

Use of an Innovative Predictive Heat Release Model Combined to a 1D Fluid-Dynamic Model for the Simulation of a Heavy Duty Diesel Engine

Mirko Baratta, Roberto Finesso, Hamed Kheshtinejad, Daniela Misul, Ezio Spessa, and
Yixin Yang

Politecnico di Torino

Massimo Arcidiacono
Centro Ricerche Fiat Scpa

ABSTRACT

An innovative 0D predictive combustion model for the simulation of the HRR (heat release rate) in DI diesel engines was assessed and implemented in a 1D fluid-dynamic commercial code for the simulation of a Fiat heavy duty diesel engine equipped with a Variable Geometry Turbocharger system, in the frame of the CORE (CO₂ reduction for long distance transport) Collaborative Project of the European Community, VII FP.

The 0D combustion approach starts from the calculation of the injection rate profile on the basis of the injected fuel quantities and on the injection parameters, such as the start of injection and the energizing time, taking the injector opening and closure delays into account. The injection rate profile in turn allows the released chemical energy to be estimated. The approach assumes that HRR is proportional to the energy associated with the accumulated fuel mass in the combustion chamber. This procedure allows an accurate calculation of different combustion parameters important for engine calibration, such as MFB50 (50% of fuel burned mass fraction crank angle).

The 0D HRR model was included in a complete 1D fluid-dynamic model of the heavy duty diesel engine, which was designed with GT-POWER commercial code.

The 0D combustion model and the 1D fluid-dynamic model were calibrated in 12 steady-state engine operating conditions acquired over the whole engine map, and the complete model was finally validated on two load transient conditions at 1100 rpm and 1400 rpm, demonstrating a very good predictive capability.

CITATION: Baratta, M., Finesso, R., Kheshtinejad, H., Misul, D. et al., "Use of an Innovative Predictive Heat Release Model Combined to a 1D Fluid-Dynamic Model for the Simulation of a Heavy Duty Diesel Engine," *SAE Int. J. Engines* 6(3):2013, doi:10.4271/2013-24-0012.

INTRODUCTION

1D computer fluid dynamic (CFD) models are currently widely used for the engine simulation by the car manufacturers and by the research community ([1, 2, 3, 4, 5, 6, 7]). This is due, on one hand, to the increasing accuracy of the specific implemented sub-models, which are dedicated to several physical aspects of IC engines, and, on the other hand, to the availability of computational resources of increasing power. These models are capable of simulating the thermo-fluid-dynamic processes in the intake and exhaust pipes, the in-cylinder combustion process, the turbocharger

performance and the engine performance and emissions. Therefore, they allow the main engine-relevant cycle-averaged parameters to be calculated, such as torque, power, indicated work, volumetric efficiency and fuel consumption.

1D simulation tools are based on the solution of the inviscid form of the conservation laws of mass, momentum and energy (Euler equations). The equations are actually written in a 'generalized' form, in which ad-hoc terms are added, in order to properly simulate the friction and heat-exchange effects at the pipe walls. Furthermore, they are written in a quasi 1D approach, as the variations of any dependent variable in a direction orthogonal to the flow

direction are neglected, and at the same time the changes in the flow cross-section along the pipe axis are accounted for.

The quasi 1D approach is adopted in the engine intake and exhaust systems, which can be considered as assemblies of pipe elements. Other engine components, such as cylinders, injectors, valves, compressors and turbines, are modeled by means of a zero-dimensional approach, usually combined with ad-hoc lumped parameter models or performance maps.

In the last two decades, several commercial 1D CFD tools have been developed for engine flow simulation. In the present paper, GT-POWER simulation tool ([8]) was used to simulate a heavy duty Fiat diesel engine, in the frame of the CORE (CO₂ reduction for long distance transport) European Research Project.

In 1D simulation models, the correct prediction of the incylinder combustion process is mandatory for a correct estimation of the engine indicated power, which in turn is fundamental for achieving the correct values of the brake power and torque, especially in transient conditions.

