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JOINVILLE TECHNOLOGICAL CENTER
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WELLINGTON BARCELOS ADÃO

SPRAY BEHAVIOR ON COMPRESSION IGNITION INTERNAL COMBUSTION ENGINES: A COMPUTATIONAL ANALYSIS USING CFD

INTERNAL COMBUSTION ENGINES LABORATORY - LABMCI
VEHICULAR SYSTEMS ENERGY EFFICIENCY
RESEARCH GROUP

Joinville

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ENGINES: A COMPUTATIONAL ANALYSIS USING CFD

Thesis submitted as partial fulfillment of the
requirements for the degree of Bachelor in
Automotive Engineering of Federal University
of Santa Catarina, Joinville Technological
Center, Brazil.

Advisor: Prof. Leonel R. Cancino, Dr. Eng.

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ENGINES: A COMPUTATIONAL ANALYSIS USING CFD

This course completion work was deemed appropriate to obtain the Automotive Engineer's degree and approved in its final form by the Undergraduate Program in Automotive Engineering of the Federal University of Santa Catarina.

Joinville, 04 November de 2018.

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Do not train children to learning by force and harshness, but direct them to it by what amuses their minds, so that you may be better able to discover with accuracy the peculiar bent of the genius of each (Plato).

ABSTRACT

Using the comparison of experiments, we analyzed the diesel injector nozzle and its main characteristics. With the aid of engineering software, AVL FIRE, we can simulate the different operating conditions and analyze the development of cavitation and erosion in the injector channel. By varying the initial conditions - initial pressure, initial temperature and injection needle elevation, it is possible to identify variations in the injected fuel range, penetration and involved phases.

Keywords: AVL. Cavitation. Diesel. Direct injection. Spray.

RESUMO

Utilizando a comparação de experimentos, analisamos o bico injetor diesel e suas características principais. Com o auxílio de software de engenharia, AVL FIRE, podemos simular as diferentes condições de operação e analisar o desenvolvimento da cavitação e erosão no canal do injetor. Variando condições inicias - pressão inicial, temperatura inicial e levante da agulha de injeção, é possível identificar variações no leque de combustível injetado, penetração e fases envolvidas.

Palavras-chave: AVL. Cavitação. Diesel. Injeção direta. Spray.

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LIST OF ACRONYMS

A/F – Air per fuel

CFD – Computed fluid dynamics

CR – Common rail

DDM – Discrete droplet method

DOE – Design of experiments

IC – Internal combustion

NOx – Nitrogen oxides

PM – Particulate matter

SAC – Area around pintel tip

SMD – Sauter mean diameter

TDC – Top dead center

VCO – Valve covered orifice

LIST OF SYMBOLS

m_d – particle mass

$\frac{du_{id}}{dt}$ – acceleration

F_{idr} – drag force

F_{ig} – forces including gravity

F_{ip} – pressure force

F_{ivm} – virtual mass force

F_{ib} – body forces

D_p – drag function

u_{irel} – relative velocity

$C_{D.|u_{rel}|}$ – drag coefficient

A_d – cross sectional area

Re_d – Reynolds number

∇_p – differential pressure operator

d_g – mean molecule diameter

Kn_p – Knudsen number

μ_g – fluid viscosity

ρ_p – density

λ – mean free path length

CONTENTS

1. **INTRODUCTION**

The use of petroleum derivates in the world is high, one of the sectors that use it most is in the transportation area. In Brazil, the transportation sector represented 77,8% of the final energy consumption of diesel in 2015 (BRASIL, 2016). In Europe, the scenery is not very different. It is expected that its demand as final energy will continue until 2030, reducing its participation slowly only during 2030-2050 (EUROPEAN COMMISSION, 2016).

Due to scarcity of crude oil and its high rising price makes a challenging job for the engine manufactures to manufacture and model less polluting and more efficient engine in order to meet its future demands (KUMAR; CHAUHAN; VARUN, 2013).

Since combustion products are sometimes harmful to the environment, the study and implementation of new technologies, to understand the phenomenon involved, increase efficiency and reduce pollutants, is of the utmost importance.

In diesel engines or compression ignition engines, the relationship between nozzle aperture, channel length, pressure and the injection time, besides the working temperatures are directly related to the spray development and characteristics in the chamber, consequently with the levels of efficiency and emission. Fuel spray atomization has an important role in the reduction of nitrogen oxides (NOx) and emission of particulates (YODA; TSUDA, 1997). Effective control of the primary break-up determines the initial droplet size, speed and any impacts to subsequent processes such as droplet collision, evaporation and inhalation in the fuel-air mixture (MOVAGHAR et al., 2017).

Experimental work which aimed at fuel economy and low pollutants emission for IC engine requires change in input parameter which is highly demanding in terms of money and time. An alternative is the numerical simulation of engine performance with the help of mathematical model and powerful digital computers reducing cost and time (KUMAR; CHAUHAN; VARUN, 2013).

By means of numerical simulation, with the help of AVL BOOST and FIRE software, in this work is analyzed the spray behavior inside the cylinder, the interactions between different input parameters, as well as the effect of cavitation on the injector, correlating the changes with the referred results. The AVL FIRE program is a simulator designed for complete

modeling of the engine, considering the entire combustion cycle (GRABOWSKI; PIETRYKOWSKI; WENDEKER, 2012).

Numerous fundamental experiments and semi-empirical relations about the general behavior of the relevant spray parameters of diesel spray as cone angle, penetration, break-up length and average droplet diameter, usually been performed with quasi-stationary sprays, most of the results can only be used to describe the main injection phase, full needle lift (BAUMGARTEN, 2005).

1.1   Objectives

1.1.1   Main objective

To analyze the diesel spray behavior inside the combustion chamber by means of numerical simulation, considering cavitation.

1.1.2   Specific objectives

- Analyze qualitatively the overall effects of droplets size on pollutant emission;
- Identify the different phase velocities for each model;
- Evaluate the effects and measure the erosion caused by cavitation in the nozzle injector;
- Correlate engine efficiency with different pressure and operating temperatures.

The numerical analysis will be done using the AVL FIRE software, taking as geometric and operation conditions an internal data-base of AVL-AST software (910 - injector nozzle diesel).

2.   THEORETICAL BASES

This chapter presents the basic topics to understanding four-stroke combustion engines, definitions, characteristics and convenient formulations for compression ignition engines. The geometry, application and concept of nozzles are shown. It exposes the basic concept of cavitation and its implications as well as the concepts of spray injected into diesel engines.

2.1   Internal combustion engines

Internal combustion engines are intended to provide mechanical energy through the chemical energy contained in the fuels. This energy was released by the oxidation of the fuel (HEYWOOD, 1988).

The conversion of chemical energy into heat is carried out through combustion, while the subsequent conversion into mechanical work is performed by the expansion of the gases burned in the combustion chamber (BOSCH, 2005).

Figure 1 shows a cylinder and the cycles of a reciprocating four-stroke engine.

Figure 1 - Four-stroke combustion engine
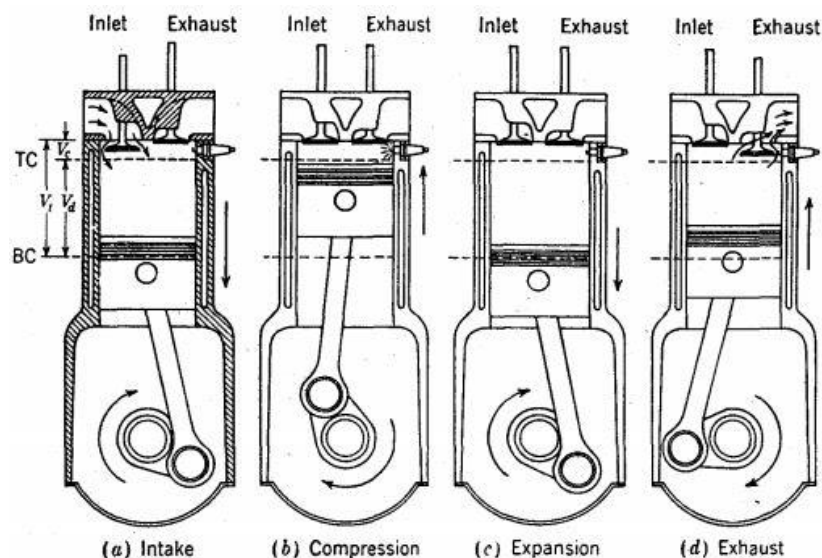
Font: Heywood, 1988.

It is possible analyze the intake (a), where the air is aspirated into the cylinder – in motors by spark and indirect injection this mixture is of air/fuel (A/F), suffering after compression (b). At this moment, the A/F mixture is under ideal pressure and temperature to

ignite through the ignition spark in spark engines and auto-ignition in diesel engines. With the combustion process in progress, the piston begins it descends through the cylinder entering the gas expansion time (c), thus ending the four-stroke cycle with the opening of the exhaust valves and the last cycle which is itself exhausted gases (d).

Combustion engines can be classified in different ways and approaches, but only the interest classifications, the cycle and the fuel supply will be analyzed in this work.

2.1.1 Spark ignition engines

Engines ignited by spark, air and fuel are admitted, previously managed (indirect injection) or formed inside the cylinders when they use direct injection and reach the flame point by an electric discharge, also known as spark, that occurs between the electrodes of spark plug (BRUNETTI, 2012).

Spark ignited engines have an excellent commercial applicability because they have a high variety of combinations and respond to a wide range of powers, being applied mainly in passenger vehicles and motorcycles.

2.1.2 Compression ignition engines

Diesel cycle engines are reciprocating engine with internal mixture formation aided by auto ignition thereof (BOSCH, 2005). The air is induced and compressed by piston movement, the fuel is injected just before the combustion process start. The load control is done by varying the fuel mass to be injected in the chamber per cycle, the air mass intake practically unchanged. The compression ratio of diesel engines is typically higher than those of spark ignition, ranging from 12 to 24, depending on the application and engine type (HEYWOOD, 1988).

The shape of the combustion chamber and the action of the piston can be used as turbulence amplifiers or to distribute liquid fuel and/or air spray/fuel vapor (BOSCH, 2005). It can be seen from figure 2 that the larger the compression ratios, the better the thermal efficiency values. For the same ratio, the Otto cycle is more efficient, but diesel engines operate at higher compression rates, as already mentioned. For the same maximum cycle temperature, the ideal diesel cycle is more efficient than the Otto cycle.

Figure 2 - Efficiency chart by compression ratio
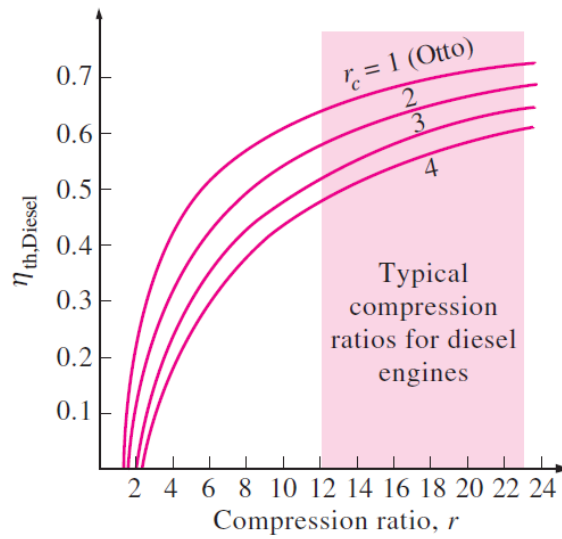
Font: Çengel and Boles, 2006.

The fuel injection process begins near the top dead center, TDC, and continues during the first part of gas expansion. Due to this longer duration, the ideal combustion process of the diesel cycle is approximated as a process of supplying heat at constant pressure as shown in figure 3 (ÇENGEL; BOLES, 2006).

Figure 3 - Diagram P-v ideal diesel cycle
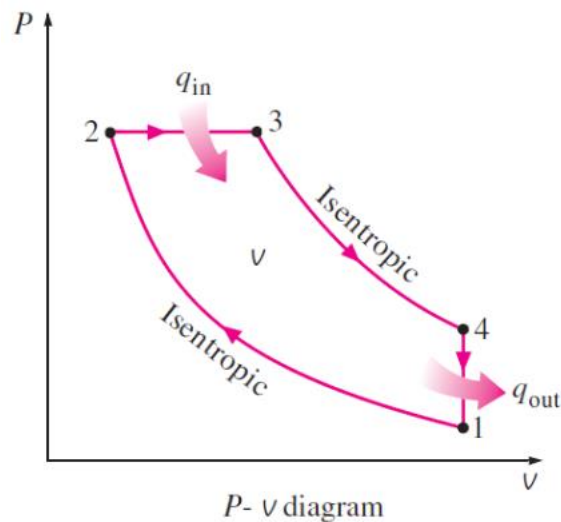
Font: Çengel and Boles, 2006.

