Influence of *ANSYS FLUENT* on Gas Engine Modeling

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ABSTRACT: *Natural gas based engines have significant contribution in different sectors and hence a very large no of studies have addressed this issue. In the current research we described the simulation process of non-premixed combustion in a direct injection natural gas engine. Direct injection natural gas engines are used in many heavy duty vehicles. Similar to diesel engines, high thermal efficiency and power density is maintained in such direct injection natural gas engines. Due to the non-premixed nature of the combustion occurring in such engines, non-premixed combustion model of ANSYS FLUENT 13 can be used to simulate the combustion process. The CFD modeling of fuel injection and, combustion were performed to simulate these phenomena and processes in a compressed natural gas direct injection engine. The details of the engine geometry chosen for the calculations were given along with and explanation for the modeling choice. The simulation we have introduced has potential applications in the forth-coming studies as combustion chamber geometry, injection timing and duration, compression ratios and piston geometry in order to increase the efficiency of the combustion process. We have described the work with good amount of experimental data.*

Keywords: CFD, Modeling, Combustion, Natural Gas Engines, Non-Premixed Combustion

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1. Introduction

The current work on gas engines start with a brief description of the fundamentals of the reduction of NOx and particulate matter (PM) emissions with high fuel efficiency. Extensive research has been conducted on various LTC combustion concepts for both light duty (LD) and heavy duty (HD) engines operating on diesel fuels, gasoline fuels, and alcohols [1,2]. Low temperature combustion (LTC) is known to be a promising approach for simultaneous reduction and this has good impact. LTC requires that fuel and air be sufficiently mixed prior to combustion to avoid soot formation in excessively rich pockets. Combustion temperatures must be low enough to avoid NOx emissions [3]. Dilution of the mixture with burned gas is one way to reduce combustion temperatures but hydrocarbon (HC) and carbon monoxide (CO) emissions typically increase. Combustion noise, load limits, and combustion phasing control are other problems that have impeded the application of LTC in production engines. However, if LTC could be achieved over the engine operating range and if combustion noise can be controlled, diesel-like fuel efficiency and compliance with stringent emissions regulations might be possible without after treatment for NOx and particulate emissions.

A gas engine is an internal combustion engine which runs on a gas fuel, such as coal gas, producer gas, biogas, landfill gas or natural gas. In the UK, the term is unambiguous. In the US, due to the widespread use of “gas” as an abbreviation for gasoline, such an engine might also be called a gaseous-fueled engine or natural gas engine[4,5]. Generally the term gas engine refers to a heavy-duty industrial engine capable of running continuously at full load for periods approaching a high fraction of 8,760 hours per year. Unlike a gasoline automobile engine, which is lightweight, high-revving and typically runs for no more than 4,000 hours in its entire life. Typical power ranges from 10 kW (13 hp) to 4,000 kW (5,364 hp).

In this paper one describe the simulation process of non-premixed combustion in a direct injection natural gas engine[6]. Direct injection natural gas engines are used in many heavy duty vehicles. Similar to diesel engines, high thermal efficiency and power density is maintained in such direct injection natural gas engines. In such engines, natural gas is injected directly into the combustion chamber. Then the gas mixes with the high pressure air in the combustion chamber and combustion occurs. Due to the non-premixed nature of the combustion occurring in such engines, non-premixed combustion model of ANSYS FLUENT 13 can be used to simulate the combustion process[7].

The paper demonstrates how to do the following:

• Set up an in-cylinder (IC) case involving only a part of compression and power stroke with only a sector of mesh.

• Set up IC non-premixed combustion in ANSYS FLUENT 13.

• Use user-defined functions (UDF) to specify initial swirl and injection flow rate.

2. Finite Volume Analisys (FVA) Model of the In-Cylinder Zone

The goal of this paper is to model process of non-premixed combustion in a direct injection natural gas engine. The finite volumes mesh will look like in Figure 1.

The fuel to be considered in simulation will be CH_4 with standard properties.

The engine functioning/geometric parameters are: Crank shaft speed 2000 rpm; Crank period 720 deg; Piston stroke 120 mm; Connecting rod radius 220 mm; Piston diameter 80 mm.

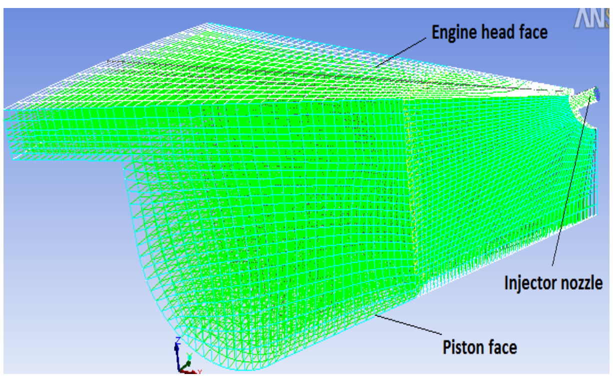


Figure 1. FV mesh

In order to re-mesh the new volumes created by the moving piston face, the Dynamic meshing option is ON. The mesh method is the Layering one.

3. CFA Simulation Results

The results analysis will cover the following crank angles (CA):

• *CA* = 717.5 (deg) just before starting the injection process.

• *CA* = 722.5 (deg) in which the injection of fuel already started

• *CA* = 737.5 (deg) where the injection is nearly finished

3.1 Results for *CA* = 717.5 (deg)

• Velocity Fields

At this time the fuel is not yet injected. This step is important to be shown in order to be used as a benchmark to analyze the future steps[8]. The maximum velocity is the velocity of air due to the compression process and has a maximum of 24.8 m/s in the zone in between the engine head and piston face.

Here the air is pressed and forced to migrate toward the zones with lower pressure.

