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Turbulent Jet Ignition (TJI) Combustion Modelling in GT-Suite

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Introduction

The purpose of this thesis work is to develop a methodology that allows to model a Turbulent Jet Ignition Predictive Combustion through the usage of the latest version of the software GT-ISE (v2020). The work is developed in collaboration with Maserati S.p.A. (located in Modena - MO) and Powertech Engineering S.r.l. (Torino - TO).

Turbulent Jet Ignition (TJI) is an advanced prechamber initiated combustion system for an otherwise standard Spark-Ignition engine found in current on-road vehicles. Turbulent Jet Ignition combustion systems produce multiple and distributed ignition sites, enabling very fast burn rates and allowing for increased levels of dilution when compared to conventional Spark-Ignition combustion. The high speed of combustion can be exploited to get significant benefits in terms of fuel consumption and pollutant emissions, making this kind of technology very promising for the development of internal combustion engines.

Due to the importance of these systems, a methodological study aimed to model the Jet Ignition combustion is necessary. A fully predictive combustion model would allow to evaluate the TJI combustion with respect to some engine design parameter variations, like the prechamber geometry or the compression ratio, or operating parameters as the spark timing or VVA strategies. This study was carried out on a turbocharged SI-engine, which was developed by Maserati company, using the GT-SUITE environment. GT-SUITE is the leading engine and vehicle simulation tool used by engine makers and suppliers. It is suitable for analyses of a wide range of issues related to vehicle and engine performance.

A methodology to be followed has been built. After an analysis of the experimental data, a simplified model was set up, in order to reduce the computational time. Later, the prechamber system was modeled. The v2020 of GT-ISE showed to be
unable to properly model a predictive combustion in the prechamber, while the Jet Ignition calibration turned out to be more robust if conducted with the aim of minimizing a user-defined error calculated on the main chamber pressure. Despite this, the degree of correlation achieved on the full load curve does not meet the expectations, but there is room for improvement, expected for the v2021.

This thesis work is structured as follows: in chapter 1, a brief overview of the Turbulent Jet Ignition combustion technology is provided. Then, a description of the main organized motions in the main chamber is given in chapter 2. Once understood the physics of the phenomena to be modeled, a wide overview of the tools provided by the software developer is given in chapter 3. In chapter 4, a brief description of the engine considered and the operating points tested are supplied, while the approach followed is depicted in chapter 5. The results of the Jet Ignition combustion calibration are shown in chapter 6. The methodology built and the main conclusions are summarized in chapters 7 and 8.
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List of Abbreviations

1D-CFD - 1-Dimensional Computational Fluid Dynamics
3D-CFD - 3-Dimensional Computational Fluid Dynamics
ANHRR - Apparent Net Heat Release Rate
CA - Crank Angle
DIPulse - Direct-Injection Diesel Multi-Pulse Model
FL - Full Load
GA - Genetic Algorithm
HRBC - Heat Release Based Calibration
HT - Heat Transfer
MC - Main Chamber
MFB50 - Mass of Fuel Burned = 50%
PC - PreChamber
PDI - Port Direct Injection
PL - Partial Load
RMSE - Root Mean Square Error
SI - Spark Ignition
SITurb - Spark-Ignition Turbulent Flame Model
TDCF - Top Dead Center of Firing
TJI - Turbulent Jet Ignition
TKE - Turbulent Kinetic Energy
TPA - Three Pressure Analysis
WG - Waste-Gate valve
Chapter 1

Turbulent Jet Ignition Combustion

The use of a prechamber in Spark-Ignition engines was already analyzed in the last part of the 20th century. Several variants of the prechamber combustion system have been developed, primarily to increase the engine’s dilution tolerance. In general, when using a prechamber, two ignition mechanisms are possible, depending on the prechamber nozzles diameter: Jet Ignition and flame ignition. In every case, the combustion is initialized burning a small quantity of stoichiometric or near-stoichiometric fuel/air mixture in the prechamber. As the prechamber pressure goes up, the jet accelerates, reaches a maximum, and then starts decelerating.

If the prechamber nozzle diameter is sufficiently small, the prechamber flame is quenched when passing through the orifice, due to the high stretch rate and heat losses through the walls, generating jets of hot combustion products. Very few intermediate species and radicals are present in the jets as demonstrated by high-speed OH chemiluminescence imaging, which can reach very high velocities and, in some cases, reach the speed of sound ([1]). In the main chamber, the jet does not provide a point-like trigger but the combustion tends to start from almost the entire jet surface, leading to a remarkable reduction of the duration of the combustion process. The main chamber ignition does not take place during the jet acceleration process. Rather, it occurs when the jet is decelerating.

In comparison with a traditional SI engine, the combustion is faster (Figure 1.1); this allows the burning of very diluted mixtures, which would have difficulties to burn in adequate times in a traditional engine.
Burning a lean mixture leads to a lower combustion temperature, in comparison with a stoichiometric mixture, and to a subsequent reduction of NOx emissions (whose formation is strongly dependent on the combustion temperature due to the Zeldovich mechanism).

If the nozzle diameter is large enough, the flame front generated into the prechamber does not quench passing through the orifices (flame ignition); this case resulted to have less advantages in comparison with a Turbulent Jet Ignition combustion.

Turbulent Jet Ignition combustion engines can be characterized by active or passive systems. In a passive system, due to the lack of an injector in the prechamber, the prechamber mixture equivalence ratio is not decoupled from the main chamber mixture equivalence ratio (a part of the main chamber fresh charge simply flows into the prechamber during the compression stroke).

**Figure 1.1:** Traditional Spark-Ignition Burn rate VS Jet Ignition combustion Burn rate

**Figure 1.2:** A passive system scheme. The PC lacks a dedicated injector
In these systems, stoichiometric or near stoichiometric mixtures conditions are typical for both chambers, due to the difficulties that occur when a lean mixture has to be ignited in the prechamber. Despite this, the higher speed of the combustion process leads to a series of advantages: a faster combustion is nearer to the ideal case (of isochore combustion), increasing thermal efficiency. Moreover, since the end gas is exposed to high pressures and temperature for less time, knock risks are mitigated and, even if a very lean mixture cannot be used, the faster combustion enables to increase EGR levels, with remarkable benefits on fuel consumption thanks to the dethrottling at partial loads.

In active systems, a further injector seeds the prechamber (Figure 1.3); this allows to realize different equivalence ratios between the main chamber and the prechamber.

**Figure 1.3:** Active system. The prechamber is fed by a dedicated injector

This advantage could be used to have a lean mixture in the main chamber, while maintaining a rich or near-stoichiometric mixture in the prechamber, a condition which enhances the start of combustion. An active system provides a higher flexibility (lean combustion is allowed), but requires a more complex hardware because the prechamber has to host an additional injector and the after-treatment system has to be able to manage NOx emissions in an oxidant environment, due to the main
chamber lean mixture. For this reason, the passive prechamber system seems to be more promising, providing a more convenient cost-benefit ratio in comparison with an active system.

Thanks to the possibility of a fast combustion, Turbulent Jet Ignition systems can also be coupled with Miller cycle strategies, which have the disadvantage of a limited turbulence close to the TDC firing.

Maserati Nettuno (Figure 1.4) is the first road car engine equipped with a prechamber system. In this engine, a further spark plug is located in the main chamber to allow an adequate mixture ignition in operating conditions in which the Turbulent Jet Ignition could be critical (i.e. low loads, in which the prechamber burned gas may not be adequately replaced by the fresh charge before the start of combustion).

Figure 1.4: Maserati Nettuno engine
Chapter 2

Charge Turbulent Motion

In Spark-Ignition engines the turbulent flow field inside the cylinder plays a major role. Multiple phenomena, such as early flame kernel development, flame propagation and gas-to-wall heat transfer, that occur during the high-pressure part of the engine cycle, are influenced by in-cylinder turbulence. Turbulence inside the cylinder is primarily generated from the flows that occur during the intake process, the high-velocity injection sprays and the destruction of macro-scale motions produced by tumbling and swirling structures close to the top dead center.

2.1 Charge Organized Motions

A short description of the in-cylinder charge organized motion can be useful to better understand the flow velocity field evolution during the engine operation. In particular, three main organized motions of the charge can be distinguished: tumble, swirl and squish.

The swirl motion is characterized by a rotation around the axis of the cylinder. This kind of motion, unlike the tumble, tends to preserve itself during the compression stroke. Especially in Diesel engines, the swirl is used to promote a more rapid mixing between the inducted air charge and the injected fuel, enhancing combustion. Usually tumble is preferred to swirl in SI engines, because the latter produces a limited degradation of the organized motion in smaller scale structures.

Swirl is created by bringing the intake flow into the cylinder with an initial angular momentum. In general, two approaches can be followed to induce the swirl motion:
either discharging the flow into the cylinder tangentially toward the cylinder wall, or generating it in the inlet port, forcing the flow to rotate about the valve axis before its entrance in the cylinder. A further description of these methods is provided in [2].

Figure 2.1: Swirl motion

The swirl motion is usually quantified on a test bench through the usage of a light paddle wheel, pivoted on the cylinder centerline (with low friction bearings), mounted between 1 and 1.5 bore diameters down the cylinder. The paddle wheel diameter is close to the cylinder bore. The rotation rate of the paddle wheel is used as a measure of the air swirl, while a torquemeter can be used to determine the swirl torque T. A representation of this experimental setup is provided in Figure 2.2.
The literature provides several parameters to describe the swirl motion; one of the simplest is:

$$C_s = \frac{2T}{\dot{m}v_0B} \tag{2.1}$$

$m$ is the air mass flow rate, $B$ the cylinder bore and $v_0$ the characteristic velocity of the flow, which depends on the pressure drop across the valve [2].

