the 3D domain involving all the cells near the coupled interface within dx, as shown in the Fig. 5.9.

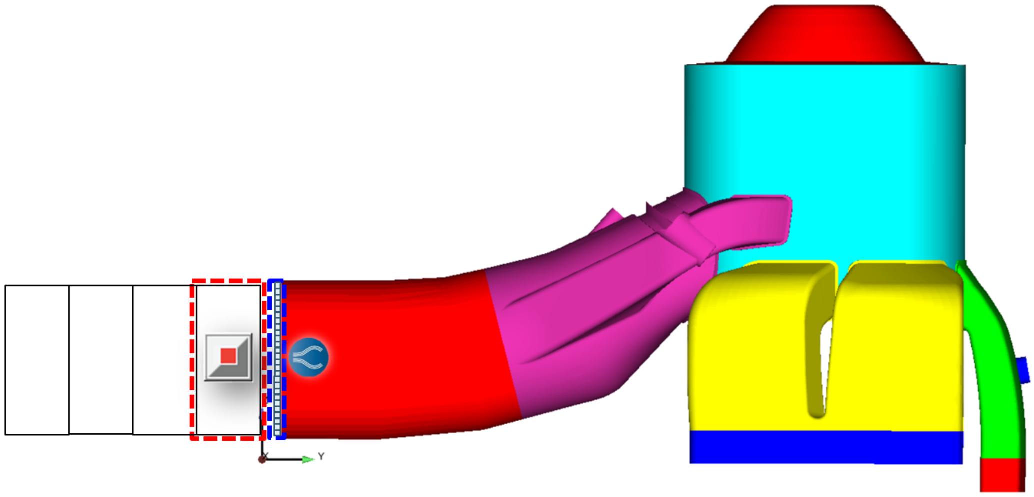


Figure 5.10: Averaging method 2: Averaged value at boundary

With the averaging method number 2, 1D results calculated in the last pipe subsection are imported in the 3D domain involving the first layer of cells of the coupled interface.

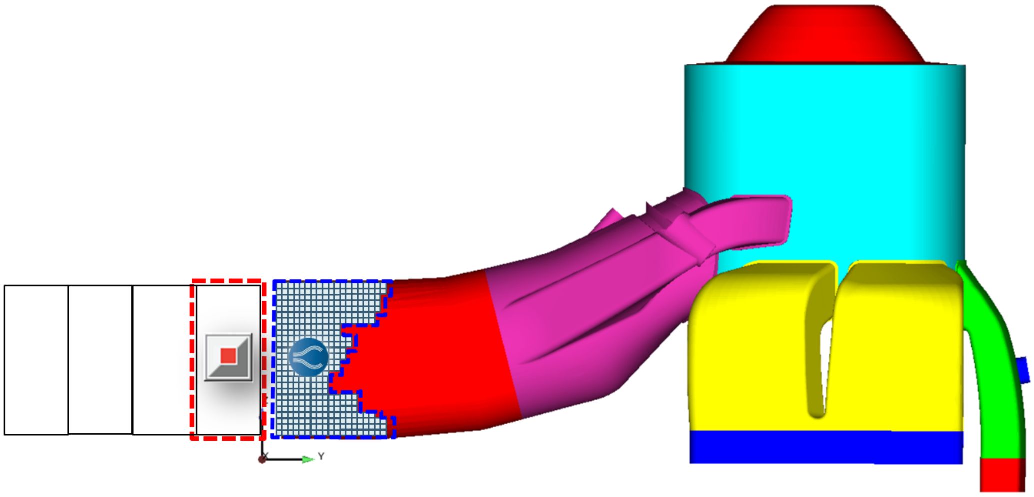


Figure 5.11: Averaging method 3: Avg. value in the subvolume defined by a virtual region

With the method number 3, 1D results calculated in the last pipe subsection are imported in the 3D domain involving the cells inside a virtual region defined iteratively by the 3D software tool taking into account to the flow field and temperature distribution. This method is the one recommended by CONVERGE Studio.

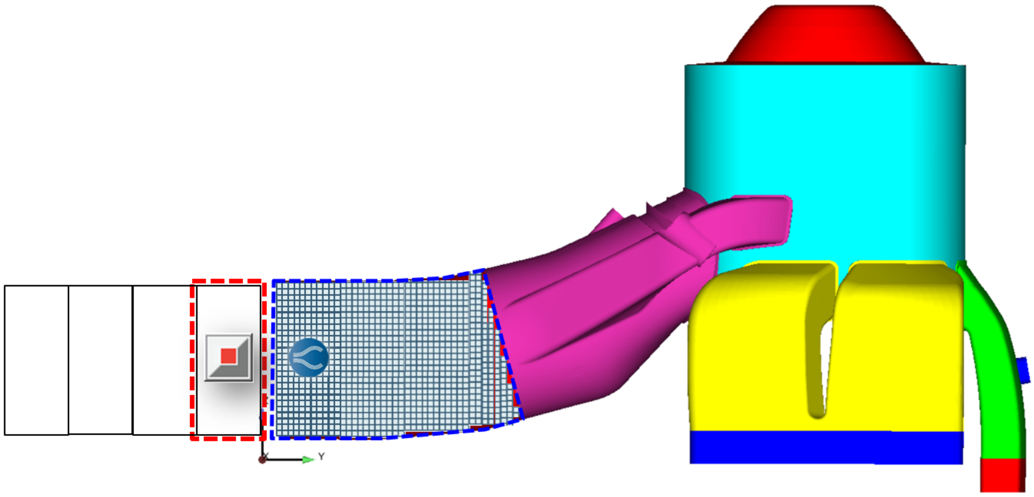


Figure 5.12: Averaging method 4: Averaged value in the subvolume defined by a region

With the averaging method number 4, 1D results calculated in the last pipe subsection are imported in the 3D domain involving the cells inside the region adjacent the coupled interface.

In the following table the main advantages and disadvantages of the different approaches are summarized:

Table 5.2: Pros and Cons of 1D and 3D CFD simulation methodologies

Simulation Methodology	
1D CFD	3D CFD
+ Computaitonal time	+ Catching scavenging effect
+ Complete engine modelling	+ Detailed flow analysis
Reproduce pressure wave phasing	+ Actual geometry evaluation
- Predictability	- Computational cost
- Scavenging modelling	- Model calibration time

As said in the paragraph 5.4, the main goal of the coupled 1D-3D simulation methodology is to have both the advantages of the 1D and 3D approaches while minimizing the drawbacks.

Table 5.3: Pros and Cons of the coupled 1D-3D CFD simulation methodology

Simulation Methodology
1D-3D CFD
+ High resolution where required
+ Lower computational time wrt full 3D
+ High flexibility
- Methodology under development

Chapter 6

Model Build-Up

The processes that are crucial for the thermodynamic and fluid dynamic effectiveness of a two stroke engine are, as already said, the scavenging process, the combustion and the mixture preparation. In order to exploit all the potentialities of the simulation in the early stage of a 2 stroke engine development process are low computational effort with sufficiently accurate results for the prediction. For sure, the more accurate is the model, the more complex and more time consuming is the model solution. At the same way the more deterministic the used model are, the less assumptions have to be made.

The geometric characteristic of the 2 stroke engine under investigation are summarized in Tab. 6.1:

Table 6.1: Engine characteristic

Bore [mm]	85
Stroke [mm]	74
Connecting rod [mm]	134
Geometric CR [-]	12.73
Power [kW]@8250 RPM	110
Torque [Nm]@8250 RPM	125

6.1 Engine Sections

In the following paragraphs the engine will be analized section by section in order to understand how the 2 stroke engine under investigation can be efficiently modelled to reach the right compromise of short calculation time and good results quality.

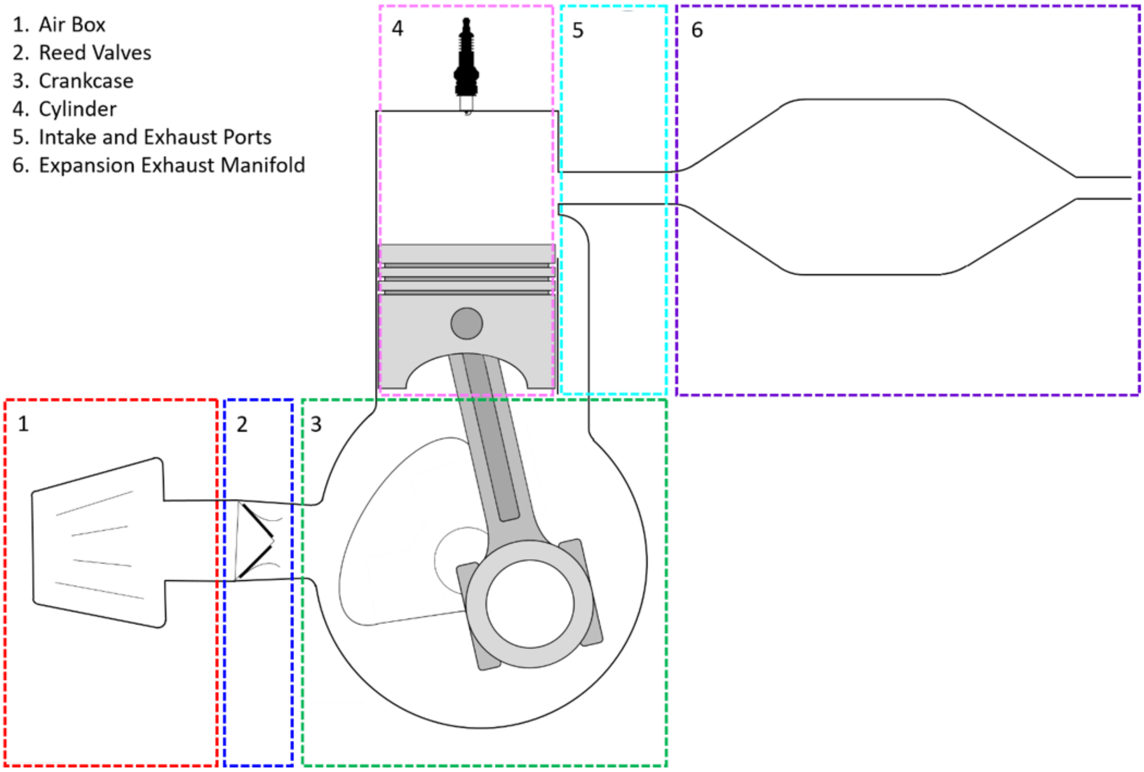


Figure 6.1: Engine subsections

6.1.1 Intake Muffler

It contains the suction pipe and connect the external environment to the reed valves. The main function is to clean the air (by means of the filter) and damp down the gas pulsation to reduce the noise. It affects the delivery ratio of the engine. The mutual influence of the cylinder gas dynamics in a multicylinder engine and the pressure losses are reproduced with good approximation in 0D/1D models. So this approach is used for the first section.