This is the only difference between the cycles Otto and Diesel, the other processes are ideally alike. In points 1-2 it is a process of isentropic compression, in 2-3 we have an addition of heat at constant pressure, in 3-4 isentropic expansion and 4-1 a heat rejection at constant volume (ÇENGEL; BOLES, 2006).

2.2   Fuel admission systems

The purpose of the fuel admission system is to vary the mass of fuel admitted to achieve the desired A/F ratio. This relation is a quotient between mass or fuel flow and the mass of air flow that makes up the mixture (BRUNETTI, 2012).

Diesel engines can be divided into two types: those with indirect injection where the chamber can be divided into two regions and fuel injected into a pre-chamber and those of direct injection that have a single open combustion chamber to which the fuel is injected directly (HEYWOOD, 1988).

The injector nozzle is one of the most important elements of a diesel engine and the spray characteristics affect directly the performance and pollutant formation (PAYRI; MARGOT; SALVADOR, 2002).

Two basic types of nozzles can be classified depending on the design of the needle seat and nozzle hole inlets. Figure 4 shown a needle SAC type – which will be the model used in the numerical experiment, on the left and valve covered orifice (VCO) on the right. Developments in the design of the injector nozzle have led to a reduction in SAC volume with the introduction of mini-SAC and VCO nozzles and a consequence improvement in engine hydrocarbon emissions (MORGAN et al., 2001).

Figure 4 - Injector types. SAC (left) and VCO (right) type fuel injection nozzles
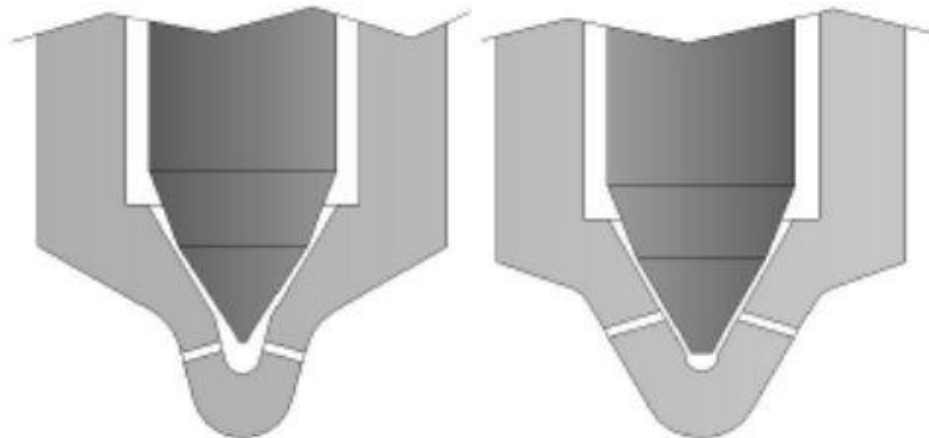
Font: Lindström, 2009.

The major difference between the two designs lies in the fact that in SAC or mini-SAC the needle closes the nozzles in a region above the holes. In VCO type the needle cover the holes when it is closed.

2.2.1 Indirect injection

The air intake generates a swirl, which is a fluid movement within the spiral cylinder, and despite the amplification of this phenomenon by the piston, the A/F is not sufficiently mixed for small diesel engines operating at high speeds that are normally used in automobiles. As shown in figure 5, indirect injections or divided into pre-chambers aid in this mixing process (HEYWOOD, 1988).

Figure 5 - Two examples of indirect injection: (a) pre-swirl chamber; (b) turbulent pre-chamber
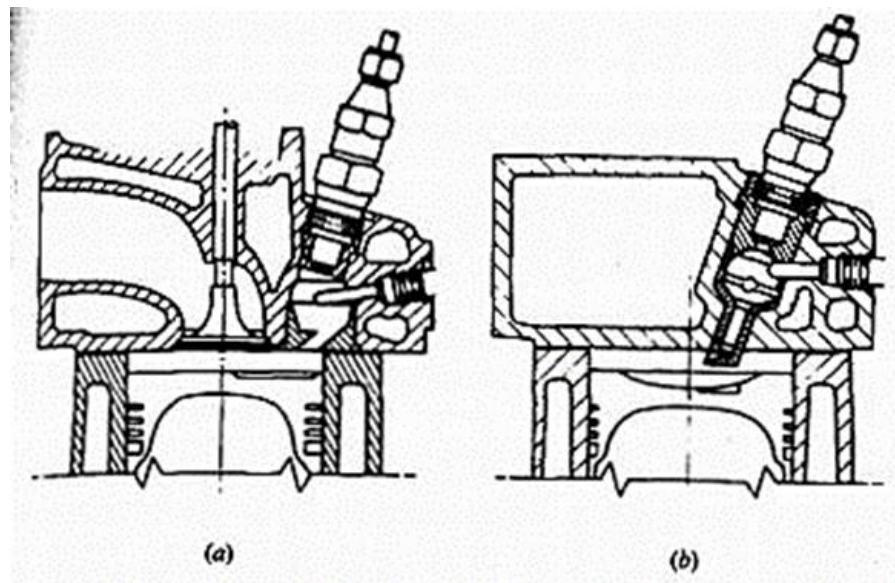
Font: Heywood, 1988.

In the compression process, air is forced against the pre-chambers above the piston where they suffer a strong induction in the flow helping the mixture. The fuel is injected by low pressure systems through single spray jet injectors (Heywood, 1988).

2.2.2 Direct injection

In high torque engines, where the mixture ratio is less important, quiescent direct injection systems are used. The figure 6 illustrates some types of direct injection, who is the model used in this numerical experiment due to the wide use.

Figure 6 - Common types of direct injection: (a) quiescent chamber with multiple holes; (b) multi hole bowl-in piston chamber; (c) single hole bowl-in piston chamber
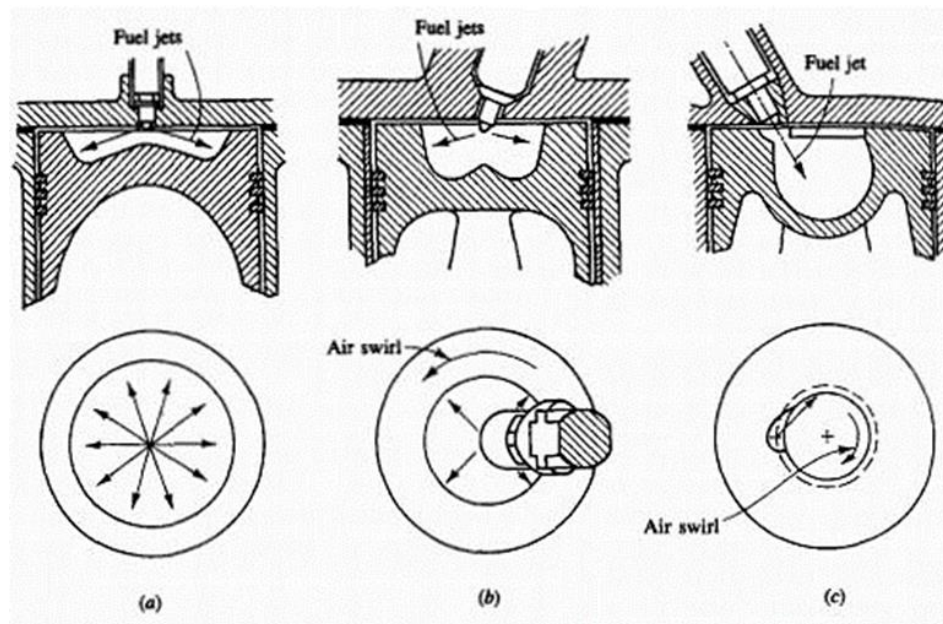
Font: Heywood, 1988.

The time and energy of the direct injection of the fuel jet are sufficient to reach the proper distribution and A/F mixture ratio. Generally, the shape of the combustion chamber looks like a shallow bowl in the piston crown, called *bowl-in*, in addition to using multi holes for injection (HEYWOOD, 1988).

According to Zhu, Zhao and Ladommatos (2003), diesel engines are increasingly used in automotive applications due to superior fuel economy. However, they have to meet stringent emission legislations on pollutants such as nitrogen oxide (NO_X) and particulate matter (PM). The multiple injections and high injection pressure offered by common rail (CR) fuel system can reduce the emission of particulate matters due to improve spray atomization and fuel mixing.

2.3   Spray

The atomizing mechanism of the liquid fuel nozzle in diesel engines, emanating from the injector is provided at a rate velocity proportional to the pressure difference between the injection system and the cylinder. Due to the relative velocity between the liquid film and the chamber environment, the aerodynamic effects (friction and pressure) induce undulating instabilities, which increase until disintegration and the consequent droplets are formed.

These droplets continue to suffer ruptures by aerodynamic effects of smaller droplets, the surface tension tries to keep the drop in spherical form, resisting the deformations. The droplets can coalesce due to the collision between them, as well as evaporate when exchanging heat with the environment (BRUNETTI, 2012).

2.3.1 Droplets theory

In theory of combustion it is assumed that droplet after break up having the same initial diameter which is equal to the Sauter Mean Diameter (SMD) having neglecting droplets size distribution and the details of the atomization process (KUMAR, CHAUHAN, VARUN, 2013).

2.4 Cavitation

If the pressure in a flowing liquid falls locally below the vapor pressure of the liquid, vapor bubbles form. This phenomenon is called cavitation. If the cavitation is extremely strong, the erosion caused by cavitation may occur in the proximity of the phenomenon.

When the surface is eroded, the process continues at a higher rate than the initial one. Increasing the roughness promoted by erosion increases the cavitation bubbles and weakens the already deteriorated surface. If this occurs within the holes of a diesel injector, the result of the geometry change would negatively influence the injection process (LINDSTRÖM, 2009).

The hydrodynamic cavitation is the most important effect in diesel engine fuel injection nozzles. At localized pressure changes inside the nozzle cavitation, bubbles and cavities originate. The visual observations of the cavitation process are extremely difficult due to high flow velocities in the spray hole and the typical spray hole dimensions (BLESSING et al., 2003).

Cavitation in such nozzles has been identified in two distinct forms. The geometric-induced cavitation is a relatively well-known phenomenon initiating at sharp corners where the pressure may fall below the vapor pressure of the flowing liquid (ANDRIOTS; GAVAISES; ARCOUMANIS, 2008).

The second form of cavitation is referred to as string or vortex cavitation, these structures have been observed in the in the bulk of the liquid inside sac and mini-sac type nozzles, where the relatively large nozzle volume allows the formation of large-scale vortical structure (MITROGLOU; GAVAISES, 2013).

Studies have provided evidence that geometric-induced and vortex cavitation structures can interact, thus increasing the inherent instability of such transient flow fields (MITROGLOU; GAVAISES, 2013).

The full-cone high pressure given in figure 7, show the development of a SAC hole and injection hole. The first break-up is called primary break-up and results in large droplets that form the dense spray near the nozzle. In high pressure injection, cavitation and turbulence are the main break-up mechanisms (BAUMGARTEN, 2005).

Figure 7 - Full cone spray development
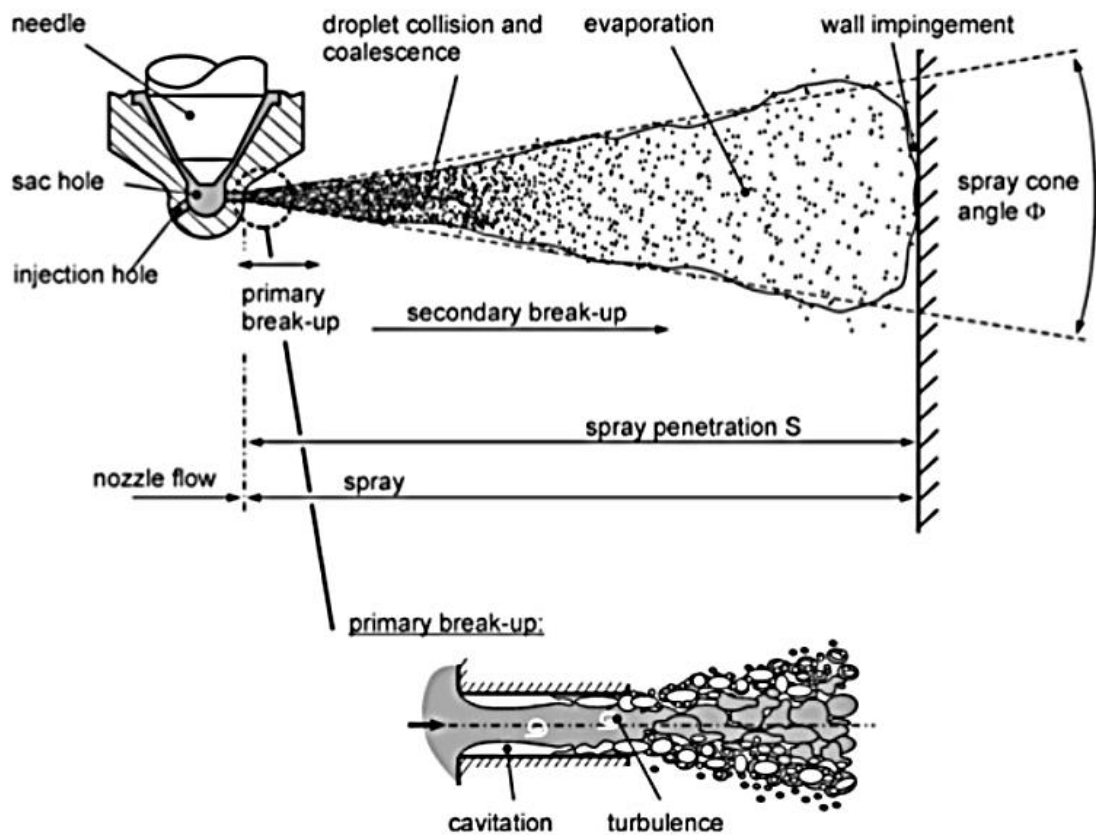
Font: Baumgarten, 2005.

