

# Educational Software for Diesel Engine Simulation Performance and Parametric Analysis\*

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The development, implementation and evaluation of a virtual instrument that allows a detailed diesel engine simulation performance and parametric analysis of the combustion process are described. The stimulus for developing this computer-aided educational tool derives from the continuing need for the engineer to understand and apply mathematical models in engine simulation. The application of software in teaching internal combustion engines can help students to improve their parametric analysis and allow them to understand diesel combustion better. The educational software of Simulation and parametric analysis are developed and visualized using MATLAB code and the designed graphical user interface (GUI) respectively.

**Keywords:** simulation; combustion; diesel engine

## 1. Introduction

Practical or experimental issues should be an important part of an engineering student's education in subjects related to internal combustion engines. Its importance is due to the complexity and high level of interconnectivity of operating variables in engine performance characteristics [1].

In the last few years, student and teacher surveys for internal combustion engine courses have shown the need for more practical content and self-study material, especially in lessons on the internal combustion engine, which has a complex behavior, influenced by many parameters. The Thermal Power Group (University of Seville, Spain) has distributed diverse practical activities across the curricula. In their fourth year, students take a 15 hour laboratory course in which the software developed described in this paper is included. The high cost of internal combustion engine laboratories and the laborious task of engine testing make using simulation software highly advisable for complementing practical training.

In this article, a powerful, versatile and easy-to-teach general purpose program that simulates a diesel engine is presented. Simulation not only includes performance results but also allows us to understand the influence of many parameters in diesel combustion. This solution makes it possible for the student to separately analyze the influence of each of the parameters in the combustion process.

A source code was necessary to create the program; the same one was used to make the graphic interface. The programming environment used is

MATLAB code, because of the student's familiarity with it [2–5].

This paper is divided into four main sections and an introduction. The first section is a brief description of diesel engine operating principles, which are essential for the program model. In the second section, the modeling of the operating principles discussed in the first section is described. Once the whole model has been constructed, a user-friendly environment to facilitate the users' interaction with the program is needed. This friendly environment is the graphical user interface described in Section 3. Finally, the program results have to be checked to see if they are close to a real diesel engine's performance and to quantify how close they are. This is done in the last section using operating data from fifteen different diesel engines; however, for the sake of space, only three cases are described in this paper.

## 2. DI diesel engine fundamentals

Diesel engines, or compression ignition engines, only breathe air with a small quantity of residual gases at the induction stroke, which is compressed at the compression stroke, the volumetric compression ratio being between 13 and 19 [6]. The diesel fuel is injected near top dead centre (TDC).

The software is for direct injection diesel engines, which are the most common type, thus the diesel spray is injected directly into the cylinder. At the injection point, the temperature and pressure are high, which causes vaporization of the fuel and combustion reaction kinetics slowly start at this point. Owing to slow velocity reactions, their effects are undetectable for a small time interval (delay

time); after this interval, there is a steep rise in temperature and pressure within the cylinder.

The ignition point in these engines appears when there is an inhomogeneous fuel, air and residual mixture in the cylinder. Furthermore, at this point, some part of the fuel is in liquid phase.

Owing to the high compression ratio of these engines, the cylinder gas temperature and pressure at the injection point are above auto ignition requirements. Under these conditions, the ignition of the mixture occurs when the local air–fuel ratio is suitable for combustion. The ignition is propagated to the adjacent zones if these can maintain combustion or diminishes delay time if the zones are not yet ready to maintain combustion.

In compression ignition engines, combustion is due to local mixture conditions rather than flame front propagation, as in spark ignition engines. This makes the combustion velocity closely related to the correct mixture formation [7], which makes the good design of the injection system and the combustion chamber fundamental in obtaining a good fuel distribution within the cylinder during injection.

### 2.1 Combustion phases

There are four periods in the combustion process [8, 9]:

- *The delay period:* This is the period between fuel injection and mixture ignition. This delay is defined by the delay time or by the delay angle.
- *The rapid pressure rise period:* During this period, the fuel injected and vaporized during the delay period burns instantly or very quickly, which causes a steep pressure rise. If the pressure rise is too high, it causes diesel combustion noise, which damages the engine. The only way to control this pressure rise is to limit the amount of fuel vaporized (injected) in the delay period. During this period, the fuel injection continues as the vaporized fuel is burned.
- *The mechanically controlled burning period:* During this period, the pressure rise diminishes and the only fuel burned is the vaporized fuel. The cylinder gas pressure and temperature are very high, thus as the fuel is injected it is burned. During this period the combustion velocity is controlled by fuel injection, continuing until injection ends.
- *The post-burning period:* During this period, the burning of the fuel and partial combustion products during the mid-expansion stroke is completed. Up to 10% of the total injected fuel can be burnt in this phase.

### 2.2 Parameters that affect combustion

The compression ignition engine combustion velocity and the pressure gradient strongly depend on

the combustion chamber and fuel injection design [10]. There are several feasible designs, which make it difficult to establish some general design rules [6].

Moreover, the large number of parameters that affect the combustion characteristics must be taken into account. What follows describes only the parameters that have a strong influence on the combustion characteristics due to the limitation of space in this paper.

