ISSN 1392–1207. MECHANIKA. 2015 Volume 21(1): 40–50

Computational thermodynamic of a turbocharged direct injection diesel engine

B. Menacer*, M. Bouchetara**
**University of Sciences and the Technology of Oran Algeria, Email: acer.msn@hotmail.fr*
***University of Sciences and the Technology of Oran Algeria, Email: mbouchetara@hotmail.com*

crossref http://dx.doi.org/10.5755/j01.mech.21.1.8690

1. Introduction

More than one century after his invention by Dr. Rudolf Diesel, the compression ignition engine remains the most efficient internal combustion engines 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 details. However, these models need a precise experimental input and substantial computational power, which make 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 of 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 the thermal loss is represented more classically in the form of a thermal conductance between the mean temperature of gases, on each transformation V = constant, p = constant, compared to the wall temperature T_{wall}. Among the objectives of this work is to conduct a comparative study of simulation results of the performances of a six cylinder direct injection turbocharged compression ignition engine obtained with the elaborate calculation code in FORTRAN and those with the software GT-Power. We also 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 modelling of internal combustion engine [4, 5]: a. thermodynamic models based on first and second law analysis, they are used since 1950 to help engine design or turbocharger matching and to enhance engine processes understanding; b. empirical models based on input-output relations introduced in early 1970s for primary control investigation; c. 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:

- thermodynamic-based models or knowledge models (so-called "white box") for nonlinear model physically-based suitable for control;
- non-thermodynamic models or "black-box" models for experimental input-output models;
- semiphysical approximate models or parametric models (so-called "grey-box"). It is an intermediate category, here, model are built with equations derived from physical laws of which parameters (masses, volume, inertia, etc.) are measured or estimated using identification techniques.

Next section focuses on category 1 with greater interest on thermodynamic models. For the second and third class of models see [6].

2.1. Thermodynamic-based engine model

Thermodynamic modeling techniques can be divided, in order of complexity, in the following groups [7]: a. quasi-stable; b. filling and emptying; c. the method of characteristics (gas dynamic models). Models that can be adapted to meet one or more requirements for the

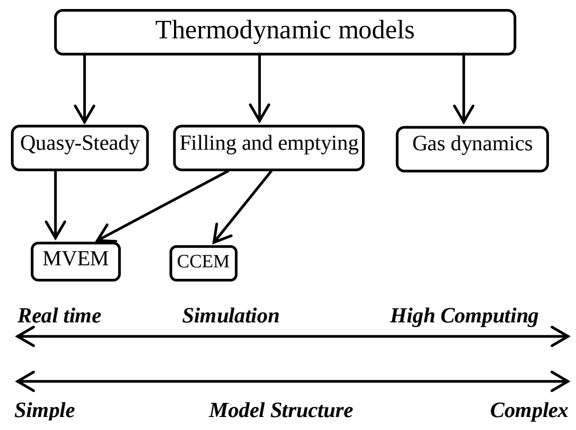

Fig. 1 Basic classification of thermodynamic models of internal combustion engines

development of control systems are: quasi-steady, filling and emptying, cylinder-to-cylinder (CCEM) and mean value models (MVEM). Basic classification of thermodynamic models and the emergence of appropriate models for control are shown in Fig. 1.

2.1.1. Quasi-steady method

The quasi-steady model includes crankshaft and the turbocharger dynamics and empirical relations representing the engine thermodynamic [8, 9]. Quasi-steady models are simple and have the advantage of short run times. For this reason, they are suitable for real-time simulation. Among the disadvantages of this model was the strong dependence of the experimental data and the low accuracy. 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

Under the filling and emptying concept, the engine is treated as a series of interconnected control volumes (open thermodynamic volume) [10, 11]. Energy and mass conservation equations are applied to every open system with the assumption of uniform state of gas. The main motivation for filling and emptying technique is to give general engine models with the minimum requirement of empirical data (maps of turbine and compressor supplied by the manufacturer). In this way, the model can be adapted to other types of engines with minimal effort. 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, assumptions of uniform state of gas cover up complex acoustic phenomena (resonance).

2.1.3. Method of characteristics (or gas dynamic models)

It 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. Its advantage is effectively understood the mechanism of the phenomena that happen in a manifold [12] and, allows 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.

3. General equation of the model

In this work we developed a zero-dimensional model proposed by Watson et al [11], which gives a satisfactory combustion heat to calculate the thermodynamic cycle. In this model, it is assumed that: engine plenums (cylinders, intake and exhaust manifolds) are modelled as separate thermodynamic systems containing gases at uniform state. The pressure, temperature and composition of the cylinder charge are uniform at each time step, which is to say that no distinction is made between burned and unburned gas during the combustion phase inside the cylinder. With respect to the filling and emptying method, mass, temperature and pressure of gas are calculated using 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 as follows (Fig. 2):

$$V_{cyl}(t) = V_{clear} + \frac{\pi D^2 L}{4}\Big(1+\beta_{mb}\big(1-cos(\omega t)\big)- \\ -\sqrt{1-\beta_{mb}{}^2 sin^2(\omega t)}\Big), \qquad (1)$$

where t is time measured with respect to TDC [s], ω is engine speed, rad/s; V_{clear} is clearance volume $V_{clear} = V_{cyl}(t)/c_r$; c_r is compression ratio; $\beta_{mb} = 2l/L$ is ratio of connected rod length to crank radius; l connecting rod length, m; L is the piston stroke, m, D is the cylinder bore, m.

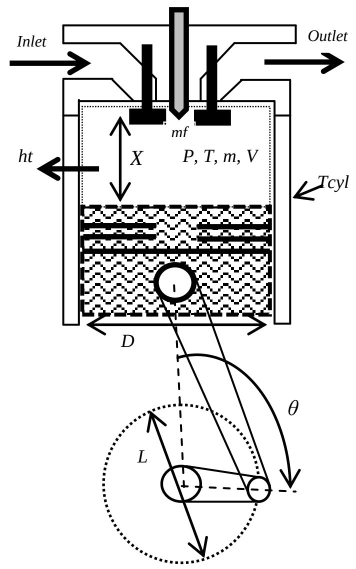

Fig. 2 Cylinder scheme and its variables: (*P* - pressure, *T* - temperature, *m* - mass, *V* - volume)

3.1. Fuel burning rate

There are two empirical models to determine the fuel burning rate: the simple Vibe law and the modified or double Vibe function following the Watson and al. model. In this simulation, we chose the single zone combustion model proposed by Watson et al. [4]. This correlation developed from experimental tests carried out on engines with different characteristics in different operating regimes. This model reproduces in two combustion phases; the first is the faster combustion process, said the premixed combustion and the second is the diffusion combustion which is slower and represents the main combustion phase.