GT-POWER includes several tools to analyze the diesel combustion process. A first approach, which is not of the predictive, but of the diagnostic type, requires the experimental in-cylinder pressure trace. This is analyzed by means of a two-zone thermodynamic model for the calculation of the heat release. This approach is useful for the in-cylinder heat transfer calibration ([1]). A second approach involves the utilization of a Wiebe function in which the coefficients have to be properly calibrated and used by means of look-up tables for the engine simulation. A third approach consists of the utilization of HRR (heat release profiles) as input quantities. During the simulation, the HRR profiles can be interpolated by means of look-up tables as a function of significant engine parameters, such as the load and the speed. Finally, the last approach involves a complex multizone model in which the spray is discretized into axial and radial volumes, and to which the energy and mass conservation equations are applied; although accurate, this approach requires several calibration parameters and increases the computational time to a great extent. GT-POWER also allows the implementation of homemade models for the prediction of the in-cylinder combustion.

In the present paper, the 0D combustion model presented by the authors in ([9, 10, 11]) was implemented in the GT-POWER code and compared to a standard approach based on the interpolation of the experimental HRR profiles. The model was tuned on 12 steady-state operating conditions which cover the whole engine map, and validated over two different load transient conditions at 1100 rpm and 1400 rpm.

ENGINE SETUP AND EXPERIMENTAL ACTIVITY

The main engine specifications are reported in [Tab. 1](#). The engine is six cylinder 8.7 dm³ heavy-duty diesel engine,

equipped with a Common Rail injection system and a VGT turbocharger.

Table 1. Engine specifications.

Displacement	8.7 dm ³
Stroke	135 mm
Bore	117 mm
Connecting Rod	200 mm
Compression ratio	16.5:1
Number of cylinders	6
Injection system	Common rail, Bosch CRIN3, 1600 bar
Turbocharger	VGT type

The experimental data were acquired at a stationary test bench at CRF (Fiat Research Centre), in the frame of the CORE (CO₂ reduction for long distance transport) European Research Project.

The test engine was instrumented with piezoresistive pressure transducers and thermocouples to measure the pressure and temperature at different locations, such as upstream and downstream from the compressor, turbine and intercooler, and in the intake manifold.

Thermocouples were also used to measure the temperatures in each of the six exhaust runners. Piezoelectric transducers were installed to measure the pressure time-histories in the combustion chamber of the six cylinders.

12 steady-state operating conditions were acquired for the model calibration, at four different engine speeds (800 rpm, 1100 rpm, 1400 rpm and 1700 rpm) and three different load conditions, corresponding to the 25%, 50% and 100% of the maximum torque which is supplied by the engine at each considered speed.

The injection strategy was made up of a pilot and a main injection for all the conditions, except for medium and high load conditions at 1700 rpm. External EGR was not used in the considered operating points, NO_x emissions being controlled by a retarded injection strategy, therefore the intake charge oxygen concentration was constant and equal to that of fresh air.

Table 2 shows a summary of the 12 analyzed steady-state conditions. For confidentiality reasons, the torque and rail pressure values were normalized to a reference value, whereas the SOI_{pil} and SOI_{main} parameters were expressed as differences with respect to the case 1 (a negative angle indicates a delayed SOI).

For the model validation, two different load transient conditions were acquired, at 1100 rpm and 1400 rpm respectively. During the transient at 1100 rpm, the position of the acceleration pedal (denoted with "Alpha") varied from 37% to 100% in 2 s, whereas at 1400 rpm it varied from 48% to 100% in 2 s.

Table 2. Summary of the steady state operating conditions.

Case	N	Torque	p _{rail}	q _{pil}	SOI _{pil}	SOI _{main}
-	rpm	-	-	mm ³	deg	deg
1	1700	0.14	0.66	1.9	0	0
2	1700	0.27	0.66	0	-	1.1
3	1700	0.54	0.73	0	-	2.5
4	1400	0.14	0.52	1.9	-2.8	-0.8
5	1400	0.28	0.55	1.9	-3.4	-1.2
6	1400	0.56	0.63	1.9	-5.7	-3.5
7	1100	0.14	0.45	1.9	-7.2	-3
8	1100	0.28	0.46	1.9	-8	-3.8
9	1100	0.56	0.49	1.9	-9.6	-5.1
10	800	0.10	0.31	1.9	-9.9	-3.7
11	800	0.20	0.39	1.9	-	-4.3
12	800	0.40	0.43	1.9	-	-8.5

During the transients the VGT controller was deactivated, and the rack position of the turbine was kept constant and equal to the nominal value optimized for the steady state engine operating condition at the end of the transient.

External EGR was not employed and the injection strategy was made up of a pilot injection shot and a main injection shot, except for the transient at 1400 rpm, where the pilot injection was deactivated when Alpha reached a value equal to 100%.