As the spray develops, the droplets with low kinetics are push to outer spray region, forming the conical design. The further away from the initial cone, the more steam and less liquid are witnessed. In the dilute spray further downstream, the main factors of influence on further spray disintegration and evaporation are the boundary conditions imposed by chamber as gas temperature and density as well as gas flow (BAUMGARTEN, 2005).

Cavitation develops inside the nozzle holes because of the decrease of static pressure due to the strong acceleration of liquid, different from boiling, where there is an increase in temperature, as in figure 8.

The two mean forms of cavitation are hydrodynamic cavitation, that occurs when the pressure decrease below static vapor pressure of liquid, and the geometric cavitation which is influenced by the stress concentrations imposed by the geometry, causing pressure drop and thus cavitation.

Figure 8 - Graph of state of matter. Difference of boiling and cavitation
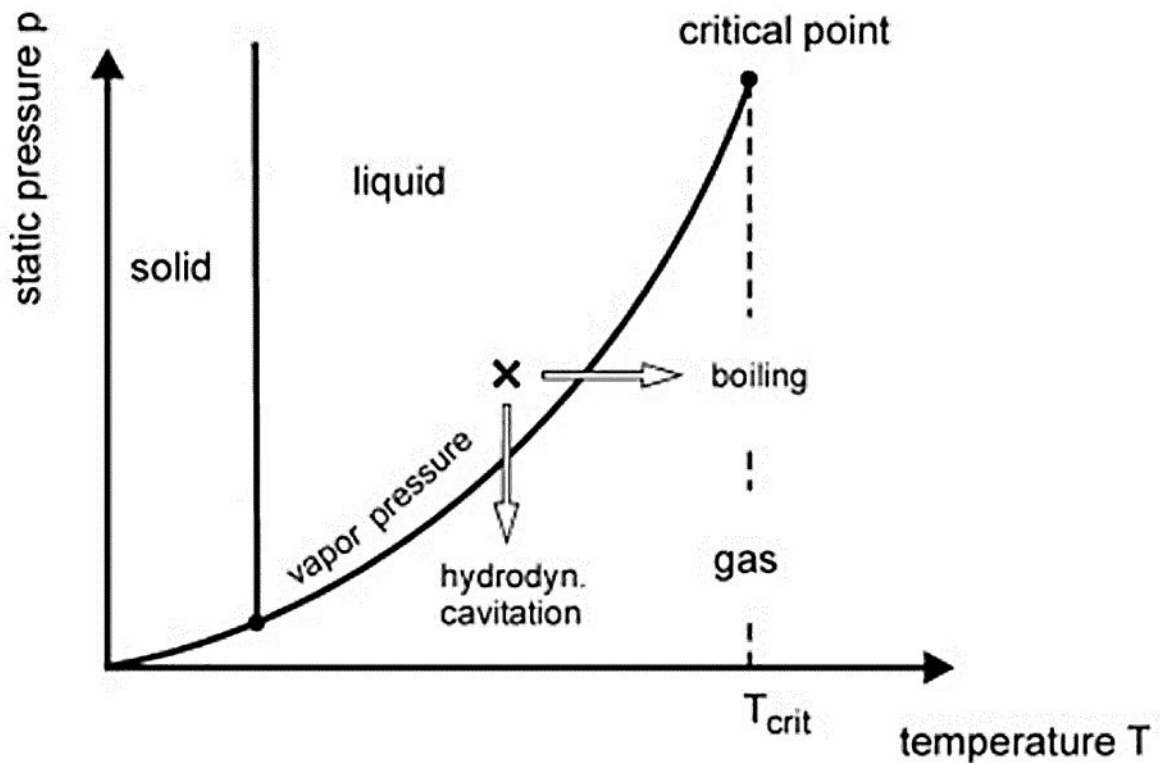
Font: Baumgarten, 2005.

These cavitation inside the nozzle increase the turbulence level and stimulate the spray disintegration.

2.5   AVL

The AVL software represents an advanced and fully integrated virtual motor simulation tool with models for predicting accurate performance (ION; ANISOR, 2011).

It is used for the design and optimization of internal combustion engines in terms of dynamic fluid. It can solve many of the demands of flow problems with their complex geometries, chemical and physical modeling, particularly developed for the calculation of problems related to internal combustion engines. Including simulations of phenomena within the cylinder, such as gas changes, fuel injection, blending generation, combustion and emission

forms, as well as other functionalities (GRABOWSKI; PIETRYKOWSKI; WENDEKER, 2012).

There are different approaches of models for IC diesel engines, this work is designed to use the model highlighted in figure 9, assuming the thin spray model too. With respect to the liquid phase, spray calculations are based on a statistical method referred to as the discrete droplet method (DDM). Using the single zone model, we can obtain from ordinary differential equations – trajectory, momentum, heat and mass transfer of single droplets, the cylinder pressure, temperature and mass. The dual zone combustion model provides better results in face of single zone. Otherwise, multizone model use the temporal and spatial variations of temperature and concentration to permits calculations of exhausted gas composition.

> In multi-dimensional model, the instantaneous conservation time average equation of mass, momentum, energy has been taken into calculation. Stochastic as well as computational fluid dynamics (CFD) model are introduced to see the effect of gas phase turbulence in many parts has also been ignored. Cylinder is considered to be divided into number of zones and behavior of each individual zone being studied. […] In order to avoid the computational efficiency in case of multidimensional model, multizone model describe better the spray penetration with the help of empirical correlations instead of solving the full momentum equation. By the implementation of this method quasi-dimensional model provides the fastest and non-expensive means of generating the spatial information required to predict pollutant emission. […] Since the multizone model based on empirical correlations to study the spray evolution, the fidelity of the spray penetration model is extremely important for accuracy (KUMAR; CHAUHAN; VARUN, 2013, p. 520).

The droplets are tracking in a Lagrangian way through the computational grid used for solving the gas phase partial differential equations. Full two-way coupling between the gas and liquid phases is taken into account (AVL LIST, 2014)

Figure 9 - Classification of diesel engine model
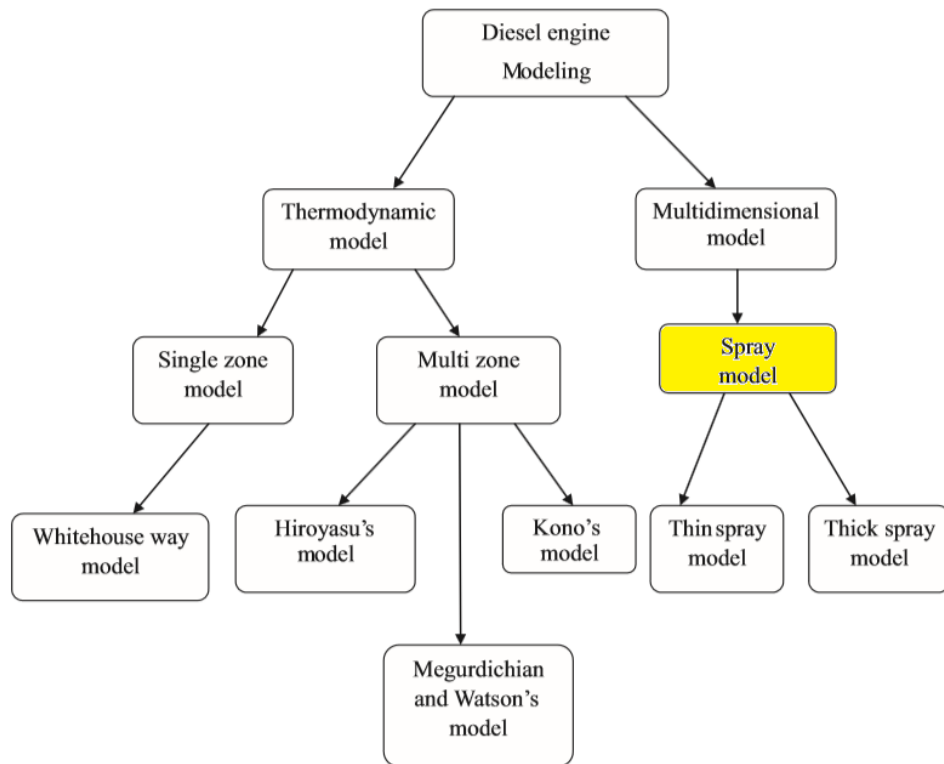
Font: Modified from Kumar, Chauhan and Varun, 2013.

Using multiphase modules, it is possible to simulate phenomena in various injection devices, including non-linear cavitation effects, taking into account the movement of the injection needle as an example (GRABOWSKI; PIETRYKOWSKI; WENDEKER, 2012).

3. METHODOLOGY

The scientific method is a set of processes that must be used in the investigations, following a line of reasoning for the research process (SILVA; MENEZES, 2005).

Using the Engine Simulation Environmental Diesel (ESE Diesel) we can configure, perform and analyze the injection and combustion process involved in diesel. As stated earlier, we will focus on the behavior of spray and cavitation evolving into de piston chamber.

With the standard example file Injection Nozzle: Diesel injector (910), we can create meshes and simulate fluid systems. Making a design of experiments (DOE), we can change model parameters like needle lift, initial pressure and initial temperature to see the difference between penetrations, pressure drops, velocities and erosion evolution.

DOE is a systematic method to determine the relationship between factors affecting a process and the output of that process. Can be defined to find cause and effect relationship (SUNDARARAJAN, 2018).

To gain more time, in our DOE, we will work with the variables that most influence the process – based on previous studies, reducing simulation time and we use the two-level factorial design, that is simple, versatile and can be used for many factors.

When using simulation tools for diesel combustion it is always necessary to perform an adjustment of the input data for the calculation models to fit the computed results to measurements or visualization data. This is an interactive and engine specific process, which makes it very difficult to develop a standardized adjustment strategy (AVL LIST, 2014).

3.1. Overview from injector nozzle initial example

Experiments are difficult to manage for injection conditions because involve small scales and high-speed flows, a computational approach seems to be a better way to understand what happens in everything that surrounds this phenomenon.

The documentation of AVL List said that, in order to take into account the impact of geometrical details on the highly transient nature of the cavitating injector flow, a

comprehensive multi-dimensional two-phase flow model is only able to provide the relevant information required as input for an internal combustion, IC, engine simulation.