Tumble motion defines a macrovortex which rotates around an axis perpendicular to the axis of the cylinder. This macrovortex is generated during the intake stroke, and it is obliged to spin around an increasingly small radius during the compression stroke, as shown in Figure 2.3.
In order to respect the angular momentum conservation, the vortex is forced to increase its angular velocity (spinning up phenomenon). This leads to the generation of high local velocity gradients which, due to the fluid viscosity, conduct to a subsequent motion disintegration in microvortex (eddies). In SI engines, this small scale turbulence accelerates the flame brush combustion, enhancing the flame front wrinkling effect, making the tumble motion particularly sought in this kind of engines.

Usually, tumble is generated through a port design which directs the flow predominantly towards the exhaust valves, as shown in Figure 2.3.

On the test bench, usually, a T joint is used for the characterization of the tumble motion (Figure 2.4). The T joint allows converting the vortex axis by leading the flow towards a light paddle wheel with a honeycomb structure which is used to measure the tumble torque $T$. Several parameters were used to define tumble. Among them, one of the most used tumble number definitions is given by the equation (2.2).
\[ C_T = \frac{2T}{\dot{m} v_0 B} \] (2.2)

\( \dot{m} \) is the air mass flow rate, \( B \) the cylinder bore and \( v_0 \) the characteristic velocity of the flow, which depends on the pressure drop across the valve [2].

The third main organized motion inside the cylinder is the squish motion; squish is typical of the pistons of Diesel engines. It occurs toward the end of the compression stroke when a portion of the piston face and cylinder head approach each other closely [2]. In this phase, the flow tends to move from the cylinder walls toward the
cylinder center, generating a toroidal vortex, as shown in Figure 2.5.

\begin{figure}[h]
\begin{center}
\includegraphics[width=0.5\textwidth]{squish_motion.png}
\end{center}
\caption{Squish motion}
\end{figure}

Due to its dynamics, squish motion is tough to characterize through a global parameter. Further information about this kind of organized motion can be found in [2].
Chapter 3

Tool Capabilities

GT-SUITE is the leading engine and vehicle simulation tool used by engine makers and suppliers. It is suitable for analyses of a wide range of issues related to vehicle and engine performance ([3]). The solution is based on one-dimensional fluid dynamics, representing the flow and heat transfer in the piping and other flow components of an engine system. A typical scenario provided by the GT-ISE interface is shown in Figure 3.1, where a simplified version of the turbocharged SI engine which was used for this analysis is shown.

![Simplified model of the mono-cylinder turbocharged engine](image)

**Figure 3.1:** Simplified model of the mono-cylinder turbocharged engine
The use of a simplified model allows to reduce the computational time of the analyses carried out. Among the objects displayed, the cylinder object mainly contains the thermal model for the determination of the wall temperatures, the heat transfer model, the flow model and the combustion model. A brief description of the flow model and the combustion model is given in the following paragraphs, while further information regarding the other models is provided in [3]. A description of the prechamber object is given in chapter 3.3.

### 3.1 Combustion models

In addition to the fluid flow and heat transfer capabilities, the code contains many other specialized models required for system analyses (i.e. flow models, aimed to describe the air motion within the cylinder, or the combustion models).

The most used combustion models are either two-zone based or multi-zone based. A two-zone combustion model characterizes two distinct zones, unburned and burned, normally modeled with a separate temperature for each zone. At each time step, a mixture of fuel and air is transferred from the unburned zone to the burned zone with a velocity defined by the burn rate. In the two-zone model, the energy equations (3.1) and (3.2) are solved in each time step:

\[ \frac{dm_u e_u}{dt} = -p \frac{dV_u}{dt} - Q_u - \left( \frac{dm_{f,b}}{dt} h_f + \frac{dm_{a,b}}{dt} h_a \right) + \frac{dm_{f,i}}{dt} h_{f,i} \]  

where the \( m_u \) is the unburned zone mass, \( m_{f,b}, m_{a,b} \), the fuel and air masses transferred to the burned zone, \( h_f \) and \( h_a \) the respective enthalpy, \( m_{f,i} \) the injected fuel mass, \( h_{f,i} \) its enthalpy, \( p \) the cylinder pressure, \( V_u \) the unburned zone volume, \( Q_u \) the unburned zone heat transfer rate and \( e_u \) the unburned zone energy.

\[ \frac{dm_b e_b}{dt} = -p \frac{dV_b}{dt} - Q_b - \left( \frac{dm_{f,b}}{dt} h_f + \frac{dm_{a,b}}{dt} h_a \right) \]  

where subscript "b" denotes burned zone. In each equation the internal energy of each zone varies because of the pressure work, the heat transfer, the combustion and a transport term relative to the fuel injection. Multi-zone models engage several unburned and burned zones. The most important multi-zone combustion model
for the purposes of this thesis work is the JetIgnition model, whose description is provided in chapter 3.1.4.

A further classification of combustion models is related to their predictability. In Non-Predictive combustion models, the burn rate is directly imposed as a simulation input and it is independent of variables such as residual fraction or cylinder pressure: the fuel and air simply burn at the prescribed rate. In a Predictive combustion model, the burn rate is calculated from the appropriate inputs (pressure, temperature, equivalence ratio, residual fraction, etc.) and then applied in the simulation. Similar to other predictive models (like the zero-dimensional flow model), these models require a calibration phase conducted by tuning some parameters.

### 3.1.1 User-Imposed Combustion Profile

The EngCylCombProfile template can be used to impose a user-defined combustion profile. It is particularly useful if the instantaneous cylinder pressure from the engine is available as measurement because the burn rate can be calculated from the cylinder pressure ([3]). A user-imposed combustion profile is shown in Figure 3.2. On the same graph, the SI Wiebe Model that best fits the imposed burn rate is represented.

### 3.1.2 Spark-Ignition Wiebe Model

When modelling a Spark-Ignition engine, the most used non-predictive model is the SIWiebeProfile. The Wiebe function, whose expression is provided by equation (3.3), approximates the typical S-shape of the fraction of fuel burned, expressed as a function of the crank angle, through an exponential function dependent on an anchor angle (typically MFB50 - the angle at which the mass of fuel burned is equal to 54%), an indication of the duration of the combustion process (the 10-90% burn duration), an exponent and eventually the overall fraction of fuel burned when the end of combustion occurs. The Wiebe function can be written as shown in equation (3.3).

\[ x_b = F \left( 1 - e^{-W(\theta-\theta_{SOC})(E+1)} \right) \]  

(3.3)

where \( \theta \) is the instantaneous crank angle, \( \theta_{SOC} \) the calculated start of combustion, \( F \) the fraction of fuel burned, \( W \) a constant dependent on the combustion duration.
and E the Wiebe exponent. A typical Wiebe profile is shown in Figure 3.2.

![Figure 3.2: Wiebe function or cumulative fraction of fuel burned (green) and resulting burn rate (blue)](image)

### 3.1.3 Spark-Ignition Turbulent Flame Model

SITurb (or Spark-Ignition Turbulent Flame Model) is the most used two-zone predictive model for traditional SI engines. When an SITurb model is used, once the spark occurs, two zones are defined: the unburned zone and the burned zone. The rate at which the mass is transferred from the unburned zone to the burned zone is given by the equation (3.4)

\[
\frac{dM_b}{dt} = \frac{M_e - M_b}{\tau}
\]  

(3.4)

where \(M_e\) is the mass entrained into the flame front, and \(M_b\) the burned zone mass. \(\tau\) is a time constant, given by the ratio between the Taylor microscale length \(\lambda\) and
the laminar flame speed $S_L$. The entrainment of fuel and air into the flame front is governed by the equation (3.5):

$$\frac{dM_e}{dt} = \rho_u A_e (S_T + S_L)$$

(3.5)

where $\rho_u$ indicates the unburned zone density, $A_e$ the flame front area, $S_T$ and $S_L$ the turbulence speed and the laminar speed respectively. The laminar flame speed is defined as shown in equation (3.6):

$$S_L = (B_m + B_\phi (\phi - \phi_m)^2) \left( \frac{T_u}{T_{ref}} \right) ^\alpha \left( \frac{\rho}{\rho_{ref}} \right) ^\beta f(Dilution)$$

(3.6)

where $\phi$ and $\phi_m$ are the equivalence ratio and the equivalence ratio at maximum speed respectively, $T_u$ the unburned gas temperature, $B_m$ the maximum laminar speed, $B_\phi$ the laminar speed roll-off value, $\alpha$ and $\beta$ the temperature and pressure exponents, $T_{ref}$ and $\rho_{ref}$ the temperature and the density at the reference conditions, and $f(Dilution)$ a function that accounts for the impact of dilution on the flame speed through a Dilution Effect Multiplier, a tuning parameter.

The turbulent flame speed $S_T$ is described by the equation (3.7):

$$S_T = C_{TFS} u' \left( 1 - \frac{1}{1 + C_{FKG} \left( \frac{R_f}{L_i} \right) ^2} \right)$$

(3.7)

$u'$ is the turbulent intensity, $R_f$ the flame radius, $L_i$ the integral length scale. The equation contains two calibration parameters: $C_{TFS}$, the Turbulent Flame Speed Multiplier and $C_{FKG}$, the Flame Kernel Growth Multiplier. The integral length scale is related to the Taylor microscale length through the equation (3.8):

$$\lambda = C_{TLS} L_i \sqrt{Re_t}$$

(3.8)

where $C_{TLS}$ is the Taylor Length Scale Multiplier and $Re_t$ the turbulent Reynolds number, whose definition is provided in equation (3.9):

$$Re_t = \frac{\rho_u u' L_i}{\mu_u}$$

(3.9)

$\mu_u$ indicates the unburned zone dynamic viscosity.
In the model shown in Figure 3.1 since a prechamber object is modeled, the cylinder does not accept an SITurb model, which can only be used in the prechamber.