Table 3 reports a summary of the two transient conditions. The torque values were normalized to a reference value for confidentiality reasons.

Table 3. Summary of the transient operating conditions.

Transient	N	Alpha	Torque	EGR
-	Rpm	[-]	[-]	[%]
1	1100	From 37% to 100% in 2s	Min:0.140 Max:0.546	0
2	1400	From 48% to 100% in 2s	Min:0.140 Max:0.555	0

1D ENGINE SIMULATION MODEL

The engine was modeled at ICEAL-PT (Internal Combustion Engines Advanced Laboratories) of Politecnico di Torino with GT-POWER v7.0, a 1D simulation tool licensed by Gamma Technologies ([8]). The model has been built according to the geometry marked by the technical drawings of the engine. Each system component has been properly characterized in terms of its specific geometric features such as length, diameter, bend radius, etc. Moreover, joints and connections have been carefully designed in order

to account for the concentrated losses induced by the sudden changes in the flow section. The distributed losses have in turn been estimated by a suitable setting of the pipes surface roughness.

The geometric features of the inlet and exhaust pipes were accurately measured at the engine test bench at CRF, in order to correctly reproduce the real experimental setup of the experimental tests which were used for the model calibration.

HRR SIMULATION MODEL

The HRR simulation model was presented by the authors in ([9, 10, 11]). The approach is based on the assumption that, at each time instant, the rate of released chemical energy (dQ_{ch}/dt) is proportional to the energy associated with the fuel mass available in the combustion chamber ([12, 13, 14, 15]). The latter can be computed as the difference between the chemical energy of the injected fuel quantity (Q_{fuel}) and the released chemical energy (Q_{ch}).

The energy associated with the injected fuel is not evaluated at instant t , but at instant $(t - \tau)$, τ being an ignition-delay correlated quantity. Furthermore, the contribution of each injection pulse (j) is resolved.

In fact, each injection event is characterized by its own ignition delay and heat release rate, according to the following equation:

$$\frac{dQ_{ch,j}}{dt} = K_j \left[Q_{fuel,j}(t - \tau_j) - Q_{ch,j}(t) \right] \quad (1)$$

where:

$$Q_{fuel,j} = \int_{t_{SOI,j}}^t \dot{m}_{f,inj} LHV dt \quad t \leq t_{EOI,j} \quad (2)$$

$$Q_{fuel,j} = \int_{t_{SOI,j}}^{t_{EOI,j}} \dot{m}_{f,inj} LHV dt \quad t > t_{EOI,j} \quad (3)$$

$$Q_{ch} = \sum_{j=1}^n Q_{ch,j} \quad (4)$$

K can be interpreted as a spray or atomization characteristic number ([13]). As K increases, the droplet size reduces and/or spray penetration is enhanced, thus the combustion rate is increased. For a given spray, K can be related to the swirl ratio or to the turbulence ([12]), according to the assumption that the rate of the fuel oxidation is determined by the rate of mixing of fuel vapor and fresh gas and thus by the local density of the turbulent kinetic energy [15].

The combustion rate coefficient of the main injection, i.e., K_{main} , can assume two distinct values corresponding to the premixed and diffusive phases respectively, as follows:

$$K_{main} = K_{main,p} \quad (t < t_{SOC,main} + \tau_{main,diff}) \quad (5)$$

$$K_{main} = K_{main,diff} \quad (t \geq t_{SOC,main} + \tau_{main,diff}) \quad (6)$$

This is a novelty with respect to the model proposed in [9, 10, 11], and was introduced in order to obtain a more accurate prediction of the premixed and diffusive burning phases of the main injection.

A similar approach to that proposed by the authors is the one presented in [16], which is based on the characteristic combustion time. In that case, laminar and turbulent timescales were considered for the modeling of the premixed and diffusive combustion. Moreover, the model proposed in [16] also suggests that crank-angle dependent combustion rate coefficients could be considered for improving the accuracy of the model. The author intend to investigate this approach in future extensions of their model.

The K_j and τ_j coefficients have to be appropriately tuned on the basis of the Q_{ch} curves derived from the experimental in-chamber pressure trace.

The fuel burned mass fraction $x_{b,f}$ can be derived from the Q_{ch} curve as follows:

$$x_{b,f} = \frac{Q_{ch}}{m_{f,inj}^* LHV} \quad (7)$$

where LHV is the fuel lower heating value and $m_{f,inj}^*$ is the total injected fuel mass per cycle/cylinder.