The differential equation that governs trajectory and velocity of particle is a momentum as follow in equation 3.1. The m_d is the particle mass and the $\frac{du_{id}}{dt}$ is the acceleration (AVL LIST, 2014).

$$m_d \frac{du_{id}}{dt} = F_{idr} + F_{ig} + F_{ip} + F_{ivm} + F_{ib} \qquad 3.1$$

In this equation of momentum, the F_{idr} is the drag force, as in equation 3.2.

$$F_{idr} = D_p. u_{irel} \qquad 3.2$$

And the drag function is defined as 3.3.

$$D_p = \frac{1}{2}\rho_g. A_d. C_{D.|u_{rel}|} \qquad 3.3$$

C_D is the drag coefficient which generally is a function of the droplet Reynolds number Re_d and A_d is the cross-sectional area of the particle. The drag coefficient of a single sphere is used formulation from Schiller and Naumann as equation 3.4.

$$C_D = \begin{cases} \dfrac{24}{Re_d C_p}(1 + 0.15 Re_d^{0.687}) & Re_d < 10^3 \\ 0.44/C_p & Re_d \geq 10^3 \end{cases} \qquad 3.4$$

Reynolds number is given in 3.5, where μ_g is the domain fluid viscosity.

$$Re_d = \frac{\rho_g |u_{rel}| D_d}{\mu_g} \qquad 3.5$$

When using the drag on small particles the Cunningham correction factor C_p, equation 3.6, based on Knudsen number, equation 3.7, reduces the drag coefficient.

$$C_p = 1 + Kn_p(2.492 + 0.8e^{\frac{-1.74}{Kn_p}})$$

<div align="right">3.6</div>

$$Kn_p = \frac{\lambda}{D_d}$$

<div align="right">3.7</div>

The mean free path length λ in gaseous phase can be as equation 3.8. With the mean molecule diameter in gas phase $d_g = 2.8e^{-10}$m.

$$\lambda = \frac{k_b T_g}{\sqrt{2}\pi d_g^2 p_g}$$

<div align="right">3.8</div>

F_{ig} is a force including the effects of gravity and buoyancy, equation 3.9.

$$F_{ig} = V_p \cdot (\rho_p - \rho_g)g_i$$

<div align="right">3.9</div>

F_{ip} is the pressure force, given in equation 3.10.

$$F_{ip} = -V_p \cdot \nabla_p$$

<div align="right">3.10</div>

F_{ivm} is the virtual mass force taking into account acceleration / deceleration of the medium surrounding the drops, particles and bubbles. This is treated by increased inertia of the Lagrangian parcels (half mass of displaced fluid is added). The additional force term finally is the $\vec{F}_a$, equation 3.11, with the m_f denoting mass of displaced fluid, $\vec{v}$ parcel velocity and $\vec{u}$ ambient flow velocity. F_{ig} summarizes other external forces like the so-called virtual mass force, magnetic or electrostatic forces, Magnus force or others (AVL LIST, 2014).

$$\vec{F}_a = -\frac{1}{2}m_f\left(\frac{d\vec{v}}{d_t} - \frac{D\vec{v}}{D_t}\right)$$

<div align="right">3.11</div>

3.2   Injector nozzle: diesel injector

In this and follow chapters we going to explain how the standard example was build and its main characteristics. The recommended meshing procedure to get an injector mesh is to split the surface mesh into two parts, where each part will be meshed separately (AVL LIST,2014).

Due to the short time, a mesh study was not performed, this being the result of the mesh generation optimized by the software.

3.2.1   Meshing strategy

The geometry should be as in figure 10. It is recommended that the injector mesh be divided in two parts, the body and the injector hole.

Figure 10 - Diesel injector mesh divided into body and hole injector
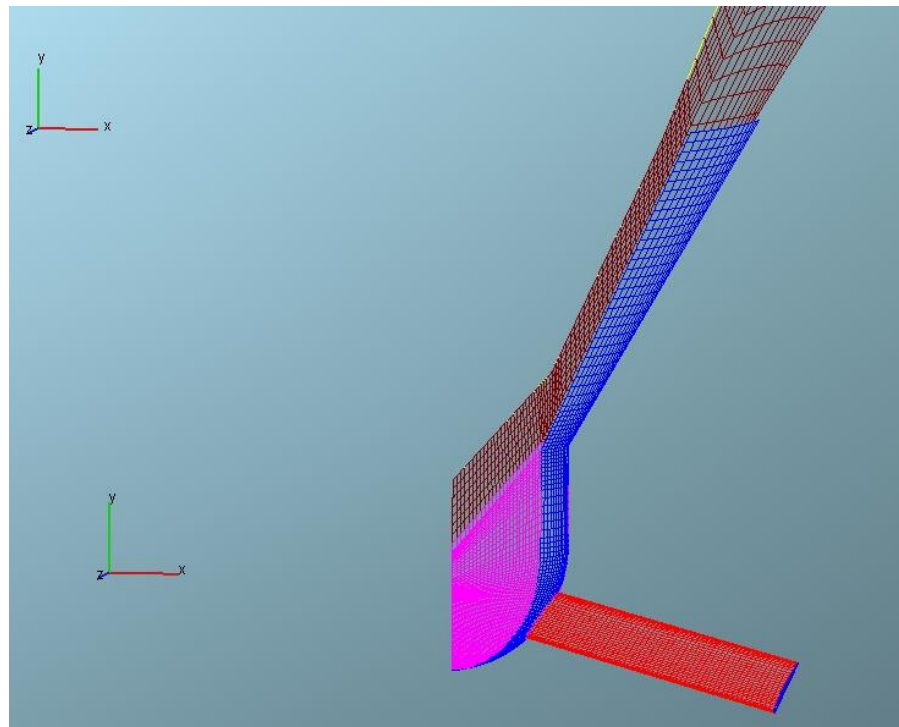
Font: Autor, 2018.

This arrangement is necessary because we need more details on hole, and the interface between both need to be the sort of that to better describe the phenomena. A necessary care is in the interaction of bodies, it is important that the meshes are connected and are proportional, resulting in a harmonious transition as in figure 11 that follows.
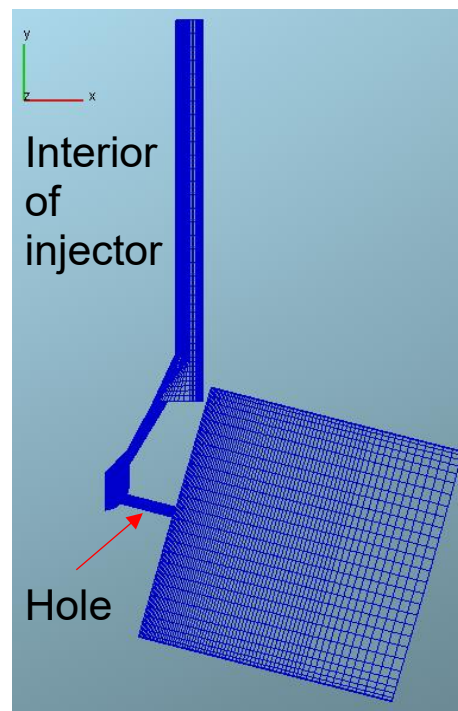
Figure 11 - Two bodies mesh interface
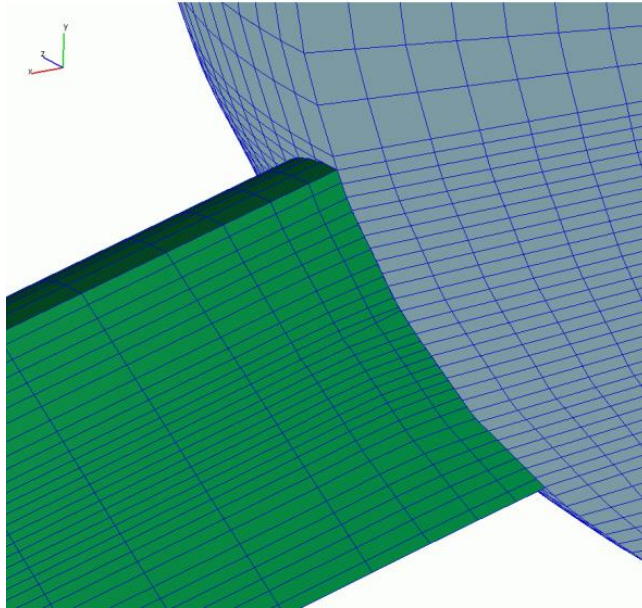
Font: AVL LIST, 2014.

3.2.2 Discharge volume

To get a more realistic condition we need a cylindrical discharge volume- figure 12, which is used for physical initialization and boundary condition (AVL List, 2014).
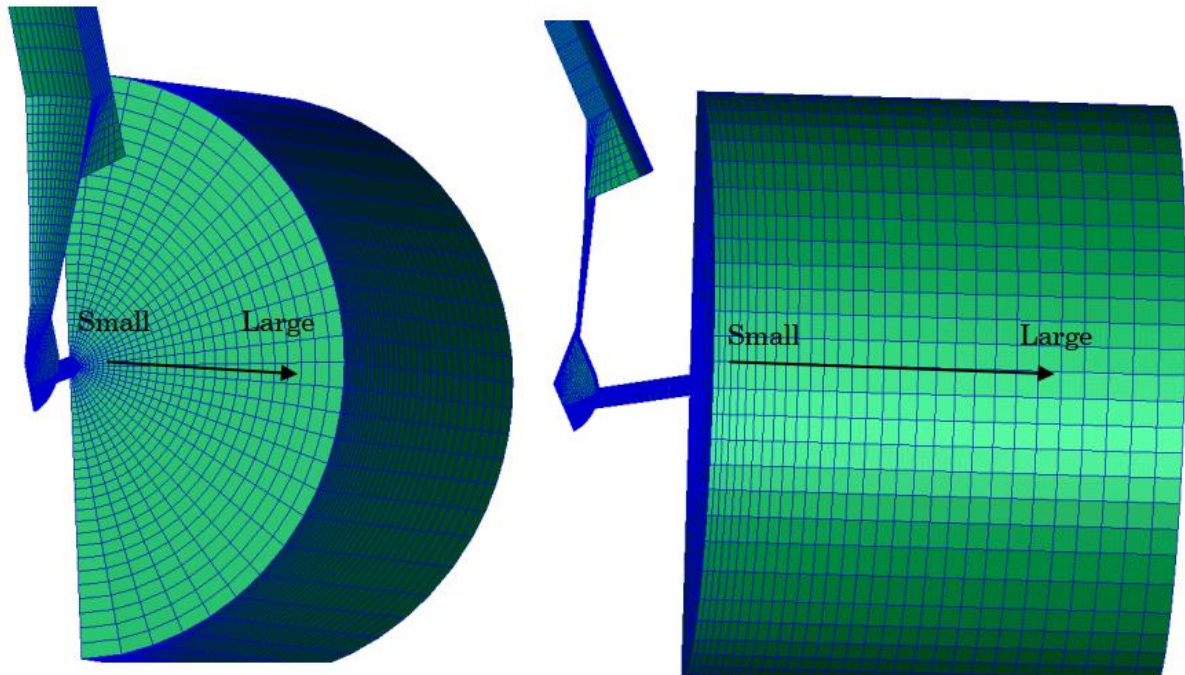
Figure 12 - Discharge volume

Font: Autor, 2018.

We need take the same care to better describe the effect of injection, the cell distributions on the discharge volume need to be with a compression rate in a way that the cell layer are smaller near the injector outlet like in figure 13.

Figure 13 - Detail of mesh near injector outlet

Font: AVL LIST, 2014.

3.2.3   Conditions of the numerical experiment

Boundary specify the physical properties of the faces on volume mesh (AVL List, 2014). This condition is the same that we expect to find in actual numerical experiments. As the number of phases has been increased to three, certain boundary conditions will require specification for 3 phases, but when we use a pressure boundary, only the first is used, because in numerical model MP, the multi-fluid model share the same pressure field by all phases (AVL List, 2014). We can see the inlet boundary conditions below in table 1.

Table 1 - Inlet boundary conditions

Description	Values from AVL example	Unit
Phase 1:		
Total Pressure	2e008	Pa
Fixed Temperature	293.15	K
Fixed Volume Fraction	0.999998	-
Phase 2:		
Fixed Temperature	293.15	K
Fixed Volume Fraction	1e-006	-
Phase 3:		
Fixed Temperature	293.15	K
Fixed Volume Fraction	1e-006	-
For all three phases:		
Turbulent Kinect energy	0.1	kg.m²/s²
Turbulent length scale	0.001	m

Font: Autor, 2018.

In multiphase simulations, the sum of volume fractions of all phases must be 1. And the outlet boundary is in table 2. The phases are fuel liquid for 1, phase 2 to fuel vapor and 3 to air.

Table 2 - Outlet boundary conditions

Description	Phase 1	Phase 2	Phase 3	Unit
Pressure	500	-	-	kPa
Fixed volume fraction	1e-006	1e-006	0.999998	

Font: Autor, 2018.

3.2.4 Initial conditions

The parameters used in the first conditions are described in table 3.

Table 3 - Initial condition for base model

Parameter	Phase 1	Phase 2	Phase 3	Unit
Pressure	2.0e008	-	-	Pa
Temperature	293,15	293,15	293,15	K
Velocity (u, v, w)	0	0	0	m/s
Volume Fraction	0.999998	1e-006	1e-006	-

Font: Autor, 2018.

4. COMPARATION

The diesel injector example was used as started point to the others numerical experiments. The comparison strategy was based on initial pressure change, initial temperature and needle lift. For temperature and pressure, we varied half on the value up and down, from 283,15K to 313,15K and 100KPa to 200KPa, respectively. The needle opening treatment should have an increase of 0,25% greater and 0,25% smaller compared to the initial opening, causing a greater or lesser closure of the passage of fluid through the nozzle hole.