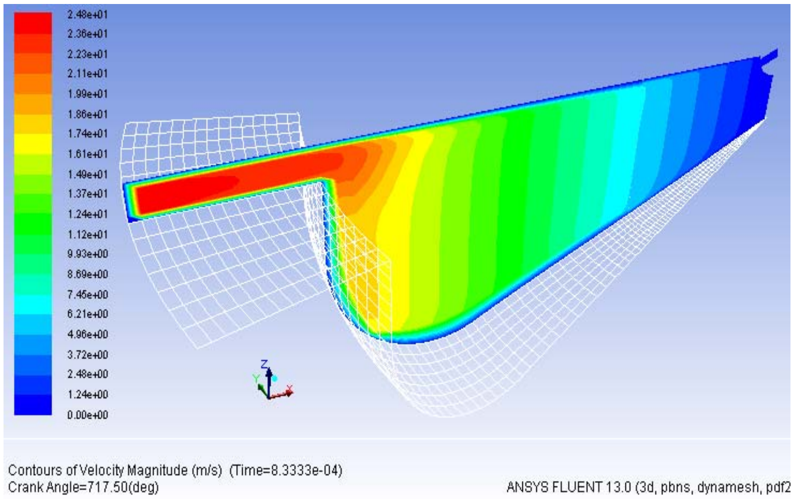


Figure 2. Velocity fields for *CA* = 717.5 deg

Total Pressure fields

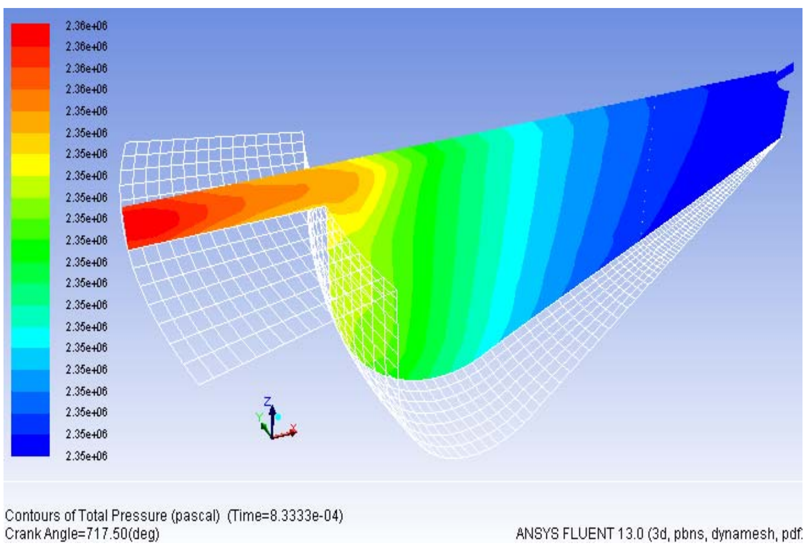


Figure 3. Total Pressure fields for *CA* = 717.5 deg

At this time the total pressure is increasing to a maximum of 2.36 e6 Pa in the same zone mentioned above.

The auto-ignition point for methane is 5800C so that before injection the compression process should ensure a temperature above this point inside the injection process.

The compression process as seen above is ensuring a temperature of 7290C inside the combustion chamber so that the condition for auto-ignition is satisfied.

• **Temperature**

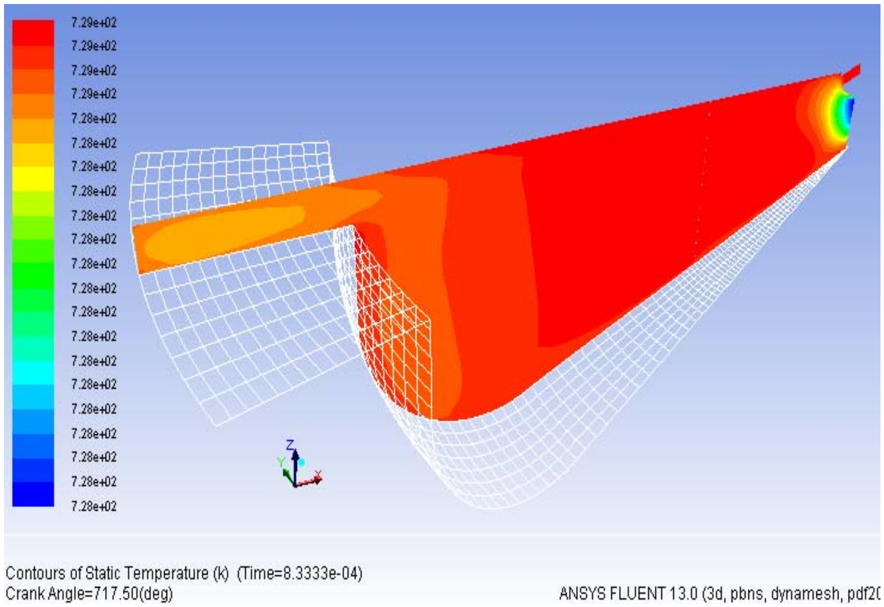


Figure 4. Turbulence kinetic energy for *CA* = 717.5 deg

• **Turbulent intensity**

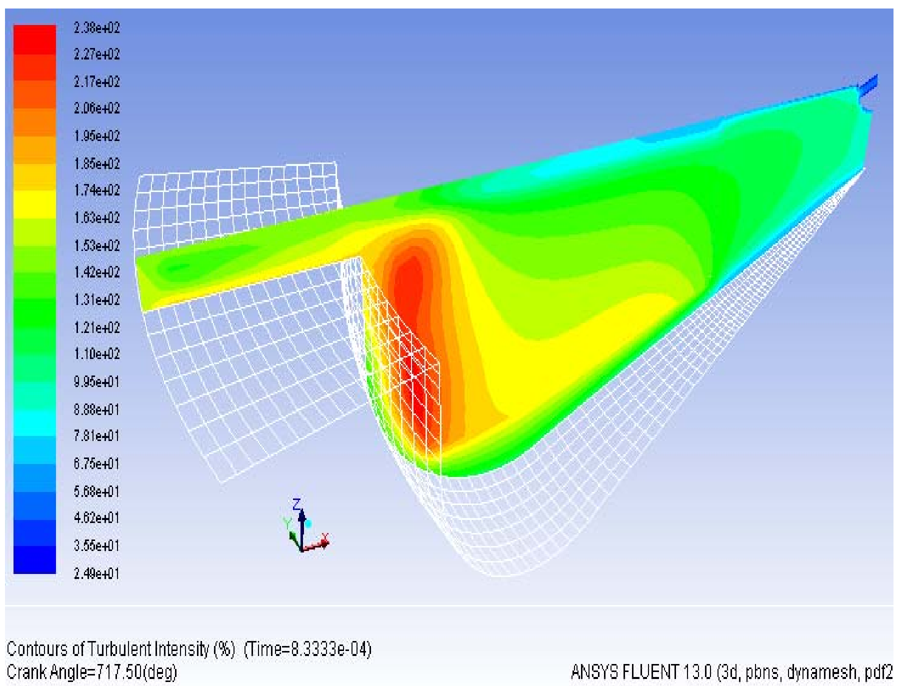


Figure 5. Turbulent intensity for *CA* = 717.5 deg

The bigger the turbulence is, the better is for the functioning of the internal combustions engines. The turbulence will ensure a good fuel-air mixture.

3.2 Results for CA = 722.5 (deg)

• Velocity fields

At this time the fuel is injected and due to the temperature existing inside the chamber is auto-igniting[9,10]. Therefore an flame front will appear with a maximum velocity of 544 m/s inside the injector nozzle.