During combustion, the amount of heat release $\dot{Q}_{comb}$ is assumed proportional to the burned fuel mass:

$$\frac{dQ_{comb}}{dt} = \frac{dm_{fb}}{dt} h_{for} , \quad (2)$$

$$\frac{dm_{fb}}{dt} = \frac{d\dot{m}_{fb}}{dt} \frac{m_f}{\Delta t_{comb}} \quad . \quad (3)$$

The combustion process is described using an empirical model, the single zone model obtained by Watson et al. [4]:

$$\frac{d\dot{m}_{fb}}{dt} = \beta \left(\frac{dm_{fb}}{dt} \right)_p + (1-\beta) \left(\frac{dm_{fb}}{dt} \right)_d , \quad (4)$$

where $\frac{dQ_{comb}}{dt}$ is rate of heat release during combustion, kJ/s; $\frac{dm_{fb}}{dt}$ is Burned fuel mass rate, kg/ s; h_{for} is enthalpy of formation of the fuel, kJ/kg; $\frac{d\dot{m}_{fb}}{dt}$ is normalized burned fuel mass rate; m_f is injected fuel mass per cycle, kg/cycle; $\left(\frac{dm_{fb}}{dt} \right)_p$ is normalized fuel burning rate in the premixed combustion; $\left(\frac{dm_{fb}}{dt} \right)_d$ is normalized fuel burning rate in the diffusion combustion; β is fraction of the fuel injected into the cylinder and participated in the premixed combustion phase. It depends on the ignition delay τ_{id} described by Arrhenius formula [14] and the equivalence ratio ϕ :

$$\beta = 1 - \beta_1 \phi^{\beta_2} / \tau_{id}^{\beta_3} , \quad (5)$$

where $\beta_1, \beta_2, \beta_3$ are empirical constants for fuel fraction in the premixed combustion ($\beta_1 = 0.90, \beta_2 = 0.35, \beta_3 = 0.40$); ϕ is fuel-air equivalence ratio.

The equivalence ratio ϕ is defined as:

$$\phi = \left(\frac{m_{fb}}{m_a} \right) \Big/ \phi_s , \quad (6)$$

where m_a is mass air participating in fuel combustion, kg; ϕ_S is stoichiometric fuel-air ratio.

In diesel engine, in which quality governing of mixture is used, the equivalence ratio varies greatly depending on the load.

The fuel burned mass m_{fb} is written as follows:

$$m_{fb} = \frac{m_{cyl} \phi_s \phi}{1 + \phi_s \phi} . \quad (7)$$

From the Eqs. (6) and (7), one obtains the state equation of the equivalence ratio [15]:

$$\frac{d\phi}{dt} = \left(\frac{1+\phi_s \phi}{m_{cyl}} \right) \left(\frac{1+\phi_s \phi}{\phi_s} \frac{dm\,fb}{dt} - \phi \frac{dm_{cyl}}{dt} \right) \quad (8)$$

The ignition delay τ_{id} is the period between injection time and ignition time and it calculated by Arrhenius formula, ms:

$$\tau_{id} = k_1 \, \bar{p}_{cyl}^{k_2} \, e^{\left(\frac{k_3}{\bar{T}_{cyl}} \right)} , \quad (9)$$

where $\bar{p}_{cyl}$ and $\bar{T}_{cyl}$ are average values of the pressure and temperature in the cylinder when the piston is at the top dead center; $k_1 = 0.0405$; $k_2 = 0.757$; $k_3 = 5473$ are these coefficients are experimentally determined on rapid compression engines and valid for the cetane number between 45 and 50, [16].

3.1.1. Fuel burning rate during the premixed combustion

The normalized fuel burning rate in the premixed combustion is [7, 11]:

$$\left(\frac{dm_{fb}}{dt} \right)_p = C_{1p} C_{2p} t_{norm}^{(C_{1p}-1)} \left(1 - t_{norm}^{C_{1p}} \right)^{(C_{2p}-1)} , \quad (10)$$

$$t_{norm} = \frac{t - t_{inj}}{\Delta t_{comb}} = \left(\theta - \theta_{inj} \right) / \Delta\theta_{comb} , \quad (11)$$

where t_{norm} is normalized time vary between 0 (ignition beginning or injection time) and 1 (combustion end); $\Delta t_{comb}, \Delta\theta_{comb}$ is combustion duration, s, °CA; t_{inj}, θ_{inj} is injection time and angle, s, °CA; t, θ is actual time and angle, s, °CA; C_{1p}, C_{2p} are constants model of the premixed combustion: $C_{1p} = 2 + 1.25 \times 10^{-8} \left(\tau_{id} N \right)^{2.4}$, $C_{2p} = 5000$.

3.1.2. Fuel burning rate during the diffusion combustion

The fuel burning rate in the diffusion combustion is calculated as [11]:

$$\left(\frac{dm_{fb}}{dt} \right)_d = C_{3d} C_{4d} t_{norm}^{(C_{4d}-1)} e^{\left(-C_{3d} t_{norm}^{C_{4d}} \right)} , \quad (12)$$

where C_{3d}, C_{4d} are constants of the diffusion combustion model, then: $C_{3d} = 14.2 / \phi_{tol}^{0.644}$, $C_{4d} = 0.79 C_{3d}^{0.25}$.

3.2. Heat transfer in the cylinder

Heat transfer affects engine performance and efficiency. The heat transfer model takes into account the forced convection between the gases trapped into the cylinder and the cylinder wall. The heat transfer by conduction and radiation in the engine block are much less important than the heat transfer by convection [17]. The instantaneous convective heat transfer rate from the in-cylinder gas to cylinder wall $\dot{Q}_{ht}$ is calculated by:

$$\frac{dQ_{ht}}{dt} = A_{cyl} h_t \left(T_{cyl} - T_{wall} \right), \tag{13}$$

where T_{wall} is temperature walls of the combustion chamber (bounded by the cylinder head, piston head and the cylinder liner). From the results of Rakapoulos et al. [13], T_{wall} is assumed constant.