Using the pressure as 100 MPa, we can see in figure 14 that near the injection hole is a value below the initial number. This is expected because with lower inlet pressure, the forces involved are smaller, resulting in lower losses due to geometry difference.

Figure 14 - Comparison between inlet injector absolute pressure at 200 MPa (left) and 100 MPa (right)
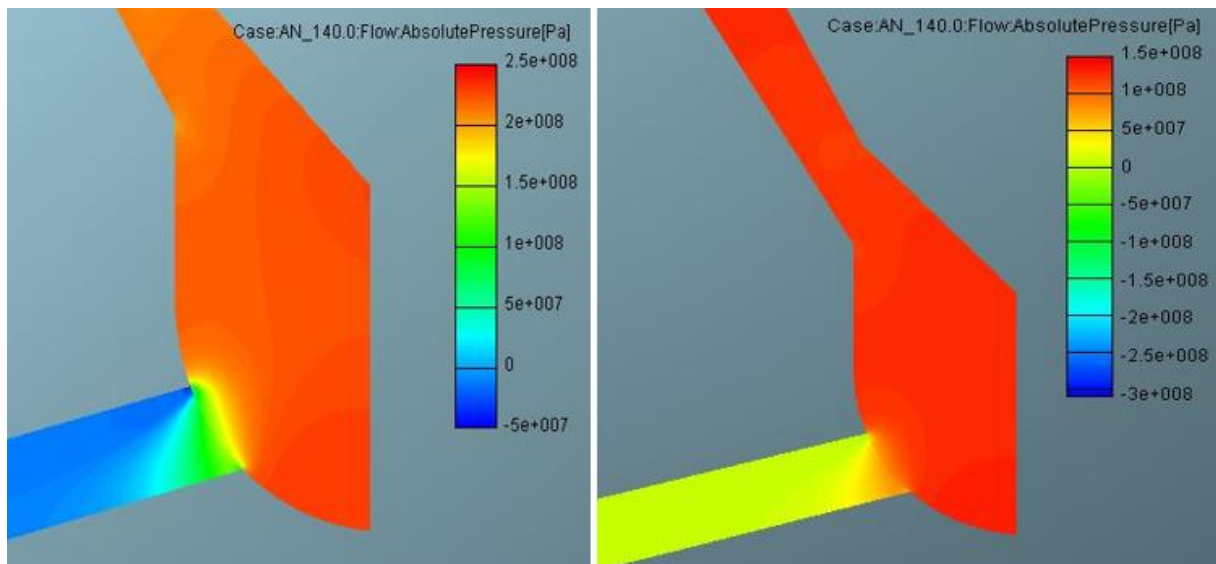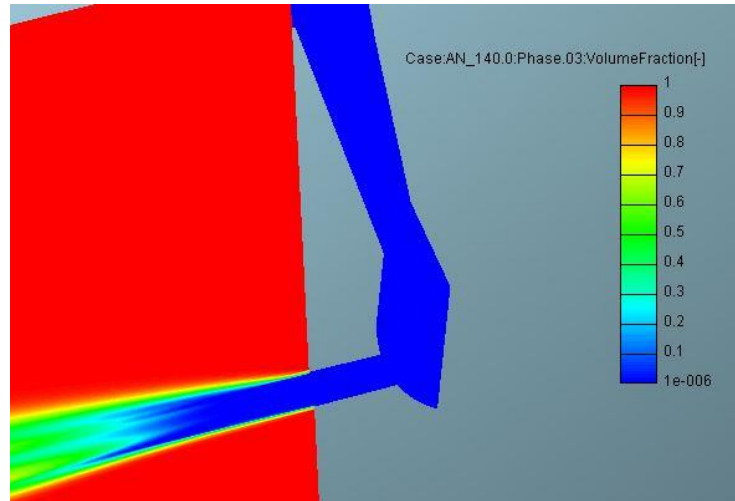

Font: Autor, 2018.

In this comparison we can assume that lower inlet pressure may help to minimize the effects of cavitation, but when we do this, we are damaging the combustion, low pressures prevent the fuel droplets from being all broken and the atomization inefficient inside the piston bowl.

The velocity involved in this case follows what was previously achieved, smaller velocities in the hole, having a reduction of 39% in liquid velocity, 15% in gas velocity and 15% in vapor velocity. This is representative in figure 15 that show the volume fraction on vapor phase, which is more pronounced than the initial model, show in figure 16.

Figure 15 - Volume fraction of vapor phase in injector hole with 200 MPa

Font: Autor, 2018.

Figure 16 - Volume fraction of vapor phase in injector hole with half of initial pressure (100MPa)
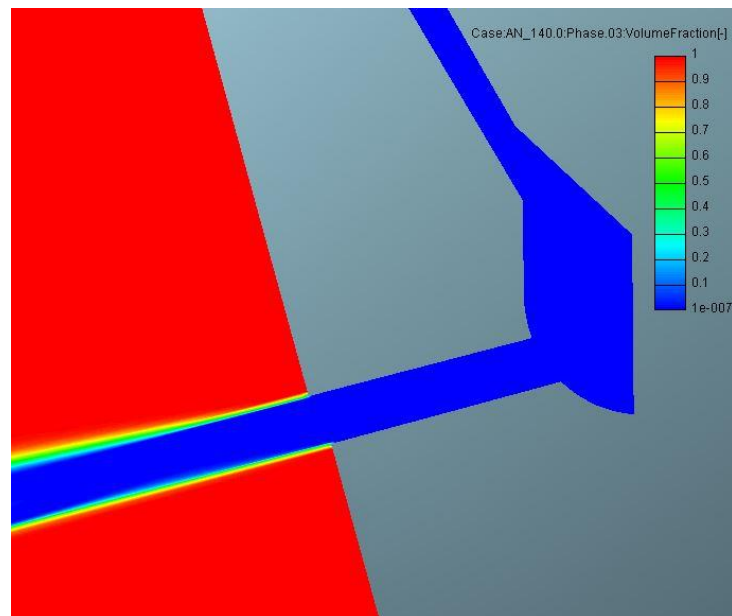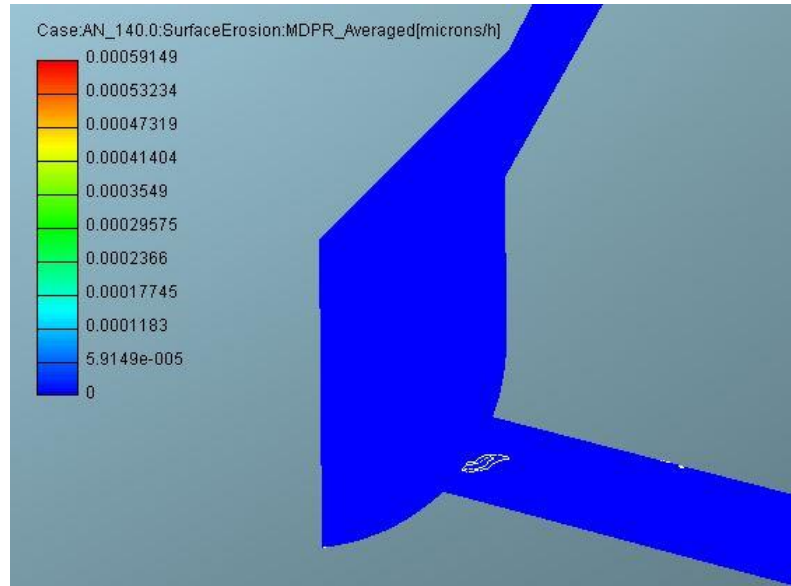

Font: Autor, 2018.

This vapor is important to combustion because this help the ambient to achieve a better mixture to completed ignite. Figure 17 represents that there are more liquid and gas molecules inside the piston chamber.

With the data obtained by the study, we can also see that the mean erosion, in micros/h, is higher where the velocity and pressure are higher, resulting from the larger forces involved, figure 18. This is acceptable because higher velocities induce greater pressure drops, facilitating the effect of dynamic cavitation.

Figure 17 -Mean depth of penetration rate of surface erosion

Font: Autor, 2018.

This variable indicates the probability for erosion due to cavitation. The values represent the amount of material removal per unit time during the steady part of the erosion process (AVL LIST, 2014).

The figure 19 show the MDPR with half pressure and the mean difference is in the scale, the position is similar.
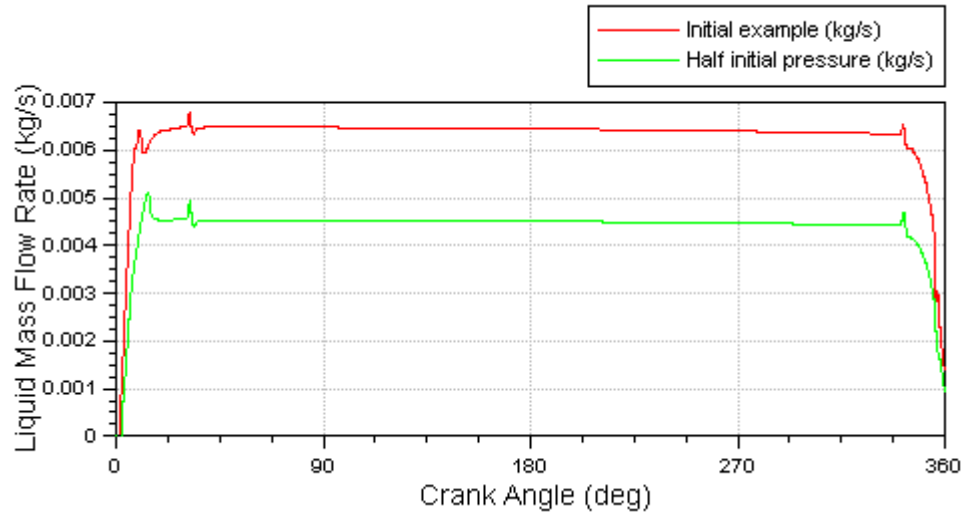
Figure 18 - Mean depth of penetration rate with half initial pressure (100MPa)
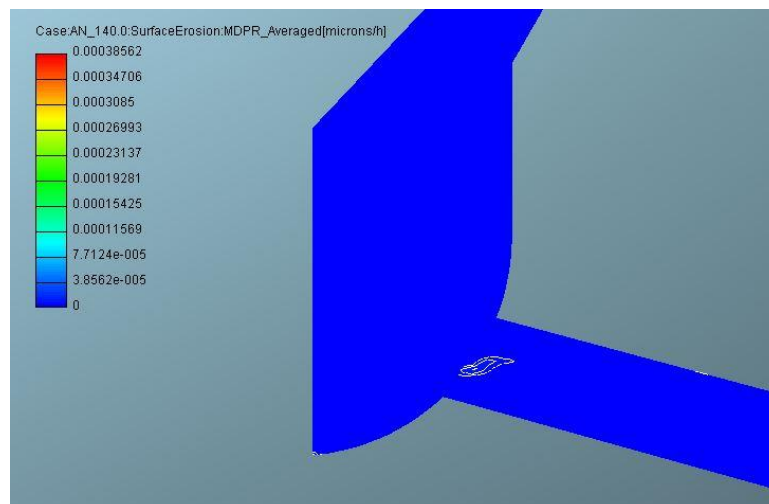
Font: Autor, 2018.