- *Fuel characteristics:* one the most important is the cetane number; the higher the cetane number, the lower the time delay.
- *Engine speed:* Turbulence intensity improves mixture and flow propagation; both increase with engine speed. The start of combustion delay diminishes with engine speed on a time basis but increases on a crank angle basis. The following relationship can be established:

$$\Delta\theta_{\text{combustion}} = w \cdot \Delta t_{\text{combustion}} \quad (1)$$

so that any  $w$  increment is greater than any  $\Delta t_{\text{combustion}}$  reduction,  $w$  being angular engine speed.

- *Engine load:* Compression ignition engines control load by injecting more or less fuel. During the first stage, the load does not affect delay time because the quantity of fuel injected during the delay period is the same independently of load. However, when more fuel is injected, higher temperatures are reached that raise the cylinder wall and cylinder head temperatures and there are more residuals within the cylinder. Therefore, as the load increases, the delay time decreases.
- *Air pressure:* A decrease in air pressure causes a decrease in combustion peak pressure.
- *Air temperature:* A higher air temperature means a higher inlet air temperature, which means a higher cycle temperature. This results in a decrease in the pressure gradient in the second combustion period and lower engine performance due to the decrease in delay time.
- *Injection point:* The injection point has a vital role in combustion characteristics as the ignition angle mainly depends on the injection angle. Thus, all the variables that depend on the ignition angle also depend on the start of the injection angle: peak cycle pressure, pressure gradients, temperatures within the cylinder, NOX emissions and other second-order parameters.

The optimum engine performance is achieved when peak cycle pressure is nearly 15–20° after TDC; this is obtained for an injection point from 10–15° before TDC.

If the injection is earlier, the pressure and temperature in the cylinder would be low, which causes a steep pressure gradient and an early

ignition angle. The result is very high pressure and temperature before TDC, which damages the engine and reduces efficiency. However, if the injection is later, the initial high pressure and temperature would decrease and cause a large delay angle. Therefore, most of the fuel is burnt during the expansion stroke, which diminishes efficiency and raises exhaust gas temperatures.

- *Cooling water temperature*: as the cooling water temperature rises, the cylinder wall and gases within the cylinder temperatures rise. This temperature increase causes a decrease of delay time.
- *Compression ratio*: For constant engine speed, injection point and air–fuel ratio, higher compression ratios mean lower delay time and nearly equal pressure gradients. However, if the injection point is controlled to achieve optimum combustion, the pressure gradient diminishes with the compression ratio.

By raising the compression ratio, a slight improvement in engine performance is obtained due to the narrow range of compression ratios obtainable. The main drawbacks of raising the compression ratio are: higher friction, a rise in blow by losses, starting torque rise and other drawbacks. Therefore, the compression ratio for an engine is the minimum compatible with engine cold start, which for a high speed DI diesel engine is between 17 and 19.

- *Cylinder size*: As the cylinder diameter increases, the nominal angular engine speed decreases, the engine is warmer, the relative heat losses are minor and the local temperatures are higher. All this causes a small delay angle; thus the proportion of fuel burnt during the rapid pressure rise period is less. This relationship is so sensitive that the more the cylinder diameter increases, the lower the requirements of the fuel (cetane number) are.

### 3. The simulation program

Combustion models can be classified into two groups: thermodynamic models and multidimensional models. Thermodynamic models are divided into one zone [11] and multizone models [12].

Here the chosen model is the one zone thermodynamic model. In this model the cylinder gas is considered to be uniform in composition, temperature and pressure; this allows the calculation of energy development via the first law of thermodynamics. Therefore, this model assumes that the mix process is instantaneous; moreover, it considers the gas within the cylinder to be a mixture of gases, with variable composition and dependant on temperature.

The one zone model consists of an ordinary differential equation system for mixture pressure, temperature and mass. Mass and heat losses are included in the equations; on the other hand, drops of fuel, combustion chamber geometry and mixture homogeneities are not considered in these equations.

#### 3.1 One zone model and intake/exhaust models

The one zone model for the engine assumes that the cylinder content is a homogeneous gas mixture. Temperature  $T$ , pressure  $p$ , composition and the amount of fuel injected—considered instantaneous vaporization—determine the state of the mixture each time. Fuel injection simulation is not like the real phenomenon because the fuel atomization and vaporization process are not considered.

The following equation applies the principle of mass conservation to the combustion chamber [6]:

$$\frac{dm}{dt} = \dot{m}_{\text{int}} + \dot{m}_f - \dot{m}_{\text{exh}} \quad (2)$$

where  $m$  is the mass within the cylinder,  $\dot{m}_{\text{int}}$  is the instantaneous intake mass flux through the intake valve,  $\dot{m}_{\text{exh}}$  is the exhaust mass flux through valve and the losses and  $\dot{m}_f$  is the theoretical injected fuel mass flow.

