EVALUATION OF MECHANICAL LOSSES AND ANALYSIS OF THE
INFLUENCE OF EGR AND TURBOCHARGER CONTROL ON
HEAT RELEASE RATE IN AN AUTOMOTIVE DIESEL ENGINE

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Abstract

A wide experimental programme was developed on the engine test bench of the Dipartimento di Macchine, Sistemi Energetici e Trasporti of the University of Genova, in order to analyse the influence of turbocharger and exhaust gas recirculation control on combustion characteristics of an automotive turbocharged DI Diesel engine and to develop correlation for the evaluation of heat release rate and mechanical losses. Indicated pressure diagrams were measured in different engine part and full load operating conditions, for different settings of turbine waste-gate and EGR valves opening degrees, and heat release rate evaluation allowed to outline the influence of these control variables.

Experimental heat release rates were then used for the development of a single zone combustion correlation, whose coefficients are expressed as functions of engine operating parameters such as engine speed, gas-fuel ratio and ignition delay, while in-cylinder pressure diagrams allowed to evaluate indicated mean effective pressure values, mechanical losses and efficiency; mechanical losses were then linked to mean piston speed and in-cylinder maximum pressure in a simple correlation for their calculation.

In the paper the analysis of the influence of turbocharger and EGR control on combustion is focused, while correlation are proposed, comparing calculated values of heat release, in-cylinder pressure and mechanical losses to experimental data.

Introduction

Notwithstanding internal combustion engine (ICE) technology is well established, further developments are still possible both in spark ignition and Diesel engines, taking into account that different technical solutions (electric and hybrid propulsion, fuel cell, etc.) seem not to be able to substitute ICE in its supremacy in automotive applications, at least for the next ten years [1, 2, 3, 4, 5]. On the other hand, in order to comply with the new limits of European legislation on pollutants emissions (EURO IV phase, in force in 2005) and with the requirements related to fuel consumption and CO\textsubscript{2} emissions reduction [6], updated configurations are being developed, with interesting breakthrough technologies, such as gasoline direct injection (DI) and electronically controlled Diesel fuel injection system (FIS) [1, 2].

In automotive field, DI Diesel engines are a widespread alternative, due to their low fuel consumption and to the improvement in torque, driveability and acoustic comfort allowed by FIS based on the common rail technology [7, 8]; specific solutions are also required, combining intake and exhaust components usually fitted, such as turbochargers and exhaust gas recirculation (EGR) system, with new devices for emissions control, i.e. deNO\textsubscript{x} catalysts and particulate filters. These elements will need integrated control strategies in order to achieve proper operating conditions and to fulfil different goals, in terms of performance, fuel consumption and emissions, taking also into account the interactions with other engine components, like FIS, and the characteristics of the regulating system, such as waste-gate (WG) valves or variable geometry (VG) devices usually fitted on automotive turbocharger turbines.

The development of new engine configurations with the relevant control strategies requires wide experimental and theoretical studies: a research on this subject is being developed at the Dipartimento di Macchine, Sistemi Energetici e Trasporti (DIMSET) of the University of Genova since many years [9, 10, 11, 12], with reference to different automotive turbocharged DI Diesel engines, equipped with mechanically or electronically controlled FIS, fitted with VG or WG turbochargers turbine and EGR system. Experimental activity is developed on a test facility which allows to evaluate operating parameters of both the engine and the turbocharger [9, 10], while theoretical investigations are focused on the development of a simulation model of the engine-turbocharger matching, suitable for control applications [11, 12].
In the paper, the main results of a wide experimental programme aimed at the analysis of the influence of turbocharger and EGR control on combustion characteristics of a DI Diesel engine and at the development of correlation for the evaluation of heat release rate and mechanical losses are presented and discussed. Indicated pressure diagrams were measured in different engine operating conditions, with particular reference to ECE15+EUDC driving cycle, for different settings of WG and EGR valves opening degrees, and heat release rate evaluation allows to point out the influence of these control variables.

Experimental heat release rates were also used for the development of a single zone combustion correlation, whose coefficients are expressed as functions of engine operating parameters (i.e., engine speed, gas-fuel ratio and ignition delay), while in-cylinder pressure diagrams allowed to evaluate indicated mean effective pressure values and, through comparison to brake mean effective pressure levels, mechanical losses and efficiency; mechanical losses were then linked to mean piston speed and in-cylinder maximum pressure in a simple correlation for their calculation.