6.1.2 Reed Valves

Reed valves affect the volumetric efficiency and the power output of the engine. In 1D codes they are simplified with check valve by means of flow resistance curves. A further refinement allows to 0D/1D models to consider the pressure on both sides of the reed valves and also estimates the eigenfrequiencies of the petals. Since the topic of the work is not focused on the complex interaction of the air flow with the moving reed petals, also here the 1D approach is adopted.

6.1.3 Crankcase

The dominating effects of the crankcase are two: the heating of the fresh charge (especially due to the heat transfer from the lower part of the cylinder), and the pumping effect of the bottom side of the piston. The crankcase can be modelled in 0D, considering only the pre-compression effect for the scavenging, the heat transfer, the conservation of mass and energy. The assumption of a perfect mixed condition is reasonable. Furthermore, with the 0D modelling, additional computational time is saved due to the reduced amount of moving parts in the 3D simulation.

6.1.4 Intake and Exhaust Ports

Intake ports connect the crankcase to the cylinder. In the loop-scavenged engine under investigation, there are one boost port and four transfer ports. The boost port, placed at the opposite side of the exhaust ports, is responsible to produce the upward flow motion inside the cylinder, while the transfer ports are responsible to produce the rotating flow motion. On the other hand the exhaust port consists of one main and two side pipes. The entrance angle, the shape end the cross section of the ports significantly affect the scavenging of the cylinder, the mixture preparation and the AFR distribution. These kind of information are lost with a 1D approach, so a 3D modeling here is needed.

6.1.5 Cylinder

For a loop-scavenged engine the intake and exhaust port are open simultaneously for a long period of time. This leads to a complex interaction between the intake and exhaust flow that hugely affect the mixture preparation and amount of residual gases. In 0D/1D simulations the scavenging is simulated by phenomenological models based on the graph of scavenging ratio versus delivery ratio or also the so called scavenging function.

The main limitation of 0D/1D models lies in the lack of simulation accuracy in predictive simulation. Scavenging and heat release rate functions have to be taken from test bench measurements or simply estimated. Contrary the 3D simulation is able to simulate with sufficient accuracy the charge motion inside the cylinder and it is possible, for example, to predict residual gas mass, fresh air mass, fuel mass as well as their spatial distribution [19]. So the 3D-approach is strongly recommended also for this section.

6.1.6 Exhaust Tuning Pipe and Muffler

As already deeply discussed, the exhaust tuning pipe is the determining part of the gas dynamics of a 2 stroke engine. For pipes with moderate geometry transitions, the prediction of the heat transfer - determinant for the temperature of gas and therefore for the pressure waves speed - with a 1D as well as with a 3D simulation approach. Furthermore, a 1D simulation can predict the gas flow sufficiently exact and offers possibility for geometry optimization. The exhaust muffler, instead, is meant to damp down the noise and is not design to influence the gas dynamics of the exhaust system.

In the end, the final model of the two stroke engine under investigation is the following:

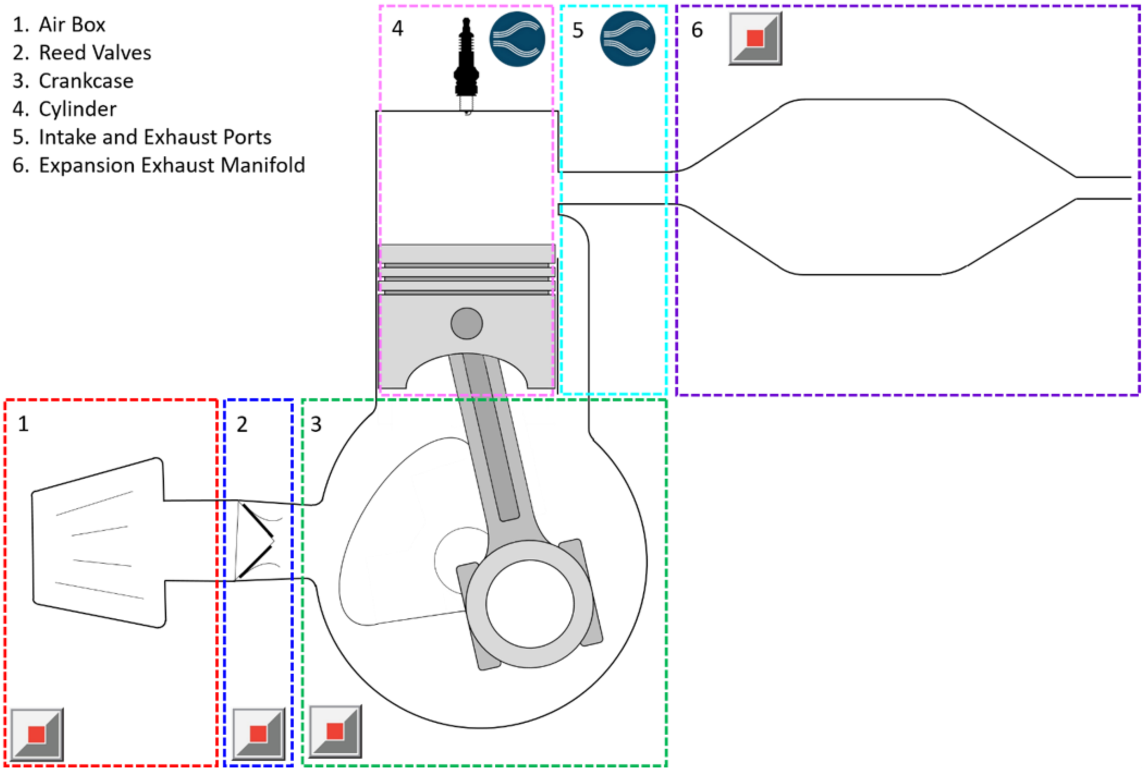


Figure 6.2: Engine subsections modelling

6.2 3D Coupled Model

At the end of the analysis performed in the previous section, it can be shown the difference in terms of size and complexity of the initial full 3D model of the engine, and the final simplified 3D model ready for the coupled simulation.