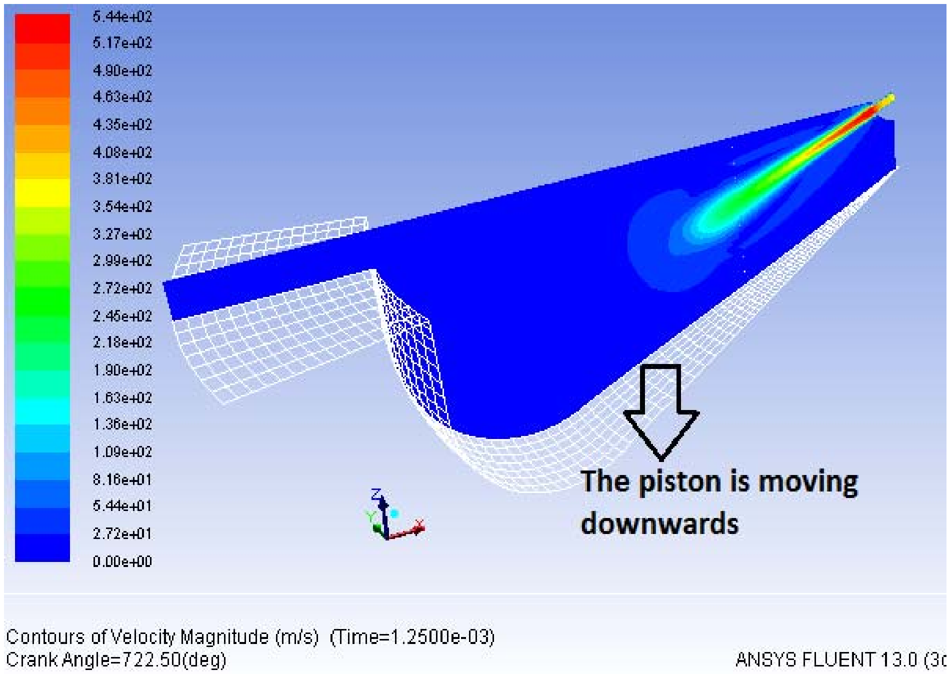


Figure 6. Velocity fields for CA=722.5 deg

• **Total Pressure fields**

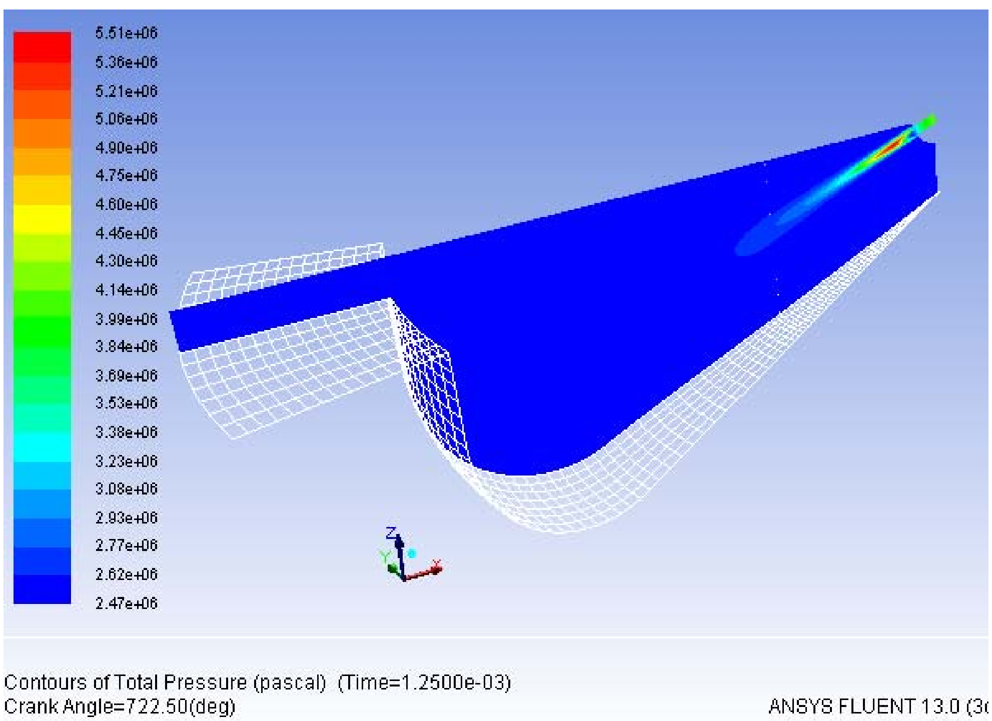


Figure 7. Total Pressure fields for *CA* = 722.5 deg

At this time the total pressure is increasing to a maximum of 5.51 e6 Pa inside the injector nozzle whereas in the reminder of the chamber will stay to an average of 2.47 e6 Pa, meaning that the flame and combustion gases started to make their affect in increasing the chamber pressure, even if the piston is moving downwards.