In the paper the analysis of the influence of turbocharger and EGR control on combustion is focused, while correlation are proposed, comparing calculated values of heat release, in-cylinder pressure and mechanical losses to experimental data.

1. Experimental set-up and operating conditions

The experimental activity was developed on an automotive DI Diesel engine, with a displacement of about 1.9 litre, fitted with a mechanical distributor injection pump, an exhaust turbocharger with a waste-gated turbine, an intercooler and an uncooled EGR system. Measurements were performed through a dedicated test bench [10], equipped with an eddy current dynamometer, exhaust gas analysers for carbon monoxide and dioxide, unburnt hydrocarbons and nitrogen oxides, a variable sample smoke meter, an automatic data acquisition and processing system and two independent pneumatic circuits for turbocharger and EGR system control, which allow to set waste-gate valve opening degree (A, equal to the ratio between the displacement of the relevant driving rod and its total displacement) and EGR rate (f_{EGR}, defined as the mass flow of recirculated gas divided by the total mass flow), which is evaluated as the ratio between intake and exhaust carbon dioxide concentrations [13].

For in-cylinder pressure diagrams and crank angle measurements, a cooled piezoelectric pressure transducer, connected to an amplifier, and a photoelectric incremental encoder were fitted on the engine: their signals were sampled by a digital storage oscilloscope, managed by a personal computer, which stored and processed experimental values. Data processing, performed through software developed at DIMSET in Fortran or in Matlab® environment, allowed to evaluate absolute pressure levels setting a proper reference value [14] and to filter pressure signals through a numerical procedure based on moving means centred on three values [15], in order to eliminate noise due to the pressure waves in the measurement duct between the sensor and the combustion chamber.

In each operating condition, the main engine operating parameters (i.e., engine rotational speed, brake torque and power, air and fuel mass flow rates, volumetric concentrations of exhaust pollutants, smoke, EGR rate, WG opening degree, etc.) were measured together with the in-cylinder pressure diagrams.

The experimental activity was developed in two step: in a first phase, measurements were performed in order to study engine mechanical losses and to define an empirical correlation to calculate friction mean effective pressure (fmep) as a function of mean piston speed (um) and in-cylinder maximum pressure (p_{MAX}); therefore pressure diagrams were measured considering different levels of engine speed (n = 1500 ÷ 4000 rev/min, with a step of 500 rev/min) and load, starting from a brake mean effective pressure (bmep) of 0.2 MPa up to its maximum level. Several engine operating conditions with bmep lower than 0.2 MPa were also considered; in this step, WG and EGR valves were kept closed.

The second step was focused on the analysis of the combustion process, with particular reference to the influence of EGR and turbocharger control on in-cylinder pressure and rate of heat release, and on the definition of a combustion correlation for the evaluation of heat release as a function of the main engine operating parameters. Selected experimental operating conditions referred to this phase are shown in fig.1, expressed through the relevant bmep and n values: a greater importance was given to engine on-vehicle use and to EGR system activation, thus preferring points at part load and low-medium engine speed. Exhaust gas recirculation technique is not generally applied to the whole engine operating field, because the reduction of inlet air mass flow rate (M_i) and air-fuel ratio (AFR) increases smoke; this effect is more significant at high engine speed and load, therefore when these parameters grow, EGR is reduced and then excluded, also in order to avoid any influence on engine maximum performance [9].

With reference to the considered engine, EGR operating field is very close to that defined through the ECE15 + EUDC driving cycle [9], for which fifteen equivalent engine steady-state points were calculated in a
previous phase of DIMSET research activity, taking account of a specific car [10]; transient cycle modes were considered in terms of average engine speed and bmep values. Part load experimental points were therefore selected between those of the ECE15 + EUDC driving cycle, or setting one of the engine regulating parameters to a constant level (bmep = 0.2 and 0.4 MPa, n = 2000 rev/min); in these operating conditions, pressure diagrams were measured considering four different EGR rate (including fEGR = 0), imposing a smoke limit of 5 FSN (Filter Smoke Number, ranging from 0 to 10) to set maximum EGR rate.

In order to deepen the influence of turbocharger control on combustion, measurements were performed also for three WG opening degrees (A = 0, 15 per cent and 45 per cent) in several points. Finally other experimental conditions were selected outside EGR operating field, at constant engine speed (n = 2000 and 3000 rev/min) and at maximum load (fig.1).