3.1.4 Turbulent Jet Ignition Combustion Model

The Turbulent Jet Ignition combustion can be described by using a multi-zone predictive combustion model provided in the main chamber (EngCylCombJetIgnition), which tracks the evolution and burning of the jet from the prechamber nozzle. A detailed description of the combustion model is given in [4]: at first, all the main chamber mass is contained in a single thermodynamic zone, the main unburned zone, where the mixture from the intake ports and the main chamber injected fuel flow. This zone also exchanges mass and energy with the prechamber. Once the combustion starts in the prechamber, a new zone in the main chamber is introduced, the jet zone. All these zones can exchange mass and energy with each other. The jet zone evolution is described by two main quantities, the penetration distance $S(t)$ and the tip velocity $u(t)$.

$$S(t) = C_s t^{1/2} \left( \frac{u_{noz} d_{noz}}{C_d} \sqrt{\frac{\rho_{noz}}{\rho_{cyl}}} \right)^{1/2}$$  \hspace{1cm} (3.10)

$$u(t) = C_u t^{-1/2} \left( \frac{u_{noz} d_{noz}}{C_d} \sqrt{\frac{\rho_{noz}}{\rho_{cyl}}} \right)^{1/2}$$  \hspace{1cm} (3.11)

where $t$ is the time elapsed, $u_{noz}$ is the velocity at the prechamber nozzle, $d_{noz}$ is the nozzle diameter, $C_d$ is the nozzle discharge coefficient, $\rho_{noz}$ is the density of the fluid in the prechamber jet at the nozzle exit, $\rho_{cyl}$ is the density of the surrounding fluid in the main chamber, $C_u$ and $C_s$ are model constants.

Even though there is the presence of more than one nozzle, at the moment the JetIgnition model is not tracking a separate flame from each prechamber nozzle hole. Instead, it calculates the total area/volume of the flame from all nozzle holes and treats it as a single flame propagating from the user specified location.
The mass entrained into the jet, \( m_{je}(t) \), is calculated through the equation (3.12).

\[
m_{je}(t) = C_e m_{noz} \left( \frac{u_{noz}}{u(t)} - 1 \right)
\]

where \( m_{noz} \) is the mass in the jet at the prechamber nozzle exit and \( C_e \) is a tuning parameter (Entrainment Rate Multiplier). This combustion model contains an ignition delay model developed using data from natural gas engines. Once the ignition delay is consumed, a third thermodynamic zone is introduced, the burned zone, to which mass and energy are transferred based on the combustion models employed. At the tip of this transient jet, a spherical flame is initiated, which propagates into the main chamber like a conventional turbulent flame. The jet flame and the spherical flame occur together until a sufficient pressure difference exists between the two chambers.

To model this behaviour, two modes of combustion are activated after ignition: jet combustion and propagating spherical turbulent flame combustion. Even after the start of combustion, the jet continues to entrain mass based on the equation (3.12). The jet burn rate is governed by the equations (3.13), (3.14) and (3.15).

\[
\frac{dm_{ju}}{dt} = \frac{dm_{je}}{dt} - \frac{dm_{jb}}{dt}
\]

\[
\frac{dm_{jb}}{dt} = C_{df} m_{ju} \sqrt{\frac{k}{V_{cyl}^{1/3}}} f(y_{O2}) + s_j m_{ju}
\]
Turbulent Jet Ignition (TJI) Combustion Modelling in GT-Suite

\[ s_f' = \frac{1}{m_u} \frac{dm_{fb}}{dt} \tag{3.15} \]

where \( m_{jw} \) is the unburned mass inside the jet, \( m_u \) is the total unburned mass in the main chamber, \( k \) is the turbulent kinetic energy, \( V_{cyl} \) is the main chamber volume, \( C_{df} \) is a tuning parameter (Diffusion Combustion Rate Multiplier). The jet combustion is mixing-controlled, and so governed by turbulence (TKE) and oxygen mass fraction. The last term of the equation (3.14) considers the contribution given by the turbulent flame combustion to the burned gas entrained by the jet. The turbulent flame combustion is governed by the equations (3.16), (3.17) and (3.19).

\[ \frac{dm_{fa}}{dt} = \rho_u A_e S_T - \frac{dm_{fb}}{dt} \tag{3.16} \]

\[ \frac{dm_{fb}}{dt} = \frac{m_{fa}}{\tau} + \dot{s}_j m_{fb} \tag{3.17} \]

\[ \tau = \frac{C_t \lambda}{S_L} \tag{3.18} \]

\[ \dot{s}_j = \frac{1}{m_b} \frac{dm_{jb}}{dt} \tag{3.19} \]

\[ S_T = S_L + C_t u' \tag{3.20} \]

\( m_{fb} \) and \( m_{fa} \) are the unburned mass entrained by the flame brush and the burned mass behind the flame brush respectively, \( m_b \) is the total burned mass in the main chamber, \( \rho_u \) is the density of the unburned gases, \( A_e \) the entrainment surface area, \( S_T \) is the turbulent flame speed of the brush, \( S_L \) is the laminar flame speed, \( \tau \) is the characteristic burning timescale, \( \dot{s}_j \) is a source term coupling the flame propagation model with the jet combustion model and \( C_t \) and \( C_l \) are tuning parameters (Turbulent Flame Speed Multiplier and Taylor Length Scale Multiplier). The equation (3.16) controls the rate of entrainment of gases by the flame brush while the equation (3.19) governs the rate of burning of the entrained mass. The equation (3.19) contains, like the equation (3.14), a source term \( (\dot{s}_j m_{fb}) \) that considers the burned gases from jet combustion. The turbulent flame speed \( S_T \) and the burning timescale \( \tau \) are functions of the main chamber turbulence quantities: the turbulence intensity \( u' \) and the Taylor microscale \( \lambda = l/Re_T \), calculated using the integral length scale \( l \) and the turbulent Reynolds number \( Re_T \). These quantities are provided by the turbulence sub-model used in the main chamber, as described in chapter 3.2. In comparison with an SITurb combustion model, the absence of the Flame Kernel
Growth Multiplier makes the flame propagation combustion turbulent as soon as it starts from the tip of the jet.

This study ([4]) does not describe the premixed combustion contribution to the jet combustion, which is modeled in GT-ISE. The complete Turbulent Jet Ignition predictive combustion model provided by Gamma Technologies can be seen as a combination of a DIPulse model and an SITurb model, which describe respectively the jet combustion and the turbulent flame combustion (the DIPulse predictive model is the most used combustion model for Diesel engines in GT-ISE). EngCylCombJetIgnition combustion can be eventually calibrated by tuning 7 calibration parameters.

<table>
<thead>
<tr>
<th>Calibration parameter</th>
<th>Symbol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Entrainment Rate Multiplier</td>
<td>$C_e$</td>
</tr>
<tr>
<td>Ignition Delay Multiplier</td>
<td>$C_{id}$</td>
</tr>
<tr>
<td>Premixed Combustion Rate Multiplier</td>
<td>$C_{pc}$</td>
</tr>
<tr>
<td>Diffusive Combustion Rate Multiplier</td>
<td>$C_{df}$</td>
</tr>
<tr>
<td>Turbulent Flame Speed Multiplier</td>
<td>$C_t$</td>
</tr>
<tr>
<td>Taylor Length Scale Multiplier</td>
<td>$C_l$</td>
</tr>
<tr>
<td>Dilution Effect Multiplier</td>
<td>$C_{de}$</td>
</tr>
</tbody>
</table>

Table 3.1: EngCylJetIgnition - Calibration parameters

The relationship between the described zones is displayed in Figure 3.4.

Figure 3.4: Jet Ignition combustion model - Mass exchange between the zones
Since this model is predictive, it requires a Flame Geometry Object characterization. The Flame Object describes the geometry of the head and the piston and permits to define the location of the prechamber nozzles with respect to the cylinder. Currently (v2020), when a Jet Ignition combustion object is used in the main chamber, it is not possible to model the main chamber spark.

3.2 In-Cylinder Turbulence Model

As a consequence of the importance of the gas motion within the engine cylinder, in the last years a lot of time and effort have been spent to develop models able to predict the turbulence as a function of engine operating conditions. Typically detailed 3D-CFD simulations are required to describe complex flow phenomena, but the main drawback of these simulations is given by the high computational time and cost that they require. Due to this, zero dimensional (0D) flow models have been developed, aiming to accurately predict the in-cylinder turbulence levels over the entire cycle. In GT-ISE, each chamber is characterized by its own zero dimensional flow model. The in-cylinder flow model (whose description is provided in [3] and [5]), that governs the evolution of the mean and turbulent flow inside the cylinder, breaks the cylinder into multiple regions: the central core region, the squish region, the head recess region, and the piston cup region. At each time step in each region, the mean radial velocity, axial velocity, and swirl velocity are calculated taking into account the cylinder chamber geometry, the piston motion, and the flow rates of the incoming and exiting gases through the valves. The main chamber flow model utilizes a K-k-\(\epsilon\) approach. K is the mean kinetic energy, defined as shown in equation (3.21):

\[
K = \frac{1}{2}U^2
\]

(3.21)

where \(U\) is the mean velocity inside the cylinder. The turbulent kinetic energy \(k\), in a zero-dimensional approach, is described by equation (3.22):

\[
k = \frac{1}{2}(u'_x^2 + u'_y^2 + u'_z^2) = \frac{3}{2}u'^2
\]