The approach requires the injection rate as an input. In [9, 10, 11] a simple triangular injection rate was considered for a light-duty engine. For the engine considered in the present paper, it was necessary to introduce a trapezoidal shape for high energizing time values, in order to obtain a better matching between the predicted and the experimental injection rate profiles. The simplified injection rate is built taking into account the nozzle opening delay (NOD), the nozzle closure delay (NCD), the slope of the injection profile and the maximum value of the injection rate. These quantities are available through specific correlations, which are functions of the energizing time (ET) and injection pressure, worked out on the basis of the injector experimental data.

For a generic injection shot, the model starts from the rail pressure, SOI (Start of Injection), ET (Energizing Time) and DT (dwell time), which are known from the engine maps, then evaluates the NOD, NCD, the opening slope and the maximum rate of injection, and finally calculates the simplified injection law.

If the time integral of the simplified injection rate does not match with the injected fuel quantity (which is known), the injection rate is scaled accordingly.

The application of Eqs. (1, 2, 3, 4, 5, 6) to a simple injection rate distribution, consisting of one pilot and one main injection, is given in Fig. 1.

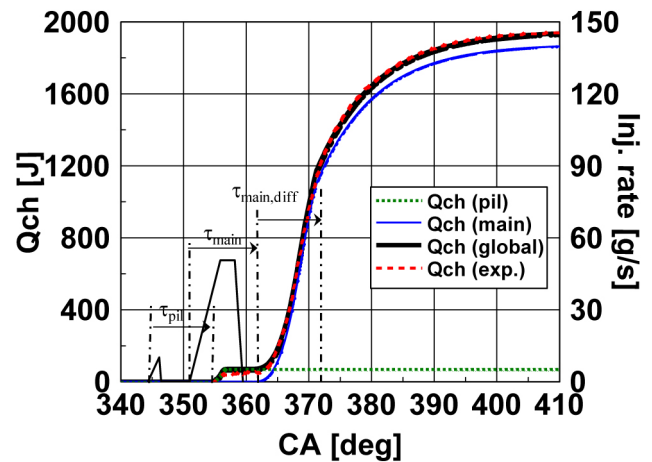


Figure 1. Injection rate and predicted vs. experimental values of Q_{ch} .

Figure 1 reports the injection rate (black thin line), the predicted Q_{ch} curves of the pilot and main pulses (green dotted line and blue thin line respectively), the predicted global Q_{ch} curve (black thick line) and the experimental Q_{ch} curve (red dashed line). The experimental Q_{ch} curve was evaluated on the basis of the measured in-cylinder pressure, by means of a two zone diagnostic combustion model embedded in GT-POWER.

A simple triangular shape was taken to simulate the injection rate of the pilot pulse, whereas a trapezoidal shape was considered for the injection rate of the main pulse. The time integral of the injection rate of each pulse is equal to the total injected fuel quantity for the same pulse, which is known.

In Fig. 1 the τ_{pil} , τ_{main} and $\tau_{main,diff}$ parameters are also reported for a better understanding.

The model calibration parameters K_{pil} , $K_{main,p}$, $K_{main,diff}$, τ_{pil} , τ_{main} and $\tau_{main,diff}$ were properly calibrated in order to obtain the best matching between the predicted and experimental fuel burned mass fraction, for the 12 steady-state operating conditions. This was carried out by means of the least square method. The identified set of optimal values was then analyzed in order to identify physical correlations to be implemented in the combustion model.

MODEL CALIBRATION

1D Engine Model

The calibration of the engine 1D model was carried out according to the procedure reported in [1, 2, 3].

It is generally accepted that 1D models need a careful calibration procedure in order to obtain accurate results ([1, 2, 3, 4, 8]), especially for turbocharged engine applications. In fact, due to their combined one dimensional and zero-dimensional nature, these models are characterized by a simplified description of many physical phenomena, such as: wall friction and heat transfer, concentrated losses, flow

through the valves, fuel injection, and so on. Thus, they are generally not capable of describing the engine fluid-dynamics accurately, unless semi-empirical input data are provided by means of the model calibration, which is usually carried out by adjusting specific model coefficients in order to accurately reproduce the experimental measurements taken at selected steady-state operating conditions.

The accurate prediction of the in-cylinder pressure trace is mandatory for a good estimation of the engine indicated power, which in turn is fundamental to achieve the correct value of the brake power and torque. To this end, the experimental in-cylinder pressure is the main reference data for the calibration of combustion and in-cylinder heat transfer models.