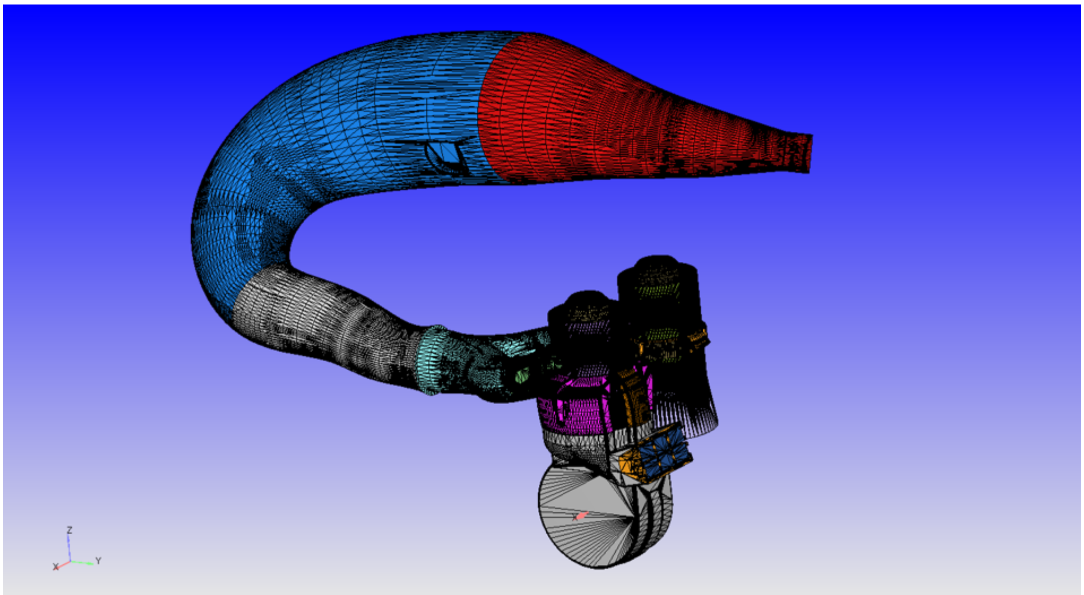


Figure 6.3: Complete 3D model of the engine

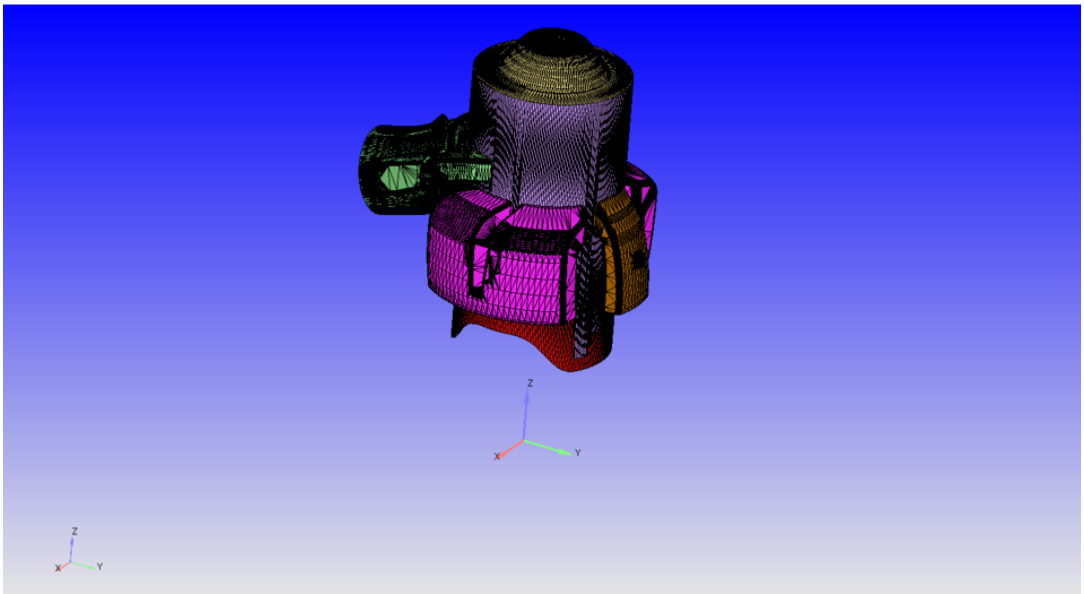


Figure 6.4: Simplified 3D model for the coupled simulation

Starting from the complete 3D model, following components has been deleted:

- Cylinder 2
- Exhaust expansion chamber

- Reed Valves
- Crankcase
- Crankshaft
- Connecting Rod
- Exhaust Y pipe

Doing so, the fluid domain has been reduced of 94%. In addition to that, it has to be considered that the computational effort has been furtherly reduced by lowering the number of moving parts. Now there is only the translation of the piston top while the complete model includes also the connecting rod and the crankshaft, as it shows the Fig. 6.5.

(a) Complete 3D model kinematics

(b) Coupled 3D model kinematics

Figure 6.5: Moving boundaries in the complete and coupled 3D model

Then, as suggested by the CONVERGE Manual for coupled simulation, additional constant section pipe have to be add to the 3D model to improve data sampling and data exchange between the 1D-3D coupling interfaces and CFD Domain Sampling Length should be equal to the pipe discretization length in the adjacent GT-SUITE pipe.

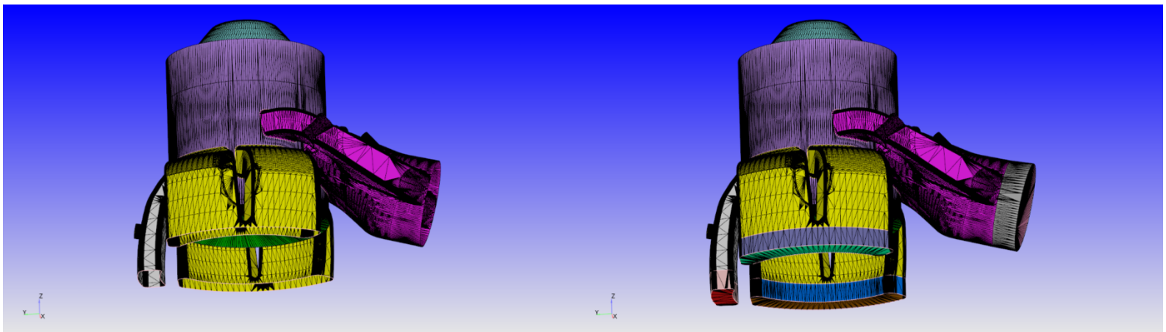


Figure 6.6: Addition of the constant section pipe

The single cylinder model has been subdivided in Converge in twentyone boundaries:

1. Piston;
2. Piston Skirt;
3. Piston Bottom;
4. Liner;
5. Head;
6. Boost Port;
7. Inflow Boost Port;
8. Inflow Transfer Port A;
9. Inflow Transfer Port B;
10. Exhaust Port;
11. ExhaustPort Coupled;
12. Exhaust Flow;
13. Injector;
14. Spark Plug;
15. Spark Electrode;
16. Crankcase Fictitious;
17. BoostPort Coupled;
18. TransferA Coupled;
19. TransferB Coupled;
20. TransferB;
21. TransferA.

Among the abovementioned boundaries, those moving are clearly the Piston, Piston Skirt and Piston Bottom. Particular attention has to be focused on the interaction between the Ports and the Piston. Since no valves are present in the model, the gas exchange process is ruled by the covering and uncovering of the

Ports by means of the Piston. For such dynamic process it is crucial to define the sealings between the Piston and the Liner, and the Piston and the Ports. Sealings in Converge remove from the fluid domain all the cells between two boundary that are close within a tolerance limit defined by the user (usually 0.2 mm).

In addition to the boundary, there have been defined also the regions: a group of boundaries that define a closed volume in the fluid domain. They are:

1. Cylinder;
2. BoostPort;
3. GT Boost;
4. TrfA;
5. TrfB;
6. GT TrfA;
7. GT TrfB;
8. Exhaust Ports;
9. GT Exhaust;

It can be noticed that some regions and boundaries may look redundant, but as said before, this is due to the fact that in correspondence of the coupling interface the length of the pipe is extended for an amount called "CFD sampling length" to define the artificial subvolume length to be referenced when sending back data to 1D environment (and viceversa).

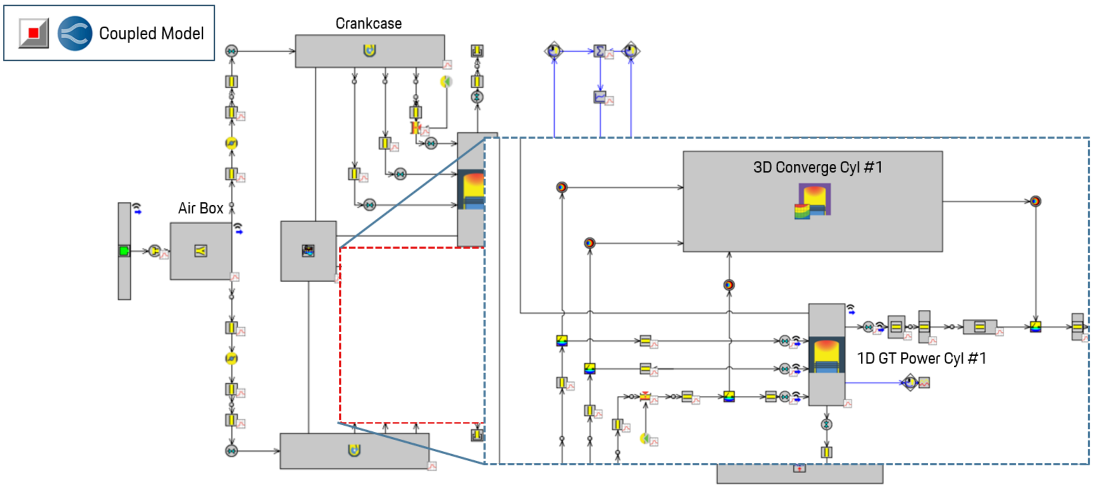


Figure 6.7: 1D coupled engine model

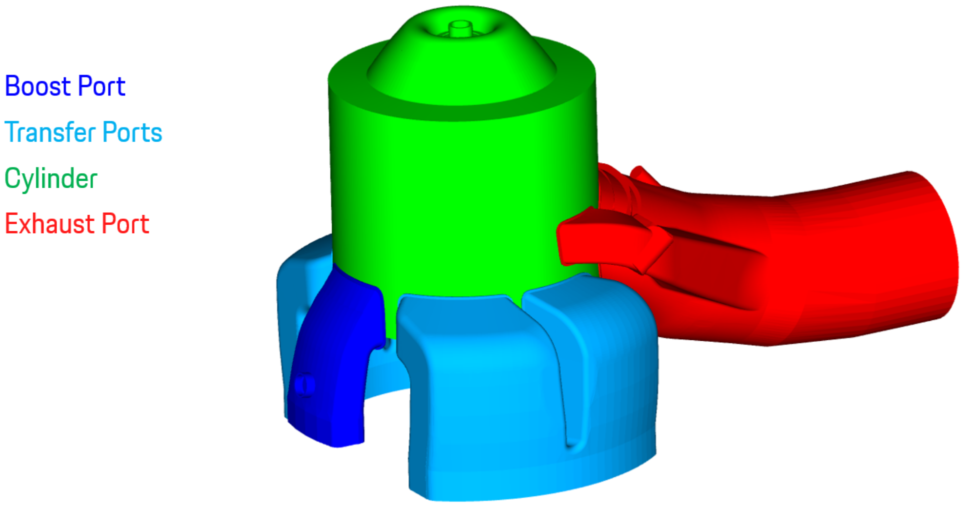


Figure 6.8: 3D coupled engine model

6.3 Boundary conditions and initialization

Boundary conditions have been specified using data from test bench and results from standalone 1D simulations. Wall temperature is a parameter of great importance since it influences the heat transfer during the combustion process, affecting the pressure trace along the cycle.

Wall temperature for the different parts are listed in Table 6.2.

Table 6.2: Wall temperature of the parts used in Converge

Part	Wall temperature at 8250 rpm [K]
Head	475
Liner	450
Piston	550
Piston Skirt	500
Exhaust port	715
Boost Port	315
Transfer Port	315
Spark Plug	900
Spark Electrode	800

On the other hand, for what concern the initial condition for the regions, they can be summarized in Table 6.3, Table 6.4, Table 6.5, Table 6.6. The simulation starts at 280 CAdeg (at the Exhaust Port Closure) with an in cylinder pressure of 2.9 bar (from experimental data).