• **Temperature**

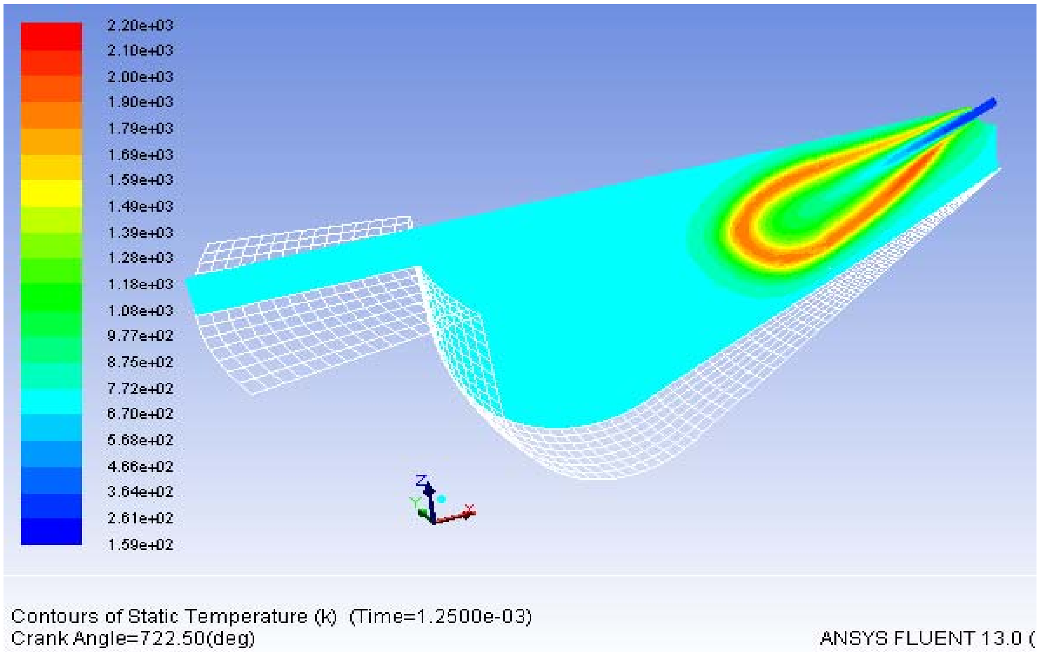
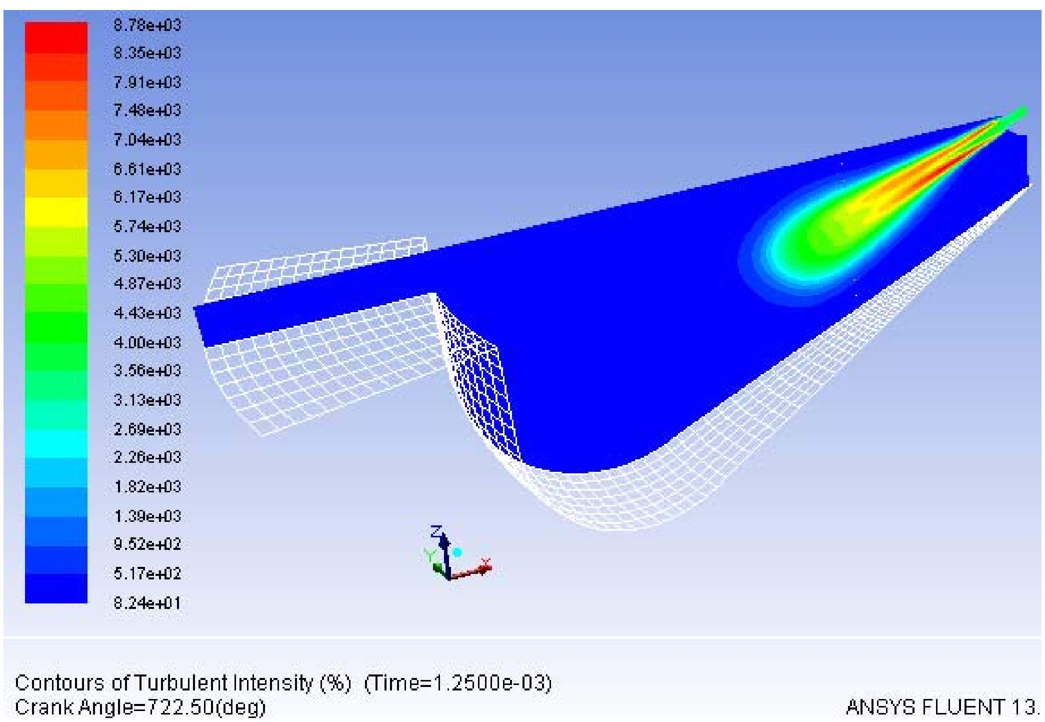


Figure 8. Temperature for *CA* = 722.5 deg

The flame front of burning fuel is clearly visible above, the maximum temperature of the front being of 2200 0C.

• **Turbulent intensity**



Figur 9. Turbulent intensity for *CA* = 722.5 deg

The turbulence inside the flame is obviously big since the combustion reactions inside the flame are taking place.

• **Mass Fraction of CH_4**

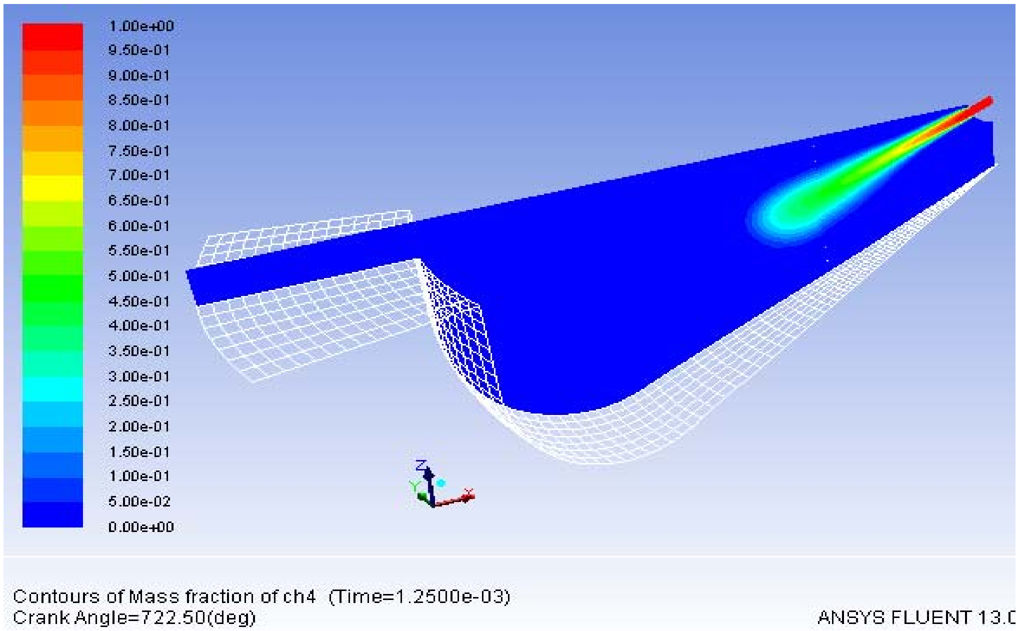


Figure 10-Mass fraction of CH_4 for CA=722.5 deg

The mass fraction of CH_4 describes the fuel yet not burnt meaning that this is the '*reserve*" for the future steps in piston evolution. Obviously the biggest quantity of CH_4 is inside the injector nozzle from where the fuel is still flowing.

