ISSN 2176-5480

22nd International Congress of Mechanical Engineering (COBEM 2013)
November 3-7, 2013, Ribeirão Preto, SP, Brazil

DETERMINATION OF THE APPARENT HEAT RELEASE FOR DIESEL ENGINES

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Abstract. *The analysis of the apparent heat release rate from an indicating diagram is one of the usual methods to study the thermodynamic characteristics of the combustion process in compression ignition engines. Given the cylinder geometry (bore, stroke, rod length, compression rate), it is possible to obtain the cylinder volume (as a function of crank angle) occupied by the gas mixture, and deduce its angle derivative. The evolution of instantaneous pressure for the cycle is measured in the laboratory. Its angle derivative can be obtained numerically. The objective of this work is to analyze and simulate the behavior of a Diesel Engine using the concept of apparent heat release rate. A computational tool was developed to analyze the apparent heat released for one cylinder in a compression ignition engine, knowing its geometric parameters and the instantaneous pressure. This first tool allows studies on the combustion process (ignition delay, effect of multiple fuel injections, duration of combustion) from experimental instantaneous pressure. In a second stage, another computational tool was developed to forecast the indicator diagram, adopting a user-defined heat release rate. This tool can explore the indicated parameters (indicated mean pressure, maximum combustion pressure, etc.) associated with a given heat release pattern.*

Keywords: *apparent heat release, diesel engines, combustion process, computational tools*

1. INTRODUCTION

Diesel engines have a heterogeneous and complex combustion process. The fuel is introduced as a spray in the compressed air, a two phase flow is established, and each droplet must be heated, evaporated, mixed with air and then suffer auto-ignition. The process is transient and turbulent. Although the jet spray direction, angle and penetration in the chamber, its interaction with swirling air, and droplet size distribution have been studied (Chang and Farrell, 1997; Hung et al., 1997; Kuo et al. 1983), the stochastic nature of the process possess a high degree of difficulty for the modeling of the reacting medium (Heywood, 1980; Ramos, 1989). For the same reasons, the same degree of difficulty is observed for the application of experimental methods to the study the diesel combustion.

One problem which is of practical importance is to correlate the fuel injection characteristics (start of injection, end of injection, injection mass flow, multiple injections, etc.) with combustion characteristics such as ignition delay, rate of pressure rise in the initial phase of the combustion, roughness and noise from combustion, maximum pressure attained, speed of combustion, and effects of the burning rate on the engine performance.

Due to these characteristics, it is important to develop approximate and indirect methods capable to shed some light on combustion process in diesel engines. A method that is often adopted is the determination of the apparent heat release from the combustion reaction. A general view of the rate of heat release during the diesel combustion can be seen in the Fig.1 and is presented in many combustion engine textbooks (Heywood, 1988; Ferguson and Kirkpatrick, 2001; Benson and Whitehouse, 1979).

The whole combustion process is divided in four typical phases: ignition delay, premixed combustion, the mixing-controlled combustion and the late combustion. During the ignition delay the first part of the droplets already in the chamber is heated by the high temperature of the compressed air, begins the evaporation of fuel droplets, the fuel vapors mix with air and the pre-flame reactions occur. The overall temperature in the chamber can decrease and there is no pressure elevation due to combustion, even if there is a considerable amount of fuel in the chamber. The second phase is characterized by a rapid burning rate taking a few crank angles to occur, when the fuel-air mixtures that are in the range of flammability limits, suffer auto-ignition. The rate of the pressure rise due to combustion during this phase is higher when the ignition delay period is longer. The third phase of the combustion is controlled by the rate of air-fuel mixing process. Now the combustion chamber is very hot and the fuel droplets are heated and evaporate very fast. But in the chamber there is an uneven distribution of air, fuel droplets and combustion products. So, the fuel droplets must find the oxygen of the air to react and the rate of combustion slows down. The fourth phase of the combustion is the late combustion, which can continue for a long period in the expansion stroke, sometimes until the exhaust valve begins to open. However, this phase is not important for the pressure build up in the cylinder, although it can be very important to

ISSN 2176-5480

consume incomplete combustion products and reduce the pollutant gases and particulates. The real pattern of the heat release rate is a characteristic of each engine and its operating conditions (speed and load).

The objective of this work is to use the concept of apparent heat release rate for diesel engines and develop two computational tools, with complementary characteristics. The first tool uses experimental data to obtain the heat release rate and then identify the ignition delay, the duration of the combustion and the pattern of the heat release rate for each engine operating condition for which the recorded instantaneous pressure is available. The second tool adopts a given heat release rate pattern and analyses the effect of this pattern on the engine performance. This approach is associated with engine simulations and can also be used to get a curve fit from experimental data.

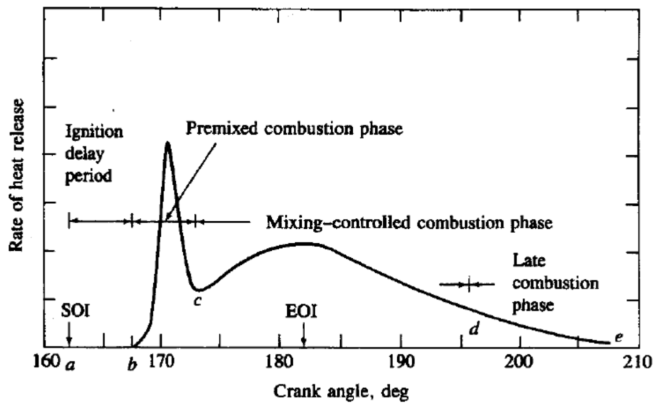

Figure 1. Identification of the phases of the combustion, from heat release rate (Heywood, 1988).

2. THE APPARENT HEAT RELEASE RATE OBTAINED FROM EXPERIMENTAL DATA

The goal of this section is to obtain the heat release rate from the recorded experimental instantaneous pressure, for each given operating condition of the engine. The pressure data were obtained using piezoelectric transducers, with the engine at full load and fifteen engine speeds ranging from 1200 to 4000 rpm. The pressure was recorded for each half degree of crank angle, after statistical treatment. The authors of this work received all these data from an engine manufacturer which asked for anonymity, since these data were obtained from a modified engine running with an experimental fuel.

Thermodynamically, there is no heat released during combustion. The increase of the temperature is due to the change in composition of the reactants (fuel and air) to the products through a reaction with the liberation of the internal energy which is present in the chemical bonds of the fuel molecules. However, the process can also be seen as the release of the heat of combustion of the fuel by each fraction of burning mass. This strategy can avoid the complex chemical description of the combustion process and works with mean fuel properties.

To obtain the heat release rate there are some effects to be considered and assumptions to be made. During the combustion, there is an intense heat transfer from the hot gases to the engine walls, from convection and radiation mechanisms. The thermodynamic model to be used to calculate the combustion heat release can contain explicit heat transfer equations, if the heat transfer is known with some degree of accuracy. Otherwise, the heat transfer effects are contained in the combustion heat release. Although the valves are closed during the combustion, there is also a small escape of mass from the cylinder to the crankcase, called blow-by. This mass loss will not be considered in this work, since its effect is small and it was not measured for the available essays. When the combustion heat release is obtained in this way, it is called *apparent* heat release.

2.1 The apparent heat release rate equation

To develop the calculation of the apparent heat release the starting point is the First law of Thermodynamics for an open system (or control volume), applied in the differential form for the combustion process:

$$\delta Q - \delta W + dm_f h_f = dU_s \qquad (1)$$

where m_f and h_f are the fuel mass and enthalpy, respectively.

The work of boundary displacement can be expressed as:

$$\delta W = pdV \qquad (2)$$

The sensible internal energy (U_s) for the gases in the cylinder can be written as:

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$$dU_s = mc_v dT \tag{3}$$

since the gas in the cylinder can be treated as an ideal gas. If h_f is the sensible enthalpy of the liquid fuel droplet, the energy associated with this term is negligible. The heat in the equation can be expressed as a sum of the real heat transfer from hot gases to the chamber walls (Q_{HT}) and a chemical energy released as heat by the combustion reactions (Q_{CH}):

$$\delta Q = \delta Q_{HT} + \delta Q_{CH} \tag{4}$$

When the heat transfer Q_{HT} is known or can be accurately modeled, the real heat released in the combustion can be determined; otherwise, only the sum can be calculated and represents the *apparent* heat release from combustion.

In the engine, the instantaneous volume and pressure are known, but there is an unknown temperature distribution in the gases during the combustion process. There are regions with compressed air, regions with cold liquid fuel droplets, regions with much hotter burned gases and flame regions with very high temperatures. The differential form of the ideal gas equation can be used to eliminate the temperature in the energy balance:

$$pdV + Vdp = mdT + Tdm \tag{5}$$

The last term of the Eq. 5 can be neglected, since the total fuel mass injected in the cylinder is below 5% of the total mass of air. This approximation is consistent with the order of magnitude of the other assumptions. Substituting the above equations expressed as functions of crank angle in the Eq. 1 and rearranging, the final form of the energy equation is:

$$\frac{\delta Q}{d\theta} = \frac{k}{k-1} p \frac{dV}{d\theta} + \frac{1}{k-1} V \frac{dp}{d\theta} \tag{6}$$

where k is the specific heat ratio (c_p/c_v). The equation can be solved in a crank angle basis, since the instantaneous volume can be calculated as a function of crank angle and the experimental pressure is available also as a function of the crank angle. It is important to note that even with unknown temperatures and total mass this method can determine the apparent heat release and its effects on some performance parameters of the engine.

From the geometry of the cylinder, the instantaneous volume for a given crank angle can be written as (Planells et al, 1995):

$$V(\theta) = \frac{\pi D^2 L}{4(\epsilon-1)} + \frac{\pi D^2}{4}\frac{L}{2}(1-\cos(\theta)) + z\left[1-\sqrt{1-\left(\frac{L}{2z}\right)^2 sin^2(\theta)}\right] \tag{7}$$

where D, L, z and ε are the cylinder diameter, piston stroke, connecting rod length and compression ratio.

The equation (6) can be solved if the derivates dp/dθ and dV/dθ are known. The instantaneous volume equation can be derived analytically and the instantaneous pressure can be derived numerically from experimental data.

2.2 Characteristics of the engine and pressure curves

The engine from which experimental data were obtained has the basic dimensions presented in the Table 1. There is no information related to the valve timing of this engine, nor about the injection timing (start of injection, end of injection, injection rate). Only the pressure as a function of the crank angle is known. Nevertheless, some important conclusions can be drawn from the pressure measurements and from the analysis of the apparent heat release.

The Fig.2 shows the pressure as a function of the crank angle, as well as its derivate, for the engine running at full load and 1800 rpm. Similar graphs are obtained for other engine speeds. The value of the maximum pressure is near 130 bar and the pressure derivate near the combustion suffers extreme variations. Since the maxima values occur during the combustion process, to have details about compression stroke, or about the exhaust stroke, it is needed a change of scale in the graph.

Table 1. Geometric data for the analyzed engine

Geometric parameters		Value
Compression ratio		19
Cylinder diameter	[mm]	96.5
Piston stroke	[mm]	102.5
Connection rod length	[mm]	177.5

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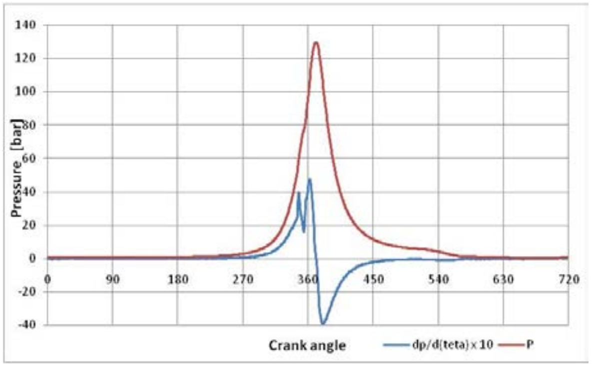

Figure 2. Experimental pressure and pressure derivate as functions of crank angle.
Engine speed: 1800 rpm. Full load.

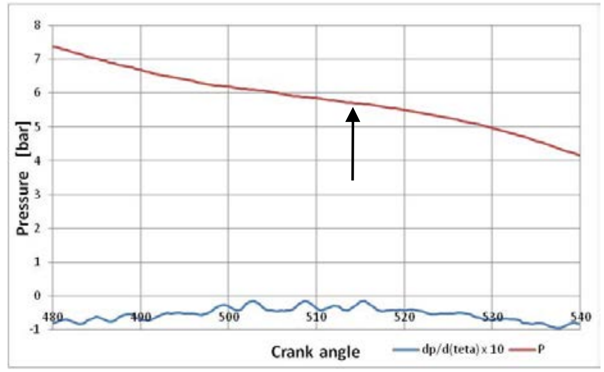

Figure 3. Locating the exhaust valve opening
Engine speed: 1800 rpm. Full load.

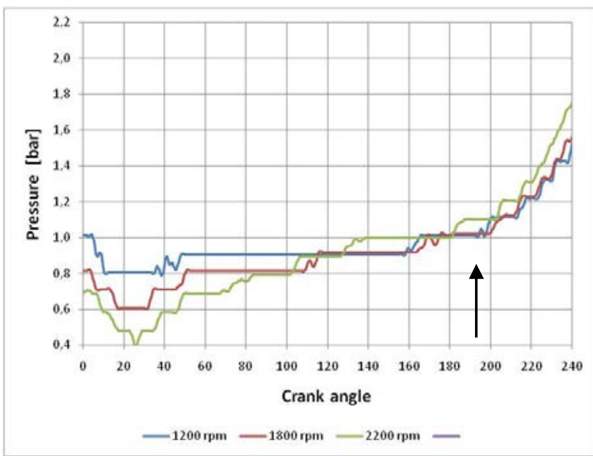

Figure 4. Locating the inlet valve closure

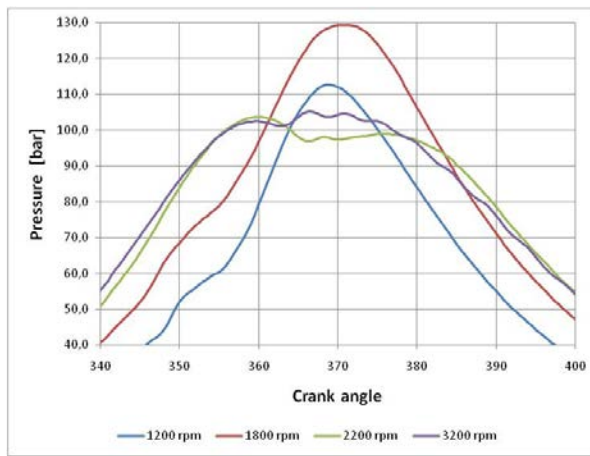

Figure 5. Details of the maximum experimental pressures

A close up view of the pressure graph can reveal that the exhaust valve must open near 515^o crank angle. Figure 3 shows the part of the expansion stroke where the exhaust valve is opened. From the Fig.3 it can be seen a change in the pressure derivate trend associated with the beginning of the blow down through the exhaust valve. A similar analysis of the pressure in the inlet stroke shows that the inlet valve closes at about 195^o crank angle, as shown in the Fig.4.

The Figure 5 shows details of the experimental pressures near TDC (360^o) for four engine speeds. When the engine speed increases, the duration of the combustion process turns the pressure curve into a more flat shape. The mixing-controlled combustion becomes more important since, for a near constant time, the range of crank angle needed to this process increases.

2.3 Apparent heat release results obtained from experimental data

The tabular data of pressure as a function of the crank angle, its time derivative, the cylinder volume as a function of the crank angle and its time derivative were the inputs to heat release rate calculations according to the Eq. 6. The calculations were performed from 250^O to 520^O crank angle. Since the exhaust valve open near 515^O, the results above 500^O crank angle must not be considered. However, as the analysis will show, this is not important.

The Figures 6, 7 and 8 show the apparent heat release for three engine speeds: 1200, 2200 and 3200 rpm. As the engine speed increases, the uncertainty on experimental pressure increases and its derivative acquires an oscillating pattern. This oscillating pattern could be minimized by a mathematical “high frequency” filter like some kind of moving average. Since the objective of this work was to develop analytical tools and not analyze in detail this particular engine, this procedure was not implemented. Apart from this, it is important to see the “pressure noise signal” as it is, to propose experimental procedures to reduce this problem.

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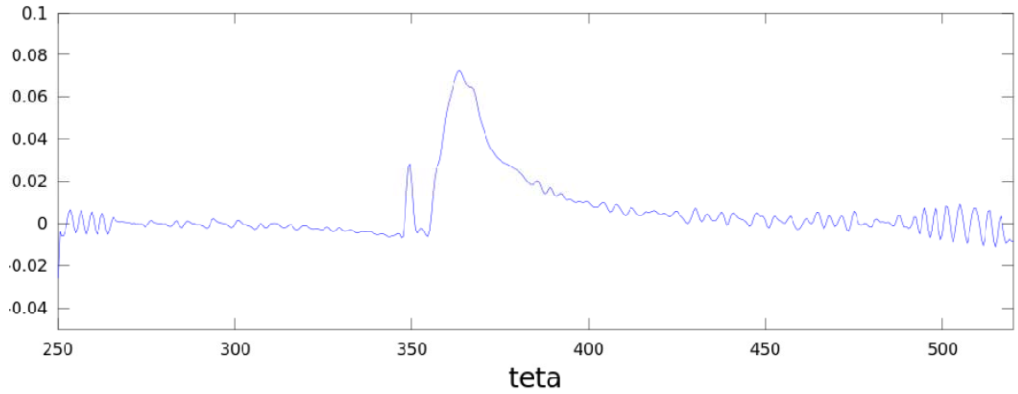

Figure 6. Calculated apparent heat release for 1200 rpm.

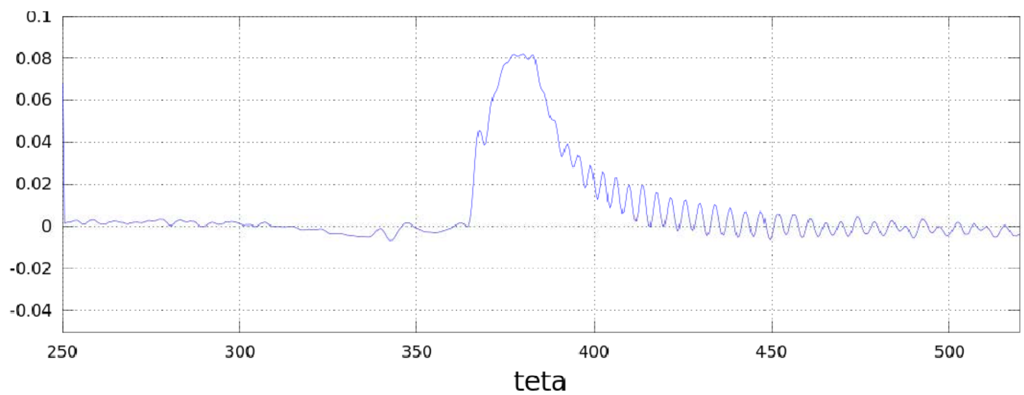

Figure 7. Calculated apparent heat release for 2200 rpm.

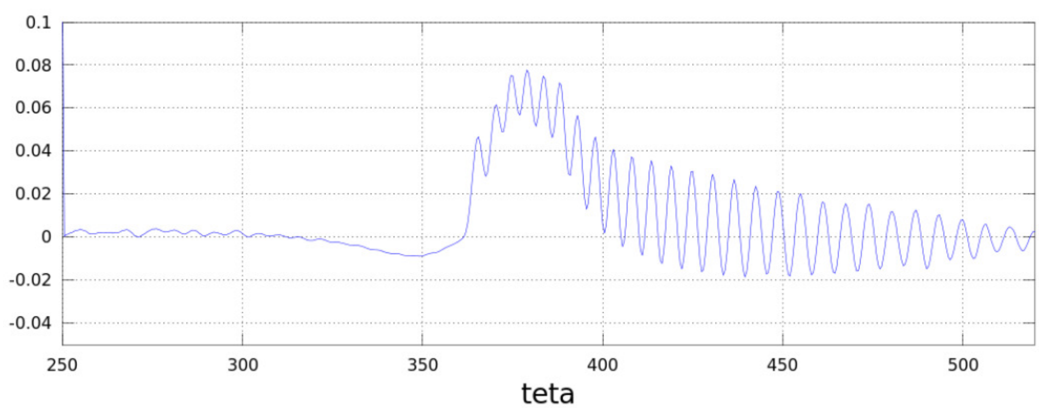

Figure 8. Calculated apparent heat release for 3200 rpm.

The analysis of the above results can show some interesting characteristics: the apparent heat release (AHR) is near zero during most part of the compression stroke. Since the combustion process did not begun yet, this means that great part of compression stroke has low heat transfer effects. The AHR becomes negative in the end of the compression, due to two effects: heat transfer from the hot compressed air to the combustion chamber walls, and the cooling effect of the fuel spray droplets during its vaporization. The sharp first peak seen in the Fig. 6 is the result of the premixed combustion, and the second one is the result of the mixing-controlled combustion. For this engine speed (and the speeds of 1400, 1800 and 2000 rpm, not shown here) this first peak is quite clear. For 2200 rpm the combustion begins later and the first peak seems to merge with the second phase of the combustion. For 3200 rpm the first peak also seems to be the fusion of the premixed combustion with the beginning of the mixing controlled phase, but, since the AHR is highly oscillating, this conclusion is not quite sure. The longer combustion duration and the late peak explain why the 2200 rpm and 3200 rpm pressure curves present a flat shape, as shown in the Fig. 5.

From the AHR is possible to evaluate the duration of the combustion process – at least its part which affects the cylinder pressure evolution. From Fig. 6, the combustion process begins about 347^o crank angle and ends about 433^o crank angle; this means a total duration of about 86^o crank angle. The same analysis for 2200 rpm (Fig.7) shows that the combustion begins latter (365^o) and also ends latter (460^o), with duration of about 95^o crank angle. Finally, for 3200 rpm, the combustion begins and ends almost at the same angles than for 2200 rpm. The analysis of the AHR could also be employed to evaluate the ignition delay of the combustion process, if the point of the start of injection were provided from experimental data.

For a crank angle about 460^O the AHR seems to be reduced to zero again for all three speeds, indicating that the last combustion phase – the late combustion, which does not affect the cylinder pressure – has begun. Comparing the Figs. 6 to 8, it is clear that, at the smaller engine speed, both combustion peaks occur sooner than for the higher speeds. In the last part of the expansion stroke the AHR is essentially zero.

Comparing the above results (specially for the small speed) with experimental results reported in the work of Ferguson and Kirkpatrick (2001) for another engine, the similitude is quite apparent: the same pattern, the first peak occurring before TDC, the second one a little bit after TDC, and even the presence of oscillating values near the end of the combustion.

The mass trapped into the cylinder, as well as the temperatures, were not employed to calculate the AHR. During the combustion process there are different zones in the chamber, with quite different temperatures: regions with pure air at temperatures not so high, regions with burned gases at high temperatures and also flame regions, with the highest temperature. However, an average temperature is frequently needed for heat transfer calculations, or to estimate the exhaust temperatures (and the exhaust gas energy) available for a turbocharger.

To evaluate an average temperature inside the cylinder, the ideal gas equation can be used, if the total mass in the cylinder is known. Adopting reasonable values for the residual gas fraction and volumetric efficiency, the total mass inside the cylinder of the engine was estimated to be 1,324 grams of air. Then, the temperature was calculated as a function of the crank angle. The Fig. 9 shows the results for 1200 rpm experimental data.

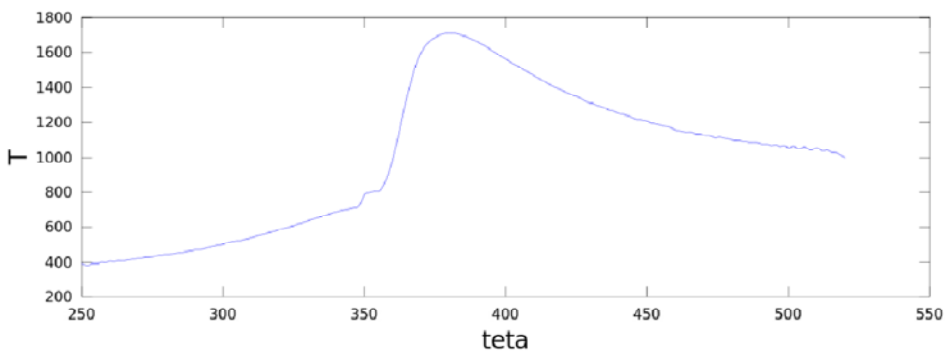

Figure 9. Calculated average temperature from experimental data for 1200 rpm.

The first sharp increase in the temperature shown in the Figure 9 is due to the first peak of AHR, and the second by the mixing-controlled combustion. The peak temperature is about 1700K and occurs for a crank angle of about 380^o. Due to the rapid combustion, the temperature decreases fast in the expansion stroke. Similar results were obtained for different engine speeds. Due to the smaller time available to heat transfer from the hot gases to cylinder walls, when the engine speed increases the gases are hotter than in the sample case shown in the Fig. 9. The higher temperature prevails also during the expansion and exhaust processes. Since the analyzed engine is naturally aspirated, the higher energy content of the exhaust gases will be dissipated in the environment. A turbocharged engine could take some advantage of the higher temperature of the exhaust gases, recovering part of this lost energy.

3. MODELING THE HEAT RELEASE RATE

3.1 Description of the model adopted

The second tool is a simplified thermodynamic model developed to simulate the diesel engine, with the purpose to evaluate the effect of the heat release rate on engine performance, that is, its mean indicated pressure, indicated work and indicated thermal efficiency. Different patterns were attempted to simulate the rate of heat release in diesel engines; this approach was initiated by the pioneer work of Austen and Lyn (1961) and has been implemented by different authors since then. Triangular patterns, sinusoidal patterns and others were proposed, but the work of Myiamoto et al (1985) is considered a milestone in this type of approach. They were the first to conduct an extensive study comparing experimental results with a pattern of heat release composed by two Wiebe functions with parameters to adjust. In this work, the same approach was adopted.

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Detailed modeling of the combustion process have been proposed by some authors (Hiroyasu and Aray, 1980; Mohammadzadeh, 1984; Desantes et al, 1989; Kouremenos et al, 1997), but the number of hypothesis to be assumed and the difficulties to validate experimentally these hypothesis are huge. These detailed models are required if exhaust emissions are to be modeled. However, to evaluate the engine indicated performance changes due to parameter variations they are too complexes; the computational time required is very high to run each simulated engine condition.

The simulation model adopted in this work and the results presented here are limited to the so called "closed phase" of the engine operation – that is, the compression, combustion and expansion processes. However, in a future work this model will be coupled with an "open phase" model for exhaust and intake processes to form a complete diesel engine model.

To simulate the compression process is adopted a polytropic process until that the combustion begins. The same procedure is done for the expansion process after the end of the combustion. This approach can be justified by the typical form of a log(p) x log(V) diagram, which show an almost straight line during compression and expansion processes. The Fig. 11 shows the log(p) x log(V) diagram constructed with one set of the experimental data employed in the first part of this work. In this example, the polytropic exponents are the slope of the respective curve: 1.32 for the compression and 1.22 for the expansion process. Typical values for the exponent in the compression process lies in the range of 1.32 to 1.38 for diesel engines. For the expansion process, typical values are from 1.18 to 1.28.

The Wiebe function can be used to represent the accumulated heat release by the combustion process both for spark-ignition and diesel engines. It varies from 0 to 1 and represents the cumulative energy liberated by the fuel combustion. It is not only a convenient equation, but was obtained from fundamental studies on chain reactions, typical for combustion processes in the internal combustion engines. Its mathematical form is:

$$x(\theta) = 1 - exp\left[-a * \left(\frac{\theta-\theta_o}{\theta_f-\theta_o}\right)^{m+1}\right] \tag{8}$$

and its derivate is:

$$\frac{dx}{d\theta} = a(m+1)\left(\frac{\theta-\theta_o}{\theta_f-\theta_o}\right)^{m} exp\left[-a * \left(\frac{\theta-\theta_o}{\theta_f-\theta_o}\right)^{m+1}\right] \tag{9}$$

where (θ_o) is the crank angle where the combustion begins, (θ_f) is the crank angle where the combustion is supposed to end, (m+1) is the form factor and (a) is an efficiency factor. The Wiebe function tends asymptotically to 1. The parameter (a) can adjust the speed with which the function tends to 1 and have typical values around 4 or 5. This parameter determines the value of the Wiebe function when θ is equal to θ_f . This can simulate to some extent the efficiency of the combustion, which is around 96 to 98% for engines operating on lean mixtures. The Wiebe function derivate represents the rate of heat release of the engine. The total energy released by the fuel during the combustion process is the injected fuel mass times it's lower heating value. Then, the rate of energy released by the combustion can be expressed as:

$$\frac{dQ_c}{d\theta} = \frac{dx}{d\theta}\, m_f PCI \tag{10}$$

The burning rate for a diesel engine was modeled as the sum of two Wiebe functions: the first function to represent the rapid heat release associated with the premixed combustion, and the second one to represent the mixing-rate controlled combustion. The percentage of the total energy released in each part of the combustion is taken into account by a weight factor attributed to each Wiebe function, as shown in the example below.

The Fig.12 shows the derivative of a combination of two Wiebe functions with the parameters defined in the Table 2. The shape of the heat release can be changed with adequate changes of the Wiebe function parameters. It is interesting to note how similar are the patterns seen on the Fig.12 and the Fig.1.

Table 2. Double Wiebe function to represent the heat release in a diesel engine

	a	θ_i	θ_f	m	weight
Wiebe function 1	5	170°	177°	3	5%
Wiebe function 2	4	170°	230°	0.8	95%

To consider the heat transfer from hot gases to the cylinder wall, an explicit semi-empirical expression for heat transfer was used:

$$\frac{dQ_{HT}}{d\theta} = h(\theta)A(\theta)[T(\theta) - T_W]/(6N) \tag{11}$$

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where N is the engine speed in rpm. The 6N term is introduced to change the time derivative to crank angle derivative.

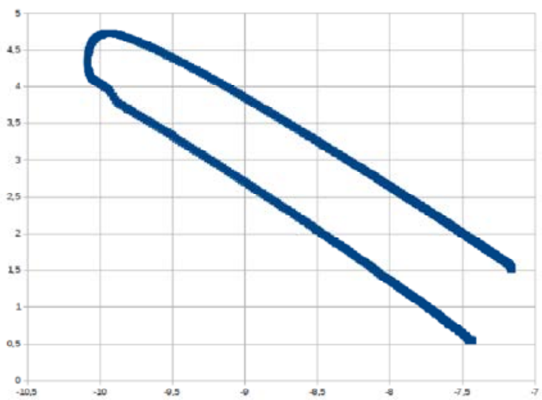

Figure 11. Log(p) x Log(V) from experimental data.

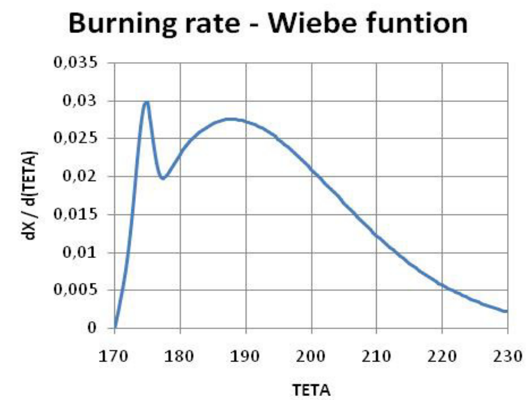

Figure 12. Diesel engine burning rate simulation.

The film coefficient h(θ) was calculated by the Hohenberg (1979) correlation:

$$h(\theta) = C_1 V^{-0,06} p^{0,8} T^{-04} \left(V_p + C_2\right)^{0,8} \quad (12)$$

where V, p and T are instantaneous values, C1 and C2 are constants. It is relevant to note that the heat transfer from gases to cylinder walls (Eq. 11) is dependant of the engine speed N (in rpm). The film coefficient also depends on the engine speed, since the piston average speed depends on N:

$$V_p = 2.L.N/60 \quad (13)$$

where L is the piston stroke.

The above presented heat transfer model was used only during the combustion process, since the heat transfer from gases to cylinder walls is implicit in the polytropic process model adopted for the compression and the expansion.

3.2 Some results from the heat release model

The effect of the injection timing can be modeled as a displacement of the crank angle where the combustion begins. Adopting the same heat release pattern shown in the Figure 12, the beginning of the combustion was set at 170 degrees as base case. If the injection timing is retarded 10°, the combustion also begins later, and follows the same pattern of the base case. The total duration of the combustion was maintained the same for both cases.

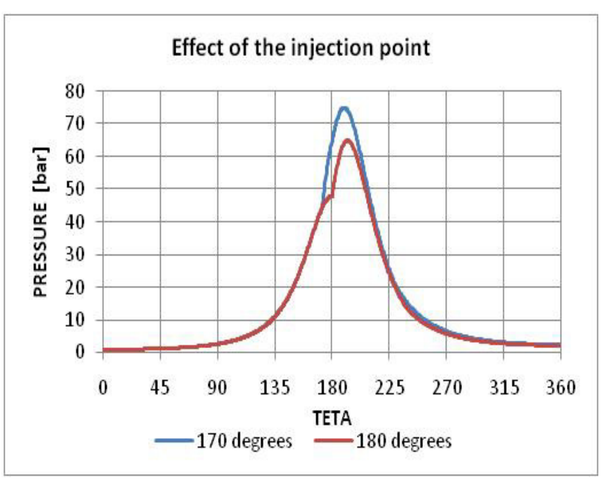

Figure 13. Pressure as a function of crank angle.

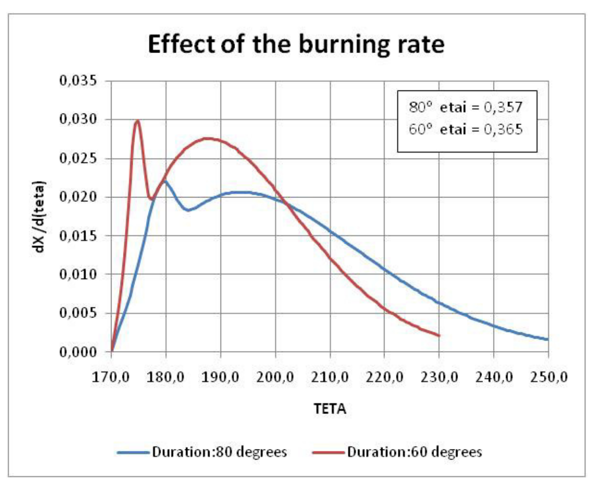

Figure 14. Heat release rates for different combustion. duration: 60° (base) x 80° crank angle

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Figure 13 shows how this delay in the combustion beginning affects pressures and temperatures as functions of the crank angle. The delayed "injection point" – that is, delayed beginning of the combustion – proved to be deleterious both for the work done and for the indicated thermal efficiency. The delayed heat release occurred when the piston was more advanced in the expansion process; the peak pressure occurred later and attained lower value than in the base case. The smaller work can be seen as a smaller area in the pressure diagram (Fig.13). The thermal efficiency of the cycle with delayed heat release is also smaller than in the base case: 0.307 versus 0.365.

The effect of the combustion duration on engine performance was also explored. This time, the start of the heat release was maintained at 170^o, but the total duration of the combustion process was increased in 20^o crank angle from base case. All other parameters in the Wiebe functions were maintained constant. Figure 14 show the heat release rate for both situations, while Fig.15 and 16 presents the effects on pressures and temperatures. Although the larger duration presented smaller work, the thermal efficiency reduction wasn't as large as with the retard of the combustion beginning: 0.357 versus 0.365. The smaller burning rate shifted the maximum temperature from 201.5^o to 211^o, but the final temperature was increased from 873 K to 909K.

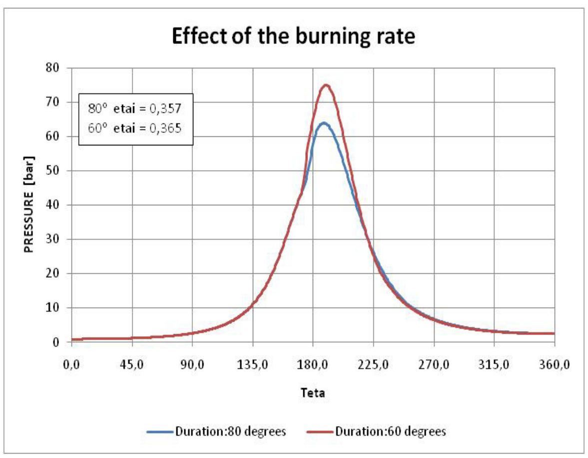

Figure 15. Effect of duration of combustion on pressure.