(3.22)

where \(u' = u'_x = u'_y = u'_z\) is the intensity of turbulent field inside the cylinder, assumed to be homogeneous and isotropic. \(\epsilon\) is the kinetic energy dissipation rate,
responsible of the dissipation of small eddies into heat. These quantities are modeled through three differential equations:

\[
\frac{d(mK)}{dt} = C_{in}(1 - \alpha_{in})E_{in} + K\dot{m}_{out} - P_k \tag{3.23}
\]

\[
\frac{d(mk)}{dt} = C_{in}\alpha_{in}E_{in} + km_{out} + P_k + C_{tumb}T - m\epsilon \tag{3.24}
\]

\[
\frac{d(m\epsilon)}{dt} = C_{in}E_{in}\sqrt{\frac{k}{L_g}} + \epsilon\dot{m}_{out} + P\epsilon + C_{tumb}T\sqrt{\frac{k}{L_g}} - 1.92\frac{m\epsilon^2}{k} \tag{3.25}
\]

where \( m \) is the in-cylinder mass. The first term of each equation indicates the contribution of the intake flow in the cylinder. \( E_{in} \) is defined in the equation (3.26).

\[
E_{in} = (1 - C_T)\frac{1}{2}\dot{m}_{in}v_{in}^2 \tag{3.26}
\]

\( \dot{m}_{in} \) is the intake mass flow rate, \( v_{in}^2 \) is the isentropic velocity of the intake flow, \( C_T \) the tumble coefficient, whose description is provided in the chapter 2.1 indicating the fraction of the inflow energy which is imparted to the tumble. In the equations (3.23) and (3.24), the term \( \alpha_{in} \) describes the contribution to turbulence given by the intake flow (and not generated by the cascade process, from the mean kinetic energy). \( C_{in} \) is a coefficient dependent on a tuning constant (\( C_1 \)) that takes into account the actual flow velocity through the valves, which is not equal to the isentropic values. The second term of each equation describes the effect of \( \dot{m}_{out} \), the mass flow rate that exits the cylinder, through a transport term. \( P_k \) and \( P\epsilon \) describe the production of turbulent kinetic energy and dissipation rate via the energy cascade process, from the large scale mean flow. A definition of these quantities is provided in [5]. They mainly depends on the turbulent viscosity, the in-cylinder density (and its rate of variation) and on two tuning constants, \( C_2 \) and \( C_3 \).

\( C_{tumb}T \) and \( C_{tumb}T\sqrt{k/L_g} \) are referring to the contribution to the turbulence given by the tumble decay that, as already mentioned in chapter 2.1 occurs during the end of compression stroke, enhancing the flame bush combustion. These terms are scaled by another model constant, \( C_{tumb} \). The last term of equations (3.23), (3.24) and (3.25) corresponds to the sink term of each equation: the conversion of mean kinetic energy to turbulent kinetic energy is controlled by the term \( P_k \); the turbulent kinetic energy dissipation into heat is described by \( m\epsilon \), while the dissipation rate is reduced by the term \( 1.92\frac{m\epsilon^2}{k} \).
Regarding the tumble, indicating the rotational component of the large scale mean flow, it undergoes the spinning up phenomenon during the compression stroke to be finally destroyed to produce turbulence. This behaviour is described by the equation (3.27), where the variation of angular momentum $L$ with respect to time is modeled by defining $\dot{L}_{in}$, the tumble production due to the intake flow, $\dot{L}_{out}$, a term that accounts for the flow out of the cylinder and a third term that considers the tumble decay.

$$\frac{dL}{dt} = \dot{L}_{in} + \dot{L}_{out} - L f(s/B) \frac{\sqrt{k}}{r_t}$$  \hspace{1cm} (3.27)

$$\dot{L}_{in} = C_T \dot{m}_{in} |v_{in}| r_t$$  \hspace{1cm} (3.28)

$$\dot{L}_{out} = \dot{m}_{out} r_t^2 \omega$$  \hspace{1cm} (3.29)

In the equations (3.28) and (3.29), $r_t = \frac{1}{4} \sqrt{B^2 + s^2}$ is the tumble macro-vortex radius ($s$ represents the cylinder stroke, while $B$ the cylinder bore), $\omega$ represents the angular speed of the tumble macro vortex and $f(s/B)$ describes the tumble decay function, whose trend is shown in Figure 3.5.

![Figure 3.5: Tumble Decay function](image-url)
Finally, since the turbulent kinetic energy and the dissipation rate were modeled, the integral length scale can be estimated as in (3.30)

\[ l = C_{\mu}^{3/4} \frac{k^{3/2}}{\epsilon} \]  

(3.30)

where \( C_{\mu}^{3/4} = 0.09 \) is a standard model constant.

This modelling of in-cylinder turbulence needs four calibration parameters (\( C_1, C_2, C_3, C_{\text{tumb}} \)), that can be encountered in GT-ISE in the calibration folder of the flow model (respectively under the name of Intake Term Multiplier, Production Term Multiplier, Geometric Length Scale multiplier, Tumble Term Multiplier) if flow model v2016 is used. Actually, a newer version of the flow model (v2019), which provides an enhanced modelling of tumble decay with respect to the v2016 described above, was used during this project. In order to better describe the tumble decay, this version supplies three further tuning parameters: Tumble Destruction Coefficient, Reverse Tumble Decay Multiplier and Outflow Tumble Decay Multiplier. An explanation of the Tumble Destruction Coefficient can be given by focusing on the definition of the tumble decay function \( f(s/B) \), given in [3]

\[ f(s/B) = C_4 \left( \frac{1}{rC_5} - 1 \right) \]  

(3.31)

where \( C_4 \) is the Tumble Destruction Coefficient (forced to be equal to 0.3 in v2016 and editable by the user in v2019), and \( C_5 \) is hard coded, in both versions, to use a value of 1.2. \( r \) is the instantaneous ratio of the piston-to-head distance to the bore (equal to \( s/B \)). If \( C_4 \) increases, the tumble decay becomes faster.

Reverse Tumble Decay Multiplier and Outflow Tumble Decay Multiplier respectively control the rate of additional tumble decay that occurs when the intake valve remains open during the compression stroke, and the rate of additional tumble decay that occurs when “reverse” tumble vortexes are formed as a result of early intake valve closure.

Finally, the last tuning parameter present in both the flow model versions, the Spray-Jet Term multiplier, is provided with the aim of considering the contribution of the injection flow and the prechamber nozzle flow to mean kinetic energy.

A brief summary of the whole set of multipliers provided by the v2019 of the 0D
The 0D flow model outputs are the following: swirl number, tumble number, turbulent kinetic energy, normalized turbulent intensity (scaled basing on the mean piston velocity) and normalized turbulent integral length scale (scaled basing on the cylinder bore).

### 3.3 Prechamber Modelling

The prechamber component contains a geometry specification of the prechamber, and some models for the heat transfer, combustion and turbulence description.

#### 3.3.1 Prechamber Characterization and Sub-models

Regarding the geometry, two options are provided: simple or advanced. The former describes a prechamber with no assumed shape characterized by imposed values of volume and surface. The latter assumes an axisymmetric geometry (defined by imposing the diameter as a function of height) which allows the software to calculate volume and surface. Only an advanced geometry description of the prechamber enable the user to exploit a predictive combustion model (SITurb).

The prechamber heat transfer coefficient can be calculated using the following three methods: "main chamber", "flow" and "profile". The first one indicates that the heat transfer coefficient is taken from the connected cylinder object by calculating an area-weighted average of the head surface heat transfer coefficient. The "flow"
heat transfer model calculates the prechamber heat transfer coefficient using the 0D prechamber flow model outputs. "Profile" indicates that the heat transfer coefficient is defined by the user as a function of the crank angle degree. The results of the heat transfer coefficient model can be tuned using the prechamber heat transfer multiplier. A "main chamber" heat transfer model was chosen for simplicity.

As regards the available combustion models, two main combustion objects can be mainly used: a non-predictive Wiebe profile and an SITurb predictive combustion. When using an SITurb combustion in the prechamber, as it was confirmed by Gamma Technologies, it is impossible to match the measured data while imposing the actual spark timing, due to two main limitations:

1) The Flame Kernel Growth Multiplier (FKGM) does not have any impact on the combustion, so that the flame is immediately turbulent - equation (3.7);
2) The modeled Flame Front is planar as soon as the spark occurs, as shown in Figure 3.6 (where a dummy prechamber geometry was used), and follows the prechamber shape during the whole combustion process always keeping perpendicular to the PC axis.

These features lead to very aggressive burn rates and pressure rises in the first phase of the combustion, as shown in Figure 3.7, where the SITurb multipliers were left to their default values.

As has been explained by Gamma Technologies, the v2021 of the software should
Figure 3.7: Prechamber - Default SITurb Model
allow the user to overcome these problems, since in this version the FKGM has an impact again and a ‘Flame Area Transition’ attribute, that makes the flame spherical in the first phase, has been added.

Due to these aspects, only Wiebe objects were feasible to be used for modelling the prechamber combustion.

The prechamber also contains its own flow model, which can be tuned through the usage of four multipliers. Please note that the flow model has no impact on the case study since a non-predictive combustion and a heat transfer coefficient from main chamber were used.

An overview of the prechamber object main folder is given in Figure 3.8.

![Prechamber Object - Main folder](image)

**Figure 3.8: Prechamber Object - Main folder**

Eventually, the prechamber object permits to perform an Heat Release Based Calibration analysis mode, basing on the pressure cycles (chapter 3.4).

### 3.3.2 Gas Exchange

The information related to the nozzle hole diameter, number of nozzle holes, forward and reverse nozzle discharge coefficients can be provided by using the prechamber connector component. The software only allows to model a number of nozzles of
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the same diameter, even though a real prechamber can be characterized by several holes of different size. An overview of the prechamber object main folder is given in Figure 3.9.

![Figure 3.9: Prechamber Connector - Main folder](image)

3.4 Heat Release Based Calibration

Gamma Technologies suggests to rely on the Heat Release Based Calibration analysis mode for the calibration of the prechamber and the main chamber combustion models.