The instantaneous heat exchange area A_{cyl} can be expressed roughly by the following relation:

$$A_{cyl} = \left(\alpha_p + \alpha_{ch} \right) \frac{\pi D^2}{4} + \pi D \frac{S}{2} \times \\ \times \left(\frac{l}{r} + 1 - cos\left(\omega t \right) - \sqrt{\left(\frac{l}{r} \right)^2 - sin^2\left(\omega t \right)} \right), \tag{14}$$

where α_p is coefficient shape of the piston head; α_{ch} is coefficient shape of the cylinder head, (for flat area $\alpha_{p,ch} = 2$ and for no flat area $\alpha_{p,ch} > 2$).

The global heat transfer coefficient in the cylinder can be estimated by the empirical correlation of Hohenberg which is a simplification of the Woschni correlation; it presents the advantage to be simpler of use and is the most adequate among all available relations to compute the heat transfer rate through cylinder walls for diesel engine [18].

The heat transfer coefficient h_t, kW/K m^2 at a given piston position, according to Hohenberg's correlation [18] is:

$$h_t(t) = k_{hoh} p_{cyl}^{0.8} V_{cyl}^{-0.06} T_{cyl}^{-0.4} (\bar{v}_{pis} + 1.4)^{0.8} \tag{15}$$

where p_{cyl} is cylinder pressure; V_{cyl} is in-cylinder gas volume at each crank angle position; k_{hoh} is constant of Hohenberg which characterize the engine ($k_{hoh} = 130$).

The mean piston speed $\bar{v}_{pis}$, m/s is equal to:

$$\bar{v}_{pis} = 2 \times S \times N \tag{16}$$

where N is engine speed, rpm.

3.3. Energy balance equations

In the filling and empting method, only the law of conservation energy is considered. The energy balance of the engine for a control volume constituted by the cylinder gasses is established over a complete cycle:

$$\frac{dU}{dt} = \frac{dW}{dt} + \frac{dQ}{dt} \tag{17}$$

where U is the internal energy, W is the external work and $\dot{Q}$ is the total heat release during the combustion.

The internal energy U per unit mass of gas is calculated from a polynomial interpolation deduced from the calculation results of the combustion products at equilibrium for a reaction between air and fuel $C_n H_{2n}$. The polynomial interpolation is a continuous function of temperature and equivalence ratio. It is valid for a temperature range T between 250°K and 2400°K and equivalence ratio ϕ between 0 and 1.6. To determine the change in internal energy, we use the expressions of Krieger and Borman [19]:

$$\frac{dU}{dT} = \left(\frac{dA}{dT} - \frac{dB}{dT} \phi \right) \Big/ \left(1 + \phi_s \phi \right), \tag{18}$$

$$\frac{dA}{dT} = C_0 + C_1 T + C_2 T^2 - C_3 T^3 + C_4 T^4, \tag{19}$$

$$\frac{dB}{dT} = -C_5 - C_6 T + C_7 T^2 - C_8 T^3, \tag{21}$$

where $\frac{dA}{dT}, \frac{dB}{dT}$ are interpolation polynomial of Krieger and Borman; $C_0, C_1, C_2, C_3, C_4, C_5, C_6, C_7, C_8$ are Krieger and Borman constants.

The work rate is calculated from the cylinder pressure and the change in cylinder volume:

$$\frac{dW}{dt} = -p_{cyl} \frac{dV_{cyl}}{dt}. \tag{20}$$

The total heat release $\dot{Q}$ during the combustion is divided in four main terms:

$$\frac{dQ}{dt} = \frac{dQ_{in}}{dt} + \frac{dQ_{comb}}{dt} - \frac{dQ_{out}}{dt} - \frac{dQ_{ht}}{dt}, \tag{21}$$

with

$$\begin{cases} \frac{dQ_{in}}{dt} = C_p \dot{m}_{in} T_a \\ \frac{dQ_{out}}{dt} = C_p \dot{m}_{out} T_{cyl} \\ \frac{dQ_{comb}}{dt} = \dot{m}_{fb} Q_{LHV} \end{cases} \tag{22}$$

where $\frac{dQ_{ht}}{dt}$ is rate of the convective heat transfer from gas to cylinder walls, kW; $\frac{dQ_{in}}{dt}$ and $\frac{dQ_{out}}{dt}$ are inlet and outlet enthalpy flows in the open system, kW; $\dot{m}_{in}$ is mass flow through the intake valve, kg/s; $\dot{m}_{out}$ is mass flow through the exhaust valve, kg/s; Q_{LHV} is lower heating value of fuel, kJ/kg; C_p is specific heat at constant pressure, kJ/kg K; C_v is specific heat at constant volume, kJ/kg K.

The rate of change of mass inside the cylinder is evaluated from mass conservation, and is as follows:

$$\frac{dm_{cyl}}{dt} = \dot{m}_f + \dot{m}_{in} - \dot{m}_{out} \tag{23}$$

From the energy balance, we can deduce the temperature of gases in the cylinder $\dot{T}_{cyl}$ [7]:

$$\frac{dT_{cyl}}{dt} = \left[\left(\frac{dQ_{ht}}{dt} + \Sigma \left(h_0 \frac{dm}{dt} \right)_{in} - \Sigma \left(h_0 \frac{dm}{dt} \right)_{out} + \frac{dQ_{comb}}{dt} - \right. \right.$$
$$\left. \left. -u \frac{dm_{cyl}}{dt} \right) \frac{1}{m_{cyl}} - \frac{RT_{cyl}}{V_{cyl}} \frac{dV_{cyl}}{dt} - \frac{\partial u}{\partial \phi} \frac{d\phi}{dt} \right] \Big/ \left(\frac{\partial u}{\partial T_{cyl}} \right). \qquad (24)$$

In Eq. (24), many terms will be zero in some control volumes all or some of the time. For examples: $\frac{dV_{cyl}}{dt}$ is zero for the manifolds; $\left(h_0 \frac{dm}{dt} \right)_{in}$ and $\left(h_0 \frac{dm}{dt} \right)_{out}$ are zero for the cylinder; $\frac{dm_{fb}}{dt}$ is zero the manifolds; $u \frac{dm_{cyl}}{dt}$ is zero for the cylinder except for mass addition of fuel during combustion; $\frac{dQ_{ht}}{dt}$ is neglected for the inlet manifolds; $\frac{\partial u}{\partial \phi}$ is zero for the cylinder except during combustion (when fuel is added, hence ϕ changes); specific enthalpies $(h_0)_{in}$ and $(h_0)_{out}$ (except the specific enthalpy of formation h_{for}) are constant values.

By application of the first law of thermodynamics for the cylinder gas, the Eq. (24) became:

$$\frac{dT_{cyl}}{dt} = \frac{1}{m_{cyl} C_v} \left(\frac{dQ}{dt} - p_{cyl} \frac{dV_{cyl}}{dt} \right). \qquad (25)$$

The state equation of ideal gas is given by:

$$p_{cyl} V_{cyl} = m_{cyl} R T_{cyl}, \qquad (26)$$

where R is gas constant, kJ/kg K.

Rearranging equations (21), (24), (25), (26); the state equation for cylinder pressure finally becomes:

$$\frac{dp_{cyl}}{dt} = \frac{\gamma}{V_{cyl}} \left[RT_{in} \ \dot{m}_{in} - RT_{cyl} \dot{m}_{out} - p_{cyl} \dot{V}_{cyl} \right] +$$
$$+ \frac{\gamma - 1}{V_{cyl}} \left[\dot{m}_{bf} Q_{LHV} - \dot{Q}_{ht} \right], \qquad (27)$$

where λ is specific heat ratio ($\lambda = C_p / C_v$).

To evaluate the differential Eqs. (24) or (27), all terms of the right side must be found. The most adapted numerical solution method for these equations is the Runge-Kutta method.

3.4. Friction losses

Friction losses not only affect the performance, but also increase the size of the cooling system, and they often represent a good criterion of engine design. The model proposed by Chen and Flynn [20] demonstrate that the value of the mean friction pressure f_{mep}, bar, be composed of a mean value C and additive terms correlated with the maximal cycle pressure p_{max} and the mean piston speed $\bar{v}_{pis}$. The mean value C, supposed constant, depends on the engine type and represents a constant base pressure which is to be overcome first. The term depending on $\bar{v}_{pis}$, reflect the friction losses in the cylinder (piston-shirt).

The maximal cycle pressure p_{max} characterizes the losses in the mechanism piston-rod-crankshaft. So the friction mean effective pressure is calculated by [7]:

$$f_{mep} = C + (0.005 p_{max}) + 0.162 \bar{v}_{pis}, \qquad (28)$$

where p_{max} is maximal cycle pressure, bar. For direct injection diesel engine $C = 0.130$ bar.

3.5. Effective power and effective efficiency

For the 4-stroke engine, the effective power is [7]:

$$b_{power} = b_{mep} V_d N_{cyls} N / 2 \qquad (29)$$

where V_d is displacement volume, m^3, $V_d = \pi D^2 S / 4$, and N_{cyls} cylinder number.

The effective efficiency is given by [7]:

$$R_{eff} = Wd / Q_{comb}. \qquad (30)$$

4. Engine simulation programs

4.1. Computing steps of the developed simulation program

The calculation of the thermodynamic cycle according to the basic equations mentioned above requires an algorithm for solving the differential equations for a large number of equations describing the initial and boundary conditions, the kinematics of the crank mechanism, the engine geometry, the fuel and kinetic data.

It is therefore wise to choose a modular form of the computer program. 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 calculates in discrete crank angle incremental steps from the start of the compression, combustion and expansion stroke. The program configuration allows through subroutines to improve the clarity of the program and its flexibility. The basis of any power cycle simulation is above all the knowledge of the combustion process. This can be described using the modified Wiebe function including parameters such as the combustion time and the fraction of the fuel injected into the cylinder.

For the closed cycle period, Watson recommended the following engine calculation crank angle steps: 10ºCA before ignition, 1ºCA at fuel injection timing, 2ºCA between ignition and combustion end, and finally 10ºCA for expansion.

The computer simulation program includes the following parts:

- ***Input engine, turbocharger and intercooler data:*** engine geometry (D, S, l, r); engine constant (N, ϕ, C_r); turbocharger constant ($\pi c, \pi t, p_{amb}, T_{amb}, m, ICE, p_{out,tur}$, $T_{out,tur}$, $p_{out,man}, T_{out,man}$) and polynomial coefficient of thermodynamic properties of species.

- ***Calculation of intercooler and turbocharger thermodynamic parameters:***

compressor outlet pressure p_c ; compressor outlet temperature T_c ; compressor outlet masse flow rate $\dot{m}_c$; intercooler outlet pressure p_{ic} ; intercooler outlet temperature T_{ic} ; intercooler outlet masse flow rate $\dot{m}_{ic}$; turbine outlet pressure p_t ; turbine outlet temperature T_t ; turbine outlet masse flow rate $\dot{m}_t$.

- ***Calculation of engine performance parameters:***

calculation of the initial thermodynamic data (calorific value of the mixture, state variables to close the inlet valve, compression ratio C_r); calculation of the piston kinematic and heat transfer areas; main program for calculating the thermodynamic cycle parameters of compression, combustion and expansion stroke; numerical solution of the differential equation (the first law of thermodynamics) with the Runge-Kutta method; calculation of the specific heat (specific heat constant pressure C_p and specific heat at constant volume C_v); calculation of the combustion heat, the heat through walls and the gas inside and outside the open system; calculation of main engine performance parameters mentioned above.

- ***Output of data block*** :

Instantaneous cylinder pressure p_{cyl} ; instantaneous cylinder temperature T_{cyl} ; indicated mean effective pressure i_{mep} ; friction mean effective pressure f_{mep} ; mean effective pressure b_{mep} ; indicated power i_{power} ; friction power f_{power} ; brake power b_{power} .

The computer simulation steps of a turbocharged diesel engine are given by the flowchart in Fig. 3.

4.2. Commercial engine simulation code

The GT-Power is a powerful tool for the simulation of internal combustion engines for vehicles, and systems of energy production. Among its advantages is the facility of use and modeling. GT-Power is designed for steady state and transient simulation and analysis of the power control of the engine. The diesel engine combustion can be modeled using two functions Wiebe [21]. GT-Power is an object-based code, including template library for engine components (pipes, cylinders, crankshaft, compressors, valves, etc...). Fig. 4 shows the model of a turbocharged diesel engine with 6 cylinders and intercooler made with GT-Power. In the modeling technique, the engine, turbocharger, intercooler, fuel injection system, intake and exhaust system are considered as components interconnected in series.

4.2.1. Injection system

The simple injection system is used to inject fluid into cylinder and used for direct-injection diesel engines. Table1 shows the parameters of the injection system.

4.2.2. Inlet manifold and exhaust manifold

In the intake manifold, the thermal transfers are negligible in the gas-wall interface. This hypothesis is acceptable since the collector's temperature is near to the one of gases that it contains.

The variation of the mass in the intake manifold depends on the compressor mass flow and the flow through of valves when they are open. In the modeling view, the line of exhaust manifold of the engine is composed in three volumes. The cylinders are grouped by three and emerge on two independent manifold, component two thermodynamic systems opened of identical volumes. A third volume smaller assures the junction with the wheel of the turbine.

Table 1

Injection system parameters [21]

Injectors parameters	Units	Values
Injection pressure p_{inj}	bar	1000
Start of injection bTDC T_{inj}	°CA	15° BTDC
Number of holes per nozzle n_{inj}	-	8
Nozzle hole diameter d_{inj}	mm	0.25

4.2.3. Turbocharger

Turbocharging the internal combustion engine is an efficient way to increase the power and torque output. The turbocharger consists of an axial compressor linked with a turbine by a shaft. The compressor is powered by the turbine which is driven by exhaust gas. In this way, energy of the exhaust gas is used to increase the pressure in the intake manifold via the turbocharger. As a result more air can be added into the cylinders allowing increasing the amount of fuel to be burned compared to a naturally aspirated engine.

4.2.4. Heat exchanger or intercooler

The heat exchanger can be assimilated to an intermediate volume between the compressor and the intake manifold. It comes to solve a system of differential equations supplementary identical to the manifold. It appeared to assimilate the heat exchanger as a non-dimensional organ (one supposes that it doesn't accumulate any gas).

5. Results of engine simulation

Thermodynamic and geometric parameters chosen in this study are:

- *Engine geometry*: compression ratio C_r , cylinder bore D and more particularly to the stroke bore ratio $R_{sb} = \dfrac{L}{D}$.
- *Combustion parameters*: injected fuel mass m_f , crankshaft angle T_{inj} marking the injection timing and cylinder wall temperature T_{wall} .

The Table 2 show the main parameters of the chosen direct-injection diesel engine [21].

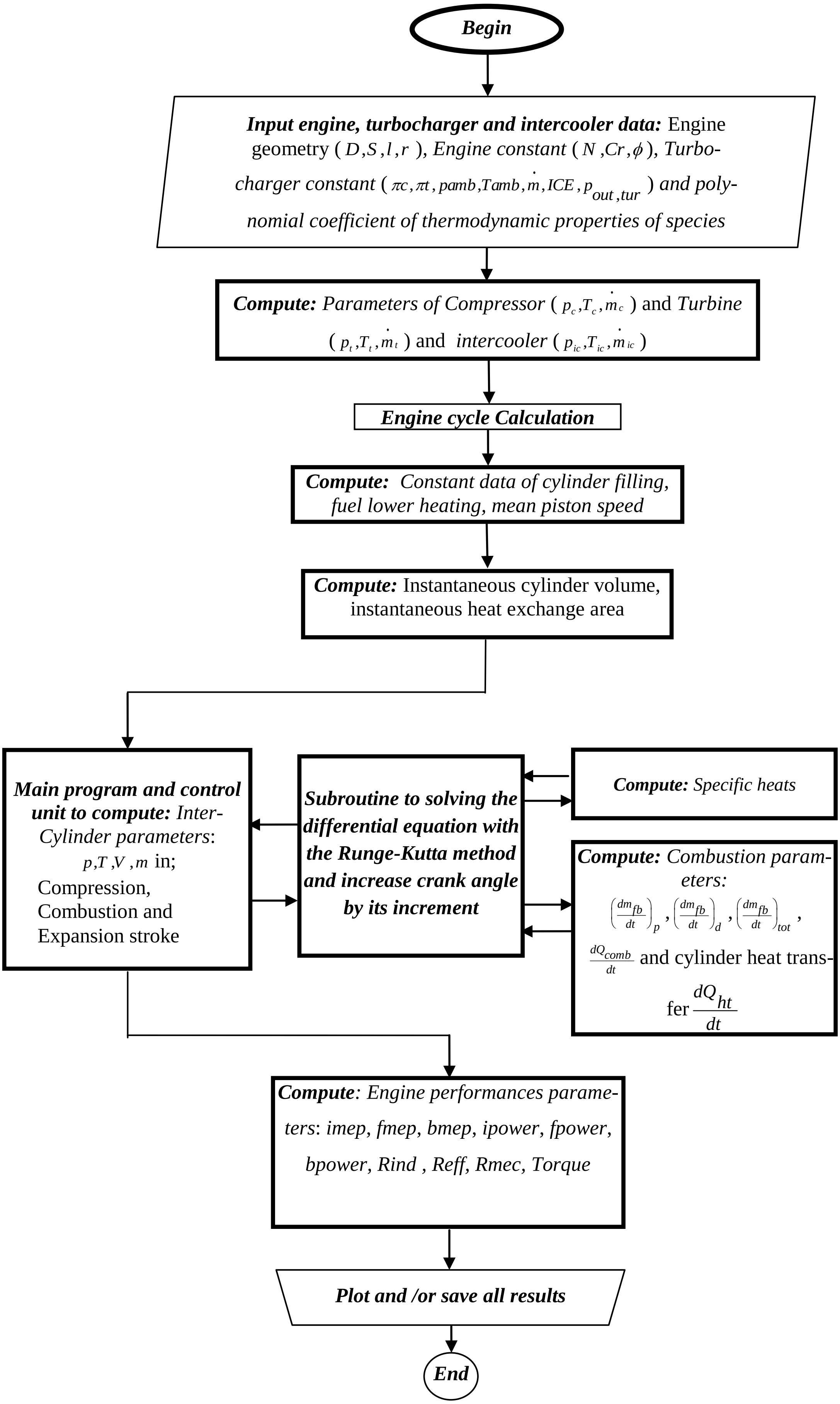

Fig. 3 Schematic Flowchart of the developed computer simulation program

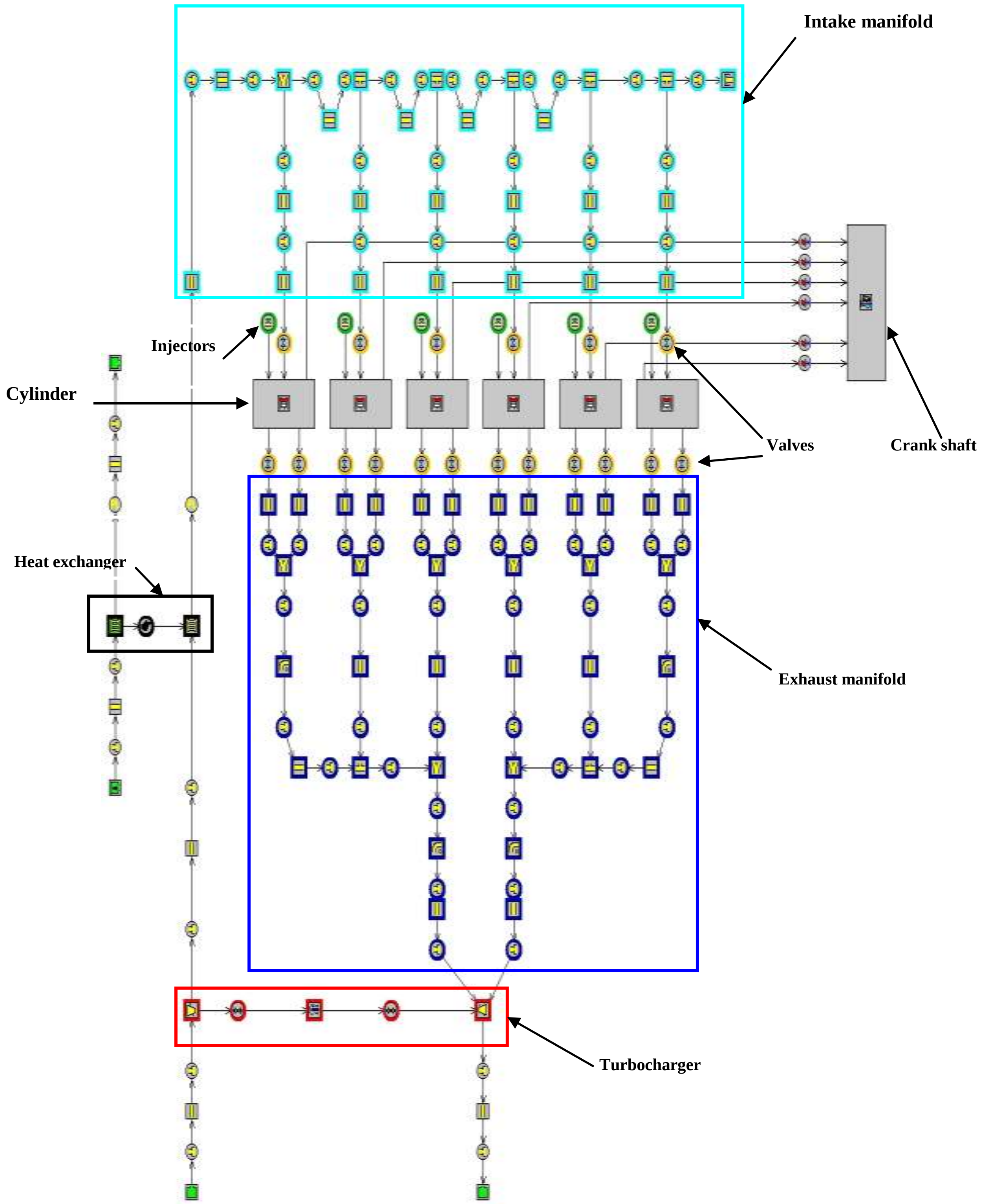

Fig. 4 Developed model in the commercial engine simulation code "GT-Power"

5.1. Influence of the geometric parameters

5.1.1. Influence of the compression ratio

In general, increasing the compression ratio improved the performance of the engine. Fig. 5 shows the influence of the compression ratio (Cr = 16:1 and 19:1) on the brake power and effective efficiency at full load, advance for GT-Power and the elaborate software. The brake efficiency increases with increase of the effective power until its maximum value, after it begins to decrease until a maximal value of the effective power. It is also valid for the effective power. If the compression ratio increase from 16:1 to 19:1, the maximal efficiency increases of 2% and the maximal power of 1.5% for GT-Power and the elaborate software.

Table 2

Engine specifications

Engine parameters	Values
Bore *D*, mm	120.0
Stroke *S*, mm	175.0
Displacement volume V_d, cm^3	1978.2
Connecting rod length *l*, mm	300.0
Compression ratio	16.0
Inlet valve diameter, mm	60
Exhaust valve diameter, mm	38
Inlet Valve Open IVO, °CA	314
Inlet Valve Close IVC, °CA	-118
Exhaust Valve Open EVO, °CA	100
Exhaust Valve Close EVC, °CA	400
Injection timing, °CA	15° BTDC
Fuel system	Direct injection
Firing order	1-5-3-6-2-4

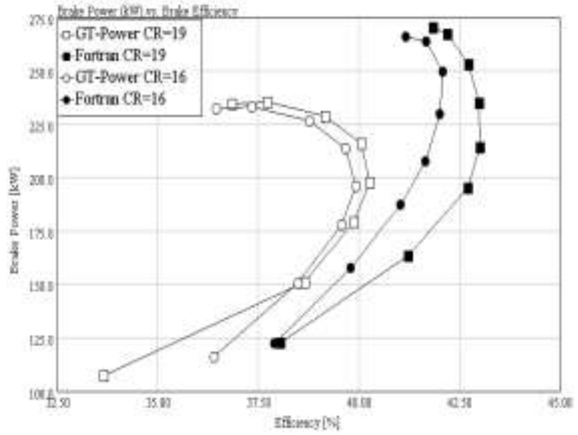

Fig. 5 Compression ratio influence at 100% load, T_{inj} = 15° bTDC, D_{cyl} =120 mm, T_{wall} = 450°K, R_{sb} =1.5

5.1.2. Influence of the stroke bore ratio

The stroke bore ratio is another geometric parameter that influences on 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 Fig. 6. If the stroke bore ratio increase, the mean piston speed is greater, and friction losses Eq. (10) are important with increasing the engine speed. The effective power and the brake efficiency decrease with the increase of the stroke bore ratio. If the stroke bore ratio augments of 0.5 (of 1.5 to 2) then, the maximum brake efficiency decreased an average of 3%, and the maximum effective power of 4%.

5.2. Influence of the thermodynamic parameters

5.2.1. Influence of the wall temperature

The influence of the cylinder wall temperature is represented also in Fig. 7, when the cylinder wall temperature is lower, then the brake efficiency increase. More the difference temperature between gas and wall cylinder is less, then the losses by convective exchange is high [21]. If the cylinder wall temperature increase by 100°K (from

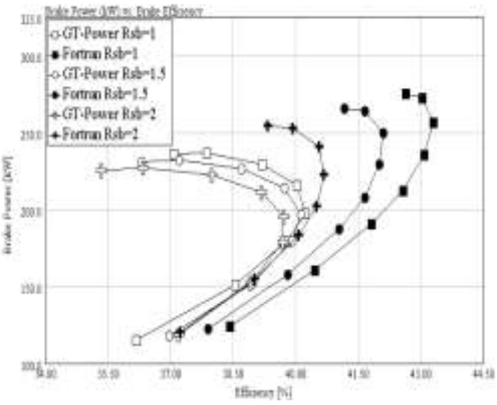

Fig. 6 Influence of Stroke bore ratio for 100% load, T_{inj} = 15° bTDC, v_{cyl} =2.0 l, C_r = 16:1

350 to 450°K), the maximum of 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.

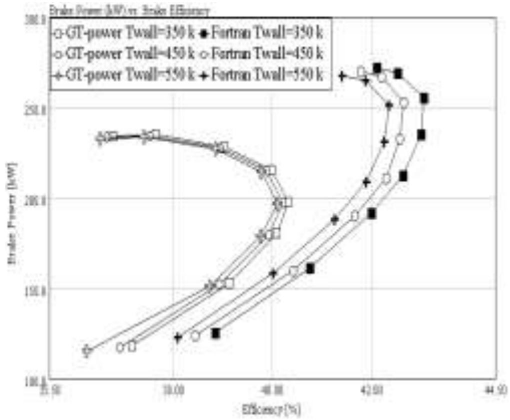

Fig. 7 Wall temperature influence for 100% load, T_{inj} = 15° bTC, D_{cyl} = 120 mm, C_r = 16:1

5.2.2. Influence of the advanced injection

Fig. 8 show the influence of different injection timing on the variation of the maximum brake power and the maximum effective efficiency for the both software; Fortran and GT-Power. This parameter has a substantial influence on the brake power and less on effective efficiency.

5.2.3. Influence of the masse fuel injected

Fig. 9 show the variation of the brake power and effective efficiency for different masse fuel injected at advance injection of 15° bTDC, compression ratio of 16:1, and n = 1400 RPM. This parameter has a strong influence on the brake power and less on the effective efficiency.

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

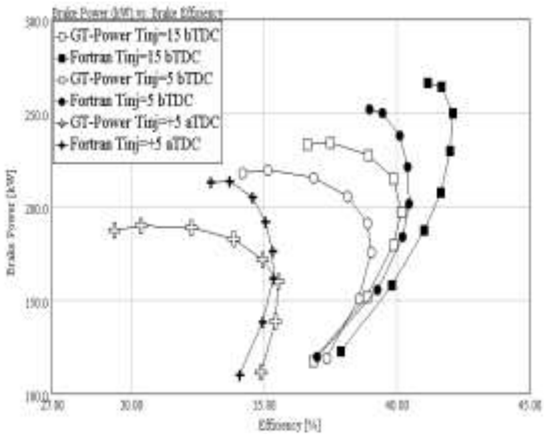

Fig. 8 Injection timing influence for 100% load, D_{cyl} = 120 mm, C_r = 16:1, T_{wall} = 450°K

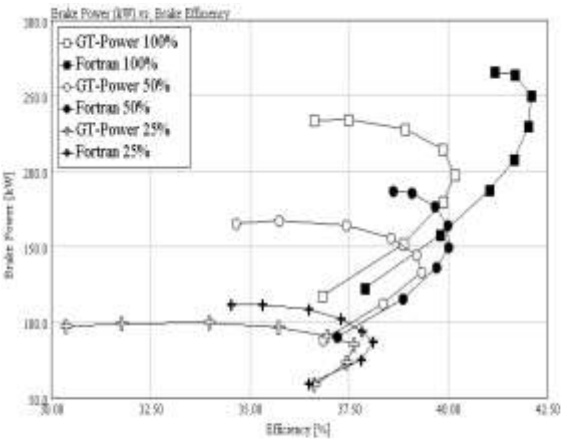

Fig. 9 Mass fuel injected influence for T_{inj} = 15° bTDC, D_{cyl} = 120 mm, C_r = 16:1, T_{wall} = 480°K

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 could develop an interrelationship between the brake power and the effective efficiency, related to the corresponding speed for different parameters studied; it results an existence of a maximum power corresponding to a state for an engine optimal speed and a maximum economy and corresponding optimal speed. We studied the influence of certain number of parameters on engine power and efficiency: The following parameters as; stroke-bore ratio and the cylinder wall temperature, have a small influence on the brake power and effective efficiency. While the angle of start injection, mass fuel injected, compression ratio have great influence on the brake power and effective efficiency. This analysis has been completed by representation of the pressure diagram for various the crankshaft angle, and the corresponding gas temperature versus crankshaft angle. The engine simulation model described in this work is valid for steady engine speed. In future work we aim to replace the simple model for fraction of mass fuel burned by a predictive model, to validate this model for transient engine speed and to take in account gas characteristics and specific heat fluctuation.