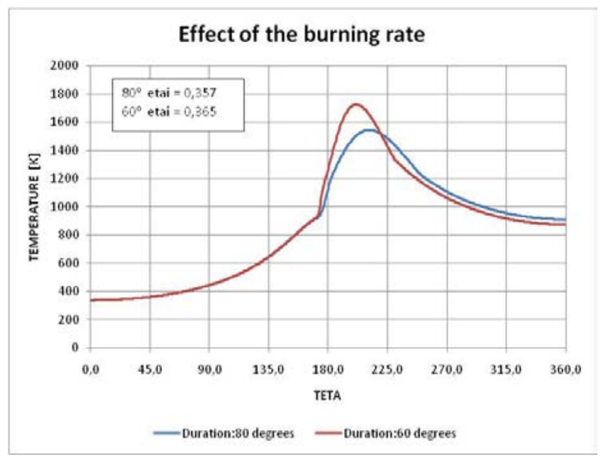

Figure 16. Effect of duration of combustion on temperature

The results presented above were obtained with the same geometry of the engine analyzed from experimental data. Since the simulation model includes an explicit model for heat transfer, the engine speed was fixed at 2200 rpm for all cases presented, to preserve the basis for comparisons changing the rate of heat release or the point of the cycle where combustion begins. The effect of the engine speed on the heat transfer from gases to cylinder walls can be obtained changing the engine speed.

4. CONCLUDING REMARKS

The use of the concept of apparent heat release can provide a powerful tool to analyze experimental data from engines and get some insight on the combustion processes. A critical aspect is the quality of data: the accurate determination of the top dead center (to trigger the pressure signal), as well as the accuracy of the pressure signal and its after-treatment to reduce "noise", which is highly amplified when the pressure derivate is calculated, giving rise to a high frequency oscillating signal.

The analysis of the rate of heat release can show the relative importance of each phase of the combustion, and the effects of strategies of multiple injections to reduce engine noise and/or emissions. When the injector needle signal is synchronized with pressure signal, the ignition delay can be estimated. The duration of the combustion provide valuable information for emission trends. Another good characteristic of this method of analysis is the possibility to estimate the average cylinder temperature during the combustion and the temperature of the gases at the end of the expansion process. It is important to note that the heat release analysis can be accomplished for naturally aspirated engines, as well as for supercharged engines.

The heat release simulation tool can be used to explore the effect of different shapes, "point of injection", combustion duration and multiple injections effects on the engine performance, as can be done experimentally changing setting parameters of the engine ECU, but at virtually no costs. The results can be seen as tendencies rather than absolute values, but can direct experimental efforts toward the correct direction, saving time and money.

The next step in the development of the simulation model will be the coupling of the present model with a model for the exhaust and intake processes. After this, the matching of the diesel engine with a turbocharger will also be implemented.

ISSN 2176-5480

5. REFERENCES

Austen A. E. W. and Lyn, W. T. 1960-61. "Relation between fuel injection and heat release in a direct-injection engine and the nature of the the combustion process". *Proc. of the Institution of Mechanical Engineers: Automobile Division*, (1):47-62.

Benson, R.S and Whitehouse, N.D., 1979. *Internal combustion engines*. Pergamon Press, Oxford.

Chang, C. ;Farrell, P.; 1997. "A study on the effects of fuel viscosity and nozzle geometry on high injection pressure diesel spray characteristics". *SAE Paper 970353*.

Desantes, J. M.; Lapuerta, M.; Tinaut, F. 1989. "Combustion dans les moteurs diesel rapides a injection directe. Modele base sur l'interaction entre le jet et le mouvement de l'air dans le cylindre". *Entropie*, (148):51-61.

Ferguson, C.R. and Kirkpatrick, A.T. , 2001. *Internal combustion engines: applied Thermosciences*. John Wiley & Sons, New Jersey, 2nd. edition.

Heywood, J. B. 1980. "Engine combustion modeling – an overview." In: Mattavi, J.S. and Amann C.A. (editors). *Combustion Modeling in Reciprocating Engines*, pages 1-38. New York: Plenum Press.

Heywood, J. B., 1988. *Internal Combustion Engine Fundamentals.* McGraw-Hill Inc.

Hiroyasu, H.; Aray, M.; 1980. "Development and use of a spray combustion modeling to predict diesel engine efficiency and pollutant emissions". *Proc. of the Fifth International Propulsion Systems Simposium*, p.264-288. SAE

Hohenberg, G. F. 1979. "Advanced Approaches for Heat Transfer Calculations". *SAE Paper 790825*.

Hung, C. C.; Martin, J. K. ; Koo, J. 1997. "Injection pressure effects upon droplet behavior in transient diesel sprays". *SAE Paper 970053*.

Kouremenos, D. A.; Rakopoulos, C. D.; Hountalas, D. T. 1997. "Mult-zone combustion modeling for the prediction of pollutants emissions and performance of DI diesel engines. *SAE Paper 970635*.

Kuo, T. W.; Yu, C. R.; Shahed, S. M. 1983. "A numerical study of the transient evaporating spray mixing process in the diesel environment. *SAE Paper 831735*.

Myiamoto, N.; Onikahisa, T.; Murayama, N.; Sawyer, T. R. 1985. "Description of diesel engine rate of combustion and performance using Vibe's functions". *SAE Paper 850107*.

Mohammadzadeh, A. 1984. *Stochastic Combustion Modelling of a Direct Injection Diesel Engine*. PhD. Thesis, University of Michigan.

Planells, R.C; Amengual, A.C.; Larruy, A.C., 1995. *Motores de combustión interna – Fundamentos*. Edicions UPC, Barcelona.

Ramos, J. I. 1989. *Internal Combustion Engine Modeling*. New York: Hemisphere Publishing Corporation.

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