We can analyze that the liquid mass flow rate pass through injector hole have the same characteristic on both cases, but the difference lies in the scale and is about 30%.
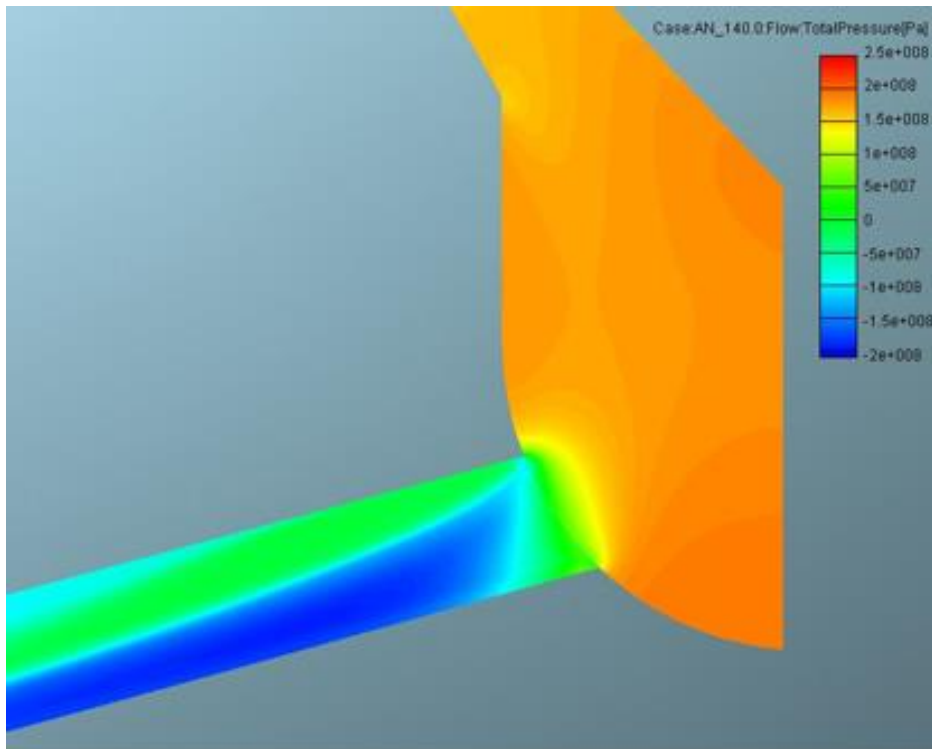
Figure 19 - Comparison between initial and first case of liquid mass flow rate
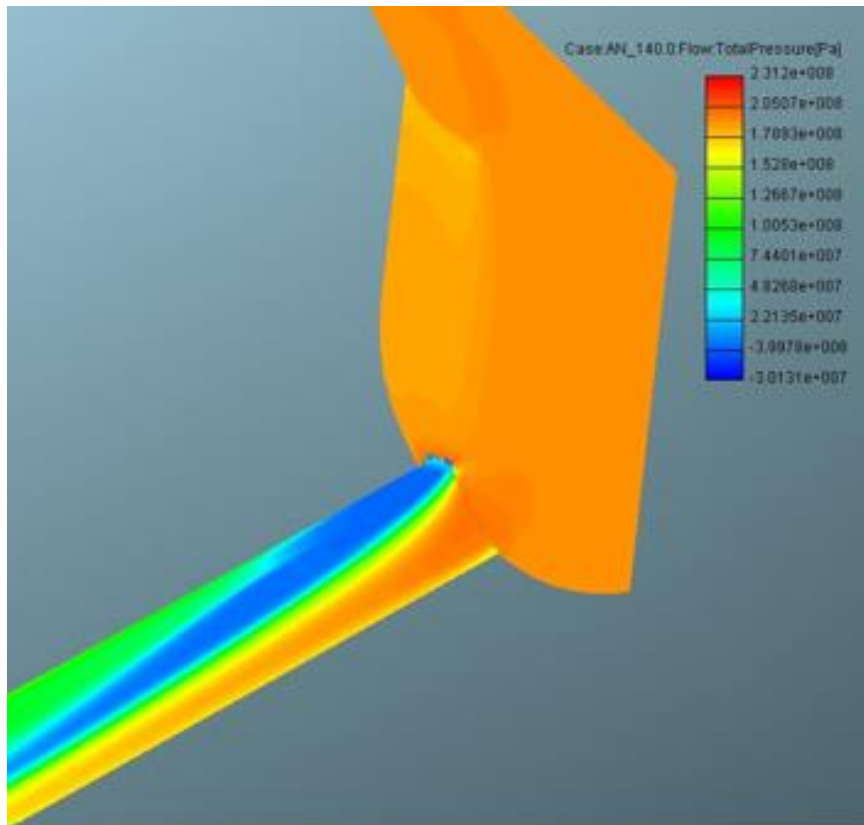

Font: Autor, 2018.

Moving in the analyzes, we now use the inlet temperature as 313,15 K. In this case the comparison between figure 20 and 21 show that now the flow profile changed.

Figure 20 - Total pressure of initial case

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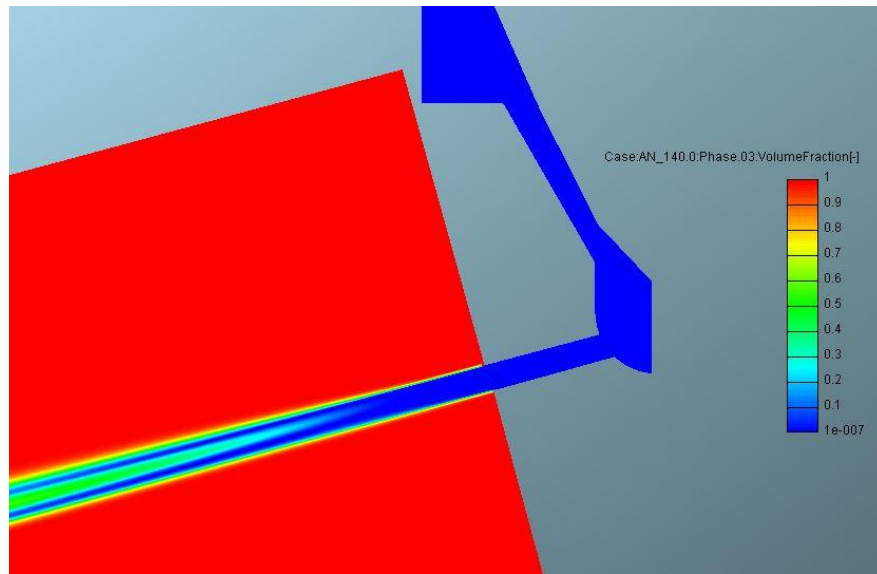Figure 21 - Experiment with doubled temperature inlet (313,15K)

Font: Autor, 2018.

The averaged pressure of all system has increased with this temperature rise. As we did not modify the inlet flow ratio, the mass was maintained, but the higher temperature provided an increase in the pressure.

Comparing the figure 22 with figure 15, we can see that the new temperature influences in volume of vapor penetration and development. The new temperature disturbed the amount of vapor who gets inside the piston because the phase of gas has increased in this case, see in figure 23.

Figure 22 - Volume fraction of vapor with doubled initial temperature (313,15K)
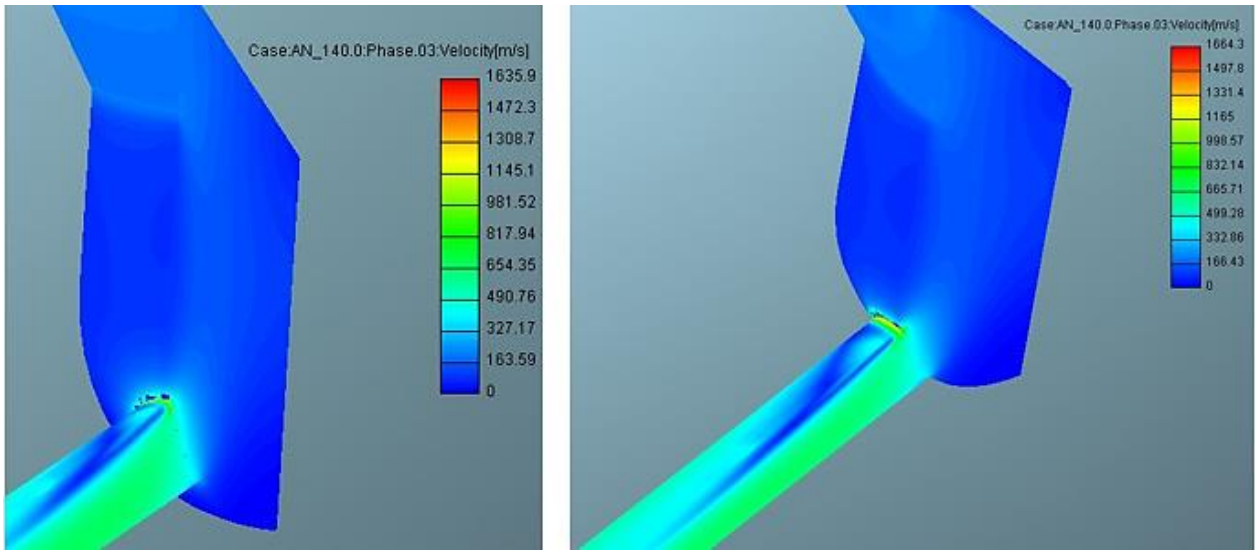

Font: Autor, 2018.

Figure 23 - Volume fraction of gas on injector hole with doubled temperature (313,15K)
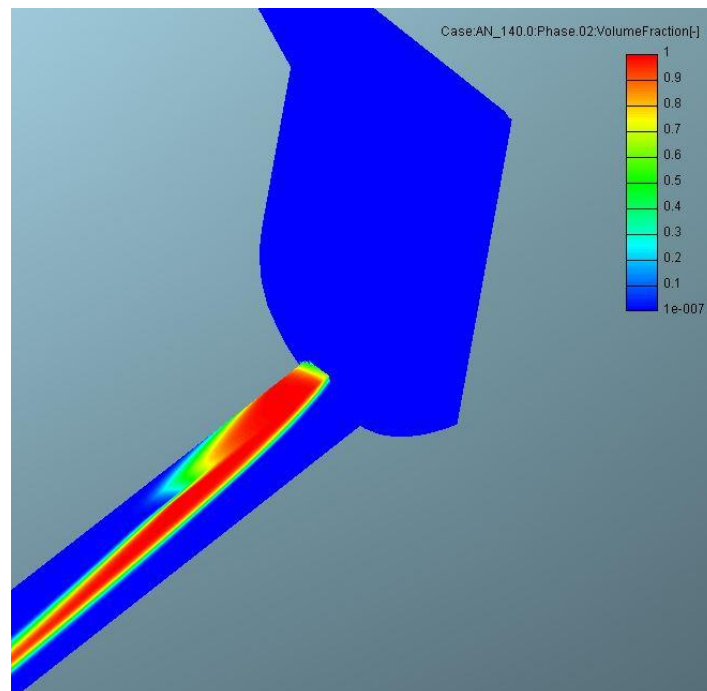
Font: Autor, 2018.

With the new pressure, when the fuel pass through the hole, the average velocity has a discrete increase, figure 24.
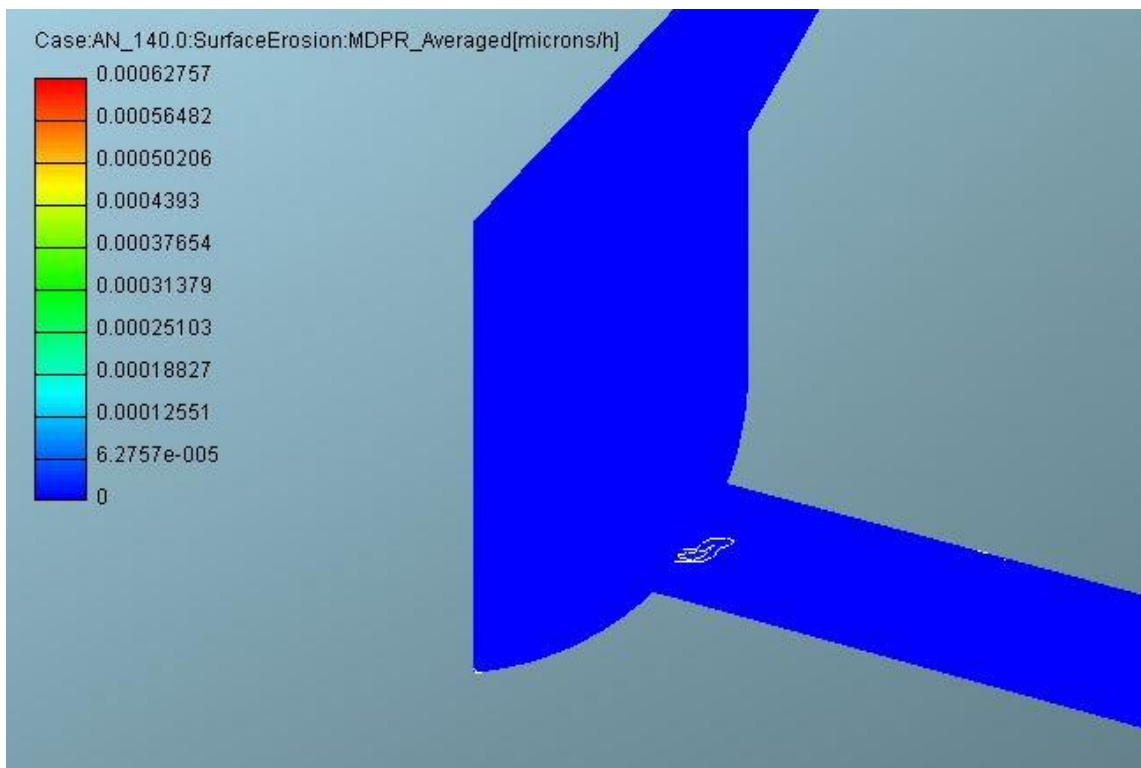
Figure 24 - Comparison of vapor phase velocity at the injector hole for initial case
(left) and doubled temperature, 313,15K (right)

Font: Autor, 2018.