2. Upgrade of a correlation for mechanical losses evaluation in Diesel engine

The evaluation of engine mechanical losses is a typical application of in-cylinder pressure diagrams measurements; generally, they are expressed in terms of friction mean effective pressure (fmep):

\[ fmep = imep - bmep \]

where indicated mean effective pressure (imep) is given by the ratio between indicated work (i.e., the area enclosed by the pressure trace on (p, V) plane, where V is the instantaneous cylinder volume) and cylinder displacement, while bmep is measured through a dynamometer. Pumping losses (due to the fact that exhaust pressure was always higher than inlet pressure for the considered engine) were included in imep calculation: therefore fmep represents the 'true engine friction' [16], that is friction due to crankshaft, piston, camshaft, etc. and engine driven accessories.

The experimental work described in point 1 allowed to compute fmep levels for a wide range of engine operating conditions and to define a simple correlation for their evaluation, suitable for the DIMSET engine-turbocharger simulation model, which requires short calculation time [11, 12]. The selected correlation is commonly applied to Diesel engine [17, 18] and allows to estimate fmep as a linear function of mean piston speed \( u_m \) and cylinder maximum pressure \( p_{MAX} \), superimposing the effects of speed and load:

\[ fmep = a + b \cdot u_m + c \cdot p_{MAX} \]

where fmep and \( p_{MAX} \) are expressed in [MPa] and \( u_m \) in [m/s]; the static term “a” represents accessories losses [17]. The updated coefficients (\( a = 0.0153 \), \( b = 0.0176 \) e \( c = 0.0159 \)) were evaluated on the basis of experimental data through a simple least-squares linear regression, obtaining a correlation coefficient \( R^2 = 0.93. \)

To compare different formulae of correlation (2) [17, 18], the relevant coefficients are reported in table 1, together with the main characteristics of the engines and the operating conditions used by the Authors for their definition. Engines and considered experimental ranges of speed and load (i.e., mean piston speed and maximum cylinder pressure) are quite different: as a consequence calculated parameters show significant differences, especially with reference to static term “a” and to the value of coefficient “b” proposed by Winterbone and Tennant [17]; the pressure coefficient is always positive, therefore an increase in peak pressure results in higher friction losses.

To complete this analysis, fmep levels evaluated with three formulae of eq.2 are compared in fig.2 with reference to experimental values, considering increasing bmep at constant speed (n = 2000 rev/min, fig.2a) and increasing engine speed at constant load (bmep = 0.4 MPa, fig.2b). Chen and Flynn 2nd correlation (which was previously used in the simulation code [11]) allows to estimate friction losses with acceptable shifts, even if calculated values are generally higher than the measured ones, while levels obtained with Winterbone and
Tennant correlation are significantly lower; it may be concluded that simplified correlation must be suited to the specific engine, because extrapolation of data between different engines may lead to unreliable results.

<table>
<thead>
<tr>
<th>Chen &amp; Flynn 1st correlation [17]</th>
<th>a = 0.0138</th>
<th>b = 0.0164</th>
<th>c = 0.005</th>
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<tbody>
<tr>
<td>• Diesel single cylinder</td>
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<tr>
<td>• ( n_{\text{MAX}} = 3200 \text{ rev/min} )</td>
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<tr>
<td>• Maximum imep = 1.8 MPa</td>
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<tr>
<td>• ( p_{\text{MAX}} = 20 \text{ MPa} )</td>
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<tr>
<td>Chen &amp; Flynn 2nd correlation [17]</td>
<td>a = 0.1172</td>
<td>b = 0.0164</td>
<td>c = 0.010</td>
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<tr>
<td>• As in 1st correlation</td>
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<tr>
<td>• Engine driven accessories</td>
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<tr>
<td>Winterbone &amp; Tennant [17]</td>
<td>a = 0.0061</td>
<td>b = 0.0098</td>
<td>c = 0.0155</td>
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<tr>
<td>• Diesel DI, turbocharged</td>
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<tr>
<td>• 6 cylinder; ( V_t = 8.2 \text{ dm}^3 )</td>
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<tr>
<td>• ( \rho = 15.4 : 1 )</td>
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<tr>
<td>• ( B = 118 \text{ mm}; S = 125 \text{ mm} )</td>
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<tr>
<td>• Maximum brake power = 190 kW @ 2400 rev/min</td>
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<tr>
<td>• Maximum bmep = 1.3 MPa @ 1800 rev/min</td>
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<tr>
<td>DIMSET</td>
<td>a = 0.0153</td>
<td>b = 0.0176</td>
<td>c = 0.0159</td>
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<tr>
<td>• Diesel DI, turbocharged</td>
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<tr>
<td>• 4 cylinder; ( V_t = 1.929 \text{ dm}^3 )</td>
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<tr>
<td>• ( \rho = 19.8 : 1 )</td>
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<tr>
<td>• ( B = 82.6 \text{ mm}; S = 90 \text{ mm} )</td>
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<tr>
<td>• Maximum brake power = 58.6 kW @ 4000 rev/min</td>
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<tr>
<td>• Maximum bmep = 1.4 MPa @ 2000 rev/min</td>
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Table 1 – Comparison of different formulae of correlation \( fmep = a + b \cdot u_m + c \cdot p_{\text{MAX}} \)