First, the measured in-cylinder pressure signals for the 12 steady state operating conditions were imposed in the engine model, which was simplified by replacing the intercooler outlet and the turbine inlet by two environments in which pressure, temperature and fluid composition were properly set on the basis of the measured data. This was done in order to calibrate the in-cylinder heat transfer independently of the turbocharger system. To this end, the diagnostic tool embedded in GT-POWER ('EngBurnRate' template) provides a few parameters that allow the heat-transfer calibration to be assessed. In particular, the heat transfer multiplier can be calibrated to minimize both the deviations of the so-called fuel 'lower heating value multiplier' (LHVM) from unity and the 'cumulative compression heat release'. The embedded diagnostic tool also allowed the fuel burned mass fraction to be calculated, which were used for the calibration of the heat release rate simulation model. Once the heat transfer model was calibrated at all the steady-state engine working conditions, the related heat-transfer multiplier values were organized in look-up tables and then included in the engine model.

The calibration of the model including the turbocharger system was made in three steps. As a matter of fact, it is not advisable to tune the entire model, consisting of the engine and the turbocharger, from the beginning ([1, 2, 3]). For this reason, an engine model including the compressor but without the turbine was built first. The turbine inlet ports were replaced by two environments in which pressure, temperature and fluid composition are properly set on the basis of the measured data. The compressor speed was identified in order to match the predicted and experimental boost pressure for the considered operating conditions. Then, a model including the turbine but without the compressor was built, and the turbine speed was imposed equal to that calibrated in the model with the compressor. The turbine mass flow multiplier (MFM) ([1, 2, 3]) and the rack position of the VGT were tuned in order to correctly simulate the pressure at the turbine inlet ports. Finally, the complete model including the compressor and the turbine was built, and the turbine efficiency multiplier (EM) ([1, 2, 3]) was calibrated in order to correctly simulate all the relevant

thermodynamic quantities at the engine intake and exhaust, as well as the turbocharger speed.

At the end, the calibrated values of the turbine MFM and EM were organized in look-up tables and included in the engine model.

HRR Simulation Model

The calibration of the HRR simulation model was carried out by tuning the calibration parameters K_{pil} , K_{main} , $K_{main,diff}$, τ_{pil} , τ_{main} and $\tau_{main,diff}$ in order to match the predicted fuel burned mass fraction profiles with the experimental ones, obtained from the GT-POWER embedded diagnostic tool, for the 12 steady state operating conditions reported in Tab. 2.

The set of the calibrated model parameters was then analyzed in order to derive physical correlations as a function of significant engine variables, following an approach which is similar to that presented in [9, 10, 11].

With reference to the ignition delay parameters τ_i , it has to be pointed out that ignition delay is made up of a physical and a chemical contribution ([17]). The former is the time required for the fuel break-up, evaporation and mixing with the in-cylinder surrounding gas. The latter is the time required by the precursor chemical reactions, which occur during and after the mixing stage. The physical contribution mainly depends on the fuel properties, injector type, combustion chamber design, injection pressure, charge thermodynamic state, and in-chamber turbulence. The chemical contribution strongly depends on the fuel properties (cetane number), charge thermodynamic state and charge composition, with specific reference to the oxygen concentration.

The prediction of the ignition delay of the pilot injection is simpler than that of the main injection, as the in-cylinder thermodynamic state at the SOI is known for the pilot shot. Moreover, considering that the SOI of the pilot injection is usually advanced, the physical contribution of ignition delay is predominant with respect to the chemical one, and an accurate correlation can be worked out as a function of the in-cylinder density ([11]). On the contrary, the in-chamber conditions at the SOI of the main pulse is strongly affected by the pilot burning, and the physical and chemical contributions to ignition delay are of the same order of magnitude, therefore a more complex correlation is required to correctly estimate τ_{main} ([11]). Finally, the parameter $\tau_{main,diff}$ was identified by analyzing the experimental fuel burned mass fraction, and identifying the instant at which a change in the heat release derivative occurs at the end of the premixed phase.

With reference to τ_{pil} , Fig. 2 shows that a clear trend exists as a function of the in-cylinder density calculated at SOI. The latter was calculated assuming that the in-chamber thermodynamic conditions at BDC are equal to those in the intake manifold (MAP, T_{int}), as follows: