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Parametric study of the performance of a turbocharged compression ignition engine

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Abstract
In this study, the thermodynamic performance of a turbocharged compression ignition engine with heat transfer and friction term losses was analyzed. The purpose of this work was to provide a flexible thermodynamic model based on the filling-and-emptying approach for the performance prediction of a four-stroke turbocharged compression ignition engine. To validate the model, comparisons were made between results from a computer program developed using FORTRAN language and the commercial GT-Power software operating under different conditions. The comparisons showed that there was a good concurrence between the developed program and the commercial GT-Power software. We also studied the influence of several engine parameters on brake power and effective efficiency. The range of variation of the rotational speed of the diesel engine chosen extended from 800 to 2100 rpm. By analyzing these parameters with regard to two optimal points in the engine, one relative to maximum power and another to maximum efficiency, it was found that if the injection timing is advanced, so the maximum levels of pressure and temperature in the cylinder are high.

Keywords
Thermodynamic, combustion, turbocharged compression ignition engine, GT-Power, performance optimization, filling-and-emptying method

1. Introduction

More than a century after its invention by Dr Rudolf Diesel, the compression ignition engine remains the most efficient internal combustion engine for ground vehicle applications. Thermodynamic models (zero-dimensional) and multi-dimensional models are the two types of models that have been used in internal combustion engine simulation modeling. Nowadays, trends in combustion engine simulations are towards the development of comprehensive multi-dimensional models that accurately describe the performance of engines at a very high level of detail. However, these models need a precise experimental input and substantial computational power, which makes the process significantly complicated and time-consuming.[1] On the other hand, zero-dimensional models, which are mainly based on energy conservation (first law of thermodynamics) are used in this work due to their simplicity and being less time-consuming in the program execution, and their relatively accurate results.[2] There are many modeling approaches to analysis and optimization of the internal combustion engine. Angulo-Brown et al.[1] optimized the power of the Otto and Diesel engines with friction loss with finite duration cycle. Chen et al.[2] derived the relationships of correlation between net power output and the efficiency for Diesel and Otto cycles; there are thermal losses only on the transformations in contact with the sources and the heat sinks other than isentropic. Merabet et al.[3] proposed a model for which thermal loss is represented more classically in the form of a thermal conductance between the mean temperature of gases on each transformation, V = constant and p = constant, compared to the wall temperature (T_{wall}). Among the objectives of this work was to compare the simulation results of the performance of a six-cylinder, direct-injection, turbocharged compression ignition engine using an elaborate calculation code in FORTRAN with GT-Power software. We also

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studied the influence of certain important thermodynamic and geometric engine parameters on the brake power, on the effective efficiency, and also on pressure and temperature of the gases in the combustion chamber.

2. Diesel engine modeling

There are three essential steps in the mathematical modeling of the internal combustion engine:[4] (1) thermodynamic models based on first and second law analysis, which have been used since 1950 to help engine design or turbocharger matching and to enhance engine processes understanding; (2) empirical models based on input–output relations introduced in early 1970s for primary control investigation; (3) nonlinear models physically based for both engine simulation and control design. Engine modeling for control tasks involves researchers from different fields, mainly control and physics. As a consequence, several specific nominations may designate the same class of model in accordance with the framework. To avoid any misunderstanding, we classify models within three categories with terminology adapted to each field as follows.

1. *Thermodynamic-based models or knowledge (so-called "white box") models* for nonlinear, physically based models suitable for control.
2. *Nonthermodynamic models or "black-box"* models for experimental input–output models.
3. *Semiphysical approximate models or parametric ("gray box") models*. This is an intermediate category; here, models are built with equations derived from physical laws of which parameters (masses, volume, inertia, etc.) are measured or estimated using identification techniques.

The next section focuses on category 1, with greater interest on thermodynamic models. For the second and third classes of models see Guzzella and Amstutz.[5]

2.1. Thermodynamic-based engine model

Thermodynamic modeling techniques can be divided, in order of complexity, into the following groups:[6] (a) quasi-stable; (b) filling and emptying; and (c) the method of characteristics (gas dynamic models). Models that can be adapted to meet one or more requirements for the development of control systems are quasi-steady, filling and emptying, cylinder-to-cylinder engine model, and mean value engine model. Basic classification of thermodynamic models and the emergence of appropriate models for control are shown in Figure 1.

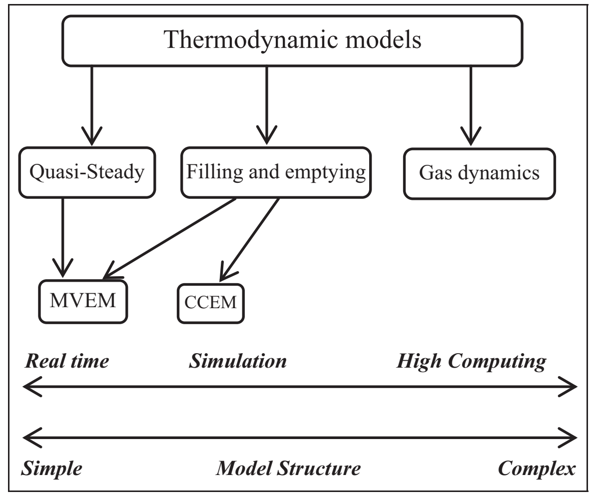

Figure 1. Basic classification of thermodynamic models of internal combustion engines. CCEM: cylinder-to-cylinder engine model; MVEM: mean value engine model.

2.1.1. Quasi-steady method. Ledger and Walmsley[7] and Benson and Baruah[8] performed the first simulations of diesel engines using the quasi-steady method in the early 1970s. The basis of this technique is to model the engine components in a steady state, so that a transient is assumed to be a sequence of steady points. The quasi-steady model includes a crankshaft and the turbocharger dynamics and empirical relations representing the engine thermodynamic. Quasi-steady models are simple and have the advantage of short run times. For this reason, they are suitable for real-time simulations. Among the disadvantages of this model are the strong dependence of the experimental data and the low accuracy. It requires a large amount of data to derive empirical relations or mapping for each engine component, and it cannot be transposed to other engines. Thus, the quasi-steady method is used in the combustion subsystem with mean value engine models to reduce computing time.

2.1.2. Filling-and-emptying method. In 1977, Winterborne et al.[9] and Watson[10] were among the first to develop the model of a diesel engine based on the filling-and-emptying method (also known as zero-dimensional). In 1981, Watson[10] showed that it is possible to reduce the computation time using several time periods depending on the engine cycle. Under the filling-and-emptying concept, the engine is treated as a series of interconnected control volumes (open thermodynamic volume). Indeed, the control volumes are defined for each manifold and each cylinder. Energy and mass conservation equations are applied to every open system with the assumption of a uniform state of gas. The main motivation for the filling-and-emptying technique is to give general engine models with the minimum requirement of empirical data (maps of the turbine and compressor supplied by the manufacturer). In this way, the model can be adapted to other types of engines with minimal effort. The filling-and-emptying model shows good prediction of engine performance under steady-state and transient conditions and provides information about parameters known to affect pollutant or noise. However, assumption of uniform state of gas covers up complex acoustic phenomena (resonance). Wave effects inside the manifold can affect engine performance and, therefore, the error introduced by the filling-and-emptying method must be considered.

2.1.3. Method of characteristics (or gas dynamic models). The method of characteristics is a very powerful method to access accurately parameters such as the equivalence ratio or the contribution to the overall noise sound level of the intake and the exhaust manifold or the amplitudes of the pressure fluctuations in the tubes according to their diameter. Its advantage is effectively understanding the mechanism of the phenomena that happen in a manifold[11] and it allows one to obtain accurately laws of evolution of pressure, speed, and temperature manifolds at any point, depending on the time, but the characteristic method requires a much more important calculation program, and the program's complexity increases widely with the number of singularities to be treated.

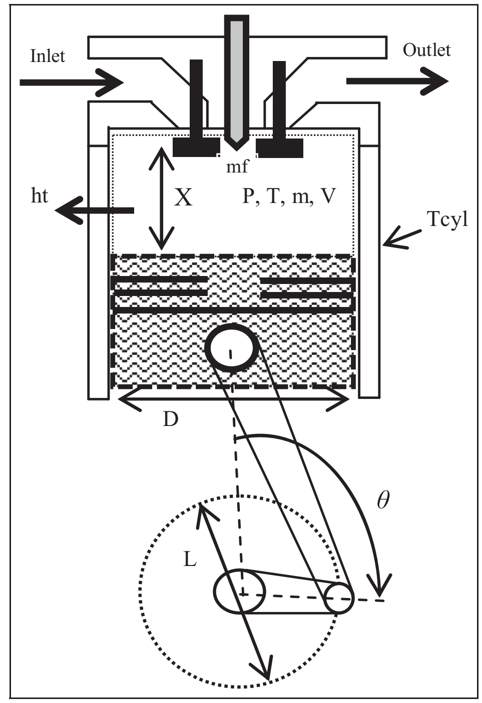

Figure 2. Cylinder scheme and its variables. P: pressure, T: temperature, m: mass, V: volume.

3. General equation of the model

In this work, a zero-dimensional model proposed by Watson et al.[12] is developed that gives a satisfactory combustion heat to calculate the thermodynamic cycle (see Figure 2). In this model, it is assumed that engine plenums (cylinders, intake, and exhaust manifolds) are modeled as separate thermodynamic systems containing gases at uniform state. With respect to the filling-and-emptying method, mass, temperature, and pressure of gas are calculated using the first law and mass conservation. Ideal gases with constant specific heats, effects of heat transfer through intake, and exhaust manifolds are neglected; compressor inlet and turbocharger outlet temperatures and pressures are assumed to be equal to ambient pressure and temperature. From the results of Rakapoulos et al.,[13] temperatures of the cylinder head, cylinder walls, and piston crown are assigned constant values. The crank speed is uniform (steady-state engine).

The rate of change of the volume with respect to time is given[14,15] as follows:

$$V_{cyl}(t) = V_{clear} + \frac{\pi D^2 L}{4} \left(1 + \beta_{mb}(1 - \cos(\omega t)) - \sqrt{1 - \beta_{mb}{}^2 \sin^2(\omega t)}\right) \tag{1}$$

where t is time (s), ω is rotational engine speed (rad/s), $V_{clear} = \frac{V_{cyl}(t)}{c}$ is clearance volume (c_r is the compression ratio), $\beta_{mb} = \frac{2l}{L}$ is the connected rod length-to-crank radius ratio (l is the connecting rod length (m), L is the piston stroke (m)), and D = cylinder bore (m).

3.1. Mass entering the cylinder

The conservation equation of the mass applied to the cylinder is

$$\frac{dm_{cyl}}{dt} = m_f + \mathrm{m}_{in} - m_{out} \tag{2}$$

3.2. Ideal gas

The ideal gas model[16] gives the relationship between the mass m_{cyl}, the cylinder volume V_{cyl}, the pressure p_{cyl}, and the temperature T_{cyl}:

$$\frac{dT_{cyl}}{dt} = \frac{1}{m_{cyl} C_v}\left(\frac{dQ}{dt} - p_{cyl}\frac{dV_{cyl}}{dt}\right) \tag{3}$$

The change of cylinder pressure with respect to Equations (2) and (3) is defined as follows:

$$\frac{dp_{cyl}}{dt} = \frac{\gamma}{V_{cyl}}[RT_{in}m_{in} - RT_{cyl}m_{cyl} - p_{cyl}V_{cyl}] \\ \frac{\gamma - 1}{V_{cyl}}[m_{fb}Q_{LHV} - Q_{ht}] \tag{4}$$

3.3. Engine heat transfer, combustion, and friction loss models

3.3.1. Heat exchange correlation. For calculating the instantaneous heat flux out of the engine (Q_{ht}), the Woschni correlation modified by Hohenberg[17] is used:

$$\frac{dQ_{ht}}{dt} = A_{cyl} h_t (T_{cyl} - T_{wall}) \tag{5}$$

The heat transfer coefficient (h_t; in kW/K·m^2) at a given piston position, according to Hohenberg's[17] correlation is

$$h_t(t) = k_{\text{hoh}}\, p_{cyl}^{0.8} V_{cyl}^{-0.06} T_{cyl}^{-0.4} (\bar{v}_{pis} + 1.4)^{0.8} \tag{6}$$

where k_{hoh} is the constant of Hohenberg, which characterizes the engine (k_{hoh} = 130).

3.3.2. Engine combustion model. The model of Watson et al.[12] is chosen in this work to develop equations for fuel energy release appropriate for diesel engine simulations. In their development, the combustion process starts from a faster premixed burning phase $\left(\frac{dm_{fb}}{dt}\right)_p$ followed by a slower diffusion combustion phase $\left(\frac{dm_{fb}}{dt}\right)_d$. So the heat release ($Q_{comb}$) is related to the burned fuel mass rate $\frac{dm_{fb}}{dt}$ and enthalpy of formation of the fuel (h_{for}):

$$\frac{dQ_{comb}}{dt} = \frac{dm_{fb}}{dt} h_{for} \tag{7}$$

The burned mass of fuel is related to the normalized burned fuel mass rate $\frac{dm_{fb}^*}{dt}$ and the injected mass per cycle (m_f) and the duration of combustion ($\Delta\theta_{comb}$):

$$\frac{dm_{fb}}{dt} = \frac{dm_{fb}^*}{dt} \frac{m_f}{\Delta t_{comb}} \tag{8}$$

The combustion process is described using an empirical model, the zero-dimensional model obtained by Watson et al.[12]:

$$\frac{dm_{fb}^*}{dt} = \beta \left(\frac{dm_{fb}}{dt}\right)_p + (1-\beta)\left(\frac{dm_{fb}}{dt}\right)_d \tag{9}$$

3.3.3. Friction losses. The friction mean effective pressure is calculated[2] by

$$fmep = C + (0.005 p_{\max}) + 0.162 \bar{v}_{pis} \tag{10}$$

3.4. Effective power and effective efficiency

For the four-stroke engine, the effective power is calculated[18] as

$$bpower = bmep V_d N_{cyls} N/2 \tag{11}$$

where $V_d = \frac{\pi D^2 S}{4}$ and N_{cyls} is the cylinder number.

Table 1. Engine specifications.[20]

Engine specifications	Unit	Value
D, cylinder bore	mm	120.0
S, stroke	mm	175.0
l, connecting rod length	mm	300.0
c_r, compression ratio	—	16.0
Inlet valve diameter	mm	60
Exhaust valve diameter	mm	38
IVO, inlet valve open	°CA	314
IVC, inlet valve close	°CA	− 118
EVO, exhaust valve open	°CA	100
EVC, exhaust valve close	°CA	400
Fuel system	—	Direct injection
Injection pressure	bar	1000
Start of injection before top dead center (bTDC)	°CA	15° bTDC
Number of holes per nozzle	—	8
Nozzle hole diameter	mm	0.25

The effective efficiency is calculated[19] as

$$R_{eff} = Wd/Q_{comb} \tag{12}$$

4. Engine simulation programs

4.1. Computing steps of the developed model

The governing equations (Equations (1)–(12)) of the comprehensive model subjected to the assumptions previously mentioned in the mathematical model are solved for different processes of the engine cycle. The fourth-order Runge–Kutta method is used to simulate the comprehensive model equations at the prescribed initial conditions. The engine cycle starts from the moment the exhaust valve closes. The developed power cycle simulation program includes a main program as an organizational routine, but which incorporates a few technical calculations, and also several subroutines. The computer program works out in discrete crank angle incremental steps from the outset the compression, combustion, and expansion stroke. For the closed cycle period, Watson et al.[12] recommended the following engine calculation crank angle steps: 10°CA before ignition, 1°CA at fuel injection timing, 2°CA between ignition and combustion end, and finally 10°CA for expansion.[15] The solution is carried out with end results of each process taken as the starting conditions for the following process and end values of the completed thermal cycle taken as the starting values for the subsequent cycle. Inputs to the program are the engine specifications and operating conditions listed in Table 1. The numerical output results are the instantaneous cylinder pressure, temperature, and engine performance variables, such as brake power and effective efficiency. For more details of the theoretical parts, see Menacer and Bouchetara.[15]

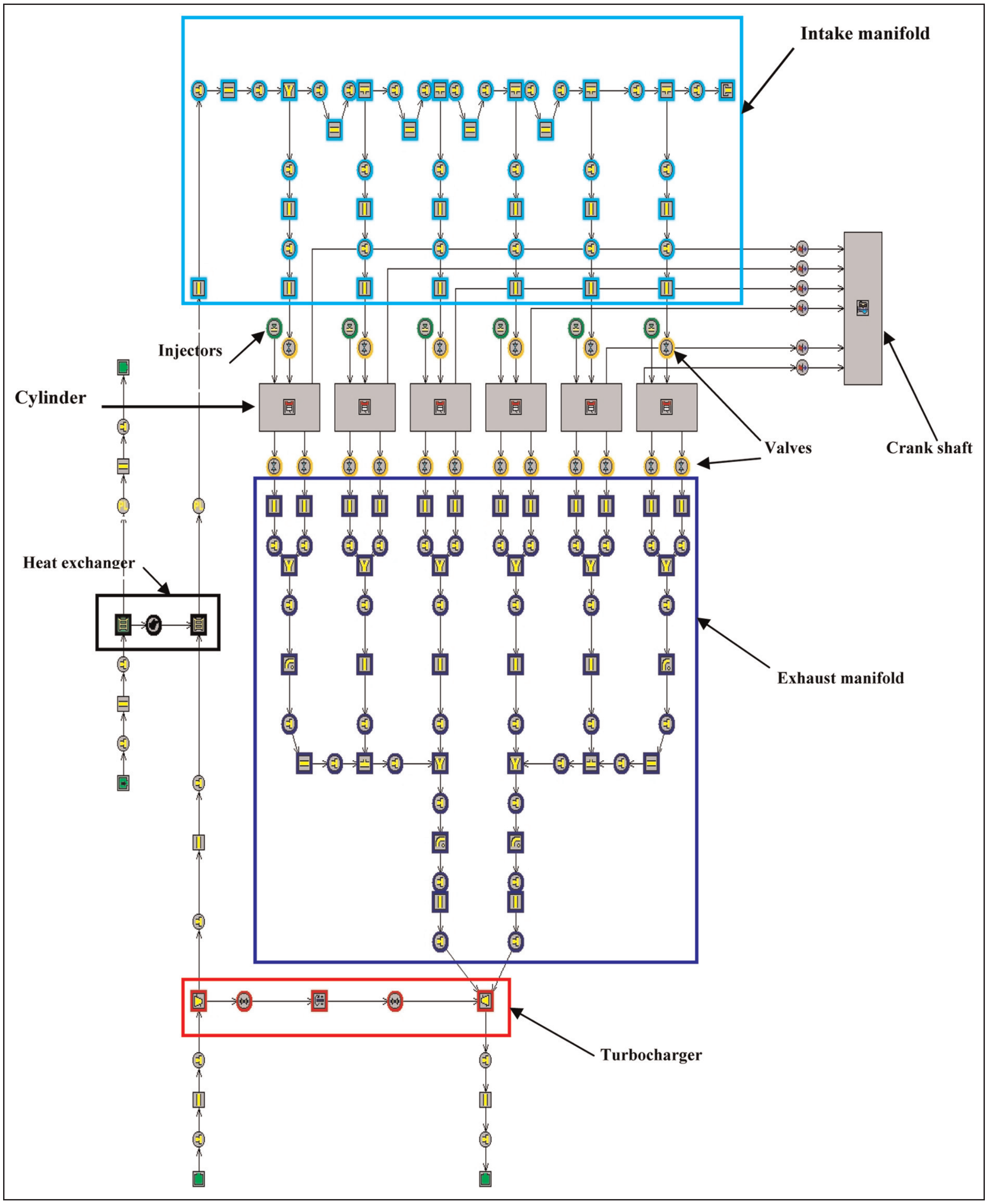

Figure 3. Developed model in the commercial engine simulation code GT-Power.[15]

4.2. Commercial engine simulation code

In recent years, a few commercial engine simulation codes, such as WAVE from Ricardo or GT-Power from Gamma Technologies, have allowed co-simulations with software dedicated to control and modeling. Optimized codes and present computer power make possible detailed engine simulation within time scales adapted to control system development. These codes give full phenomenological representation of the engine system with one-dimensional compressible flow formulation. In addition, the code contains several models for heat transfer or combustion. Diesel engine combustion can be modeled with simple combustion rate calculation (two functions Wiebe model) or with a more sophisticated diesel jet model.[20] GT-Power is an object-based code including object libraries for engine components (pipes, cylinder, cranktrain, compressor, valve, etc.). Building an engine model with GT-Power consists of parameterization and connection of these

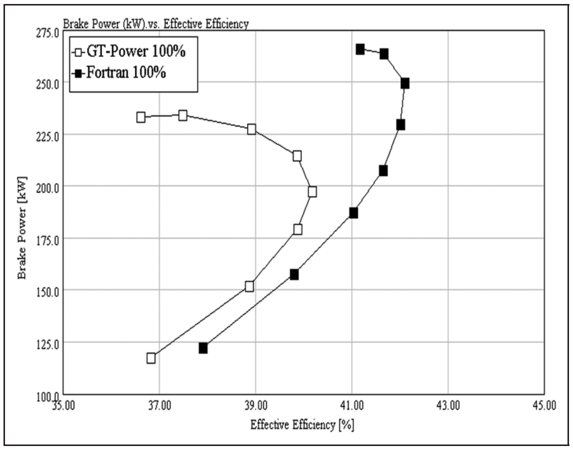

Figure 4. Brake power versus effective efficiency for full load: T_{inj} = 15° bTDC; D_{cyl} = 120 mm; c_r = 16:1; T_{wall} = 450 K; R_{sb} = 1.5.

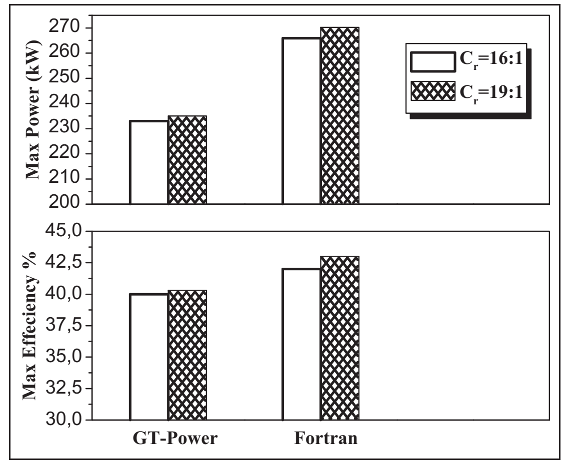

Figure 5. Maximum power and maximum efficiency for different compression ratios.

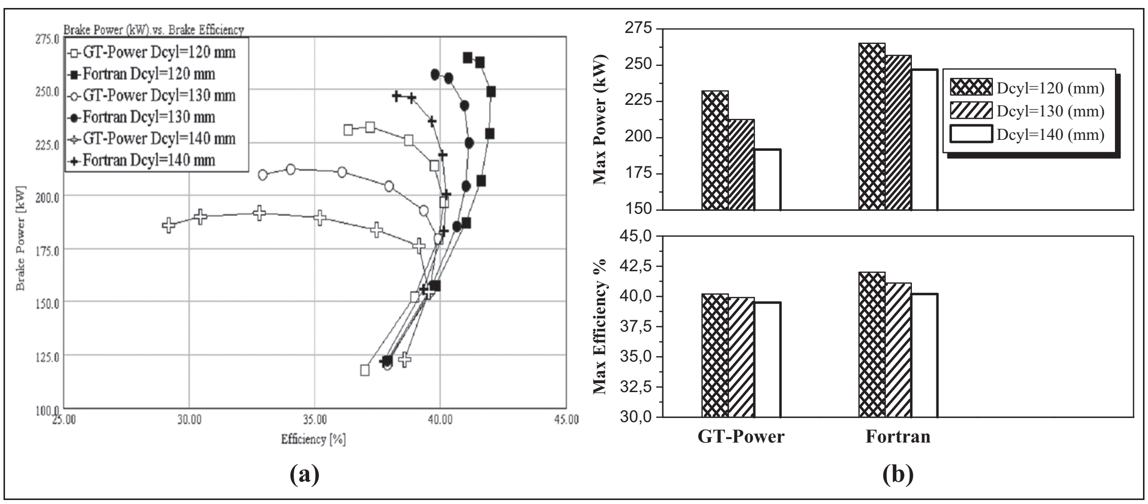

Figure 6. (a) Cylinder bore influence for 100% load: T_{inj} = 15° bTDC; D_{cyl} = 120 mm; c_r = 16:1. (b) Maximum power and maximum efficiency for different cylinder bore: D_{cyl} = 120, 130, and 140 mm.

objects. Figure 3 shows the developed model in the commercial engine simulation code GT-Power. In the modeling view, the line of the exhaust manifold is composed in three volumes: the cylinders are grouped by three and emerge on two independent manifolds—these two components are considered as open thermodynamic systems with identical volumes—and the third control volume is used to ensure the junction with the inlet of the turbine. The turbocharger consists of an axial compressor connected to a turbine by a shaft; the turbine is driven by exhaust gas to power the compressor. So more air can be added into the cylinders, allowing an increasing amount of fuel to be burned compared to a naturally aspirated engine.[21] The heat exchanger can be assimilated to an intermediate volume between the compressor and the intake manifold; it solves a system of differential equations supplementary identical to the manifold. It appeared to assimilate the heat exchanger as a nondimensional organ.

5. Results of engine simulation

The thermodynamic and geometric parameters chosen in this study were as follows:

- *engine geometry*: compression ratio (c_r), cylinder bore (D), and more particularly the stroke–bore ratio $\left(R_{sb} = \frac{L}{D}\right)$;
- *combustion parameters*: injected fuel mass (m_f), crankshaft angle marking the injection timing (T_{inj}), and cylinder wall temperature (T_{wall}).

Table 1 shows the main parameters of the chosen direct-injection diesel engine.[15,20]

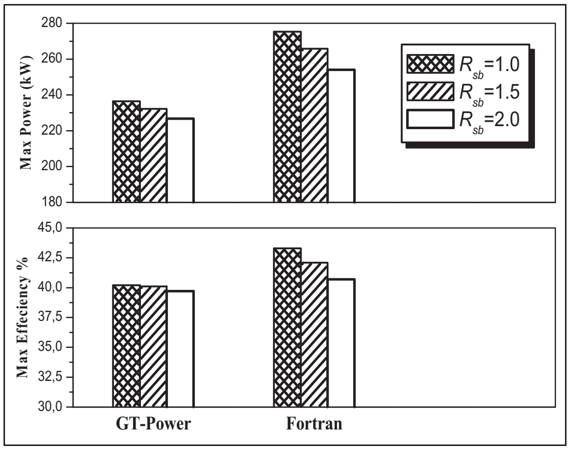

Figure 7. Maximum power and maximum efficiency for different stroke–bore ratios: R_{sb} = 1.0, 1.5, and 2.0.

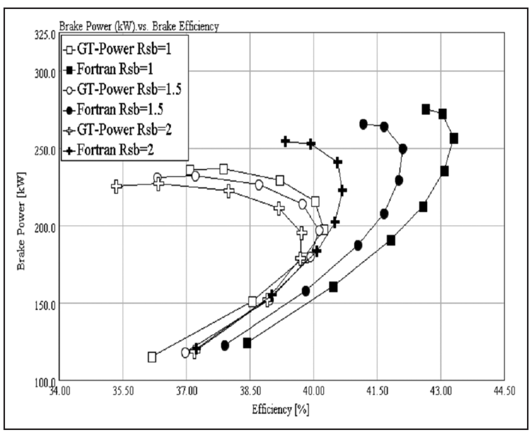

Figure 8. Influence of the stroke–bore ratio for 100% load: T_{inj} = 15° bTDC; v_{cyl} = 2.0 L; c_r = 16:1; T_{wall} = 450 K.

Figure 4 presents two operating points for the engine: the maximal effective efficiency and the maximal brake power.

5.1. Influence of the geometric parameters

5.1.1. The compression ratio. In general, increasing the compression ratio improved the engine performances.[15] Figure 5 shows the compression ratio influence (c_r = 16:1 and 19:1) on the maximum brake power and maximum effective efficiency at full load, advance injection of 15° bTDC (before top dead center) for GT-Power, and the elaborate software.

5.1.2. The cylinder bore. Figures 6(a) and (b) show the influence of the cylinder bore on the brake power and effective efficiency at full load 100%, a compression ratio of 16:1, and advance injection of 15° bTDC. If the cylinder bore increases by 10 mm (from 130 to 140 mm), the brake efficiency decreases by 2% and the effective power by 9%. From Figure 6(b), the cylinder bore has a strong influence on the maximum brake power and a small influence on effective efficiency.

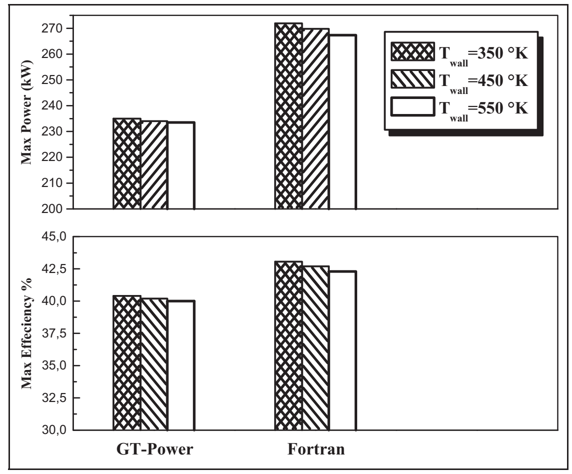

Figure 9. Maximum power and maximum efficiency for different cylinder wall temperatures.

5.1.3. The stroke–bore ratio. Figure 7 shows the variation of the brake power and the effective efficiency for different stroke–bore ratios: R_{sb} = 1.0, 1.5, and 2.0.

The stroke–bore ratio is another geometric parameter that influences the performances of a turbocharged diesel engine. The cylinder volume of 2.0 L can be obtained by a different manner while varying this parameter; its influence is shown in Figure 8. If the stroke–bore ratio increases, the mean piston speed is greater, and friction losses (Equation (10)) are important with increasing the engine speed. The effective power and brake efficiency decrease with the increase of the stroke–bore ratio. If the stroke–bore ratio augments are 0.5 (of 1.5–2) then the maximum brake efficiency decreased by an average of 3%, and the maximum effective power by 4%.

5.2. Influence of the thermodynamic parameters

5.2.1. The cylinder wall temperature. Figure 9 shows the variation of the maximum brake power and the maximum effective efficiency for different cylinder wall temperature, T_{wall} = 350, 450, and 550 K.

The effect of T_{wall} is represented also in Figure 10; when the cylinder wall temperature is lower, then the brake efficiency increase. When the difference in temperature between the gas and cylinder wall is small, then the losses by heat convective are high.[20] The effective efficiency increases until its maximum value; afterwards it starts to fall until a maximal value of the effective force is reached.

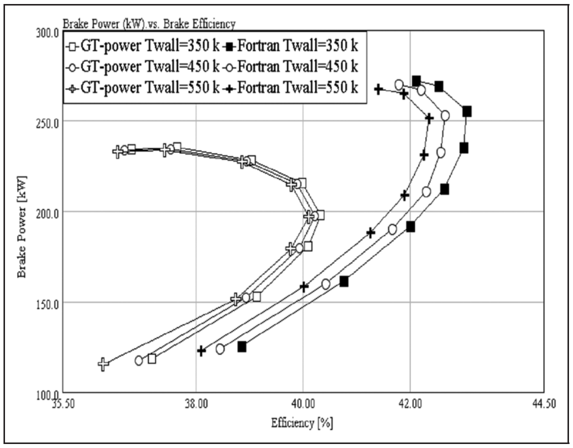

Figure 10. Wall temperature influence for 100% load: T_{inj} = 15° bTDC; D_{cyl} = 120 mm; c_r = 16:1; R_{sb} =1.5.

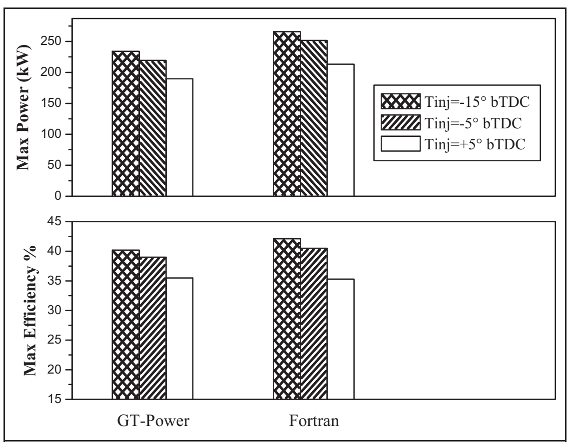

Figure 11. Maximum power and maximum efficiency for different injection timings.

It is also valid for the effective power. If the cylinder wall temperature increases by 100 K (from 350 to 450 K), the maximum brake power and effective efficiency decrease respectively by about 0.7%. The maximum operating temperature of an engine is limited by the strength and geometric variations due to thermal expansion, which can be a danger of galling. Improved heat transfer to the walls of the combustion chamber lowers the temperature and pressure of the gas inside the cylinder, which reduces the work transferred to the piston cylinder and reduces the thermal efficiency of the engine. It is thus advantageous to cool the cylinder walls provided they do not do it too vigorously.

5.2.2. The advanced injection. Figure 11 shows the influence of different injection timings on the maximum brake power and maximum effective efficiency as calculated by Fortran and GT-Power software systems. This parameter had a substantial influence on the maximum power and a smaller influence on maximum efficiency.[15]

Figures 12(a) and (b) present the influence of the injection timing and its impact on the thermodynamic cycle of the pressure and temperature in the cylinder. When the injection starts at 15° bTDC the maximal pressure and temperature are higher, and the temperature at the exhaust is lower than if the injection timing occurs at 5° after TDC (top dead center).[22,23] In this case the combustion begins, whereas when the piston starts its descent the duration of heat exchange losses is lower, and then the exhaust temperature is higher.

5.2.3. The injected fuel mass. Figures 13 and 14 shows the variation of the brake power and effective efficiency for different fuel masses injected at advance injection of 15° bTDC, compression ratio of 16:1, and N = 1400 rpm. This parameter has a strong influence on the maximum power

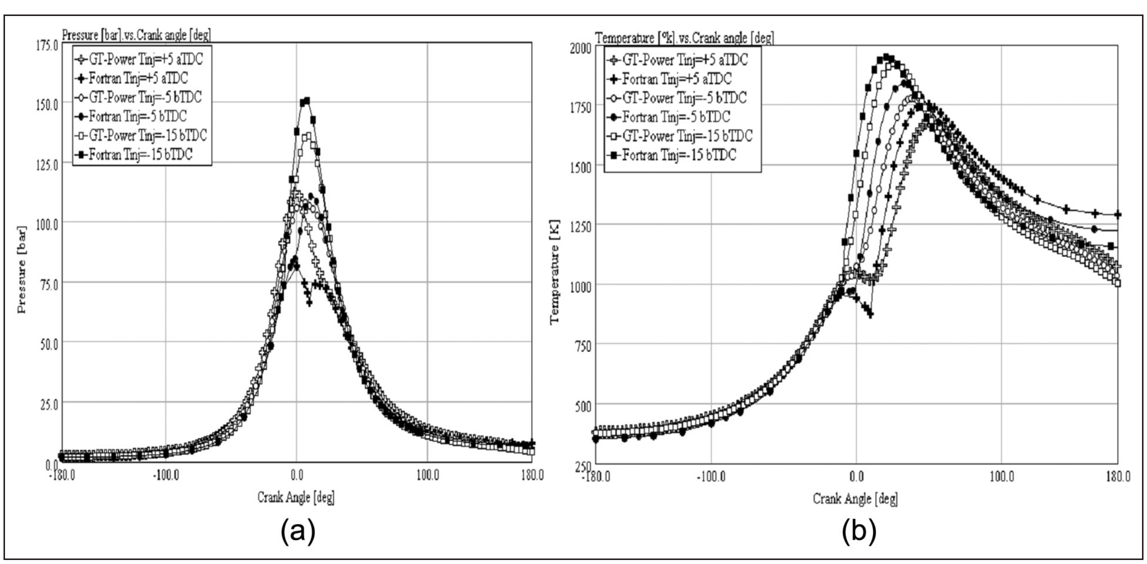

Figure 12. (a) Injection timing influence on gas pressure versus crankshaft for 100% load: D_{cyl} = 120 mm; c_r = 16:1; T_{wall} = 450 K; R_{sb} = 1.5; N = 1400 rpm. (b) Injection timing influence on gas temperature versus crankshaft for 100% load: D_{cyl} = 120 mm; c_r = 16:1; T_{wall} = 450 K; R_{sb} = 1.5, N = 1400 rpm.

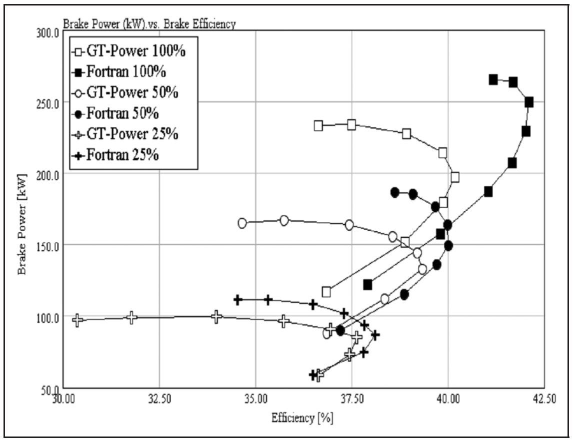

Figure 13. Influence of injected fuel mass for T_{inj} = 15° bTDC, D_{cyl}= 120 mm, c_r = 16:1, T_{wall} = 480 K, and β_{mb} =1.5.

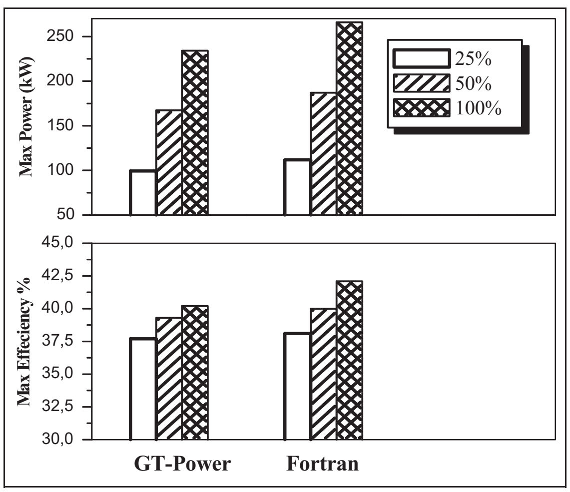

Figure 14. Maximum power and maximum efficiency for different fuel masses injected: 25%, 50%, and 100%.

and less on the maximum efficiency. The brake power and effective efficiency increases with increasing the quantity of fuel injected. If the mass fuel injected in the cylinder increases by 50% (from 50% to 100%), the maximum effective efficiency increases by 3.5% and the maximum brake power by 28.5%. This shows the importance of the variation of the quantity of injected fuel on the effective power and the brake efficiency.

6. Conclusion

This work describes a turbocharged direct-injection compression ignition engine simulator. Effort has been put into building a physical model based on the filling-and-emptying method. The resulting model can predict the engine performances. From the thermodynamic model we were able to develop an interrelationship between the brake power and the effective efficiency that was related to the corresponding speed for different parameters studied; as a result, we determined a maximum power corresponding to the state of an engine's optimal speed and a maximum economy corresponding to the optimal speed. We studied the influence of a number of parameters on engine power and efficiency: the stroke–bore ratio and the cylinder wall temperature have a small influence on the brake power and effective efficiency, while the cylinder bore, angle of start injection, mass fuel injected, and compression ratio have a great influence on the brake power and effective efficiency. This analysis has been completed by representation of the pressure diagram for the various crankshaft angles, and the corresponding gas temperature versus crankshaft angle. The engine simulation model described in this work is valid for a steady engine speed. In future work, we aim to replace the simple model for the fraction of mass fuel burned by a predictive model, to validate this model for transient engine speed and to take into account gas characteristics and specific heat fluctuation.

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