Fig.2 – Comparison of experimental and calculated values of friction mean effective pressure

3. Influence of EGR and turbocharger control on heat release rate

Turbochargers and exhaust gas recirculation systems are usually fitted to automotive Diesel engines in order to improve specific power and torque and to reduce NO\(_x\) emissions, respectively; their influence on engine performance and emissions are quite known [19, 20, 21], even if some aspects have to be further investigated and their potential is not fully exploited, since the application of electronics offers new possibilities of improving their control, also in an integrated way [10, 12], and the related effects on engine.

As it is well known, EGR leads to the substitution of part of inlet oxygen with carbon dioxide and water (in vapour form): combustion process is then modified, due to an increased thermal capacity of the charge,
Fig. 3 – Influence of turbocharger and EGR control on engine operating parameters (operating condition: \( n = 1550 \text{ rev/min}, \text{ bmep} = 0.2 \text{ MPa} \))
which results in a flame temperature reduction (thermal effect), and to the dilution of the inlet charge with a
decrease in oxygen concentration (dilution effect) [22, 23]; though other phenomena are involved (chemical
effects, changes in inlet charge temperature and in flame structure [22, 23]), thermal and dilution effect are
considered the most significant to explain EGR influence on pollutants emissions variations, reducing NO\textsubscript{x} and
increasing particulate, HC and CO [22, 23, 24, 25].

Air-fuel and gas-fuel ratios (AFR and GFR) are useful to describe in-cylinder charge composition and to
define relative proportion of air, fuel and recirculated gas [20]; with reference to a single engine cycle, GFR is
given by the ratio between the total mass of gas in the cylinder (m\textsubscript{t}) and the injected fuel mass (m\textsubscript{f}) and can be
evaluated through [20]:

\[
GFR = \frac{AFR}{1 + \frac{f_b}{1 - f_r}}
\]

where burned gas fraction (f\textsubscript{b}) is calculated with:

\[
f_b = f_{EGR} \cdot (1 - f_r) + f_r
\]

Residual gas fraction (f\textsubscript{r}) can be estimated on the basis of thermodynamics conditions at engine inlet
and exhaust and it is generally a few percent in Diesel engines [20].

In order to explain the influence of EGR and turbocharger control on pressure diagrams and heat
release, it can be useful to analyse the behaviour of engine and turbocharger parameters measured during
experimental activity, represented in fig.3 considering a part load operating condition (n = 1550 rev/min, bmep
= 0.2 MPa), as a function of EGR rate and for three different WG settings (A = 0, 15 per cent and 45 per cent);
measured data are expressed in terms of relative change, referred to the value obtained for EGR rate = 0 and
A = 0 (i.e., WG fully closed).

From fig.3 it is apparent that the matching condition is affected not only by WG opening but also by
EGR increasing, which have quite similar effect on the different parameters; generally, the most significant
variations are obtained when A changes from 0 to 15 per cent, even if air mass flow rate M\textsubscript{a}, air-fuel ratio AFR
and inlet temperature T\textsubscript{a} are much affected by EGR control. It has to remark that there is not a linear
relationship between A and relative changes in parameters, since the further WG opening from 15 to 45 per
cent has a very little effect, compared to the first step: this is due to the different sensitivity to the turbine
regulating system, which results in a non-linear relationship between control variable A and turbine flow area,
as shown in a previous study [10].

As the regulating device opens at constant EGR rate, the higher turbine swallowing capacity results in
a reduced turbocharger speed (fig.3f), with decreased levels of both turbine inlet and engine intake (i.e., boost)
pressure (p\textsubscript{3} and p\textsubscript{A}, figs.3c and 3a, respectively). Engine air mass flow rate (M\textsubscript{a}, fig.3g) is consequently lower:
since there was no significant variations in fuel mass flow rate, air-fuel ratio shows the same trend (AFR,
fig.3h). A reduction of total engine mass flow rate and gas-fuel ratio (M\textsubscript{t} and GFR, figs.3i and 3l, respectively)
also occurs; turbine inlet temperature increase (T\textsubscript{3}, fig.3d) is probably related to AFR decrease. Finally, lower
values of maximum cylinder pressure (p\textsubscript{MAX}, fig.3e) can be explained considering that p\textsubscript{A} reduction causes
lower pressure levels during the compression stroke; this trend is not modified by heat release during the
premixed phase and the first part of diffusion phase of combustion, which also influence p\textsubscript{MAX}, probably
because the shifts between the different WG settings are small (see fig.5).

Similar considerations can be developed for increasing EGR rate at constant A level: in this case it
is apparent the growth of T\textsubscript{A} (fig.3b), due to the mixing of inlet air with hot recirculated gas, which, together
with AFR decrease, leads T\textsubscript{3} (fig.3d) to higher levels; decreasing trends of M\textsubscript{a} and GFR show lower
gradients with respect to M\textsubscript{f} and AFR, because they take account of EGR mass flow rate, which of course
increases in these conditions.

Combustion ratio (β, equation 5), is defined as
the ratio between fuel mass burned in the premixed
phase of combustion (m\textsubscript{p}) and the total fuel mass
burned in a cycle (m\textsubscript{f}); its values were evaluated
through the processing of experimental heat release
rates described in point 4 and are presented in fig.4,
with reference to the same operating condition, EGR
rates and WG settings of fig.3. Since its definition,
combustion ratio is strictly related to ignition delay and to the peak value of heat release during the premixed phase of combustion: at fixed engine speed and load, in fact, a longer ignition delay (due for example to a reduction in pressure and/or temperature at the start of the fuel injection phase) results in a higher value of $\beta$ (because more fuel mass enters the cylinder during delay, thus increasing $m_p$) and in a higher peak level of heat release at the start of combustion.

At constant EGR rate, increasing A results in a decrease of $p_A$, while $T_A$ is nearly constant (figs.3a and 3b): the higher levels of $\beta$ shown in fig.4 are therefore probably due to lower pressure during compression stroke and injection phase, resulting in longer ignition delay. A different trend can be noted if $f_{\text{EGR}}$ is increased at constant A setting: ignition delay is probably shorter, because inlet charge temperature growth due to the uncooled EGR compensates for inlet pressure diminishing [20, 23], thus resulting in a reduced premixed phase of combustion and lower levels of $\beta$.

These considerations are confirmed by heat release rates, presented in figs. 5 and 6, always referred to the same operating condition ($n = 1550 \text{ rev/min and bmep} = 0.2 \text{ MPa}$): in order to point out the influence of
the control variables, the same data are shown at constant \( f_{\text{EGR}} \) for the three considered WG settings (fig.5) and for fixed A levels with EGR rate as parameter (fig.6). As it might be expected after discussing figs.3 and 4, maximum heat release in premixed phase of combustion grows as WG opens (that is, for higher A levels, fig.5), while it reduces when EGR rate grows: the highest value is obtained for \( A = 45 \) per cent and no EGR, the lowest for EGR rate = 30 per cent and \( A = 0 \).

Combustion duration is longer for lower A settings and higher EGR rates; in the first case the smaller fuel mass burned during premixed phase requires a longer diffusion phase to complete the combustion, while EGR activation probably leads to a change in the flame structure (flame length, number of ignition sites and their spatial distribution, etc. [22]) which results in a reduced velocity of oxidation processes.

Fig.7 – Influence of EGR control on in-cylinder pressure and heat release for three engine speed at constant load

It may be concluded that both turbocharger and EGR control strongly affect engine behaviour and combustion phenomena, while showing an interesting potential to improve engine performance and emissions [10, 12]; anyway, in order to develop a combustion correlation (point 4), in a first step it seemed more significant to take account of EGR influence only, since its effects on combustion process are complex [20, 21, 22, 23, 24, 25]: the wider part of the experimental activity was therefore dedicated to in-cylinder pressure
measurements with different EGR rates in an extended range of engine speed and load (fig.1); in a further step, it could be interesting to investigate if turbocharger control requires to appear more directly in the correlation, as it happens for EGR through gas-fuel ratio GFR (eqs.3 and 4), or its influence could be simply described through the variation of engine operating parameters included in the relevant functions (table 2).

A sample of experimental results is shown in fig.7, with reference to three levels of engine speed (n = 1800, 2250 and 2500 rev/min) at constant load (bmep = 0.4 MPa): in each considered point, EGR increase leads to a reduction of both compression stroke pressure levels and maximum cylinder pressure, to a reduced premixed phase of combustion and to a longer combustion duration. Measured data were then used in developing a combustion correlation for heat release evaluation which takes account of exhaust gas recirculation, described in the following point.

4.A combustion correlation for heat release rate calculation with EGR

The final step of this study aimed at the definition of a combustion correlation to evaluate heat release rate as a function of engine parameters in any specified operating condition, starting from a single zone correlation proposed by Watson and al. [26] with reference to a medium speed Diesel engine not equipped with EGR. Experimental heat release $(\Delta X/\Delta \theta)_e$ was computed by processing measured pressure data for each condition, that is for fixed levels of engine speed, bmep and EGR rate; $\Delta X$ is the amount of heat released during a crankshaft rotation $\Delta \theta$, divided by the total heat released in a cycle: the calculation is based on the first thermodynamics law, applied to the gas in the cylinder, considering a complete combustion and neglecting the variation of mass due to fuel injection; intake charge composition, depending on EGR rate, and in-cylinder changes in components concentrations and temperatures during combustion are taken into account for the evaluation of specific heats and their ratio, based on the thermodynamics properties functions of different gas reported in [27].

Intake air and fuel mass flow rates, measured on the test bench with a viscous flow meter and a gravimetric balance, respectively, were equally divided between the four cylinders; heat release was then evaluated starting from in-cylinder pressure, instantaneous cylinder volume related to each measured pressure level, computed as a function of crank angle and engine geometric characteristics, and heat transfer to the coolant, estimated with Woschni correlation [28].

$(\Delta X/\Delta \theta)_e$ was further processed through a non-linear regression procedure in order to evaluate combustion ratio $\beta$ (defined in point 3) and the coefficients $c_{p1}$, $c_{p2}$, $c_{d1}$, and $c_{d2}$, included in the theoretical heat release function $(\Delta X/\Delta \theta)_t$ proposed by Watson [26]:

$$ (\Delta X/\Delta \theta)_t = \beta \cdot f_p(\theta) + (1 - \beta) \cdot f_d(\theta) $$

(5)

where $f_p(\theta)$ and $f_d(\theta)$ represent combustion premixed and diffusion phases contribution to heat release, expressed respectively by [26]:

$$ f_p(\theta) = c_{p1} \cdot c_{p2} \cdot \left[ \left( \theta - \theta_i \right) / \left( \theta_e - \theta_i \right) \right]^{(c_{p1} - 1)} \cdot \left[ 1 - \left\{ \left[ \left( \theta - \theta_i \right) / \left( \theta_e - \theta_i \right) \right]^{c_{p1} - 1} \right\} \cdot \left[ 1 / \left( \theta_e - \theta_i \right) \right] \right] $$

(6)

$$ f_d(\theta) = c_{d1} \cdot c_{d2} \cdot \left[ \left( \theta - \theta_i \right) / \left( \theta_e - \theta_i \right) \right]^{(c_{d1} - 1)} \cdot \exp \left\{ -c_{d1} \cdot \left[ \left( \theta - \theta_i \right) / \left( \theta_e - \theta_i \right) \right]^{c_{d2}} \right\} \cdot \left[ 1 / \left( \theta_e - \theta_i \right) \right] $$

(7)

Ignition angle $\theta_i$ was evaluated considering the shift between the measured pressure trace and the politropic curve of compression, which has a coefficient variable from 1.25 to 1.35 obtained by experimental data [14]; combustion start can be placed where the two curves diverges significantly [29], due to the initial pressure gradient.

Combustion end angle $\theta_e$ corresponds to the first zero value of heat release: experimental values of combustion duration angle $(\theta_e - \theta_i)$, ranging between 20 and 50 crank angle degrees, are typical of the considered engine [20, 29]. Anyway, in order to simplify the correlation use, combustion duration is kept constant and equal to 60 crank angle degrees: this value was chosen after a comparison of different levels. To the same aim, $c_{d2}$ coefficient is set to a constant value too (equal to 100), calculated as a mean of $c_{d2}$ levels computed in a first series of regressions.

With these simplifications, correlation coefficients $R^2$ obtained processing experimental heat release rates were included in a 0.96 - 0.98 range: a non-linear regression procedure developed in Matlab® and based on function minimization through the simplex method of Nelder-Mead [30] was used to evaluate $\beta$, $c_{p1}$, $c_{d1}$ and $c_{d2}$ with $c_{p2} = 100$; this method is a direct one and it doesn’t require any information on functions gradients.
In the following phase the independent variables of the correlation were selected: many engine parameters were considered (engine speed, air-fuel ratio, gas-fuel ratio, ignition delay, air mass flow rate, bmep, inlet temperature and pressure, ignition pressure) expressing their relationship with $\beta$, $c_{p1}$, $c_{d1}$ and $c_{d2}$ through different analytical function (exponential, power, etc.); the same non-linear regression procedure based on Nelder-Mead simplex method was used in this step. It must be noted that in this experimental work ignition delay was not measured, but it was calculated on the basis of engine injection timing: its values were therefore roughly estimated, because the real injector behaviour was not taken into account.

Since the correlation would be applied in the engine-turbocharger simulation model, the choice of the independent variables (engine speed $n$, gas-fuel ratio GFR and ignition delay $\tau_{id}$) takes account not only of the best data fitting but also of their availability in the simulation code: the relevant functions are reported in table 2, together with $R^2$ values.

<table>
<thead>
<tr>
<th>Functions</th>
<th>$R^2$</th>
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<tbody>
<tr>
<td>$\beta = 0.47 \cdot \exp \left[-1.44 \cdot n^{0.505} / (\tau_{id}^{0.587} \cdot GFR^{1.406})\right] \cdot \beta^{0.176} / (GFR^{0.119} \cdot \tau_{id}^{0.170})$</td>
<td>0.856</td>
</tr>
<tr>
<td>$c_{p1} = 0.577 + 1.042 \cdot n^{0.044} \cdot \beta^{0.176} / (GFR^{0.119} \cdot \tau_{id}^{0.170})$</td>
<td>0.761</td>
</tr>
<tr>
<td>$c_{d1} = 100$</td>
<td>-</td>
</tr>
<tr>
<td>$c_{d2} = 13.86 \cdot n^{0.022} \cdot \beta^{0.607} \cdot GFR^{0.143} / \tau_{id}^{0.193}$</td>
<td>0.847</td>
</tr>
<tr>
<td>$c_{d2} = 0.015 \cdot \exp \left[3.217 \cdot n^{0.039} \cdot c_{d1}^{0.074} / (\tau_{id}^{0.014} \cdot GFR^{0.029})\right] \cdot \beta^{0.176} / (GFR^{0.119} \cdot \tau_{id}^{0.170})$</td>
<td>0.934</td>
</tr>
</tbody>
</table>

Table 2 – Heat release rate correlation

The correlation appears to be more complex than Watson one [26]: this is probably due to EGR, since it seems quite difficult to describe its influence on combustion by a mathematical point of view. Anyway it is possible to evaluate theoretical heat release rate for any considered engine operating condition and, by applying first thermodynamics law, to estimate cylinder pressure during combustion.

![Fig.8 – Comparison between experimental and calculated values of in-cylinder pressure and heat release (operating condition: $n = 2430$ rev/min, bmep = 0.47 MPa)](image-url)
A comparison between experimental and calculated values of heat release and pressure was then performed: results for one part load operating point (n = 2430 rev/min, bmep = 0.47 MPa) are shown in fig.8, with reference to the considered EGR rates; estimated values are in good agreement with measured ones, and the correlation seems to be able to reproduce some of the characteristics due to EGR activation, such as the reduction of peak pressure and of maximum heat release in the premixed phase when \( f_{\text{EGR}} \) increases. The higher shift between experimental and calculated heat release is evident in fig.8d (EGR rate = 24.1 per cent), during the premixed phase and in the first part of the diffusion phase: as a consequence maximum pressure shows the higher difference, being measured \( p_{\text{MAX}} \) equal to 8.91 MPa, while calculated \( p_{\text{MAX}} \) is 8.53 MPa, with a 4.3 per cent error.

This is generally true for each considered operating condition: shift between experimental and calculated values are more significant for the highest EGR rates; a wider range of \( f_{\text{EGR}} \) in the experimental activity could help to improve this aspect of the correlation behaviour.

Notwithstanding measurements were performed on a DI Diesel engine fitted with a mechanical distributor pump and with some limitations in the experimental apparatus, the proposed correlation seems to be quite interesting, because its development allows to deepen some phenomena involved in combustion processes and to define a procedure which can be applied, with proper enhancements, to different engines, in order to evaluate theoretical heat release rate, also in presence of exhaust gas recirculation.

Conclusions

Automotive turbocharged DI Diesel engine are a widespread choice in European market and a further growth of their diffusion is expected in a few years [2]: typical layout are quite complex by now, and simulation models are useful tools in order to study the different aspects involved in their working conditions and to develop updated configurations; particular interest is devoted to control applications models, since components such as fuel injection, turbocharging and EGR systems require proper regulating strategies in order to fulfil the different objectives (performance, fuel consumption and emissions), to exploit their potential and to manage the reciprocal interactions. Since required calculation time is short and a wide range of operating conditions are to be simulated (including different turbine VG or WG settings, EGR rates, etc.), it is essential to describe involved process in a simple but reliable way.

The work described in this paper goes in this direction: with reference to an automotive turbocharged DI Diesel engine, fitted with a mechanical distributor injection pump, an exhaust waste-gated turbocharger, an intercooler and an uncooled EGR system, it was aimed at the development of two correlation, the first to evaluate mechanical losses, in terms of friction mean effective pressure, as a function of maximum cylinder pressure and mean piston speed, the second to calculate heat release rate as a function of engine speed, gas-fuel ratio and ignition delay. These correlation are based on in-cylinder pressure diagrams measurements, developed during a wide experimental activity performed on the engine test bench of the Dipartimento di Macchine, Sistemi Energetici e Trasporti (DIMSET) of the University of Genova.

Moreover, the study allowed to deepen the influence of turbocharger and EGR control on engine operating parameters and on combustion process, which is extensively described in the paper: trend of several quantities (intake and exhaust pressures and temperatures, maximum cylinder pressure, turbocharger rotational speed, intake air and in-cylinder total mass flow rates together with the related air-fuel and gas-fuel ratios) are presented with reference to different WG opening degrees and EGR rates: the analysis of their behaviour permit to justify the observed changes in heat release rate, obtained by processing measured pressure diagrams.

The proposed procedure will be improved and applied to a DI Diesel engine recently installed on DIMSET test bench, which is equipped with an electronically controlled common rail FIS and with an open control system: in-cylinder pressure diagrams will give interesting information on engine control, whose investigation will be extended to other control variables such as fuel injection parameters (i.e., injections number, timing and rate).

Nomenclature

<table>
<thead>
<tr>
<th>Notation</th>
<th>Description</th>
<th>Symbol</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>bmep</td>
<td>brake mean effective pressure</td>
<td>M</td>
<td>mass flow rate</td>
</tr>
<tr>
<td>f</td>
<td>function, mass fraction</td>
<td>P</td>
<td>power</td>
</tr>
<tr>
<td>fmep</td>
<td>friction mean effective pressure</td>
<td>S</td>
<td>stroke</td>
</tr>
<tr>
<td>imep</td>
<td>indicated mean effective pressure</td>
<td>T</td>
<td>temperature [K]</td>
</tr>
</tbody>
</table>
m mass
n engine speed, rotational speed
p pressure
u piston speed
A waste-gate valve opening degree
AFR air-fuel ratio
B bore
EGR exhaust gas recirculation
FSN Filter Smoke Number
GFR gas-fuel ratio

\( A \) waste-gate valve opening degree
\( \beta \) combustion ratio
\( \theta \) crank angle
\( \rho \) volumetric compression ratio
\( \tau \) time
\( \Delta \) variation

Subscript

\( 3 \) turbine inlet
\( a \) air
\( b \) burned
\( d \) delay, diffusion
\( e \) end, experimental
\( f \) fuel
\( i \) ignition
\( m \) mean
\( p \) premixed
\( r \) residual
\( t \) total, theoretical
\( A \) engine intake
\( EGR \) exhaust gas recirculation
\( MAX \) maximum
\( TC \) turbocharger

References