![Figure 3.10: HRBC analysis mode - Overview](image)

In the prechamber, when this analysis is selected, the solver calculates the Measured Apparent Net Heat Release Rate from the imposed prechamber and main chamber pressure cycles (reverse run), a Predicted Apparent Net Heat Release Rate from the imposed combustion model (forward run), and an RLT that reports the RMS
error between the two. The apparent net heat release rate does not report the rate of chemical energy release, but rather the difference between the chemical energy release and the heat transfer rate to the walls.

In order to calibrate a combustion model, an optimization aimed at minimizing the ANHRR RMS error by sweeping the combustion parameters can be set up.

The HRBC analysis mode can be performed independently in the prechamber and in the main chamber. When it is carried on in the PC, the only analysis modes allowed in MC are the Three Pressure Analysis (TPA) and the HRBC. The TPA calculates the apparent burn rate from the MC measured pressure trace, the inlet and the outlet imposed pressure, independently from the PC; it can be useful during the calibration of the PC combustion and the nozzle discharge coefficients since it allows to have the best possible match between the measured and predicted MC pressures. The TPA analysis does not allow to impose a combustion model in MC, since its only purpose is calculating the burn rate. When a Jet Ignition model is used in main chamber, the only available analysis mode is the HRBC.
Chapter 4

Input data

All the simulation work that is described in this thesis has been based on the experimental tests performed on a research single-cylinder gasoline engine characterized by a prechamber passive system. Some engine features are summarized in Table 4.1.

<table>
<thead>
<tr>
<th>Feature</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>N° of cylinder</td>
<td>1</td>
</tr>
<tr>
<td>Displacement</td>
<td>0.5 L</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>11</td>
</tr>
<tr>
<td>Injection System</td>
<td>PDI</td>
</tr>
<tr>
<td>Turbocharger</td>
<td>Yes</td>
</tr>
<tr>
<td>Prechamber System</td>
<td>Passive</td>
</tr>
</tbody>
</table>

**Table 4.1:** Main characteristics of the engine considered

The Port Direct Injection (PDI) allows to split the fuel injection between a port injector and a direct injector. The former is preferred at partial load (PL) in order to improve the air/fuel mixing. The latter is used at full load (FL) in order to exploit benefits as the lower temperature at the end of the compression stroke and the increased volumetric efficiency.

The main chamber is equipped with a second spark: when the load is low, the combustion has difficulties to start from the prechamber due to the abundant presence of residual gases. In these cases, the main chamber spark occurs before the prechamber spark and the combustion is similar to that of a traditional SI engine, while the PC spark operates in order to complete the combustion process.

For what concerns the operating conditions which have been tested in the test rig,
all the points at full load from 1000 RPM to 7000 RPM with 500 RPM steps have been recorded, as depicted in Figure 4.1, where the experimental time-average data were compared with the 1D-CFD detailed engine model output.

A complete model of the single-cylinder engine, where the Waste-Gate valve was controlled in order to target the measured torque, was already available at the beginning of this thesis work (Torque CTRL).

For the available partial load operating points, which are shown in Figure 4.2, a further controller on the throttle valve opening was added in order to target the intake manifold measured pressure, while the WG was controlled to match the boost pressure.
The circled points in Figures 4.1 and 4.2 were also simulated through 3D-CFD. The boundary conditions imposed in the 3D-CFD simulations were inherited from the 1D-CFD model (since a CA-based experimental measurement of the intake and exhaust pressure was not available). The prechamber pressure profile was not measured on the test bench, so that the 3D-CFD simulations were useful also to evaluate this quantity: as it was mentioned in chapter 3.4, in GT-ISE the HRBC analysis mode routine requires the measured pressure cycle of both chambers. Due to the lack of the experimental pressure cycles in the prechamber, the pressure cycles imposed as "measured data" during the HRBC analyses were inherited from 3D-CFD. Because of this, the PC and MC combustion calibration was limited to only two operating points, where the 3D-CFD data were available and the Jet Ignition combustion oc-
Turbulent Jet Ignition (TJI) Combustion Modelling in GT-Suite

curs (the PC spark takes place before the MC spark). This was not a limitation for the study that was mainly focused on developing the calibration methodology.

In the following chapters, the load of the operating points considered will be indicated as a percentage of the maximum value for that RPM. The 3D-CFD analyses carried out at 2500x54% (PL) and 6500x100% (FL) allowed to calibrate the in-cylinder flow model (chapter 5.1), while the combustion model was calibrated at 3000x100% and 6500x100% (chapter 6).

As it was mentioned in chapter 3, a simplified model was built from the detailed one. In the simplified model, it is possible to impose the CA-resolved intake and exhaust port pressure profiles, inherited from the detailed model, for each operating point (Figure 4.3). Moreover, the simplified model enable the use of a Three Pressure Analysis (TPA) mode in the main chamber, useful during the followed procedure.

![Figure 4.3: Simplified model of the single-cylinder engine - Main input data](image-url)
The "Switch" objects allow to apply a different friction multiplier depending on the valve lift. The detailed model simulation allowed to calculate the instantaneous pressure profiles (CA-based) in the "IntPort-Y1" and "Exh-Port-1" objects. These pressure profiles, which are affected by the fluctuations caused by the reflections of the pressure waves on the valves during their closure period, were imposed in the "Intake" and "Exhaust" objects of the simplified model.

It is desirable to reduce the magnitude of these reflections in the simplified model, so that the actual pressure behavior within the port is as close to the measured behavior as possible. This is accomplished using the friction multiplier within the last pipe attached to each "EndEnvironmentTPA" (the "Intake" and "Exhaust" objects). During the valve closed period, this multiplier is increased to a very large value to damp out these fluctuations. During the valve open period, this value is set to its normal value. This is accomplished using a simple control system.

A typical pressure cycle of a Jet Ignition combustion is shown in Figure 4.4. Here, the first peak of pressure that takes place in the prechamber is related to the prechamber combustion. Before the main chamber start of combustion, the space-average MC pressure is not affected by the prechamber pressure rise. After the start of combustion in the main chamber, the second peak of pressure takes place in both chambers.

Figure 4.4: Pressure Cycles - 6500x100%
Chapter 5

Procedure

In order to develop a methodology for modelling and calibrating a Jet Ignition combustion in the GT-ISE environment, several paths have been explored and different tests have been conducted. This chapter aims to provide an overview of the methodology defined and of the main obstacles that were encountered. The main reference for the following calibrations was the available 3D-CFD dataset.

With reference to Figure 5.1, at first, the main chamber flow model has been calibrated, since it has an evident impact on the predictive combustion. The flow model calibration procedure is explained in the section 5.1. The Workflow 1 has been followed, due to the PC SITurb model limitations.

Figure 5.1: JI combustion modelling - Workflow
5.1 Main Chamber Flow Model Calibration

Since at that time no information was available regarding the prechamber geometry, the flow model was calibrated before the prechamber modelling. When using the predictive model, the most important factors in these calculations are the swirl and tumble generated by the fluids entering the cylinder through the valves. The swirl and tumble coefficients specified in the valves are used to calculate the swirl and tumble torque, which are applied to the in-cylinder gases (increasing the magnitude of the swirl and tumble coefficients will increase the generation of swirl and tumble through valve flow). Due to the absence of an intense swirl motion in the engine considered, swirl was not modeled. Tumble coefficients are provided as input data through the valve objects and are defined in the software as the ratio of the angular momentum flux to the linear momentum flux. These were calculated as shown in equation (5.1):

\[ C_t = \frac{2T}{\dot{m}U_{is}B} \]  

(5.1)

where \( T \) is the tumble torque, defined as the torque around an axis located between the intake and exhaust valves, \( \dot{m} \) the mass flow rate through the intake valves, \( U_{is} \) the isentropic velocity (equation (5.2)), and \( B \) the cylinder bore.

\[ U_{is} = \sqrt{RT_0}\left[\frac{2\gamma}{\gamma-1}\left(1-P_R^{\frac{1}{\gamma}}\right)\right]^{1/2} \]  

(5.2)

\( P_R \) is the imposed pressure ratio (static outlet pressure/total inlet pressure) and \( T_0 \) the upstream stagnation temperature.

The input data for (5.1) and (5.2) were obtained from 3D-CFD simulations. The flow model was also equipped with a coarse geometric description of the piston cup, but it resulted to have negligible effects on the flow model outputs, since the swirl coefficients were not imposed.

In order to properly consider the contribution to turbulence given by the direct injector spray, it can be useful to fill in the fields describing the injector nozzles geometry (Figure 5.2).
The turbulence model was calibrated using the seven tuning parameters which were described in chapter 3.2, aiming to obtain a good match between 0D flow model outputs and 3D-CFD results, and focusing the attention near the combustion start event.

Since the in-cylinder flow model calibration was accomplished before the prechamber modelling, the contributions to turbulence given by the prechamber jet was not considered in this phase.

Two operating points were taken into account (2500x54% - PL and 6500x100% - FL).

In order to gain more awareness about the effects of each multiplier on the flow model outputs, a sensitivity analysis was carried out before starting with the calibration.

### 5.1.1 Sensitivity Analysis

The sensitivity analysis of the flow model (in terms of tumble number, turbulent kinetic energy, normalized turbulent intensity, normalized turbulent length scale) to its calibration parameters was accomplished by sweeping each parameter individually around its default value.

A summary of the flow model tuning parameters and the corresponding default value is given in Table 5.1.
Likewise the calibration, the sensitivity analysis was performed on two operating points (2500x54% and 6500x100%); the majority of the multipliers resulted to have a similar percentage incidence on the flow model outputs at both engine speeds, except for the Outflow and Reverse Tumble Decay Multipliers which, as explained below, were not used during the calibration phase. The effect of these parameters resulted to be dependent not only on the operating points, but also on the valve timings. The impact of each multiplier on the turbulence model for the 6500x100% operating point is shown in Figure from 5.3 to 5.10 and summarized in Figure 5.11.

Table 5.1: EngCylFlow - Calibration parameters - Default values

<table>
<thead>
<tr>
<th>Calibration parameter</th>
<th>Default value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Production Term Multiplier</td>
<td>1</td>
</tr>
<tr>
<td>Geometric Length Scale Multiplier</td>
<td>1</td>
</tr>
<tr>
<td>Intake Term Multiplier</td>
<td>1</td>
</tr>
<tr>
<td>Spray/Jet Term Multiplier</td>
<td>0.1</td>
</tr>
<tr>
<td>Tumble Term Multiplier</td>
<td>1</td>
</tr>
<tr>
<td>Tumble Destruction Coefficient 1</td>
<td>0.3</td>
</tr>
<tr>
<td>Outflow Tumble Decay Multiplier</td>
<td>1</td>
</tr>
<tr>
<td>Reverse Tumble Decay Multiplier</td>
<td>1</td>
</tr>
</tbody>
</table>

Figure 5.3: Flow model sensitivity analysis - Production Term Multiplier
Turbulent Jet Ignition (TJI) Combustion Modelling in GT-Suite

Figure 5.4: Flow model sensitivity analysis - Geometric Length Scale Multiplier

Figure 5.5: Flow model sensitivity analysis - Intake Term Multiplier
Figure 5.6: Flow model sensitivity analysis - Spray/Jet Term Multiplier

Figure 5.7: Flow model sensitivity analysis - Tumble Term Multiplier
Figure 5.8: Flow model sensitivity analysis - Tumble Destruction Coefficient 1

Figure 5.9: Flow model sensitivity analysis - Outflow Tumble Decay Multiplier
The absence of a prechamber object during these analyses meant that the impact on the turbulence given by the jet from PC to MC was not considered so far. If the prechamber and its combustion model are built, the effect of the jet on the main chamber flow model outputs is given in Figure 5.12 (each flow model calibration parameter was left to its default value).
Moreover, a Spray-Jet Term Multiplier sweep performed when a PC combustion is modelled leads to the results in Figure 5.13.

Figure 5.12: Flow model sensitivity analysis - effect of the PC modelling

Figure 5.13: Flow model sensitivity analysis - Spray/Jet Term Multiplier (with PC jet)
Due to the effect of the prechamber jet on the MC turbulence, it could be useful to model the prechamber before starting with the MC flow model calibration phase, in order to improve the match with the 3D-CFD data. Despite this, the results obtained from the calibration are good.

5.1.2 Calibration Results

Due to the definition of Outflow Tumble Decay and Reverse Tumble Decay multiplier (chapter 3.2), these parameters seems to be useful to model the contribution to tumble decay that occurs when either an early intake valve closure or a late intake valve closure strategies are used. Moreover, regarding the spray-jet term multiplier, the results of the sensitivity analysis showed that it has an effect on the contribution given to turbulence by the jet from the prechamber, which was not modelled at that time. Due to these reasons, OTDM, RTDM and SJTM were not tuned, but left to their default values.

The results of the calibration of the flow model, which was designed to match the 3D-CFD data before the TDCF, are shown in and Figure 5.14 and 5.15. Despite the absence of the prechamber object, a good result has been obtained.
Figure 5.14: Calibrated flow model - 2500x54%

Figure 5.15: Calibrated flow model - 6500x100%
5.2 Prechamber Modelling

The main features of the prechamber object are summarized in Figure 3.8. An advanced geometry specification was used in order to assess subsequently the usage of an SITurb predictive model, which requires a detailed description of the prechamber. After that, the focus was moved on the nozzle discharge coefficients ($C_d$) tuning; during this phase a Three Pressure Analysis (TPA) mode was activated in the main chamber, in order to match the main chamber pressure as much as possible, reducing the errors on the prechamber combustion calibration phase. With regard to the $C_d$, as no reference was available, the forward and reverse nozzle discharge coefficients were considered equal for simplicity. The $C_d$ impacts the prechamber pressure and so on the main chamber - prechamber gas exchange, as shown in Figure 5.16, where a fixed Wiebe profile was used in all scenarios.

Figure 5.16: Prechamber - $C_d$ impact on PC Pressure and instantaneous mass PC-MC
The prechamber pressure and the mass flow rate through the orifices depends both on the $C_d$ and on the prechamber combustion. In principle, some attempts have been made to design a prechamber Wiebe combustion and a $C_d$ able to match the measured pressure and the MC-PC gas exchange, but this approach resulted to be not robust due to the high number of variables involved (several combinations of Wiebe models and $C_d$ could have been defined and considered correct). In order to simplify the problem, as first step, a reasonable $C_d$ was used to reduce the number of degrees.

Once the nozzle discharge coefficients were defined, the prechamber combustion could have been modeled either by using a non-predictive Wiebe combustion, or by calibrating a predictive SITurb. Since the main goal of this thesis work was to obtain a fully predictive model, the prechamber SITurb object was investigated, but it was found not to be able to properly describe the prechamber pressure, as shown in chapter 3.3.1.

### 5.3 Prechamber Combustion

Regarding the prechamber combustion, Gamma Technologies suggests to calibrate the Wiebe profile parameters with the aim of minimizing the PC ANHRR RMS error. This RLT is provided when the HRBC analysis mode is active. It is defined as shown in equation (5.3).

$$\text{RMSE}_{\text{ANHRR}} = \sqrt{\int_{t_0}^{t_f} (\text{ANHRR}_{\text{pred}} - \text{ANHRR}_{\text{meas}})^2 \, dt}$$  \hspace{1cm} (5.3)

where $\text{ANHRR}_{\text{pred}}$ is the Apparent Net Heat Release Rate calculated from predicted pressure, $\text{ANHRR}_{\text{meas}}$ the Apparent Net Heat Release Rate calculated from measured pressure, $t_f$ the time at end of integration (it is the time at which the measured cumulative heat release reaches 90% of the peak value). The integration is stopped at measured 90% cumulative heat release to avoid the inclusion of long "tails" that may include excessive noise from the measured pressure. $t_0$ is the time at the beginning of integration, corresponding to the predicted combustion start angle. The prechamber Wiebe profile parameters were swept in order to minimize the (5.3). When this approach was attempted on the 3000x100% operating point, the match between the measured and the predicted data resulted to be inadequate,
as shown in Figure 5.17.

![Graph showing prechamber pressure and apparent net heat release rate](image)

**Figure 5.17:** PC Wiebe ANHRR RMSE-based calibration - 3000x100%

The unsatisfactory result provided by the optimizer on the 3000x100% point can be ascribed to the particular shape of the pressure cycle. This is characterized by two main peaks (attributable to the two ANHRR peaks) that are hard to reproduce through the simple shape of a Wiebe profile.

Hence, in order to obtain a good match with the prechamber measured pressure, the Wiebe combustion was calibrated with the aim of replicating the second peak of pressure (and ANHRR), which is the nearest to the main chamber start of combustion. This kind of approach led to a delayed 1D-predicted jet in comparison with 3D-CFD data (Figures 5.18 and 5.19 - green curves), despite the acceptable result in terms of predicted prechamber pressure.
Figure 5.18: PC Wiebe definition - 3000x100%
Figure 5.19: PC Wiebe definition - 6500x100%
On account of the nature of the Jet Ignition model (paragraph 3.1.4), the approach followed could be potentially incorrect on those operating points where the prechamber pressure cycle is characterized by a slower combustion in the first phase, due to the mismatch on the jet timing, and it could lead to an incorrect calibration of the main chamber combustion model.

Hence, a further approach to the definition of the prechamber Wiebe could be based on the match on the instantaneous mass from PC to MC. In this perspective, the Wiebe profiles previously defined were shifted aiming to reproduce the jet timing, as shown in Figures 5.18 and 5.19 - blue curves (main impact on 3000x100%).

The Wiebe profiles defined in order to match the timing of the jet were used during the Jet Ignition calibration phase.

### 5.4 Main Chamber Combustion

Once the prechamber combustion was calibrated, the main chamber Jet Ignition combustion model was set up. As mentioned in chapter 3.1.4, a Flame Object characterization containing information related to the combustion chamber geometry was necessary. Regarding the calibration phase, in principle, the GA optimizer settings were inherited from the SITurb calibration procedure described in the GT manual ([3]). At first, as suggested by Gamma Technologies, the Jet Ignition model was initially optimized with the objective of minimize the main chamber ANHRR RMS error between measured and predicted data (equation (5.3) applied to the main chamber). After several tests, a routine aimed to calculate a main chamber user-defined pressure error was manually built (the software does not provide an RLT that reports the pressure RMS error). The user-defined RMS pressure error is described in equation (5.4).

\[
RMSE_{pres} = \sqrt{\int_{\theta_0}^{\theta_f} (P_{pred} - P_{meas})^2 d\theta} \\
\theta_f = 60 \text{ CAD ATDCF} \quad \text{and} \quad \theta \text{ is the generic crank angle degree.}
\]

In general, the angle at the end of integration should be defined considering the end of combustion, in order to focus the error evaluation on the zone of maximum interest.
The Jet Ignition combustion calibration phase and the main results obtained are widely described in chapter 6.
Chapter 6

Main Chamber Jet Ignition Calibration

As already mentioned in chapter 3.1.4, the Jet Ignition model contains 7 calibration parameters. Similarly to what was done for the flow model, a sensitivity analysis was performed on the 2500x54% (PL) and 6500x100% (FL) operating points before starting with the calibration phase. The JI model also requires a Flame Geometry Object, which provides the detailed combustion chamber geometry (comprehensive of "spark location", head, valves and piston geometry), to predict the combustion rate. When a Jet Ignition model is used, the "spark location" indicates the prechamber position with respect to the cylinder. It is not possible to model the main chamber spark when a prechamber object is modeled.

6.1 Sensitivity Analysis

The JI combustion model can be seen as a combination of a DIPulse model and an SITurb model, then the calibration parameters are inherited from these two models. The common upper and lower limits of each multiplier were extracted from the indications of the GT-manual for the DIPulse and the SITurb and are summarized in Table 6.1.
The sensitivity analysis was carried out by sweeping independently each multiplier between the limits shown in Table 6.1, for a fixed prechamber Wiebe combustion. The results for the 6500x100% peak power point are shown in figures from 6.1 to 6.7.

With the increase of the Entrainment Rate Multiplier, the mass entrained into the jet increases (equation (3.12)) and the combustion becomes faster (a higher quantity of mass is burned during the jet combustion). Due to the faster combustion, the main chamber pressure goes up.
If the Ignition Delay Multiplier increases, the combustion is delayed and a higher quantity of fuel is burned during the first phase.

The Premixed Combustion Rate Multiplier growth leads to faster burn rates, enhancing the premixed combustion.
The Diffusion Combustion Rate Multiplier results to be the less impacting calibration parameter. It enhances the jet combustion (equation (3.14)).

Increasing the Turbulent Flame Speed Multiplier results in faster flame combustion (equation (3.20)).
With increasing of the Taylor Length Scale Multiplier the combustion becomes slower, due to the higher characteristic burning timescale (equation (3.18)).

The Dilution Effect Multiplier increases the impact of the dilution on the laminar flame speed, which affects the mass entrained into the flame front.
A summary image which reports the effect of each multiplier on some main chamber combustion parameters is given in Figure 6.8.

<table>
<thead>
<tr>
<th>Multiplier</th>
<th>Peak Pressure</th>
<th>SoC</th>
<th>Duration 10-90</th>
<th>Duration 0-10</th>
</tr>
</thead>
<tbody>
<tr>
<td>Entrainment Rate Multiplier</td>
<td>4</td>
<td>1</td>
<td>3</td>
<td>5</td>
</tr>
<tr>
<td>Ignition Delay Multiplier</td>
<td>5</td>
<td>5</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Premixed Combustion Rate Multiplier</td>
<td>2</td>
<td>0</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Diffusion Combustion Rate Multiplier</td>
<td>1</td>
<td>0</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Turbulent Flame Speed Multiplier</td>
<td>5</td>
<td>0</td>
<td>5</td>
<td>2</td>
</tr>
<tr>
<td>Taylor Length Scale Multiplier</td>
<td>3</td>
<td>0</td>
<td>2</td>
<td>3</td>
</tr>
<tr>
<td>Dilution Effect Multiplier</td>
<td>3</td>
<td>0</td>
<td>1</td>
<td>3</td>
</tr>
</tbody>
</table>

**Figure 6.8:** Jet Ignition sensitivity analysis - MC - summary

This sensitivity analysis does not take into account the cross sensitivities between the calibration parameters, which resulted to be non-negligible from the optimization successively carried on. Due to this, the whole set of multipliers was used to tune the combustion model.

### 6.2 Genetic Algorithm

The calibration was carried out by using the Integrated Optimizer provided by Gamma Technologies, which proposes the use of a Genetic Algorithm (GA) for calibrating other predictive combustion models, likewise the SITurb. A genetic algorithm is a search heuristic that is inspired by Charles Darwin’s theory of natural evolution. Five phases are considered in a genetic algorithm:

1. Initial population
2. Fitness function
3. Selection
4. Crossover
5. Mutation
The process begins with a set of individuals which is called a Population. Each individual is a solution to the problem that has to be solved. An individual is characterized by a set of parameters (variables) known as Genes. The fitness function gives to each individual a fitness score, concerning the probability that an individual will be selected for reproduction is determined. The idea of the selection phase is to select the fittest individuals and let them pass their genes to the next generation. After the selection phase, hybrid solutions are generated through the crossover phase, during which some parts of the genes of the selected individuals are mixed to gain new solutions. Basing on some coefficients previously defined, a mutation is applied to some individuals of the population obtained by casually changing some Genes. The mutation is applied to either improve the value of the fitness function or extend the research space to avoid local optimum configurations. After the mutation phase, the procedure is restarted from point 2.

There are some parameters that characterize the GA setup (i.e. the ones shown Table 6.2). If the optimization setup is not adequate, the evolution of the optimization could be unsuccessful. As an example, the generic function shown in Figure 6.9 can be considered. Assumed that a GA is used for finding the global minimum of this function, if the GA setting is not adequate, it is possible that the local minimum is given as a result of the optimization. This condition is enhanced by the high complexity of our case study, characterized by a high number of calibration parameters (7).

![Figure 6.9: Case Sweep Optimization - min. ANHRR RMSE](image-url)
In our case, the fitness function aims to minimize either the MC ANHRR RMSE (eq. (5.3)), or the MC Pressure RMSE (eq (5.4)). The software allows to run either an independent optimization for each case considered, or a case-sweep optimization. The former is able to find a set of calibration parameters for each operating point that minimizes the ANHRR/Pressure RMSE of that point; the latter permits to define a unique set of calibration parameters able to minimize the value of RMSE averaged over all the operating points.

### 6.3 Calibration Results

As described in chapter 5, the MC Jet Ignition model was tuned through the usage of the Integrated Design Optimizer with the NSGA-III genetic algorithm. NSGA-III algorithm performs a broad search of the design space and is the recommended choice for problems with any moderate to high complexity: when using three or more factors, for multi-modal problems, and/or when working with very non-linear problems ([3]). The calibration was performed on two operating points: 3000x100% and 6500x100%.

At first, since no reference was available, the Jet Ignition combustion was calibrated by imposing the upper and lower limits shown in Table 6.1. The set-up of the optimizer is described in Table 6.2, and it is inherited from the GT-manual guidelines for the SITurb combustion calibration ([3]).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Population Size</td>
<td>30</td>
</tr>
<tr>
<td>Number of Generations</td>
<td>34</td>
</tr>
<tr>
<td>Crossover Rate</td>
<td>default (=1)</td>
</tr>
<tr>
<td>Crossover Rate Distribution Index</td>
<td>default (=15)</td>
</tr>
<tr>
<td>Mutation Rate</td>
<td>0.5</td>
</tr>
<tr>
<td>Mutation Rate Distribution Index</td>
<td>15</td>
</tr>
</tbody>
</table>

**Table 6.2:** Genetic Algorithm Optimization - Setup

The number of combinations investigated by the optimizer is given by the multiplication of the population size and the number of generation (1020 cases). As suggested by Gamma Technologies, the calibration was initially conducted with the aim of minimizing the main chamber Apparent Net Heat Release Rate RMS error. In the
prechamber, the Wiebe profiles able to match the timing of the jet were chosen (Figures 5.18 and 5.19 - blue curves). The results of the case-sweep optimization, which have room for improvement, are shown in Figure 6.10.

\[ \text{Measured} \]
\[ \text{Optimizer – Sweep - min. ANHRR RMSE} \]

\[ \text{Main Chamber Pressure - 3000x100\%} \]
\[ \text{Crank Angle [deg]} \]

\[ \text{Main Chamber Pressure - 6500x100\%} \]
\[ \text{Crank Angle [deg]} \]

\[ \text{MC Apparent Net Heat Release Rate - 3000x100\%} \]
\[ \text{Crank Angle [deg]} \]

\[ \text{MC Apparent Net Heat Release Rate - 6500x100\%} \]
\[ \text{Crank Angle [deg]} \]

**Figure 6.10:** Case Sweep Optimization - min. ANHRR RMSE

In order to evaluate the differences of the Jet Ignition combustion model between the operating points considered, an independent optimization aimed to minimize the ANHRR RMSE was run (Figure 6.11 - blue lines).

As shown at 6500x100\%, the set of multipliers that best fits the ANHRR provides a bad fit of the main chamber pressure, despite the strong relationship between the pressure and the heat release rate profile.

This behaviour can be explained by the equation \( t_0 \): the integration starts from \( t_0 \), the predicted combustion start, without considering the mismatch with the measured
ANHRR of the first phase. For this reason, the optimizer attempts to fit the peak of the release rate by delaying the predicted start of combustion.

Due to this, a user-defined routine, which is able to calculate an error between the measured and the predicted main chamber pressures, was manually built (Figure 6.12 - equation (5.4)).

The results of an independent optimization carried on with the aim of minimizing the pressure RMSE are displayed in Figure 6.11 - green lines. As shown, the fit on the 6500x100% operating point improves, being the pressure RMSE objective function more adequate than the ANHRR RMSE to describe the match between the measured and the predicted combustion.
In both cases, the results obtained on the 3000x100% point are unsatisfactory, making necessary an assessment of the GA setting described in Table 6.2. In order to understand if the optimizator provided a set related to a local minimum, a DoE analysis was run. The DoE analysis allows to assess several combinations of calibration parameters.

The DoE setup is summarized in Table 6.3

<table>
<thead>
<tr>
<th>Calibration Parameter</th>
<th>Lower Lim.</th>
<th>Upper Lim.</th>
<th># of levels</th>
</tr>
</thead>
<tbody>
<tr>
<td>Entrainment Rate Multiplier</td>
<td>0.95</td>
<td>1.25</td>
<td>4</td>
</tr>
<tr>
<td>Ignition Delay Multiplier</td>
<td>1.00</td>
<td>1.60</td>
<td>5</td>
</tr>
<tr>
<td>Premixed Combustion Rate Multiplier</td>
<td>0.05</td>
<td>0.25</td>
<td>3</td>
</tr>
<tr>
<td>Diffusive Combustion Rate Multiplier</td>
<td>0.40</td>
<td>1.00</td>
<td>3</td>
</tr>
<tr>
<td>Turbulent Flame Speed Multiplier</td>
<td>0.50</td>
<td>0.90</td>
<td>9</td>
</tr>
<tr>
<td>Taylor Length Scale Multiplier</td>
<td>1.50</td>
<td>3.00</td>
<td>6</td>
</tr>
<tr>
<td>Dilution Effect Multiplier</td>
<td>1.50</td>
<td>2.50</td>
<td>6</td>
</tr>
</tbody>
</table>

The number of levels indicates the number of cases investigated for a given calibration parameter. The cases explored are evenly-spaced (i.e. the PCRM can take the values: 0.05 - 0.15 - 0.25). The upper and lower limits and the number of levels of each multiplier were accurately chosen in order to limit the number of cases to be run. The ranges of values considered were defined on the basis of the results obtained from several optimizations previously carried out. The DoE setup led to the assessment of 58320 cases, hence this approach to the minimum search requires a large amount of time and is suitable only when the operating points are few.

The set of calibration parameters which minimizes the pressure RMSE (for each case) obtained through the DoE resulted in the green lines (Figure 6.13).
In the meanwhile, a pressure RMSE-based independent optimization was run, considering the same ranges of values of the DoE analysis (Table 6.3), resulting in the blue lines (Figure 6.13). An important results can be earned from Figure 6.13: for the ranges of values shown in Table 6.3, the optimizer setup is robust enough, being the optimum configuration provided by the optimizer very close to the optimum from the DoE analysis.

The results obtained from two optimizations carried on aiming at minimizing the pressure RMSE on the ranges of values shown in Table 6.1 and Table 6.3 are displayed in Figure 6.14.
If the ranges are more extended (fuchsia curve), at 3000x100%, the optimizer provides a result related to a local minimum (the pressure RMSE of the fuchsia curve is higher than the one of the blue case). This occurs because the optimizer has to explore a wider region of the objective function, increasing the likelihood of finding a local minimum. The combinations of calibration parameters which give the fuchsia curves is external to the ranges of the DoE analysis.

Hence, the GA setting used (Table 6.2) should be improved when the ranges of values to be explored are wider (likewise the one shown in Table 6.1).

The set of calibration parameters, obtained from the DoE analysis, which minimizes the pressure RMSE averaged over the two cases, results in the blue curves of Figure 6.14, which allows to state that for the ranges of values shown in Table 6.3, a single set of calibration parameter values able to properly describe the Jet Ignition
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Combustion at both operating points does not exist in this version of the software (v2020).

Figure 6.15: DoE analysis - Case sweep optimum - min. Pressure RMSE

Finally, a case sweep optimization was conducted on both operating points, with the aim of minimizing the pressure RMSE, considering the ranges shown in Table 6.1. The results are displayed in Figure 6.16. In this case, the pressure RMSE is lower than the one of Figure 6.15, but the results have ample room for improvement.
Figure 6.16: Case sweep Optimization - min. Pressure RMSE
Chapter 7

Methodology

In this chapter a brief description of the methodology developed is provided. This workflow takes into account the tool potentialities, the limitations encountered and the results obtained. It is assumed that a detailed model of the engine considered is already available and validated.

1. Building of a simplified model

The construction of a simplified model (likewise the one shown in Figure 7.1) from the detailed one, allows to:

- Reduce the computational cost;
- Set a Three Pressure Analysis (TPA) in the Main Chamber, useful for the following steps.

The main inputs of the simplified model are the prechamber and the main chamber measured pressure traces (obtained, in our case, from 3D-CFD), and the intake and the exhaust instantaneous pressures. The predictive combustion model in the main chamber resulted to be dependent on the simulation time-step; for this reason, the simulation time-step used in the simplified model should be similar to that resulting from the detailed one. It is possible to impose a user-defined time-step through the General folder of the Advanced Setup. Further information regarding the simplified model is given in chapter 4.
2. Prechamber Modelling

The SITurb is not able to properly describe the PC combustion in GT-ISE v2020, as confirmed by Gamma Technologies. The prechamber geometry has to be modelled; since the only adequate prechamber combustion model is non-predictive (SI Wiebe Model), a simple geometry description of the prechamber (volume and surface) is sufficient. In case of predictive combustion in the prechamber, the advanced setting should be used coupled to a calibrated flow model. A procedure similar to chapter 5.1 can be used either for the prechamber.

3. Prechamber Connector Modelling

The nozzle number and the nozzle diameter have to be defined in the prechamber connector (Figure 7.3). Before starting with the prechamber Wiebe combustion calibration phase, a reasonable value of nozzle discharge coefficients \( C_d \) can be defined. The software considers the prechamber nozzles as a single equivalent hole. If sufficient data from 3D-CFD are available, the prechamber connector will be defined in order to define an effective passage area equivalent to that of 3D-CFD. During this work, the reverse and forward \( C_d \) were considered equal.
4. Prechamber Wiebe Combustion Calibration

The Wiebe combustion models have been calibrated. The Wiebe profile has a very simplified shape in comparison with the measured data, so the match on the gas exchange PC-MC can be searched in order to avoid excessive impacts on the Jet Ignition combustion. During this phase, a TPA analysis mode should be active in the main chamber, with the scope of replicating the measured MC pressure cycle as well as possible. For the future versions, in the case of an adequate SI-Turb model for the prechamber, an RLT that describes the match between the measured and the predicted data better than the ANHRR RMSE shown in equation (5.3) should be defined also for the prechamber combustion. To this aim, the user-defined pressure RMSE (equation (5.4)) or a more adequate start of integration for the ANHRR error should be used.

5. Main Chamber Flow Model Calibration

The Main Chamber Flow Model should be calibrated, basing on the 3D-CFD data (section 5.1). This step is useful due to the evident impact of the turbulence on the predictive Jet Ignition combustion. Since the prechamber combustion was modelled in the previous steps, the impact of the jet PC-MC on
the turbulence is taken into account.

6. **Building of a routine for the calculation of the MC pressure RMSE**
   For performing the next steps, a routine for the calculation of the Root Mean Square error between the measured and the predicted pressure traces in the main chamber has to be built. This calculation can be inherited from this work, with reference to Figure 6.12 and equation (5.4). An alternative could have been to build a user-defined ANHRR RMSE considering a more adequate start of integration, but the pressure RMSE was preferred, being this output less noisy. Generally, the model has to show a good level of correlation (trapped mass, thermal status, etc.) for using the pressure RMSE with reliability.

7. **Main Chamber Jet Ignition Model Setup**
   Once calibrated the prechamber combustion, the Jet Ignition combustion can be modelled (chapter 6). Since a predictive model is employed, a Flame Object containing information regarding the combustion chamber geometry has to be defined. The Flame Object also contains a "spark location"; since the software treats the combustion as a single flame propagating from this "spark location", this latter indicates the average position of the nozzles with respect to the head.

8. **Main Chamber Jet Ignition Model Optimization**
   The optimization should be carried on with the aim of minimizing the user-defined pressure RMSE, defined at step 6. The GA algorithm can be used for the scope. Note that its robustness is improved by using the user defined error function, but a special attention should be deserved since the number of independent variables is high and the GA settings were not deeply assessed on others operating points combinations.
Chapter 8

Conclusions

The aim of the thesis work was to investigate the JetIgnition model in GT-ISE v2020 and build a methodology for its calibration. In this perspective, several analyses were performed defining a step-by-step workflow (Figure 8.1) and showing the current modelling limitations.

1) The objective of having a fully predictive model (with an SITurb model in the prechamber and a Jet Ignition model in the main chamber) cannot be accomplished, due to the impossibility to properly model a predictive combustion in the prechamber: as shown in chapter 3.3.1, the Flame Kernel Growth Multiplier (FKGM) does not have any impact when the SITurb is modeled in the prechamber, so the flame is fully turbulent as soon as the spark occurs. Moreover, it was assumed that the flame was planar immediately after the spark, which resulted in a significant overestimation of the flame area. The combination of these two issues made it impossible to match the measured data when the actual spark timing input is used. As has been explained by Gamma Technologies, the v2021 of the software should allow the user to overcome these problems, since in this version the FKGM has an impact again and a ‘Flame Area Transition’ attribute, which makes the flame spherical during the first phase of the combustion, has been added.

2) The Jet Ignition calibration based on the MC Apparent Net Heat Release, which is advised by Gamma Technologies, resulted to be not robust due to the definition of the ANHRR RMS error (given in equation (5.3)), as explained in chapter 6.3. In order to overcome this problem, the optimizer has to be set up with the aim of minimizing the error between measured and predicted pressure. An RLT that
reports the pressure RMS error is not provided by the software when using the HRBC, so a user-defined routine for calculating the error between predicted and measured MC pressure was built (equation (5.4)).

3) Also, when considering the pressure error-based optimization of the Jet Ignition model, the DoE analysis carried out on the 3000x100% operating point (FL) showed the existence of better sets of calibration parameters than the one provided by the optimizer. Hence, when the ranges of values shown in Table 6.1 are considered, the optimizer setup described in Table 6.2 should be improved. Further investigations on the optimizer setup are necessary.

4) The DoE analysis confirmed that a single set of Jet Ignition multipliers able to properly model a TJI combustion in both the operating points considered (3000x100% and 6500x100% - FL) does not exist, for the ranges of values shown in Table 6.3. The results obtained have room for improvement which is expected for the v2021, since the Jet Ignition modelling in GT-ISE seems to be very promising.

The next steps are the following:

1) The GT-SUITE v2021 potentialities assessment, with reference to the predictive combustion (SITurb) in the PC and the Jet Ignition model in the main chamber.

2) An investigation on the Genetic Algorithm setting parameters, aimed at gaining a more robust MC Jet Ignition calibration.

Finally, the methodology built and the main limitations encountered are summarized in Figure 8.1.
Figure 8.1: JI combustion predictive model - methodology
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Bibliography