Table 6.3: Initial condition Transfer Ports

Region	Boost Port
Pressure [bar]	0.895
Temperature [K]	305
O_2 [%]	15.6
N_2 [%]	52.21
C_8H_{18} [%]	32.3

Region	Transfer Ports
Pressure [bar]	0.895
Temperature [K]	305
O_2 [%]	23
N_2 [%]	77

Table 6.4: Initial condition Cylinder

Region	Cylinder
Pressure [bar]	2.9
Temperature [K]	520
O_2 [%]	21.5
N_2 [%]	72.1
C_8H_{18} [%]	6.4
Lambda [-]	0.965

Region	Exhaust Port
Pressure [bar]	1.8
Temperature [K]	800
O_2 [%]	5.4
N_2 [%]	74.81
CO_2 [%]	9.11
CO [%]	3.31
NO_2 [%]	1.1
H_2O [%]	7.26

Chapter 7

Results

The 1D model was run in advanced uncoupled for 100 cycles until it correctly reached the convergence. Once the uncoupled phase finished, the boundary conditions from the the 1D were transferred to the 3D domain which run another 12 cycles. Then, coupled simulation results have been compared with the exp. data.

7.1 Calibration

For the 1D-3D coupled model, the calibration of the pressure wave has been already performed in a previous work by the use of the standalone 1D model and no further work is needed. As it can be seen, the Figg. 7.1, 7.2 and 7.3 below show the results relative to the pressure wave at the inlet and exhaust. It is clear that the simulation matches with good approximation the results from test bench.

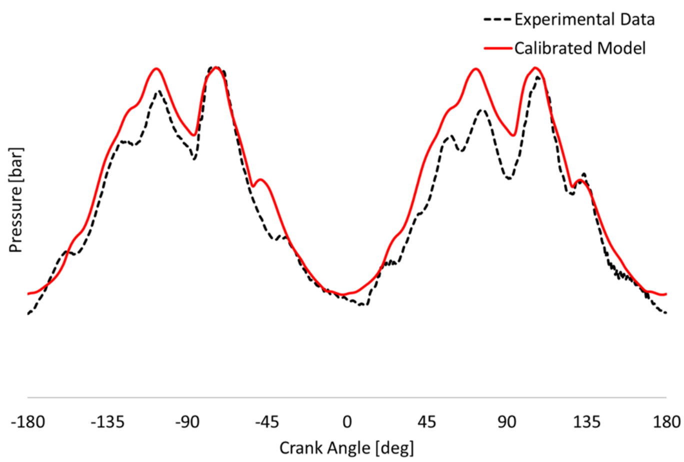


Figure 7.1: Indicated Pressure at Exhaust Port

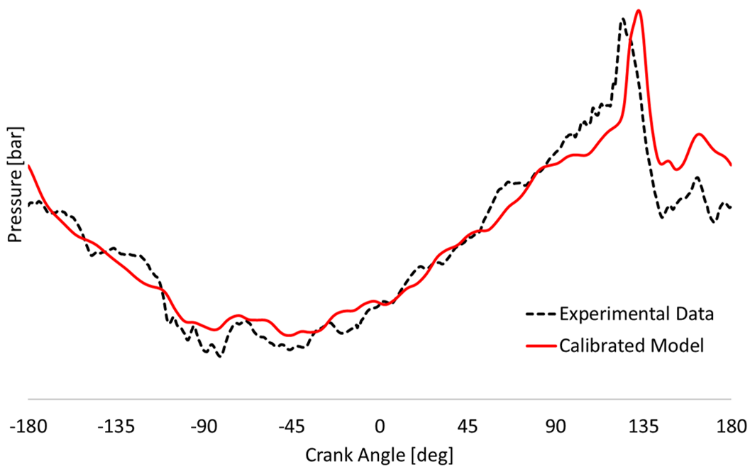


Figure 7.2: Indicated Pressure at Transfer Port

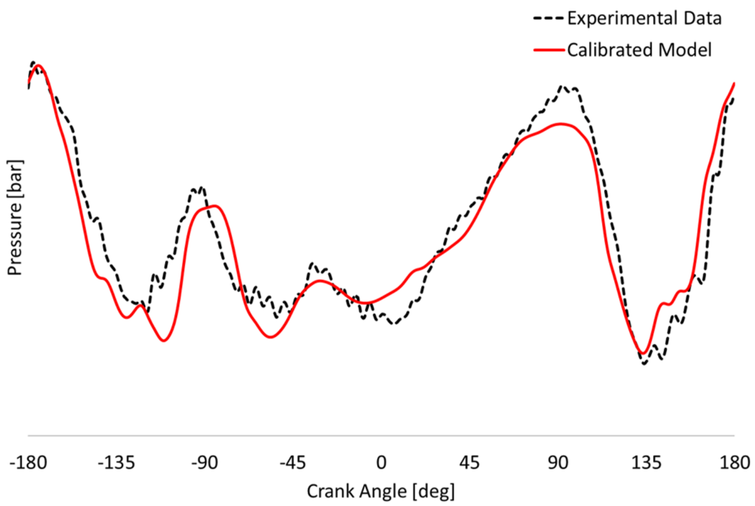


Figure 7.3: Indicated Pressure at Throttle Body

On the other hand, the calibration of the in-cylinder pressure have been performed acting on parameters related to the 3D part of the coupled model. Especially:

- The wall temperature, to calibrate the heat transfer between the walls and the gas mixture.

- The reaction multiplier to calibrate the heat release of the combustion

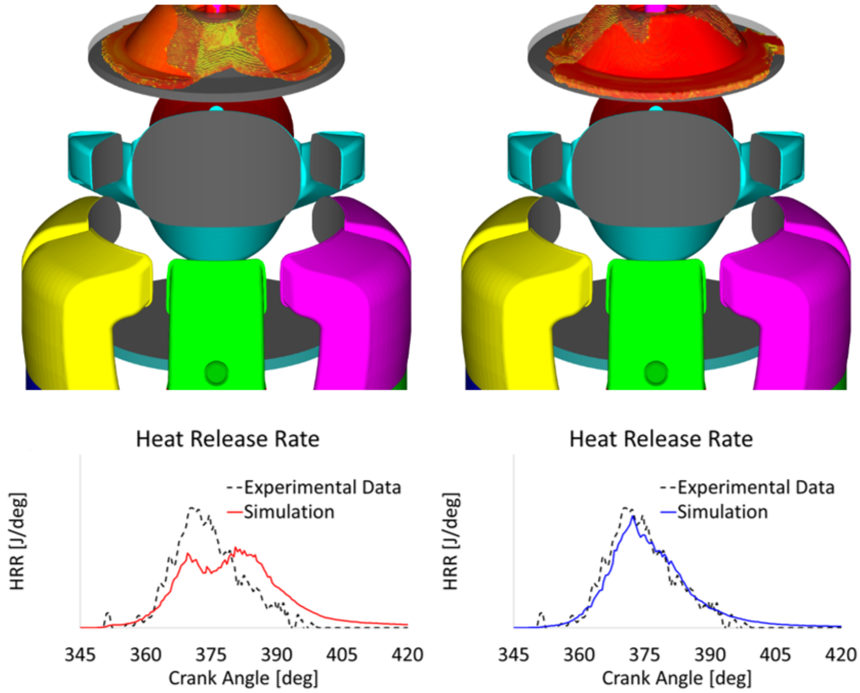


Figure 7.4: Effects of the reaction multiplier on the HRR

- The spark modelling in order to optimize the stability of the combustion occurrence

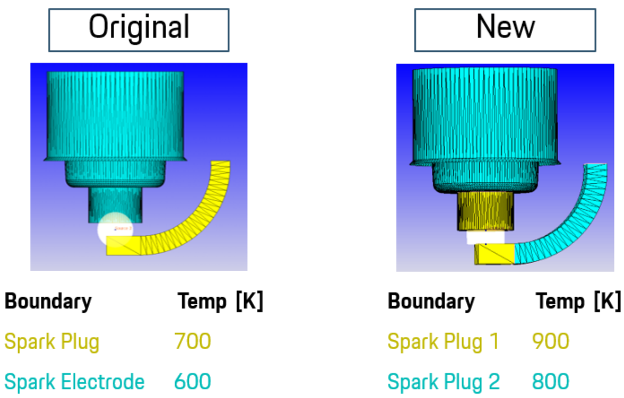


Figure 7.5: Spark modelling

From Fig. 7.5 it can be seen that the shape of the volume in which there is the energy release by the spark has been modified from spherical to cylindrical. The former imply that some of the energy is lost since not all the source volume is included in the fluid domain. In addition to that, the temperature of the spark plug has been modified in order to have higher temperature in the region closer to the spark release.

Summing up the effect of all these parameters, we have the following results:

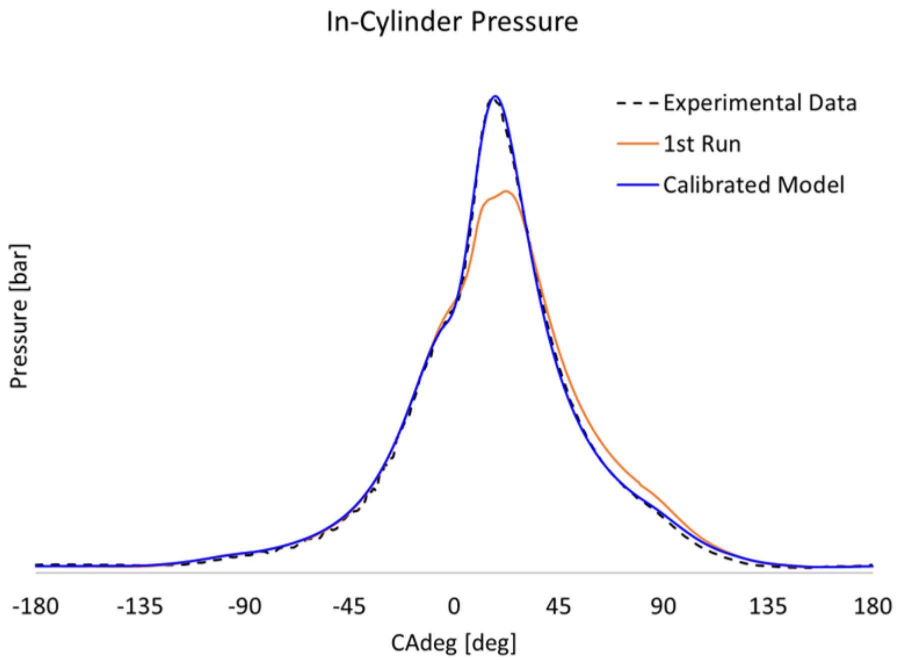


Figure 7.6: In-cylinder pressure

7.2 Mesh sensitivity

Once the model has been calibrated, next step is to reduce the computational time while having still accurate results. This can be done performing the so-called mesh sensitivity.

The 3D fluid domain is made up of cells, and for each of them the partial differential equation of the mass conservation, momentum and energy are solved. The set of cells constitutes the mesh. The smaller is the base dimension of the cell,the higher is the flow field resolution and the computational time. So the dimension of the cell has to be set in order to have the best compromise between accuracy and computational cost.

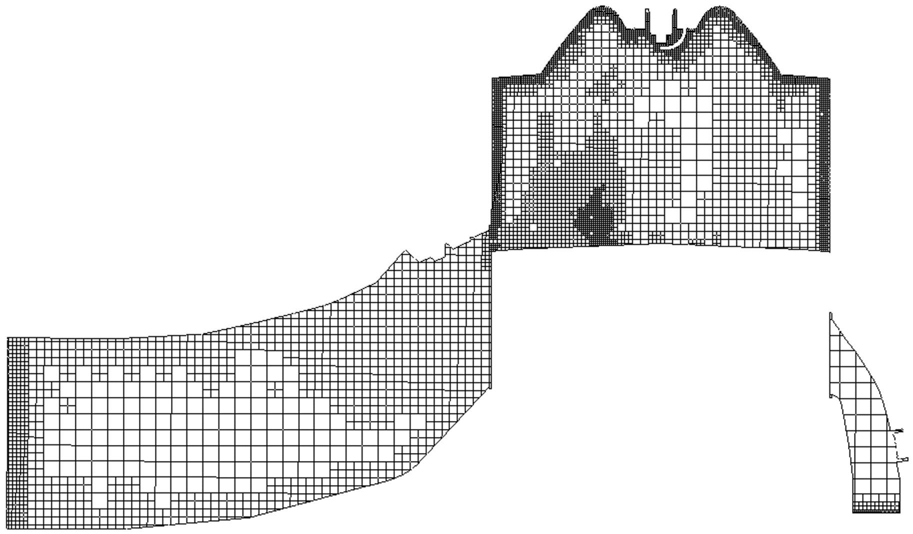


Figure 7.7: Variable size mesh

In order to reach this target, two critical aspects have to be taken into consideration:

Higher density of cells is required in the region adjacent to the coupled interface for the correct communication between the software. To do so, the Fixed Embeddings can be used in order to refine the mesh along the cycle. The mesh is refined automatically at runtime according to the grid scale as:

$$\text{scaled mesh} = \frac{base\, mesh}{2^{grid\, scale}} \tag{7.1}$$

In addition to the fixed embedding, another useful tool to increase result accuracy is the so-called Adaptive Mesh Refinement (AMR). It is able to automatically refine the mesh when high temperature gradients (e.g. between cylinder walls and burned gases) or velocity gradients (e.g. at exhaust port opening) are detected between neighbour cells.

As it can be seen from Table 7.1 and Fig. 7.8, the model with 6 mm mesh base size didn't reach the convergence while a good compromise between computational time and accuracy of the results can be found in the model with 4 mm mesh base size.

Table 7.1: Mesh sensitivity

Mesh base size	Time/Cycle [h/cyc]	Time to convergence [h]
2 mm	22	154
3 mm	13	104
4 mm	7	70
5 mm	5	60
6 mm	4	Instable

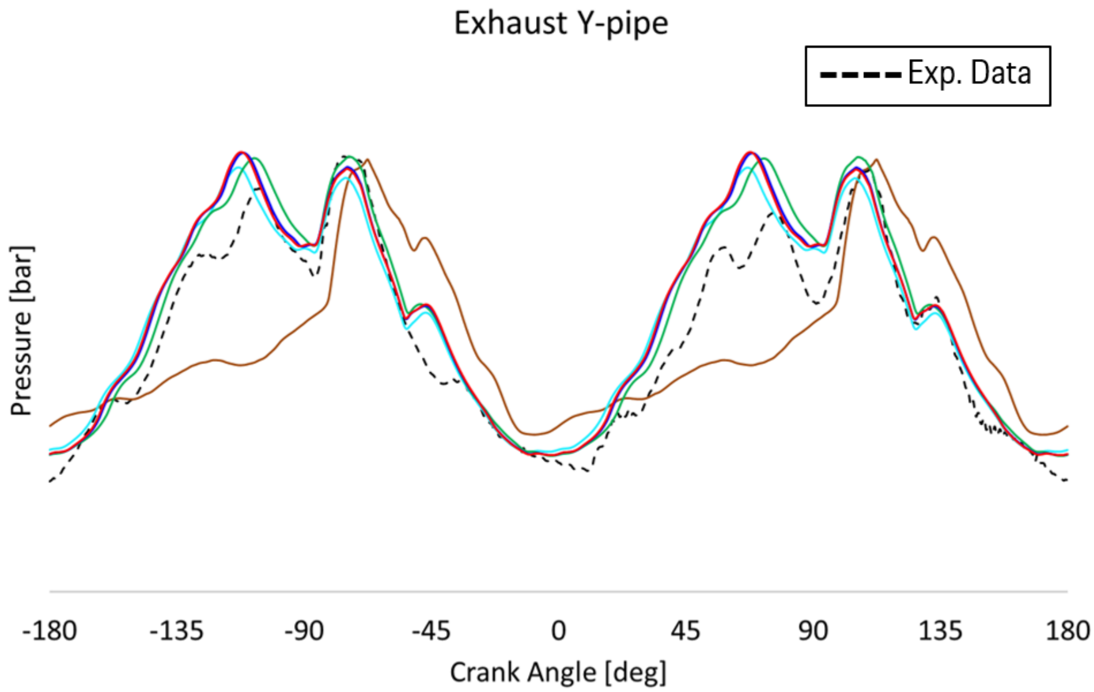


Figure 7.8: Effects of mesh sensitivity on pressure waves

7.3 Methodologies comparison

The following table shows the difference in terms of results and computational time of the three simulation methodologies. All 3D simulations have been run on 4 nodes of 24 CPUs each.

Table 7.2: Methodologies comparison

Simulation Methodology	Time for 1 cycle	Time for convergence
1D CFD	Seconds	Minutes
3D	33 hours	14 days
1D-3D	5 hours	60 hours

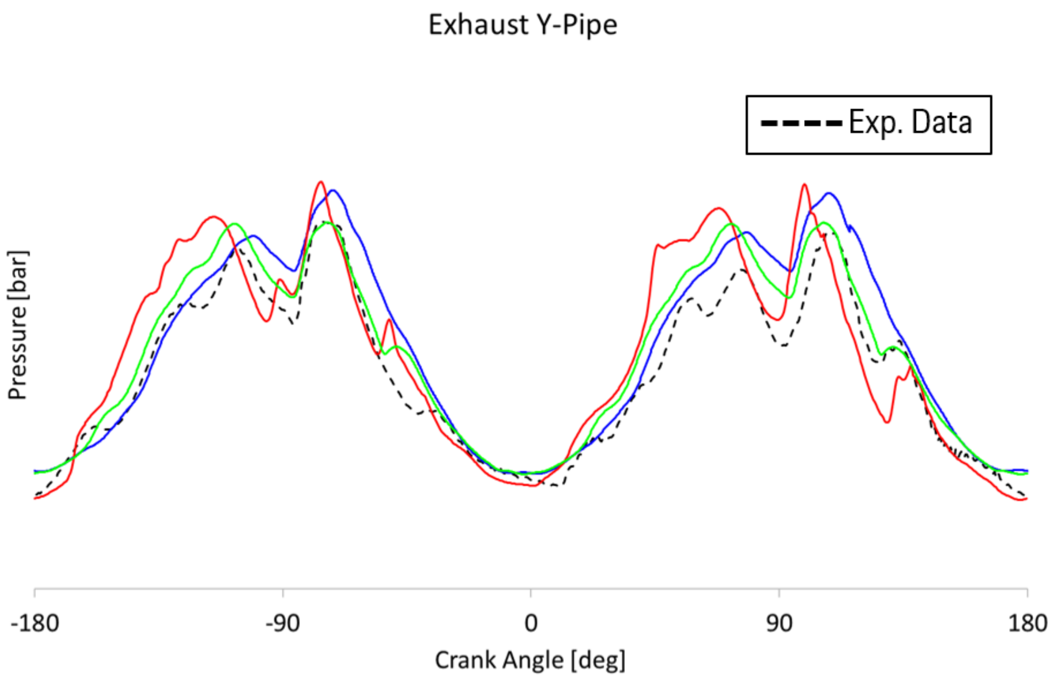


Figure 7.9: Pressure wave calibration for different methodologies

The 1D CFD methodology, as already stated before, is characterized by the fastest computational time and quite good approximation of experimental pressure data at the exhaust. On the other hand, since the scavenging process is imposed with the scavenging function, the 1D modelling is not able to predict the effects of the variation of geometric parameters (e.g. exhaust expansion manifold shape and length).

The full 3D CFD methodology provides the most detailed results among all, but the great computational time needed due to the model complexity (high number of moving components, fluidmechanical interaction between reed-valves and air, etc.) do not allow a meticulous calibration of the model. Time for convergence for this kind of model is two weeks.

The coupled 1D-3D CFD methodology provides much faster computational time with respect to the full 3D model and at the same time is able to predict the effects on the gas exchange process by modifying, for example, some geometric characteristic of the engine. This aspect is deeply investigated in the next section.

7.4 Optimization

Two different modifications have been implemented in the coupled model in order to verify the predictability and flexibility of this methodology.

One optimization is performed in the 1D side of the coupled model and it involves the exhaust expansion manifold. The goal is to have a better tuning of the engine at 8250 rpm.

The second optimization is performed by modifying the 3D design of the boost port. The goal is to minimize the amount of fuel shortcircuited at the exhaust.

7.4.1 1D Exhaust expansion manifold

As mentioned in the section 4.4 of this work, the exhaust system for a two stroke engine is the mechanical component that mostly contributes towards achieving high level of performance by proper arranging pressure waves.

The original design and the modified one are shown in the Fig. 7.10 and Fig. 7.11. The modified design is characterized by a longer central section of 100 mm. This means that the blowdown-pulse has to travel longer before being reflected in what we call stuffing pulse. The effect of this change is explained below.

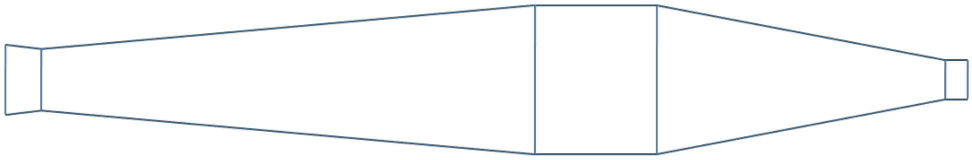


Figure 7.10: Original exhaust expansion manifold design

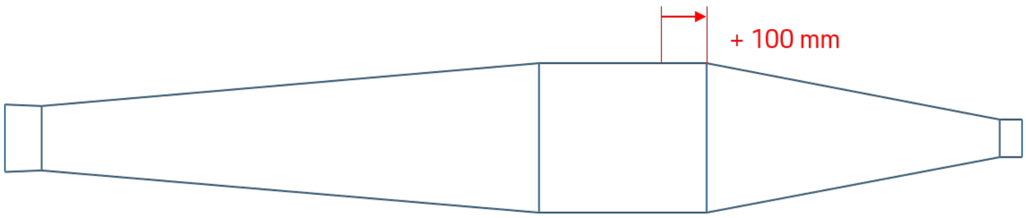


Figure 7.11: Modified exhaust expansion manifold design

The postprocessing pictures catch, from the left to the right, the exhaust port, the cylinder and the boost port. They show the direction and magnitude of the flow velocity, while the plots on the right shows the pressure at the exhaust and the fuel mass in the cylinder for the 2 model under investigation.

In Fig. 7.12 it can be seen that the peak pressure of the stuffing pulse is reached by the original model at 235 CAdeg. While (Fig. 7.13) for the modified design it is reached, as expected, 15 CAdeg later. In fact, at the exhaust gas temperature of 950 K, the speed of sound is equal to:

$$c = \sqrt{\gamma RT} = 632\, m/s \tag{7.2}$$

So 200 mm are covered in 0.316 ms, that at 8250 rpm corresponds to 15.6 CAdeg. This means that for the original model, the pressure level at the exhaust port reduces more than the modified one leading to a backflow of the fresh charge from the cylinder to the tailpipe just before the exhaust port closure (Fig. 7.14). This leads to a lower efficiency and a higher unburned hydrocarbon emissions.

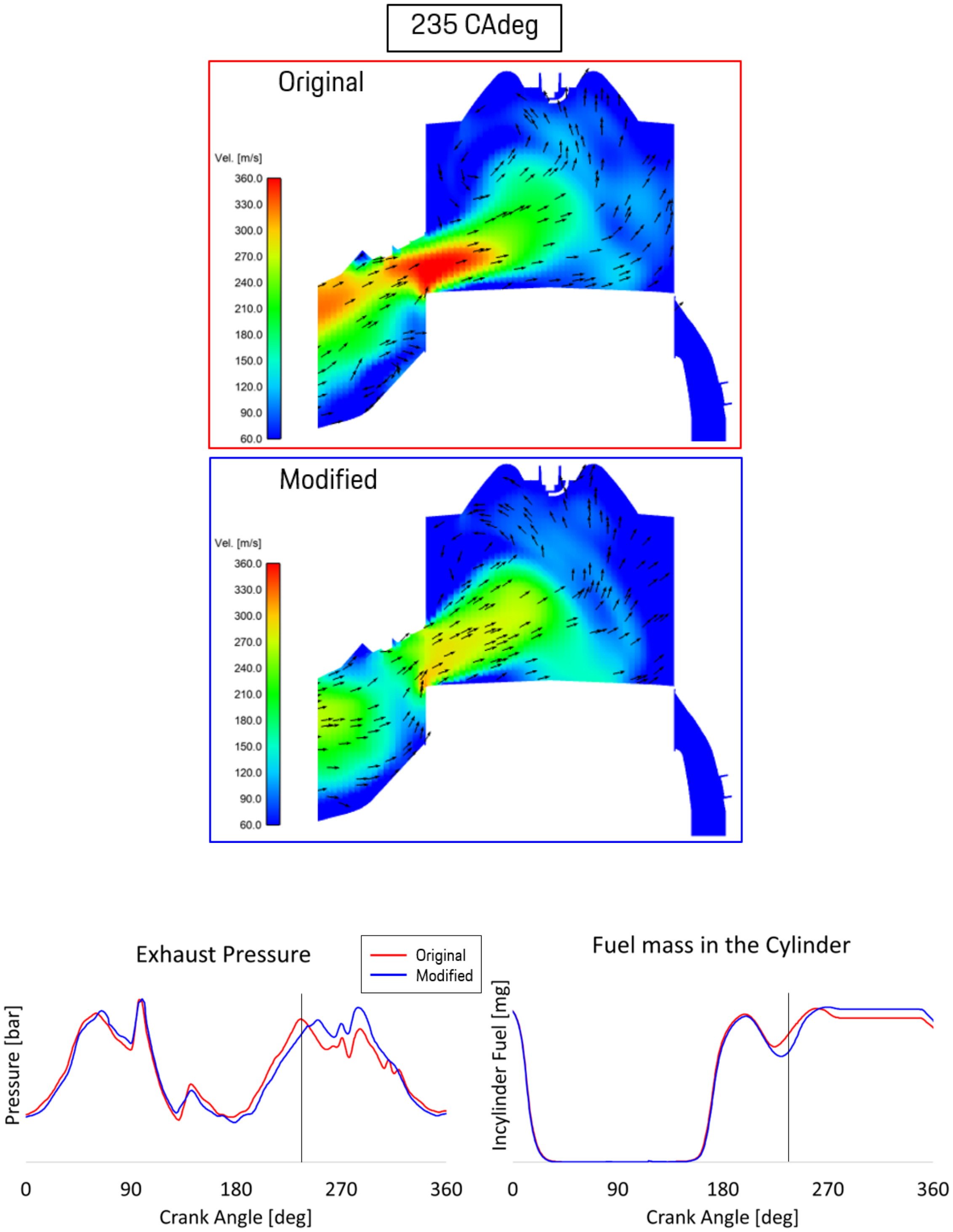


Figure 7.12: Velocity vectors at 235 CAdeg

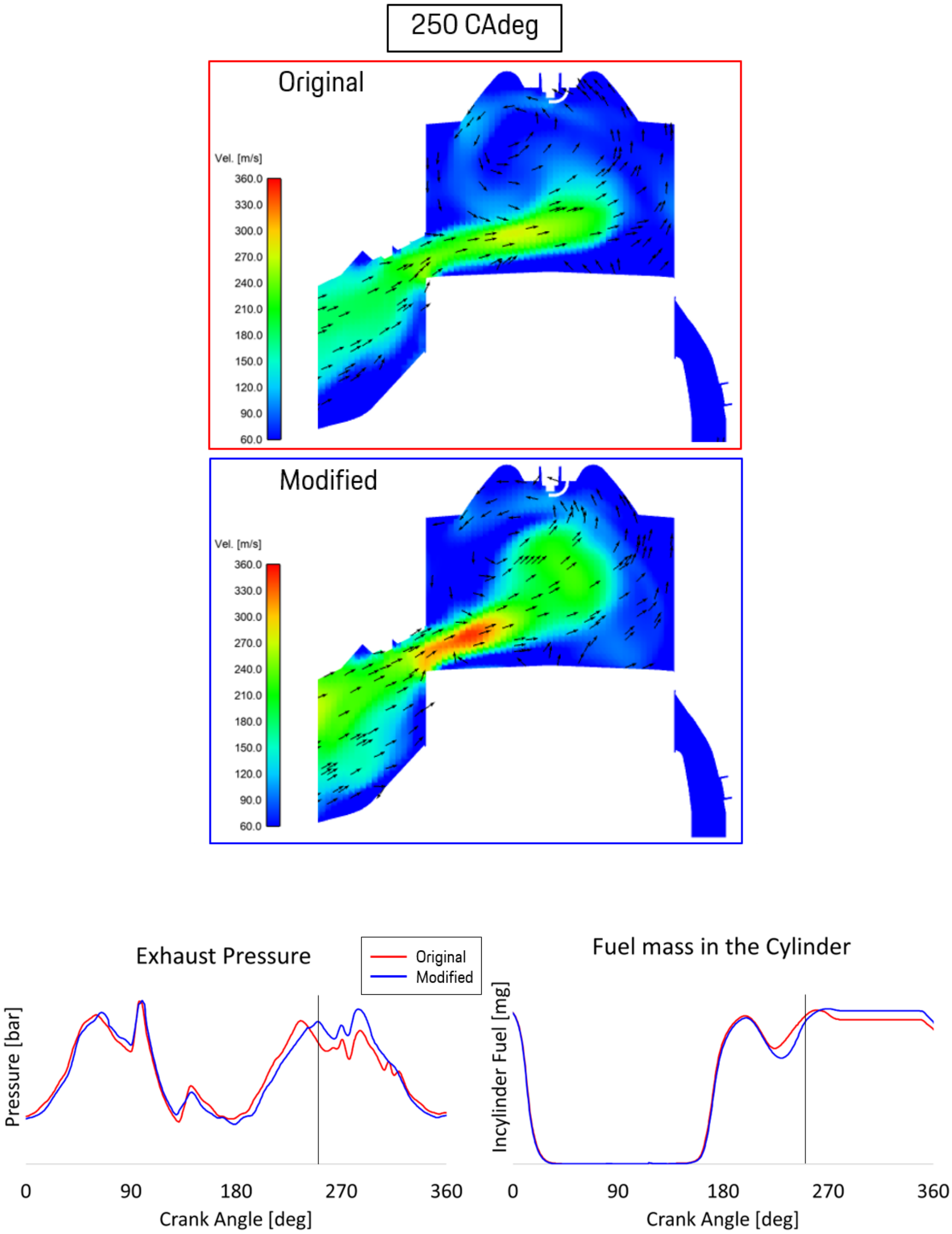


Figure 7.13: Velocity vectors at 250 CAdeg

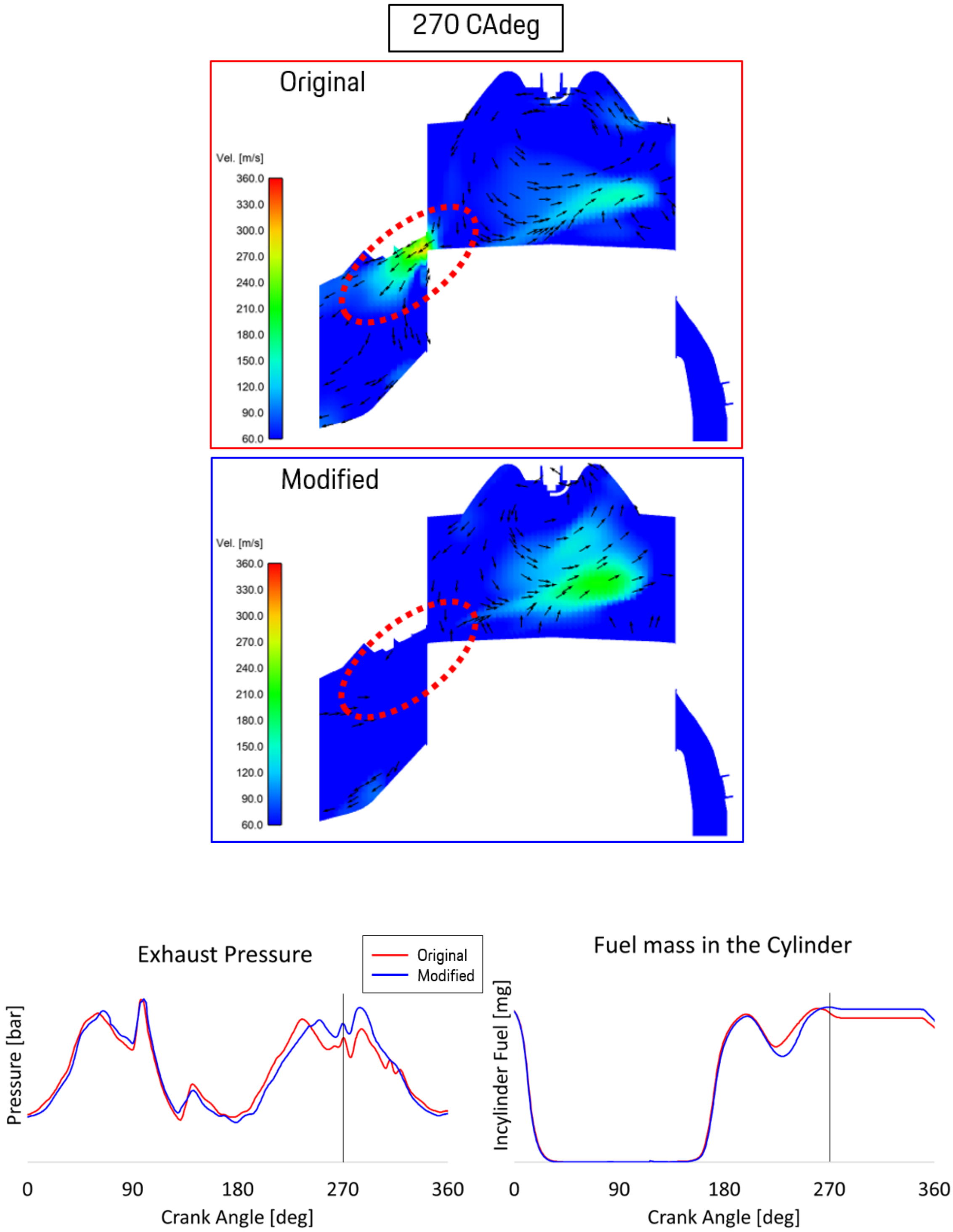


Figure 7.14: Velocity vectors at 270 CAdeg

7.4.2 3D Boost port

A variation of the 3D Boost Port has been implemented in order to minimize the fuel lost at the exhaust. As it can be seen from the Fig. 7.16, the modified design, whose producibility has to be proven, is characterized by sharp edges that direct the fresh charge as much as possible towards the upper part of the cylinder.

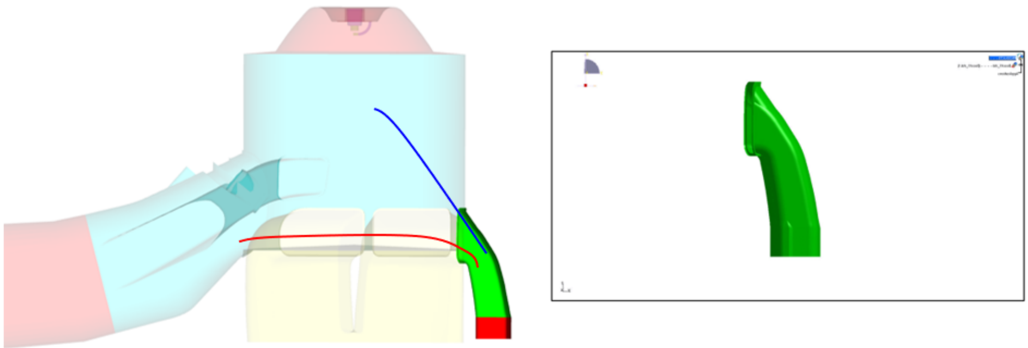


Figure 7.15: Original boost port design

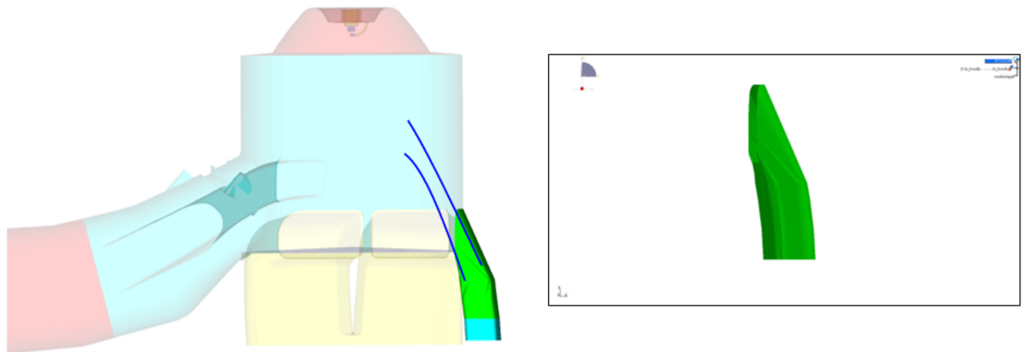


Figure 7.16: Modified boost port design

The post processing pictures shows the equivalence ratio of the mixture. Red indicates rich mixture while blue lean mixture.

Fig. 7.17 and Fig. 7.18 show that for the modified boost port, the fuel is almost attached to the cylinder wall and it penetrates less towards the exhaust port with respect the original model.

In addition, as shown in Fig. 7.19, just before the exhaust port closing, it can be seen for the original model a region of rich mixture shortcircuited at the tailpipe.