This new case provides a worse scenario to cavitation, facilitating this phenomenon,
as show in figure 25 below.

Figure 25 - Surface erosion with doubled temperature (313,15K)

Fonte: Autor, 2018.

The aspect and shape are the same in all three cases, but the value of erosion is much higher in this case due to the velocity, implying a pressure drop and consequently favoring cavitation.

The next comparison is about needle lift, figure 26 shows a lifting comparison between the ¼ % plus and ¼ % less, respectively left and right. With this comparison we notice that with other conditions maintained the total pressure do not have an expressive difference.

Figure 26 - Comparison of absolute pressure between 1/4 % plus of lifting (left) and 1/4 % less lifting (right)
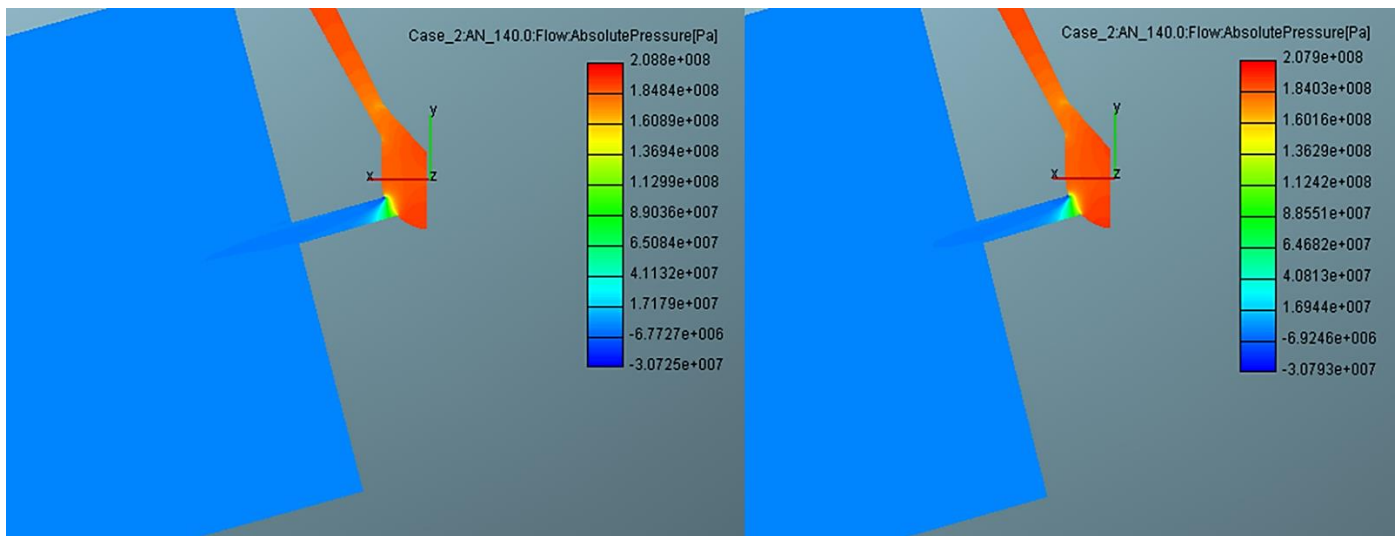
Font: Autor, 2018.

A good comparison of indicators is surface erosion which although appears in figures 27 and 28, the respective scales have very large orders of magnitude, showing that the restricted flow has an enormous influence on the geometric cavitation.

Figure 27 - Surface erosion of 1/4 % greater than initial condition
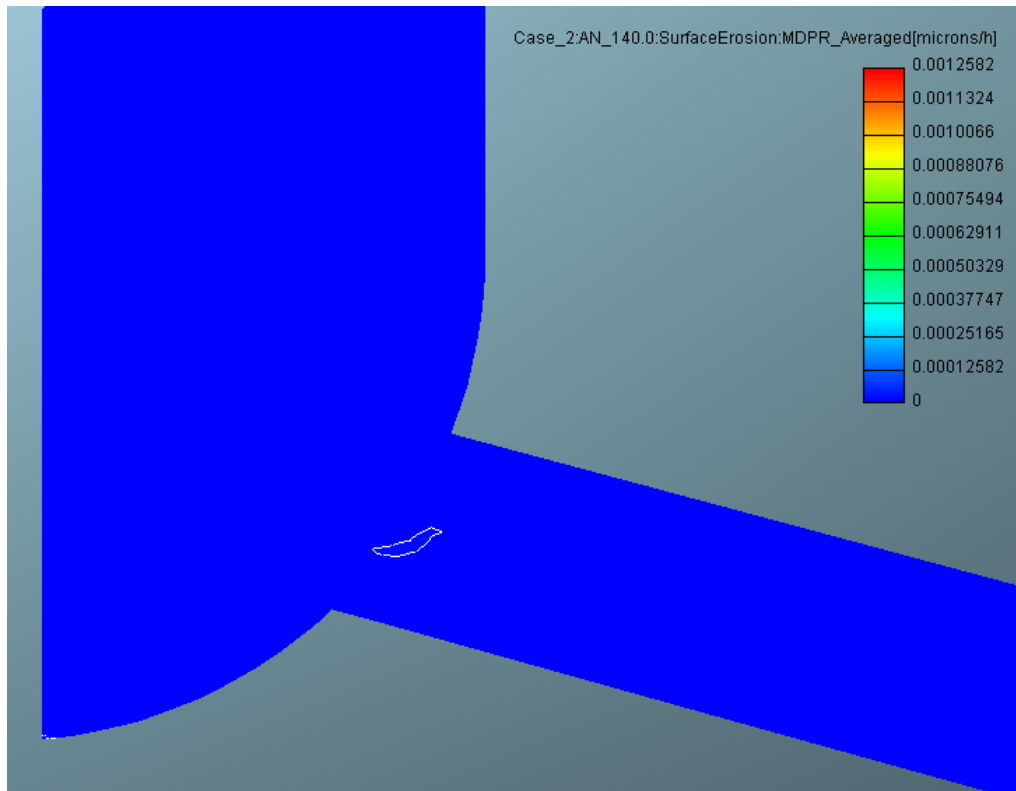
Font: Autor, 2018.

Figure 28 - Surface erosion of 1/4 % less than initial condition

Font: Autor, 2018.

Table 04 shows the interaction of the experiments. This table is based on the initial conditions, so line 9 for example, shows an experiment with half the initial pressure – 100KPa, and opening needle ¼ % larger than the base.
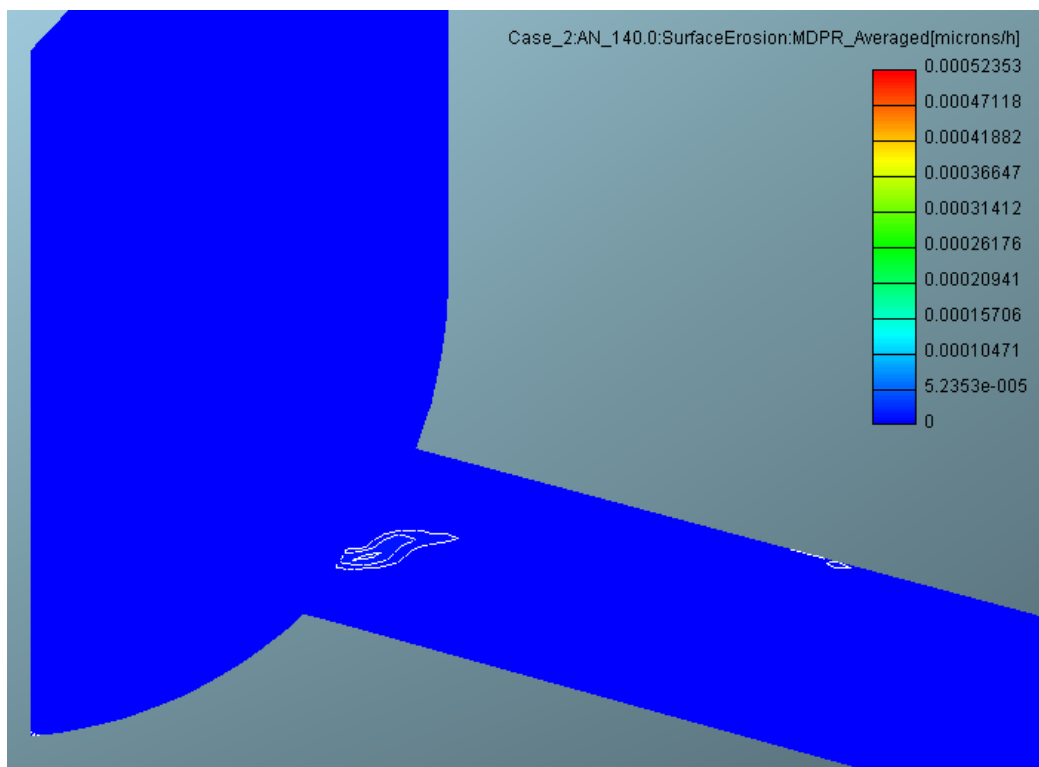
Table 4 - Design of experiments (DOE)

Position	Pressure	Temperature	Needle lift
1	0,5		
2		2	
3	2		
4		0,5	
5			0,25
6			-0,25
7		2	-0,25
8		0,5	0,25
9	0,5		0,25
10	2		-0,25
11	0,5	0,5	0,25
12	2	2	-0,25
13	2	2	0,25
14	2	0,5	-0,25
15	2	0,5	0,25

Font: Autor, 2018.

When we begin to analyze the images of the results, because of the measured variables, the changes in the characteristics are not visually perceptible. However, when analyzing the scales and the order of magnitude we can see how the numerical experiments are differentiated.

The development of pressure throughout the experiments followed a pattern where the highest pressure is in the injector body, passing at the intersection of the injector hole with SAC volume we have a pressure in the order of 208 MPa. It may be noted that for increasingly higher pressures, absolute pressure values tend to increase.

From the variations performed, the data that had the greatest impact were the speed of development of the vapor and erosion in the injector hole.

We can see in figure 29, 30 and 31 the difference in phase velocity of numerical experiment 13, 14 and 15, respectively.

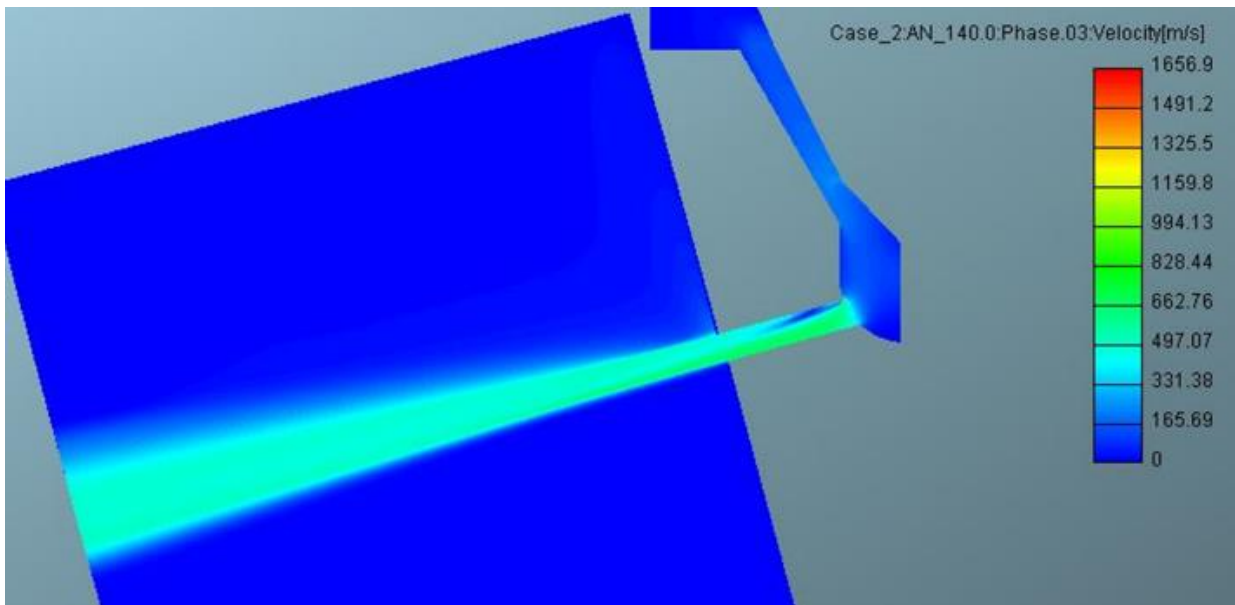The development of the jet is somewhat similar, but the values involved differ. In experiment 13, with high pressure and high temperature and higher rise of needle, we have a more gas phase and decelerated in the final parts of the injected plume.