- Mass Fraction of O_2

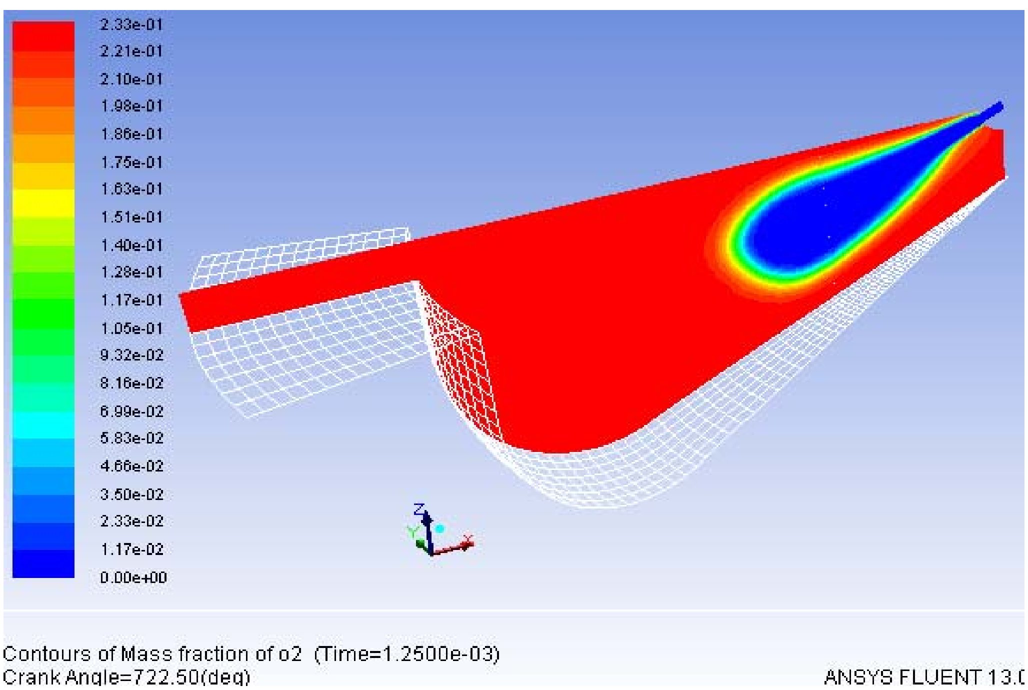


Figure 11. Mass fraction of O_2 for *CA* = 722.5 deg

The mass fraction of O_2 describes the still untouched zones of the chamber still waiting for the flame to evolve. This is the reserve for the combustion reactions.

• Mass Fraction of CO_2

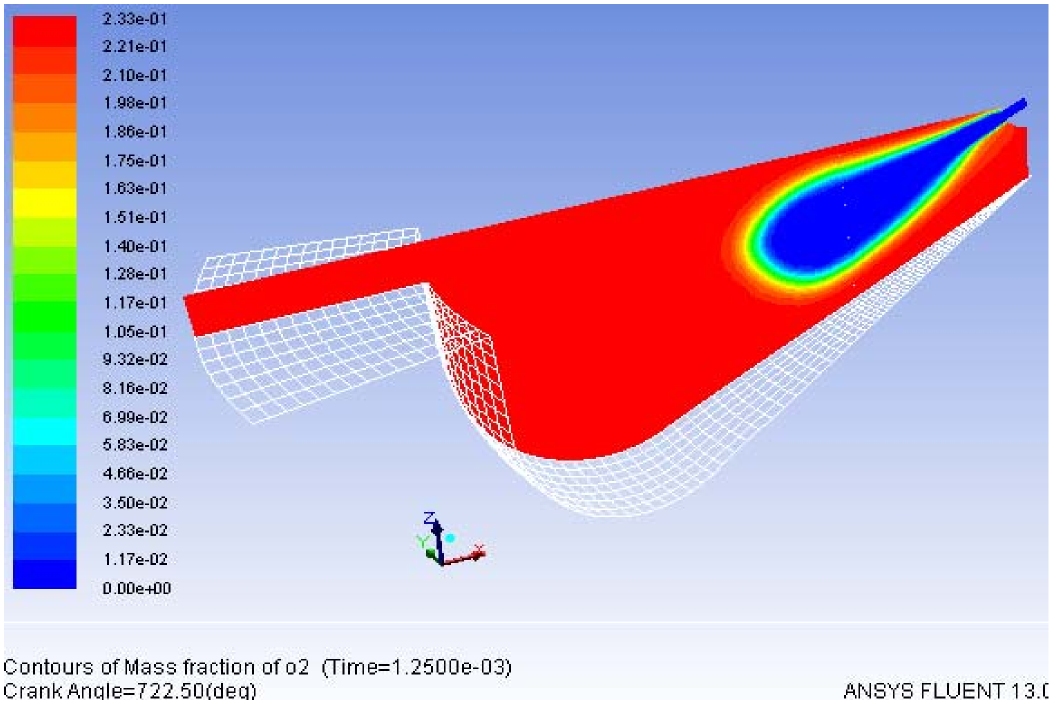


Figure 12. Mass fraction of CO2 for *CA* = 722.5 deg

The mass fraction of CO_2 describes the combustion zone where the combustion process already took place. The flame front is visible since here is the place where the chemical reactions are intense.

• Mean mixture fraction

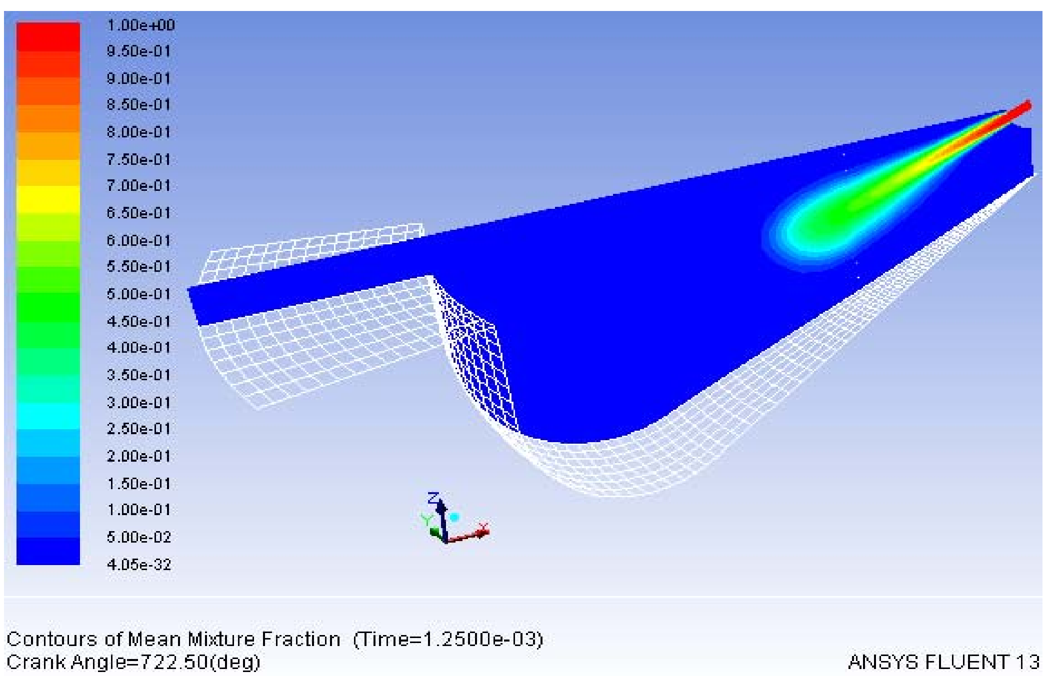


Figure 13. Mean mixture fraction for *CA* = 722.5 deg

The mean mixture fraction is a non-dimensional scalar useful to see the rate in which fuel and oxidizer are mixed creating the conditions for combustion. As seen above the flame body has values above zero, whereas in the rest of the chamber the scalar is near to zero.

3.3 Results for *CA* = 737.5 (deg)

• Velocity fields

By now the injection process is almost finished. The speed of the injected fuel is decreasing to 357 m/s and the flame is approaching its maximum extent.

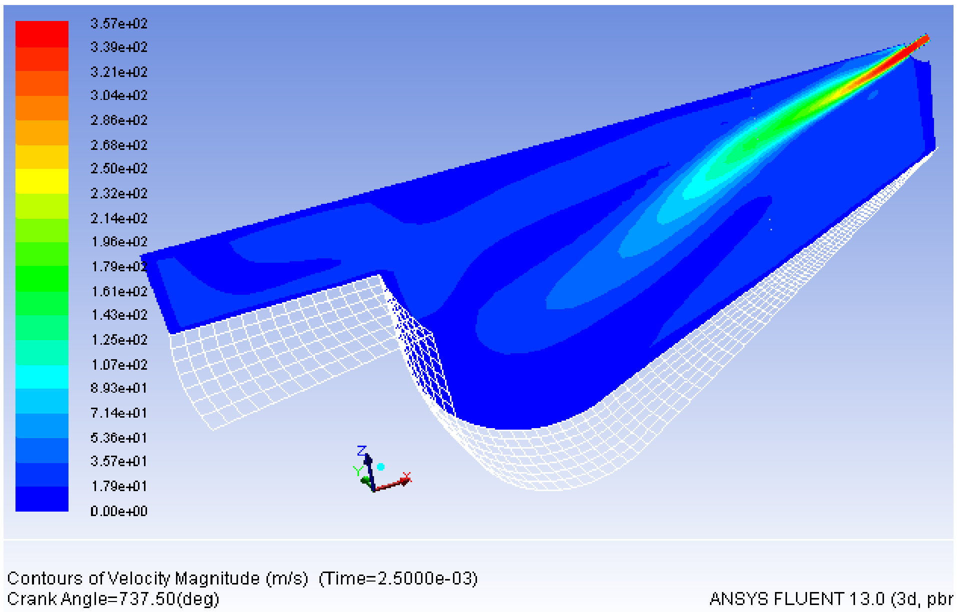


Figure 14. Velocity fields for *CA* = 737.5 deg

• Total Pressure Fields

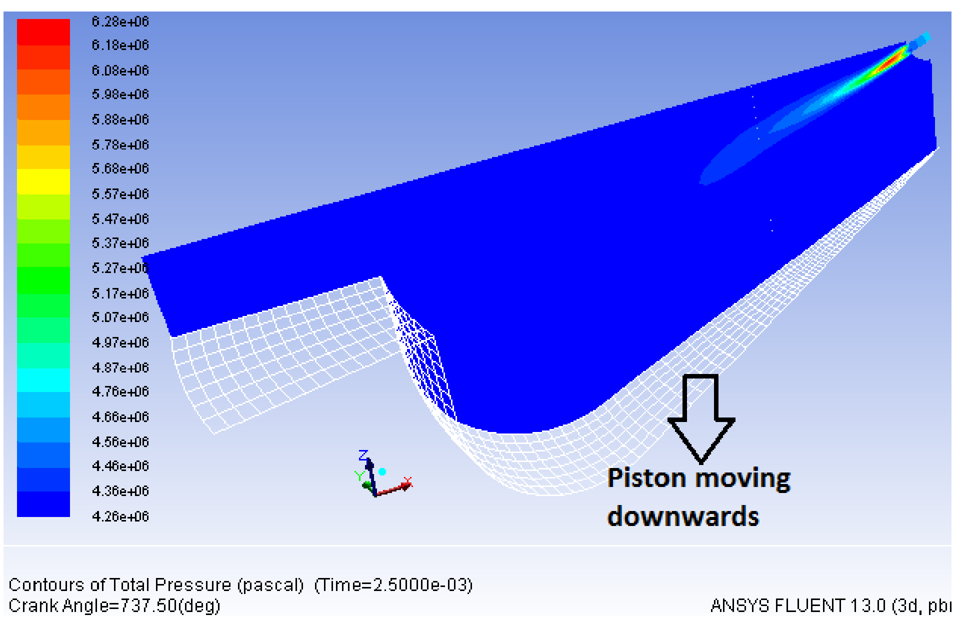


Figure 15. Total Pressure fields for *CA* = 737.5 deg

Despite the increasing volume of the combustion chamber due to the piston motion, the average pressure inside is increasing to 4.26e6 Pa and that for the combustion process still taking place.

• Temperature

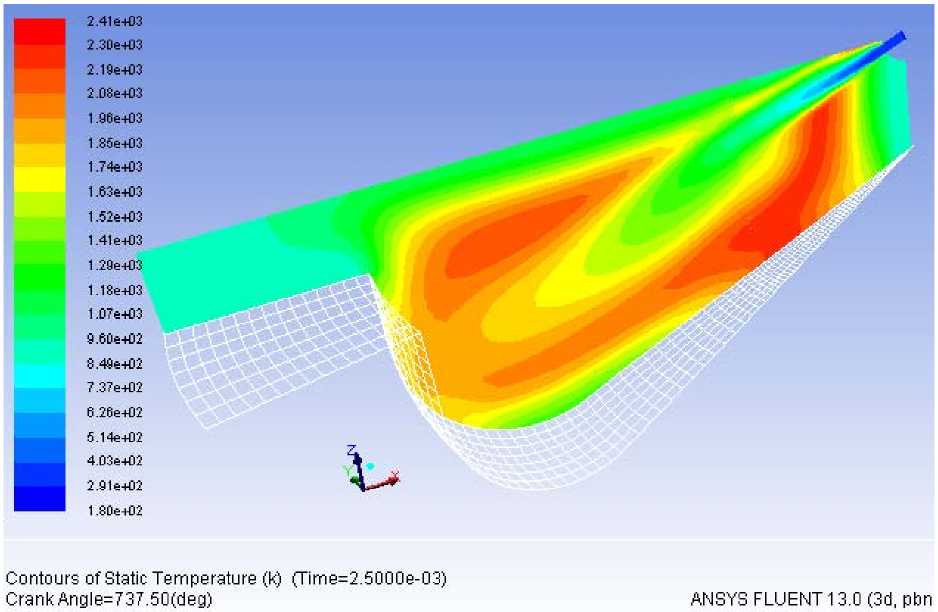


Figure 16. Temperature for *CA* = 737.5 deg

Due to combustion reactions the temperature fields rise to a maximum of 2410 ^{0}C and the fields are wider and spreading almost the entire volume of the chamber following the flame front.