References

1. **Angulo-Brown, F.; Fernandez-Betanzos, J. and Diaz-Pico, C.A.** 1994. Compression ratio of an optimized Otto-cycle model, European journal of physics 15: 38-42.
http://dx.doi.org/10.1088/0143-0807/15/1/007.
2. **Chen, L.; Lin. J.; Lou, J.; and Wu, C.** 2002. Friction effect on the characteristic performances of Diesel engines, International Journal of Energy Research 26: 965–971.
http://dx.doi.org/10.1002/er.820.
3. **Merabet, A.; Feidt, M. and Bouchoucha, A.** 2002. Effet du transfert de chaleur sur les performances d'un moteur a combustion interne atmosphérique fonctionnant suivant un cycle mixte, Termotehnica 2 : 43–46.
http://dx.doi.org/10.2516/ogst/2011135.
4. **Watson, N.; Pilley, A.D.; and Marzouk M.** 1980. A combustion correlation for diesel engine simulation, In: SAE Technical Paper: 1980-800029.
http://dx.doi.org/10.4271/800029.
5. **Gogoi, T.K.; Baruah, D.C.** 2010. A cycle simulation model for predicting the performance of a diesel engine fuelled by diesel and biodiesel blends, Energy 35(3): 1317-1323.
http://dx.doi.org/10.1016/j.energy.2009.11.014.
6. **Tschanz, F.; Amstutz, A.; Onder, C.; Guzzella, L.** 2012. Control of diesel engines using NOx-emission feedback, International Journal of Engine Research 18: 53–71.
http://dx.doi.org/10.1177/1468087412442323.
7. **Heywood, J.B.** 1988. Internal Combustion Engine Fundamentals, McGraw-Hill, New york 1:135–164.
http://dx.doi.org/10.1016/S0082-0784(75) 80383-3.
8. **Frank, W.; Frank, K.; George, R.; Emanuel, F.** 2013. Integrated Energy and Emission Management for Diesel Engines with Waste Heat Recovery Using Dynamic Models, Oil & Gas Science and Technology – Rev. IFP Energies nouvelles: 173-178.
http://dx.doi.org/10.2516/ogst/2013210.
9. **Benson, R.S.; and Baruah, P.C.** 1973. Some further tests on a computer program to simulate internal combustion engine, SAE Technical Paper : 730667.
http://dx.doi.org/10.4271/730667.
10. **Dec, JE.** 2009 . Advanced compression ignition engines-understanding the in-cylinder processes, Proc. Combust. Inst 32 : 2727–2742.
http://dx.doi.org/10.1016/j.proci.2008.08.008.
11. **Watson, N.** 1981. Transient performance simulation and analysis of turbocharged diesel engines, SAE Technical Paper 810338 : 162-166.
http://dx.doi.org/10.1093/comjnl/24.2.162.
12. **Galindo, J.; Arnau, F.J.; Tiseira, A.; and Piqueras, P.** 2010. Solution of the Turbocompressor boundary

condition for one-dimensional gas-dynamic codes, Mathematical and Computer Modelling 52: 1288-1297. http://dx.doi.org/10.1016/j.mcm.2010.05.003.

13. **Rakopoulos, C.D.; Rakopoulos, C.D.; Mavropoulos, G.C.; Giakoumis, E.G.** 2004. Experimental and theoretical study of the short-term response temperature transients in the cylinder walls of a diesel engine at various operating conditions, Applied Thermal Engineering 24: 679-702. http://dx.doi.org/10.1016/j.applthermaleng.2003.11.002
14. **Abbe, H.; Rottengruber, S.; Seifert, M.; Ringler, J.** 2013. Dynamic heat exchanger model for performance prediction and control system design of automotive waste heat recovery systems, Applied Energy 105, 293-303. http://dx.doi.org/10.2516/ogst/20132115.
15. **Gunter, P.; Christian, S.; Gunnar, S.; and Frank, O.** 2006. Simulation combustion and pollutant formation for engine-development; New York Springer-Verlag Berlin Heidelberg, Printed in Germany. ISBN: 978-3-540-25161-3 (Print) 978-3-540-30626-9 (Online) http://dx.doi.org/10.1007/3-540-30626-9.
16. **Sakhrieha, A.; and Abu-Nada, E.** 2010. Computational Thermodynamic Analysis of Compression Ignition Engine, International Communications in Heat and Mass Transfer 37: 299–303. http://dx.doi.org/10.1016/j.icheatmasstransfer.2009.11.002.
17. **Semin, R.B.; and Ismail, R.** 2008. Investigation of Diesel Engine Performance Based on Simulation, American Journal of Applied Sciences 5: 610-617. http://dx.doi.org/10.3844/ajassp.
18. **Hohenberg, G.F.** 1979. Advanced approaches for heat transfer calculations, SAE Technical Paper: 1979-790825. http://dx.doi.org/10.1016/j.applthermaleng.2003.11.002
19. **Krieger, R.; Borman G.** 1966. The computation of apparent heat release for internal combustion engines, Proceedings of Diesel Gas Power, ASME; 66-WA/DGP-4.
20. **Chen, S.; Flynn P.** 1965. Development of a single cylinder compression ignition reseach engine. SAE Transaction: 650733. http://dx.doi.org/10.4271/650733.
21. Gamma, Technologies. 2009. *GT-Power User’sManual,* GT-Suite Version 7.0; GT-Power product flyer, the Industry Standard. Available from Internet: http://www.gtisoft.com/img/broch/broch_gtpower.pdf.

B. Menacer, M. Bouchetara

COMPUTATIONAL THERMODYNAMIC OF A TURBOCHARGED DIRECT INJECTION DIESEL ENGINE

S u m m a r y

The purpose of this work is to provide a flexible thermodynamic model based on the filling and emptying approach for the performance prediction of turbocharged compression ignition engine. To validate the model, comparisons between the developed a computer program in FORTRAN language and the commercial GT-Power software results operating under different conditions. The comparisons show that there is a good concurrence between the developed program and the commercial GT-Power software. The variation of the speed of the diesel engine chosen extends from 800 RPM to 2100 RPM. In this work, we studied the influence of several engine parameters on the power and efficiency. Moreover, it puts in evidence the existence of two optimal points in the engine, one relative to maximum power and another to maximum efficiency; it was found that if the injection time is advanced, so the maximum levels of pressure and temperature in the cylinder are high, it was found that if the injection time is advanced, so the maximum levels of pressure and temperature in the cylinder are high.

Key words: Thermodynamic, combustion, turbocharged compression ignition engine, GT-Power, performance optimization, filling and empting method.

Received June 12, 2014
Accepted January 12, 2015