The first law of thermodynamics applied to the mixture within the cylinder is described by the following equation [13]:

$$\frac{dp}{dt} = -\gamma \left( \frac{p}{V} \right) \frac{dV}{dt} + \frac{p}{(\gamma - 1)} \frac{d\gamma}{dt} + \frac{(\gamma - 1)}{V} (q_w + Q) + \frac{(\gamma - 1)}{V} \sum h_j \dot{m}_j \quad (3)$$

where  $p$  is the combustion chamber pressure,  $V$  is the combustion chamber volume,  $\gamma$  is a function of the temperature,  $q_w$  is the heat to the coolant,  $Q$  is the fuel heat release rate and  $h_j \dot{m}_j$  represent the intake flux, exhaust flux and injected fuel enthalpy.

This equation calculates the pressure time evolution from heat loss through the cylinder wall, from the heat release rate and from the mixture properties variation as the process advances.

The intake and exhaust models are the “filling and emptying” [14] type, thus they do not include fluid dynamic effects along the intake and exhaust pipes.

#### 3.2 Delay angle

Pressure and temperature change in the engine during the delay period due to the piston movement. In the program, the Hardenberg and Hase correlation [15] has been implemented to calculate the delay period that is suitable for a wide range of operating conditions. The correlation calculates the

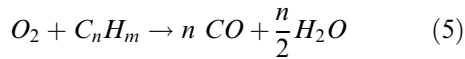
delay angle (crank angle degrees) as a function of temperature  $T_{TC}$  (K) and pressure  $p_{TC}$  (bar) [7]:

$$t_d(CA) = (0.36 + 0.22S_p) \cdot e^{\left[ \frac{E_A}{R} \left( \frac{1}{RT_{TC}} - \frac{1}{17.190} \right) \left( \frac{2.12}{p_{TC} - 1.4} \right)^{0.63} \right]} \quad (4)$$

where  $R$  is the ideal gas constant,  $S_p$  is the piston mean speed (m/s) and  $E_A$  is the apparent activation energy.

### 3.3 Properties variation

The in-cylinder fluid properties have to be known in order to analyze the combustion process. Before fuel injection, there are fresh air and residuals from the previous cycle in the cylinder. When injection starts, there is another substance: the vaporized fuel. The chemical reactions that take place in the combustion process in diesel engines can be summarized as  $\frac{(2n+m)}{2}$  [11]:



Therefore, the inclusion of this equation for the model lets us know the mol number of each species every moment. The problem now is to calculate species properties. The most accurate model for calculating the properties of a species is based on the polynomial adjustment of their thermodynamic data, assuming that the unburnt gas mixture is constant in composition and that the burnt gas mixture is in equilibrium. The specific heat for species  $i$  as a function of temperature, standard enthalpy and gamma (for the mixture, not only for one species) are calculated by the following expressions:

$$C_{p,i}/R = a_{i,1} + a_{i,2} \cdot T + a_{i,3} \cdot T^2 + a_{i,4} \cdot T^3 + a_{i,5} \cdot T^4 \quad (6)$$

$$h_i/RT = a_{i,1} + (a_{i,2}/2) \cdot T + (a_{i,3}/3) \cdot T^2 + (a_{i,4}/4) \cdot T^3 + (a_{i,5}/5) \cdot T^4 + a_{i,6}/T \quad (7)$$

$$\gamma = \frac{\sum n_i \cdot C_{p,i} / \sum n_i}{\left( \sum n_i \cdot C_{p,i} / \sum n_i - R \right)} \quad (8)$$

where  $R$  is the gas constant,  $n_i$  is the number of moles of specie  $i$  and the other coefficients have been obtained from the NASA program [16].

### 3.4 Gas loss model

The model used to determine gas losses between the combustion chamber and the crankcase is based on

the Eweis flux model [17]. This model considers that the volumes separated by the piston rings are connected by orifices whose properties are defined by a flux coefficient. Taking into consideration axial symmetry, quasi-static and isentropic thermodynamic equations, the mass flux is determined by the following expression [6]:

$$\dot{m}_{12} = \frac{\varphi_{12} A_{12} p_1 \sqrt{2}}{(RT_1)^{1/2}} \left( \left( \frac{\gamma}{\gamma - 1} \right) \left( \left( \frac{p_2}{p_1} \right)^{2/\gamma} - \left( \frac{p_2}{p_1} \right)^{\frac{\gamma+1}{\gamma}} \right) \right)^{1/2} \quad (9)$$

where conditions 1 and 2 are the conditions before and after each piston ring. These are solved by taking the combustion chamber as the condition before the first piston ring and taking the crankcase as the condition after the last piston ring. This model gives the flux coefficient. The flux coefficient for the model has been determined by an exhaustive investigation of model behavior that changes with the degree of engine wear, selected by the user.

### 3.5 Heat released rate

The heat release rate curve shape is fundamental because all problem variables directly depend on it. This is the reason for giving the user the possibility of setting and saving these coefficient values in the program.

To estimate radiant and convective heat loss through the cylinder wall, the Woschni [18, 19] correlation is used.

$$\dot{q}_w = h \cdot A \cdot (T - T_w) \quad (10)$$

where  $h$  is the heat transfer coefficient,  $A$  is the transfer area and  $T_w$  is the wall temperature.

### 3.6 Mechanical losses

Mechanical losses are divided into pumping losses, friction losses and losses associated with auxiliary elements. Pumping losses are calculated within the model equations. Friction losses are evaluated through Bishop semi-empirical correlations.

Distribution losses

$$p_{m1} = 0.701 \cdot 10^{-3} \cdot (30 - 40n/1000) \cdot D_v^{1.75} / (D^2 \cdot S) \quad (11)$$

Friction losses

$$p_{m2} = 5.5 \cdot 10^2 \cdot D \cdot n / (1000 \cdot S) \quad (12)$$

Piston-piston ring losses

$$p_{m3} = 3.69 \cdot 10^{-3} \cdot S \cdot a \cdot K / D^2 + 7.56 \cdot 10^{-3} \cdot b \cdot c_m / (D \cdot S) \quad (13)$$

where  $a$  is the number of piston rings,  $b$  the piston

skirt height,  $D$  the diameter,  $c_m$  the mean linear piston speed,  $n$  is the rotational speed and  $K$  is a constant between 2 and 3.

The total mechanical engine loss is given by (11) + (12) + (13).

#### 4. Graphical user interface

The diesel direct injection engine model previously described is implemented in MATLAB code and, at the same time, the graphical user interface (GUI) is also implemented in MATLAB. There are several codes for GUI applications; the MATLAB code has been used due to specialized MATLAB engineering libraries, and the familiarity that engineering students have with MATLAB being GUI implemented with the same code [20, 21].

The GUI is formed by five main windows that are described in the following paragraphs. Figure 1 shows the diagram flow of the GUI.

The design parameter window is where geometrical and operational engine parameters have to be included within a range indicated by the program. The data can be typed in each box or loaded from an archive. For inexperienced students, several input data archives have been created that can be loaded by pushing the EXPLORE button. Each data archive contains the data of a real direct injection diesel engine. For experienced users, the program

has the option of setting the values and then saving this input data with the button SAVE.

As the results of heat release rate coefficients are so fundamental and depend on the type of engine (obtained experimentally for a particular engine), the program can be set by the user or automatically set by the program itself. For a better understanding of heat release rate (HRR) coefficients, it shows how the HRR is simulated. The HRR is determined by two Wiebe functions [22]:

$$\frac{dQ}{d\theta} = \frac{dQ_p}{d\theta} + \frac{dQ_d}{d\theta} \tag{14}$$

The first term of the equation ( $\frac{dQ_p}{d\theta}$ ) models the rapid pressure rise period and the second ( $\frac{dQ_d}{d\theta}$ ) models the other combustion period called the diffusive period. Both terms follow an exponential law. The coefficients of the exponential laws are experimental values, thus they would be estimated experimentally as function of the engine. However, for educational purposes, typical values can be used. Each period is described by the following Wiebe correlations [23]:

$$\frac{dQ_p}{d\theta} = \left(\frac{Q_p}{\theta_p}\right) \cdot (M_p + 1) \cdot \left(\frac{\theta}{\theta_p}\right)^{M_p} \cdot \exp\left[-K_1 \cdot \left(\frac{\theta}{\theta_p}\right)^{M_p+1}\right] \tag{15}$$

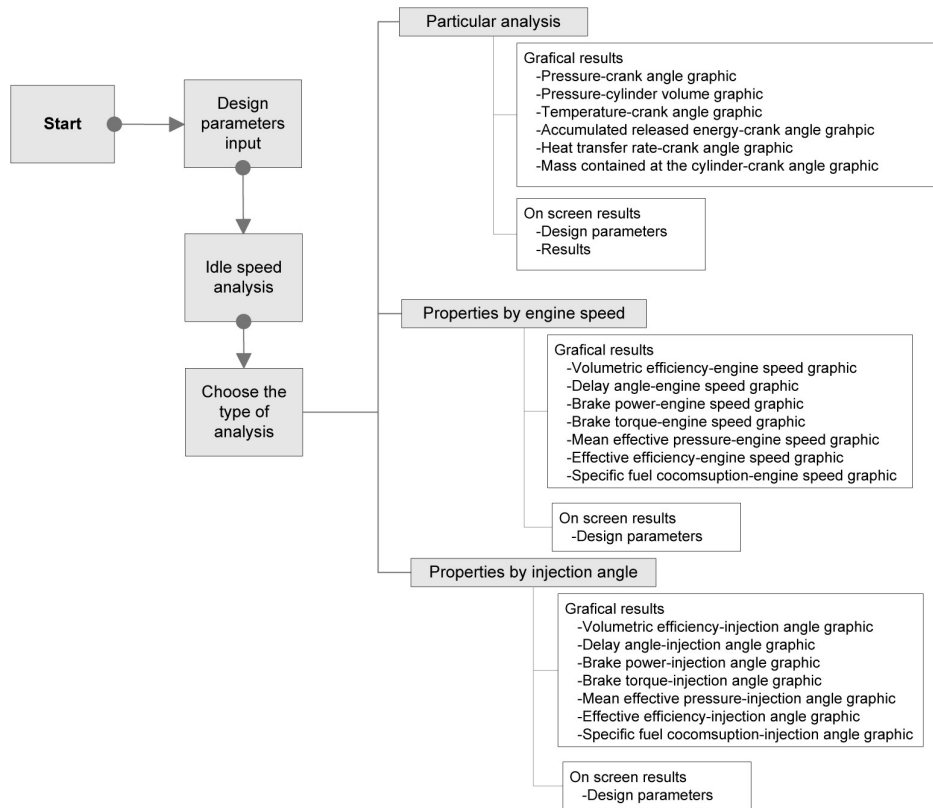


Fig. 1. GUI flow diagram.

$$\frac{dQ_d}{d\theta} = \left(\frac{Q_d}{\theta_d}\right) \cdot (M_d + 1) \cdot \left(\frac{\theta}{\theta_d}\right)^{M_d} \cdot \exp\left[-K_1 \cdot \left(\frac{\theta}{\theta_d}\right)^{M_d+1}\right] \quad (16)$$

where the sub-indices p and d refer to the phases of high pressure rise and diffusive;  $M_p$  and  $M_d$  are shape factors;  $\theta_p$  and  $\theta_d$  are the duration of phases; and  $Q_p$  and  $Q_d$  characterize the heat released in each phase.

Finally, the program gives the user the option to perform, or not, a mass loss analysis.

Once all the design parameters are set, the button ACCEPT is pressed to start the analysis. If a parameter value is out of range, an alert window appears informing the user which parameter value is misleading. After pressing the OK button, the program returns to the input parameter window.

When all the parameters are within range and the ACCEPT button is pressed, the idle speed analysis appears. The progress of the idle speed analysis is shown in this window. In the idle analysis, the amount of fuel required to overcome mechanical losses at idle speed is determined. During this process, the program can be aborted by pressing the CANCEL button.

Once the idle analysis is finished, the analysis type window appears that offers three different analyses and also the option to cancel the process by pressing the CANCEL button.

#### 4.1 Particular analysis

In this particular analysis window (Fig. 2), the user has to select the value for the program's free variables, which are: engine speed, EGR, injection point and engine load. These values are selected on the bottom slide bars in the variables section. After selecting these values, the button ACCEPT is pressed to proceed with the analysis. If any of the buttons in the graphics frame are pressed before running the analysis, an error message appears.

The progress of the analysis is shown in the frame below the push button. The analysis results can be shown in two different formats by the program:

##### 4.1.1 Graphical results

To the right of the window is the frame "GRAPHICS". In the frame, there are several buttons, each one showing the values of the variable as function of crank degree when pressed. The following graphics are shown in the axes located at the top of the window:

- *Pressure*: Pressure within the cylinder as function of crank angle. If loss analysis is selected in the design parameter window, this graphic also shows the pressure variation between the different piston rings.
- *P-V*: Pressure as a function of cylinder volume for all of the cycle.
- *Temperature*: Gas temperature within the cylinder as a function of crank angle.
- *HRR*: Heat release rate as a function of crank angle.

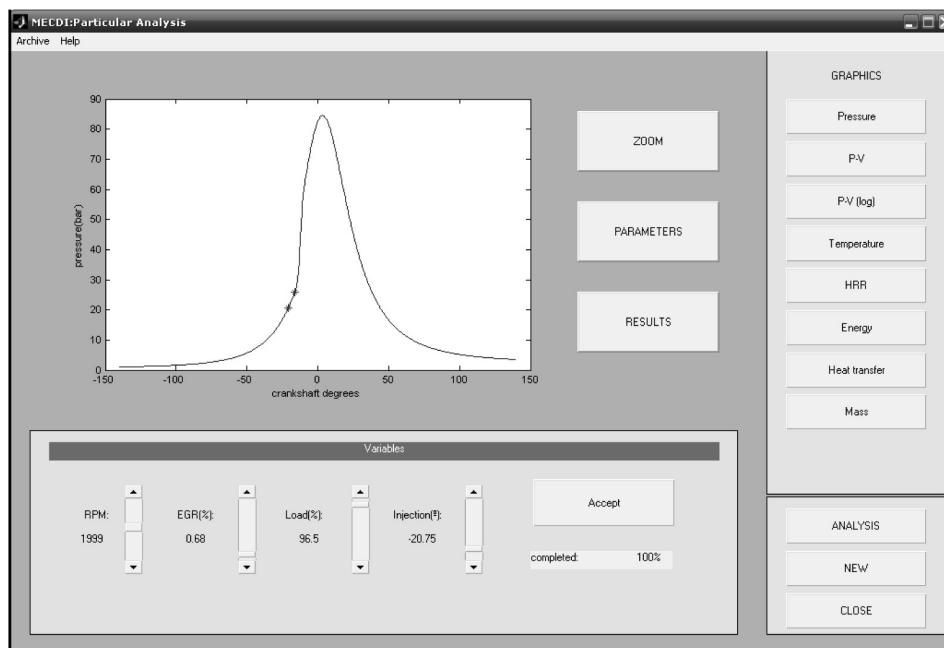


Fig. 2. Particular analysis window.

- *Energy*: Accumulated released energy as a function of crank angle. The net accumulated released energy is also shown in these axes. The difference between these two accumulated energies is the energy loss through the cylinder walls.
- *Heat transfer*: Heat transfer rate through the cylinder walls to the coolant.
- *Mass*: mass contained in the cylinder as a function of crank angle.

Finally, to focus on one point of the graphic there is the ZOOM button, located on the right of the graphic.

#### 4.1.2 On-screen results

In the window there are PARAMETERS and RESULTS buttons. The parameter window appears when pressing the first button where the fixed parameters in the analysis can be seen. On the other hand, if the RESULTS button is pressed, the results window appears where the most representative engine variables are displayed: engine speed, EGR, injection angle, air fuel equivalence ratio, delay angle, volumetric efficiency, specific fuel consumption, indicated mean effective pressure, brake mean effective pressure, friction mean effective pressure, indicated power, indicated torque, brake power, brake torque, residuals and gas losses.

#### 4.2 Properties by rpm

Engine performance is analyzed as a function of engine speed for a certain load, EGR and injection angle, whose values are selected by moving the slider bars in the frame variables, see Fig. 3.

All of the other options in this window are the same as in the particular analysis window, except for the RESULTS button, which does not appear in this window.

Furthermore, the graphics are:

- *Volumetric efficiency*: volumetric efficiency variation as a function of engine speed.
- *Delay angle*: delay angle variation as a function of engine speed
- *Brake power*: brake power as a function of engine speed
- *Brake torque*: Brake torque as a function of the engine speed
- *Brake mean effective pressure*: brake mean effective pressure as a function of the engine speed
- *Effective efficiency*: effective efficiency as a function of engine speed
- *Specific fuel consumption*: Specific fuel consumption as a function of engine speed.

#### 4.3 Properties by injection angle

The window in Fig. 4 is similar to the analysis of the function of engine speed window. The difference between these two windows is that the graphics refer to the fuel injection angle from the TDC instead of angular speed.

### 5. Program validation

This program is intended to be an alternative to engine tests for educational purposes; therefore, its results have to be close to the experimental results [24]. To check the accuracy of the program, two

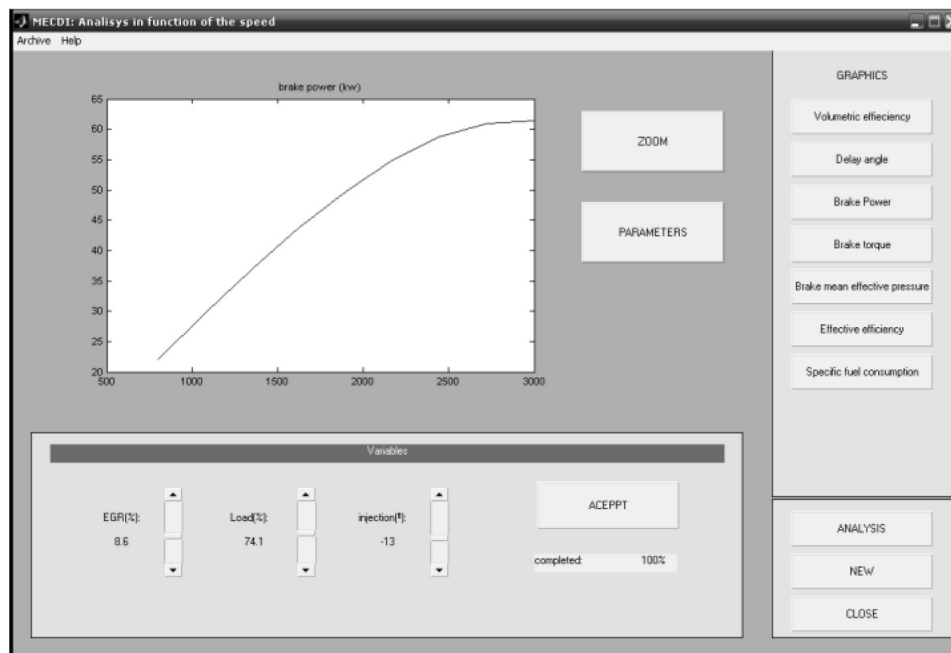


Fig. 3. Analysis of the function of the engine speed window.

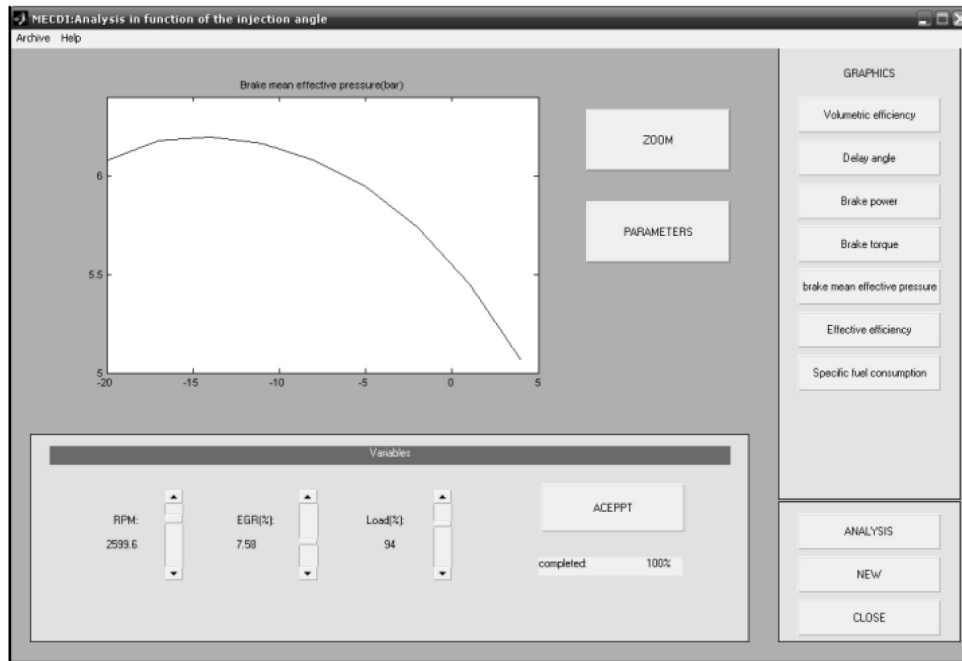


Fig. 4. Analysis of the function of the injection angle window.

Table 1. Engine characteristics

Engine type	Mercedes-Benz OM 442	MAN D0826 GF
No. cylinders	V8	4 inline
Diameter/ Stroke	130/140 mm	108/125 mm
Engine cubic capacity	15.078 cm <sup>3</sup>	4580 cm <sup>3</sup>
Power	213 kw (290 HP) at 2100 rpm	75 kW (102 CV) at 2700 rpm
Maximum torque	1100 Nm (112 kpm) at 1000–1600 rpm	310 Nm at 1200–1500 rpm
Cooling	Water	Water

simulations were carried out for two different engines and compared with the experimental data. The two engines' characteristics are given in Table 1.

The engines are tested at maximum load, which means operating with a fuel air ratio of 0.8 for the two engines. In Fig. 5, the net power, net torque and specific fuel consumption for each engine is shown.

First one can see that the differences between the simulated data and the real data for the two engines are less than 10%, except at one point where the difference is 16% for the second engine. These small differences are within the concordance interval.

Comparing the two engines, the simulated brake power curve for the first engine is under the real data, both data match for the second. This is due to the efficiency increase with increase in engine volume, which makes the engine warmer.

In the graphs one can also see that the largest differences occur at low and high engine speeds, which is the result of not taking into consideration the admission and exhaust systems and the gas dynamic effects.

## 6. Evaluation of software

The software is part of the educational resources that have been used in a graduate course [25–27]. Owing to the innovate approach, it was interesting to know the opinion and the results of the students taking part: what needs to be kept as it is, what needs improving, and to what extent the initial goals have been achieved. In order to obtain this information, a questionnaire was completed by the students at the end of the course [28]. The main objective in educational software testing is to measure the usefulness of the software in the teaching–learning process. An anonymous online survey was conducted after students obtained their grades for their laboratory reports to aid the formative evaluation of the introduction of educational software for the diesel engine. Only students who had completed the course with the integrated software were surveyed. A questionnaire using ten statements in the form of a listed table was designed for this survey (Table 2). Students were requested to respond to each item in the questionnaire using a 5-point scale: strongly



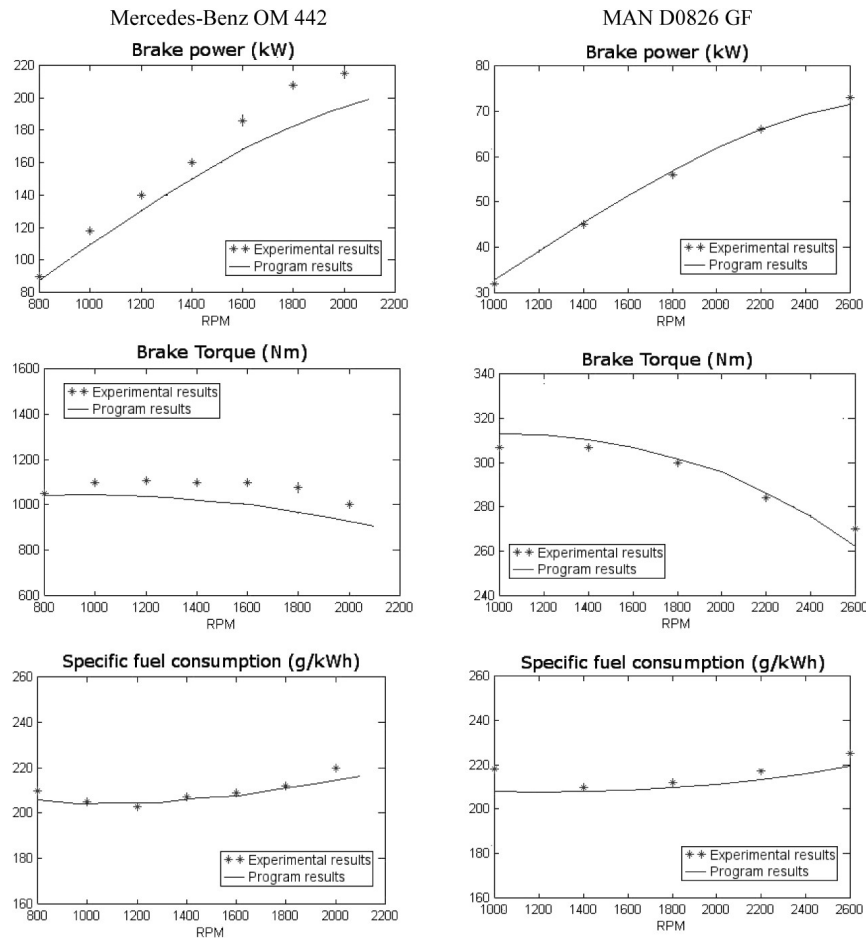


Fig. 5. Net power, net torque and fuel specific consumption for each engine, real and simulated.

agree, agree, neutral, disagree, and strongly disagree, plus a column for 'no opinion'. An opportunity was also provided for students to comment on their experience at the end of the questionnaire to collect qualitative feedback.

Table 2. List of Questions/Statements used in the evaluation of the software

No.	Question/Statement
1	I found the software easy to use.
2	I have used a similar software before.
3	The software enhances my understanding of the theory.
4	I think this software is a useful tool in my learning.
5	I now have a complete knowledge of the internal combustion engine.
6	I understand the main parameters' influence on the engine performance.
7	The applicability of the software allows me to avoid laboratory tests.
8	The software provides great potential for self-criticism and criticism.
9	The software allows me to verify numeric models by applying theoretical concepts.
10	The software provides the opportunity to put concepts into practice.

A study of the histogram that was plotted with quantitatively rated answers is extremely useful. With regard to the questions, the detailed analysis conducted on the meaning of the answers showed how important they are both in their own right and in connection to particular answers to the others questions.

Generally, student feedback surveys have very low response rates. However, the response rate was high (> 90%) with 68 responses received per question and, overall, the results from the survey were highly positive.

The responses to the survey are shown in Fig. 6 and they indicate that the students felt that they had benefited from their exposure to the educational software. It was noted that the students liked the hands-on and self-discovery approach, although at times some frustration was also noted. Once a demonstration of the software was given, the students wished to learn by themselves.

Study of the histograms plotted with quantitatively rated answers is extremely useful. Although the testing method developed here proved to be effective in rating educational software, we consider

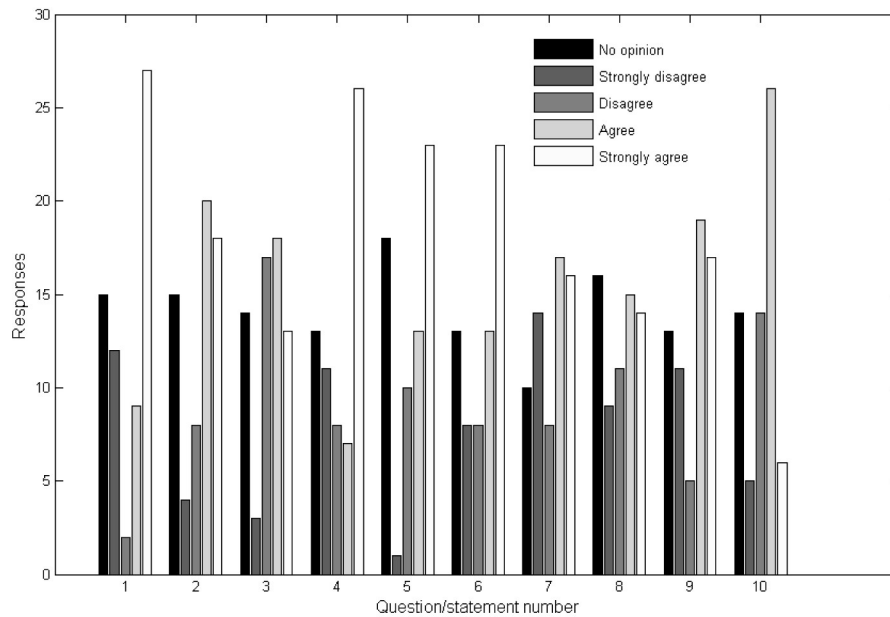


Fig. 6. Chart showing survey results (total completed questionnaires = 0.68).

that some new questions should be incorporated to allow the users to suggest improvements in the software being tested.

## 7. Conclusions

The benefits of using educational software for diesel engine simulation as a didactic tool is well accepted by students who gain confidence in their abilities and, on the whole, they improve their understanding of the theory. This software is based on integrating and reusing MATLAB to develop an alternative GUI, in order to have a user-friendly and educationally featured software to be exploited in the classroom and in laboratory activities. This possibility of integrating and reusing MATLAB to provide the internal functionality to our educational software allows for development at reduced cost, in a short time, and without losing functionality regarding the proprietary environment. The software also makes it easier to link theory with practice, the simulation program developed overcomes the enormous problems of commercial programs for the internal combustion engine.

The simulation model has been shown to be effective as an educational program in improving relevant results. The software is intended to serve as a teaching aid in internal combustion engine discipline.

This program provides a computational tool in which one of the main features is a user-friendly interface, making it easy to learn and use the software. After a short training period, the student can analyze the influence the different parameters for

diesel combustion performance and input the necessary data to solve them in a fast and precise manner.

The software presents a library of different engines that allows the students to probe most engine types that can be found in the real world. Therefore, this work represents a very interesting and efficient educational tool for students and instructors that decisively contributes to the teaching-learning process and the achievement of its objectives.

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