• Turbulent intensity

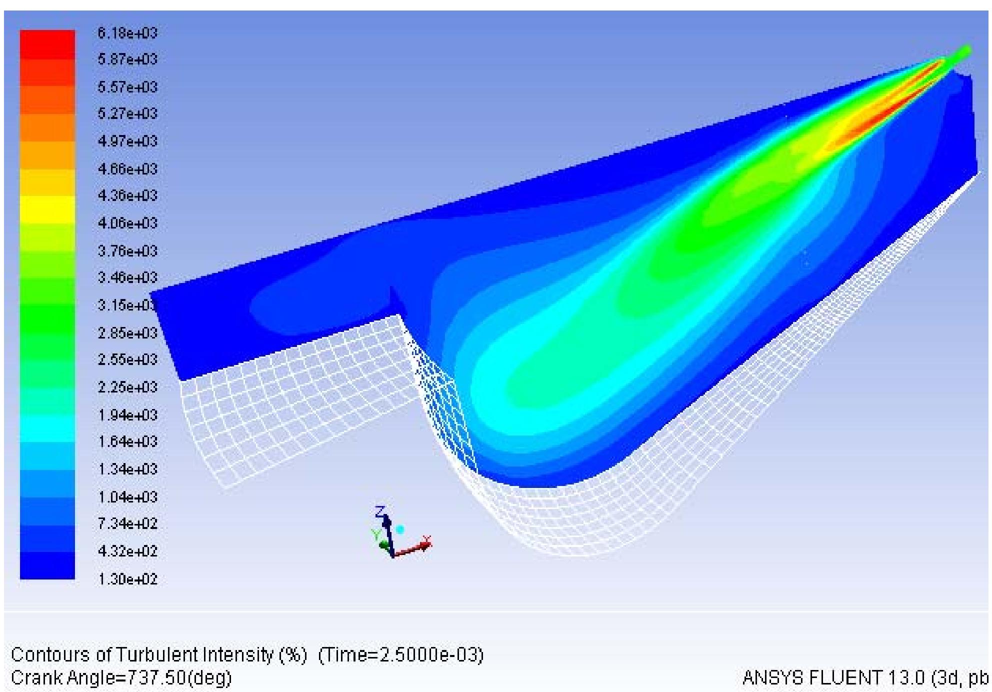


Figure 17. Turbulent intensity for *CA* = 737.5 deg

The turbulence inside the cylinder is widening as volume tending to embrace the entire IC as seen above.

• Mass Fraction of CH_4

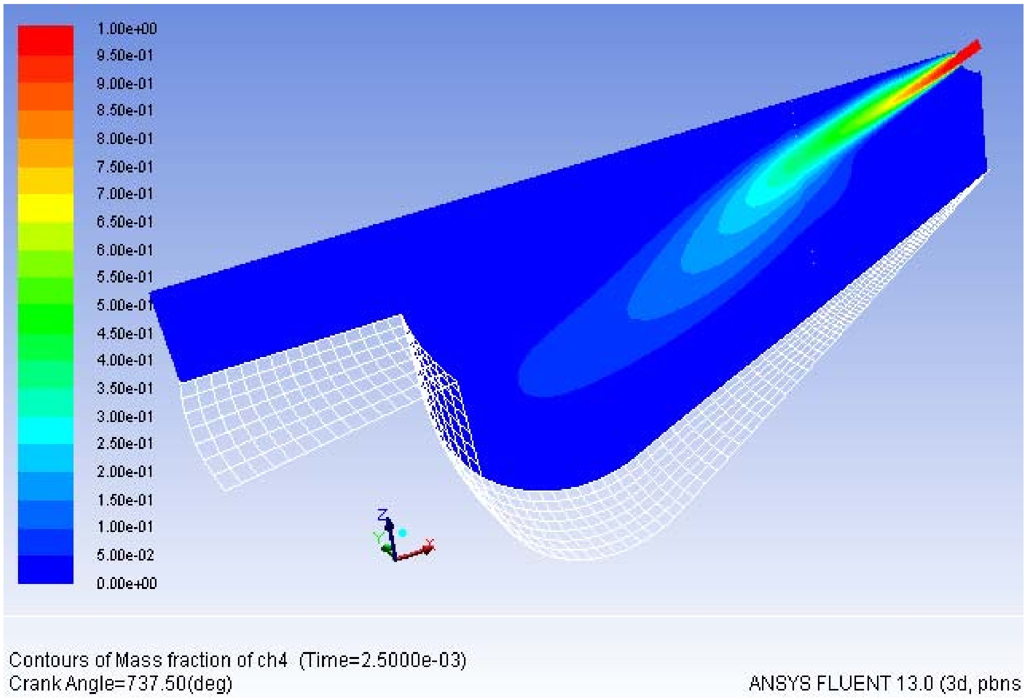


Figure 18. Mass fraction of CH4 for *CA* = 737.5 deg

The mass fraction of CH_4 is penetrating inside the chamber following the fuel jet..

• Mass Fraction of O_2

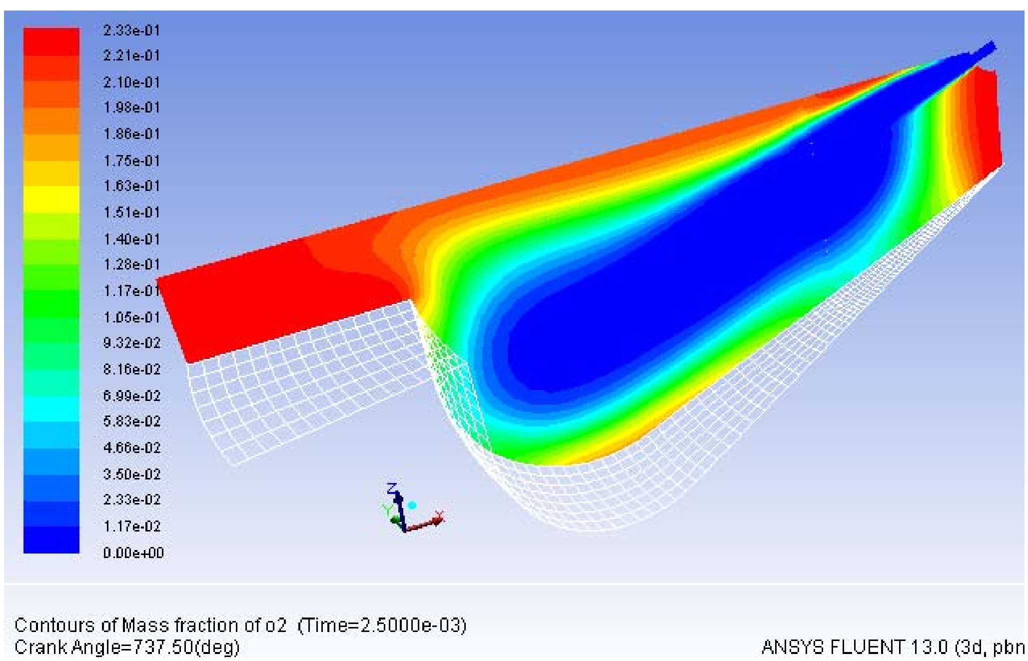


Figure 19. Mass fraction of O_2 for *CA* = 737.5 deg

The mass fraction of O_2 is becoming smaller since the O_2 is consumed progressively by the combustion process.

• Mass Fraction of CO_2

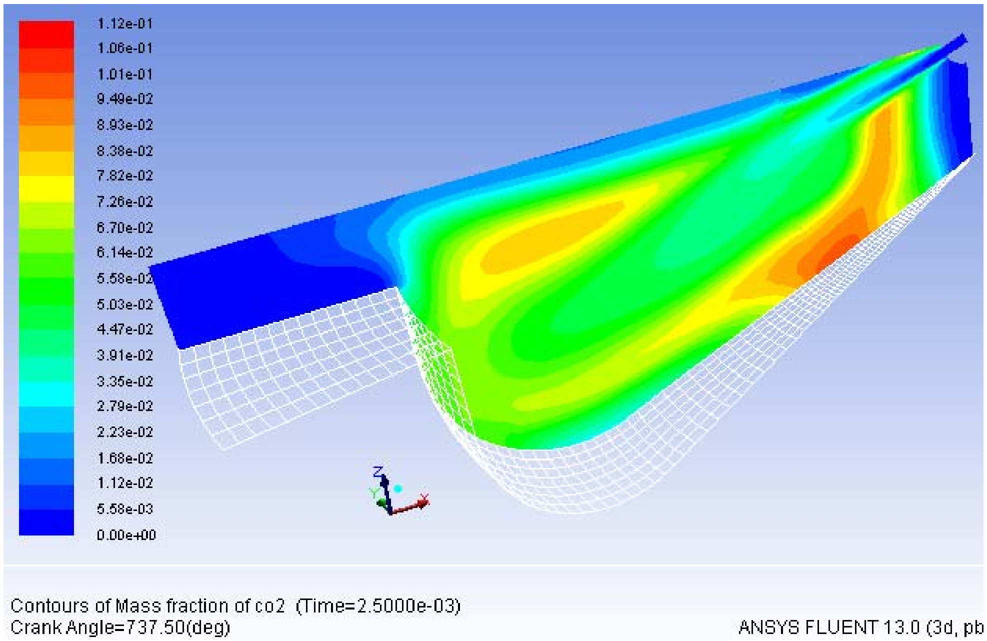


Figure 20. Mass fraction of CO_2 for *CA* = 737.5 deg

The mass fraction of CO_2 is becoming wider as the chemical reactions of combustion took place in wider areas.

• Mean mixture fraction

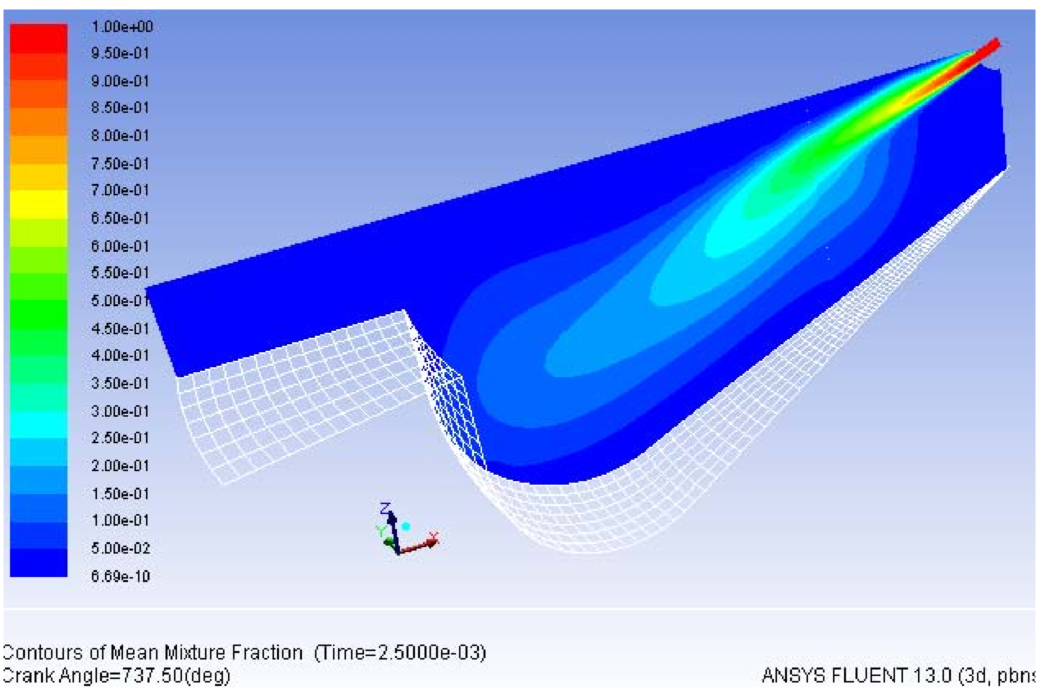


Figure 21. Mean mixture fraction for *CA* = 737.5 deg

The mean mixture fraction is becoming wider as well following the flame front.

4. Conclusions

The CFD modeling of fuel injection and, combustion were performed to simulate these phenomena and processes in a compressed natural gas direct injection engine. The details of the engine geometry chosen for the calculations are given along with and explanation for the modeling choice.

The simulation can be used for further optimization studies as combustion chamber geometry, injection timing and duration, compression ratios and piston geometry in order to increase the efficiency of the combustion process.

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