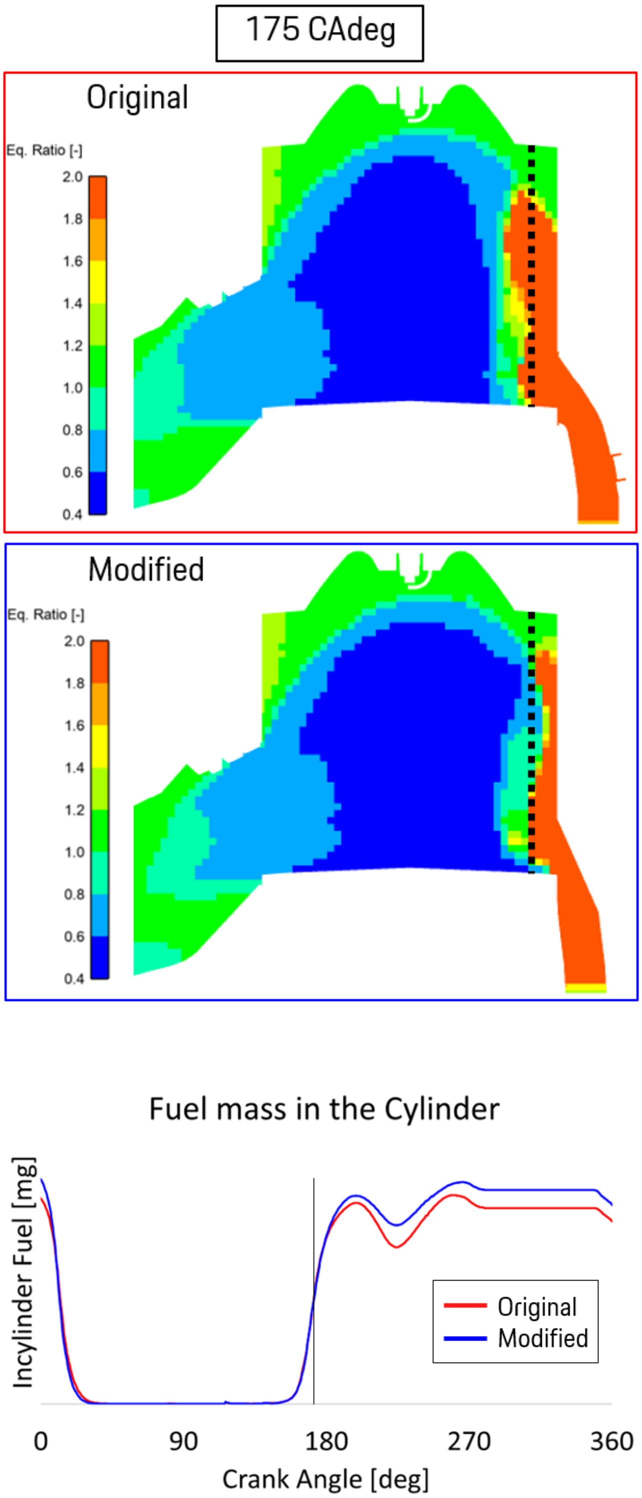


Figure 7.17: Mixture preparation at 175 CAdeg

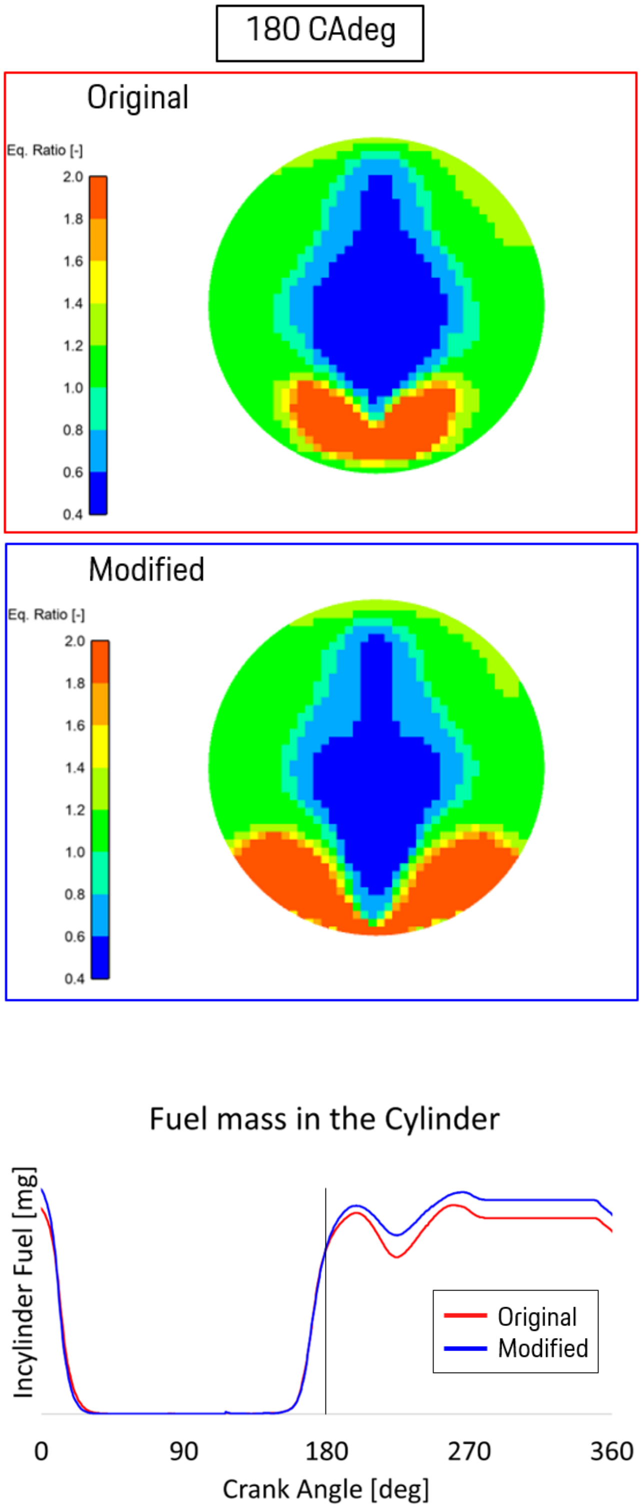


Figure 7.18: Mixture preparation at 180 CAdeg

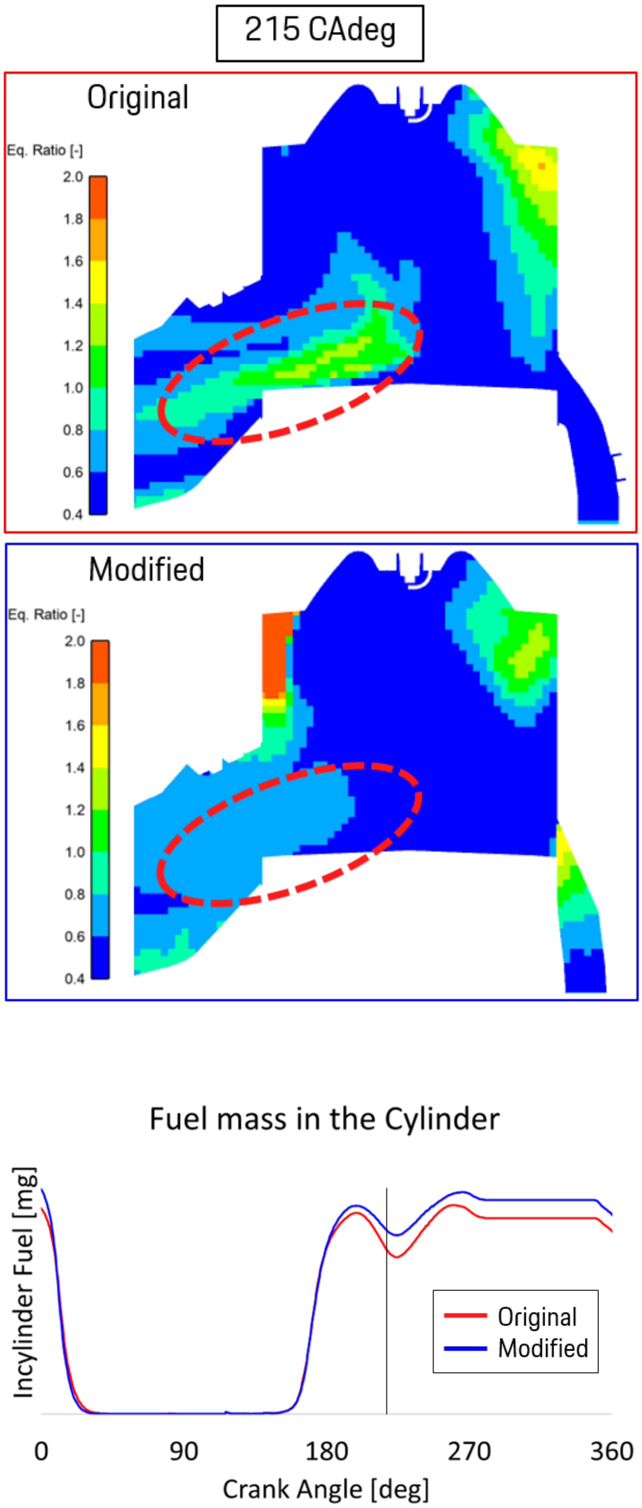


Figure 7.19: Mixture preparation at 215 CAdeg

Chapter 8

Conclusions

CO_2 and pollutant emission regulations increase the number of challenges for engine engineers. This leads to an increased need of advanced simulation tool able to support the development of a new engine at the test bench. This thesis shows that the coupled methodology is a reliable numerical tool able to well describe the interactions between scavenging and combustion processes and it is an opportunity to combine the advantages of both 1D and 3D methodology: higher amount of results in shorter calculation time, due to the reduced number of cells, and required flow details where needed. After a meticulous calibration, the agreement with the experimental data was quite satisfactory with reference to the instantaneous pressure profile.

In the second part of the thesis, it has been proven that the combination of the approaches can be used also for predictive simulation. The gas exchange process has been optimized performing two modifications: the exhaust expansion manifold in 1D and the boost port in 3D. In both cases it has been improved the efficiency of the engine and reduced the emission of unburned hydrocarbons and carbon dioxide.

For future applications, since in the coupled simulation the boundary conditions needed to the 3D model are quickly provided with the 1D model, it can be potentially used to simulate different operating points of a new two stroke engine in its early development phase and be afterward validated by test bench measurements.

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