Figure 29 - Vapor velocity of experiment 13
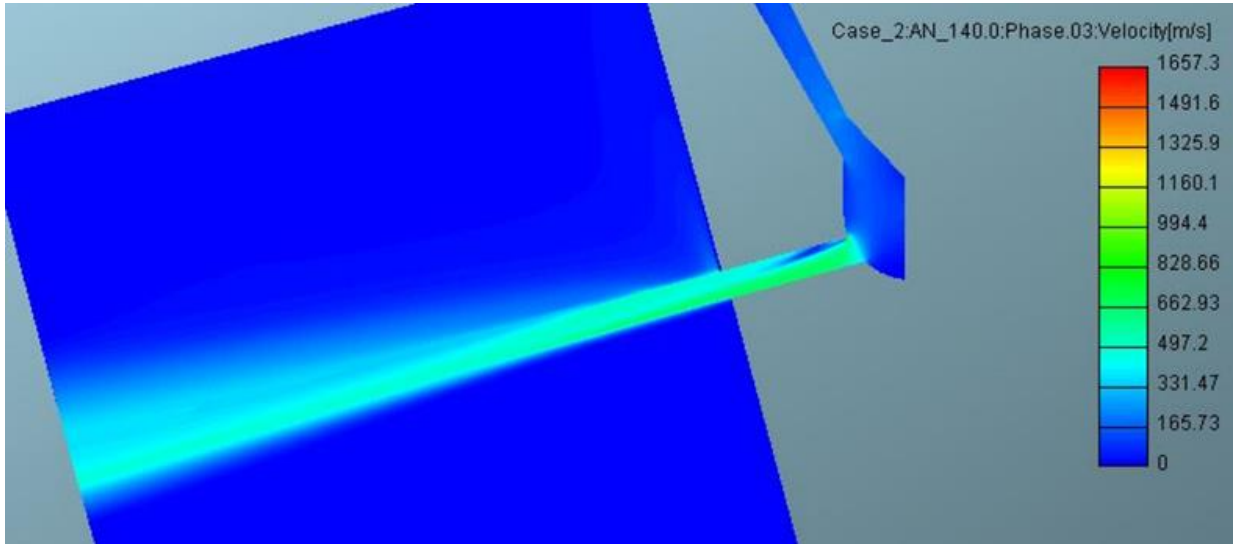
Font: Autor, 2018.

Figure 30 - Vapor velocity of experiment 14

Font: Autor, 2018.

Figure 31 - Vapor velocity of experiment 15
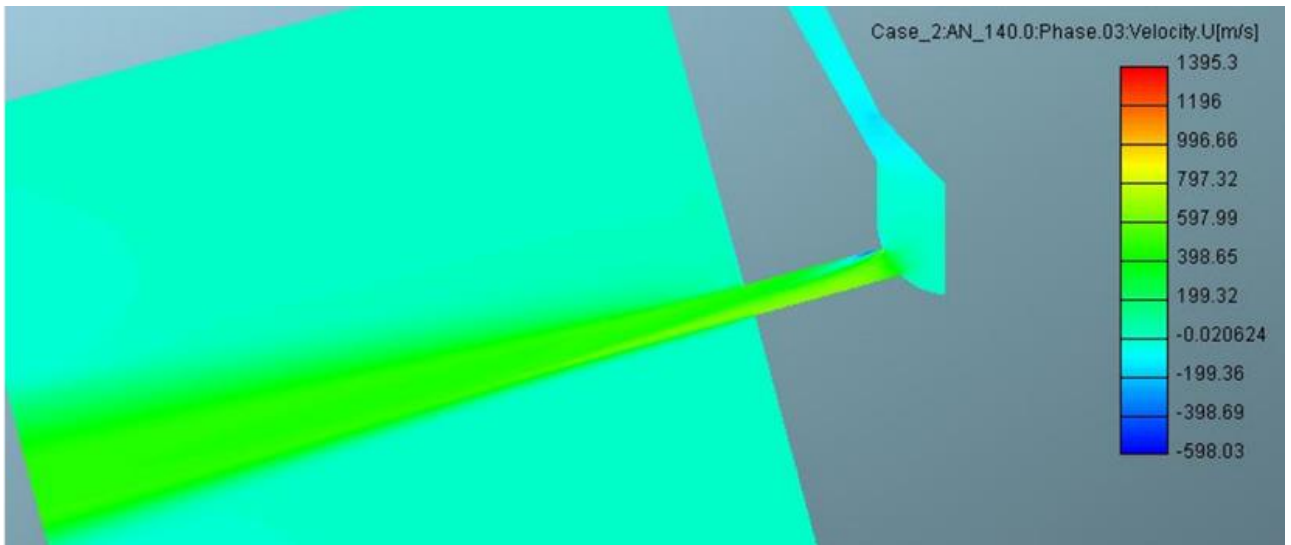
Experiment 14 follows the same pattern as the others, with little final variation, but what we established in experiment 15 is different. The reduced temperature and the higher rise caused slower vapor phase velocity development than the other experiments, with a central velocity of 20% less.

The main part about erosion is that the higher the lift associated with higher pressures as in figure 32, the magnitude reaches levels of 10 times the final value.

Figure 32 - Experiment 15, surface erosion averaged
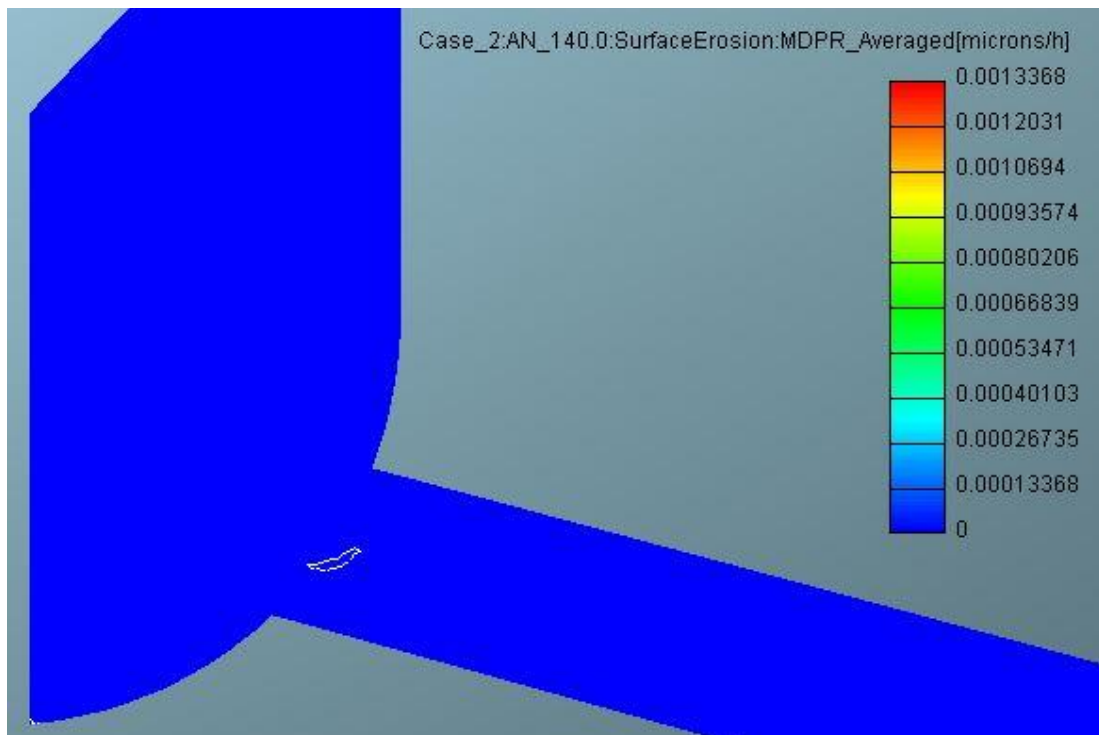
The figure 33 shows that difference. This was expected because with higher leakage, high pressure, the hole geometry would have a great influence on cavitation, generating a mean of high erosion.

Figure 33 - Experiment 14, surface erosion on injector hole
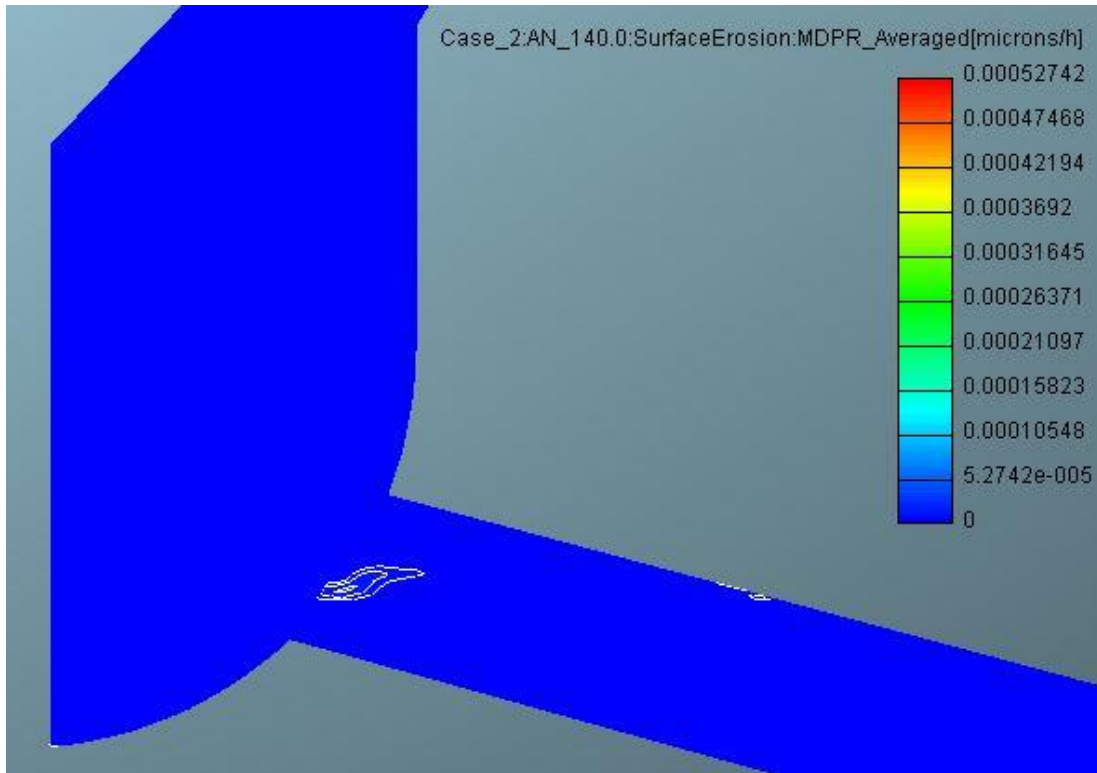
Font: Autor, 2018.

Situations like this in a truck in the routine generate sharp wear and flaws in pollution control systems.

5. CONCLUSION

The objective of this study was to understand how initial parameters influence the fuel injection phenomenon, how the effects were affected and how they behaved in terms of cavitation.

When we analyze the data obtained by the numerical experiments, high pressures and greater leakage have great influence on the injection and especially on cavitation. The higher velocities involved aid in the effect of pressure drop in the injection hole, added to the geometric effect of the flow failure between the SAC and the channel, resulting in an environment conductive to cavitation due to a marked pressure drop.

This tendency to cavitation develops other problems, such as the lower energy efficiency of the system, since the mixture of air-fuel is impaired when it does not reach an ideal penetration and a compatible jet aperture. As a result of these effects, the pollutants at the combustion outlet are expected to be higher than under normal operating conditions. The characteristics of fuel results in higher emissions of pollutants, particularly the formation of soot.

It is possible to notice that the manufacturers adopt the posture of increasing the injection pressure, reducing the amount of fuel injection and in more injections phases, tending to eliminate the effects of the failure of the mixture, but increasing the tendency to cavitation and erosion of the system. The balance between temperature and pressure is delicate in diesel engines, for that reason understand phenomena such as cavitation, which can alter the geometry of the nozzle and thus damage all the fuel injection, is of extreme importance for the industry in general, aiming reducing pollutants and increasing efficiency.

Understanding the mechanisms involved in the combustion phenomenon is important, given its great use, especially in the road transport part. Means that help us reduce pollutants and improve the environmental condition for all are increasingly exploited and valued.

As a continuation of the work, some suggestions of topics follow:

- Variation of geometric parameters associated with increase and decrease of temperature and pressure;

- Implement a stationary diesel engine and perform the tests for validation of results;
- Perhaps integrate the Boost and Fire modules with the same parameters, evaluate mechanisms such as EGR – exhausted gas recirculation.

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