

A New Tool to Perform Global Energy Balances in DI Diesel Engines

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ABSTRACT

The generalization of exhaust aftertreatment systems along with the growing awareness about climate change is leading to an increasing importance of the efficiency over other criteria during the design of reciprocating engines. Using experimental and theoretical tools to perform detailed global energy balance (GEB) of the engine is a key issue for assessing the potential of different strategies to reduce consumption. With the objective of improving the analysis of GEB, this paper describes a tool that allows calculating the detailed internal repartition of the fuel energy in DI Diesel engines. Starting from the instantaneous in-cylinder pressure, the tool is able to describe the different energy paths thanks to specific submodels for all the relevant subsystems. Hence, the heat transfer from gases to engine walls is obtained with convective and radiative models in the chamber and ports; the repartition of the heat flux throughout the engine metal elements towards the oil and coolant is estimated with a lumped capacitance model; finally, the auxiliary systems and friction losses are obtained through specific semiempirical submodels. The validation of the tool is performed in a 4-cylinder DI Diesel engine instrumented to perform detailed experimental GEB. Finally, a simple analysis of combined internal and external analysis in the complete engine map shows the effect of operating conditions on each energy term. Thus it is demonstrated the utility of the proposed tool, that complements the experimental heat flow measurements in Diesel engine researches oriented to the reduction of energy consumption.

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INTRODUCTION

Internal Combustion Engines (ICE) are, by far, the most widespread technology to supply energy in the automotive sector. Nevertheless, the success of the ICE has its counterpart, hence the environmental impact of the spread use of ICEs have forced to new regulations that have progressively limited the amount of chemical emission of the vehicles. Thus, in the last two decades the engine research has been mainly focused on reducing the emissions by means of different techniques oriented to limit the pollutant formation during the combustion process or reducing the engine tailpipe emissions. Some of the methods used to optimize the combustion are based on high pressure fuel injection systems [1], multiple injections [2], high boost pressure [3], EGR [4], variable valve timing [5], high swirl ratios [6], new clean fuels [7] or new engine control systems [8]. Nevertheless, the very stringent limits are leading to generalize the use of exhaust gas aftertreatment systems [9] because the possibilities of optimizing the thermo-fluid-dynamic processes in the engine to reduce emission formation are not enough to overcome the regulations. This has lead, during the last years, to revitalizing the interest in the other big challenge of the ICE: the engine consumption.

In this scenario, although alternative non ICE-based powertrains are being evaluated, the ICE still has room to increase its efficiency and reduce its Greenhouse Gases (GHG) emissions in traditional or Hybrid Electric powertrains, remaining a cost-effective solution for the transport problem [10]. Hence, the market pressure along with the increasing warning due to the effect of GHG emissions are pushing the researchers and manufacturers to search for more efficient engines with lower consumption and CO₂ emissions. The efforts are oriented to different issues such as thermal management [11, 12, 13, 14], indicated cycle optimization [15, 16, 17], in- cylinder heat transfer (HT) reduction [18,19], friction and auxiliaries losses reduction [20, 21, 22, 23] or engine downsizing [24] between others.

To assess the benefits of different engine strategies or configurations, the use of Global Engine Balances (GEB) is a useful tool that allows identifying the repartition of the chemical fuel energy into the final energy destination, thus improving the understanding of the mechanisms affecting the engine consumption. Different works dealing with the GEB can be found in literature, both in SI and CI engines, with conventional or alternative fuels [16,19,25, 26, 27]. Usually, the basis of those analyses is the first law of thermodynamics and, in some cases, it is combined with an energy analysis [27, 28]. The most usual approach is taking into account three major terms

in the GEB: the brake power, the cooling losses and the exhaust gas losses. A miscellanea term can also be included to consider undetermined energy losses that are difficult to measure, such as convection and radiation to the ambient or HT to the oil [9,19]. Thus, in the simplest approach, a very limited number of variables are required to carry out the GEB. However, many terms that are not considered in the simple analysis are needed to perform a detailed GEB, hence it is interesting to include specific terms for the heat flow to the oil, heat flow in the intercooler or EGR cooler, or decompose the effect of different mechanical losses. This detailed point of view requires considering a global methodology that allows analyzing the path followed by the different energy terms, starting from the fuel energy, and detailing the energy degradation due to the different heat and mechanical losses up to the final destination.

This approach includes internal and external analysis of the engine, and requires the use of both experimental measurements and internal modeling of the engine. However, very few works has been found in the literature survey where these different approaches were combined to provide a complete GEB description. Thus, it can be found:

- Some works dealing with the experimental approach from the external point of view: in [16, 19, 25, 26] GEB is performed, with different levels of complexity, measuring some fluid temperatures and mass flows along with the engine torque and speed. In some cases [16], the in-cylinder pressure allowed calculating the mechanical and pumping losses.
- A lot of works dealing with the calculation of the HT in the cylinder chamber [18, 19, 29, 30, 31, 32] and detailed finite elements models or lumped conductance models [33, 34, 35] to estimate the wall temperatures and the heat fluxes repartition.
- Several works describing detailed friction model to estimate the friction in the piston assembly, crankshaft bearing and valve train along with the power to drive the auxiliaries system [20, 21, 22, 23].

However, no work has been found in the literature where the detailed models of in-cylinder HT, HT throughout the metallic parts, friction and auxiliaries losses are linked to provide a detailed description of the engine thermal balance. The tool described in this work, which is called CALMEC, is focused on modeling internal energy terms that can hardly be measured with conventional or specific instrumentation, and it deals with all the relevant engine subsystems involved. It includes a combination of physical and semi-empirical submodels that are fast and simple enough to be linked with the in-cylinder combustion analysis. The external GEB measurements will be used to validate the internal models and to complete the energy balance with experimental terms that are not modeled

(turbocharger, intercooler, heat transfer to the ambient...), thus providing detailed information of the effect of different operating conditions on energy repartition.

EXPERIMENTAL SETUP

The submodels adjustment and the parametrical studies were conducted in the experimental setup shown in Fig. 1. The experimental tests were carried out in a 4-cylinder 1.6 liter DI Diesel engine, Euro 5 compliant (Engine 1), that is currently in production. The engine main characteristics are given in Table 1.

The engine was directly coupled to an electric dynamometer that allows controlling the engine speed and load, while the installation includes a complete instrumentation to measure different fluid temperatures and mass flows for external thermal balance. The list of the relevant instrumentation is given in Table 2.

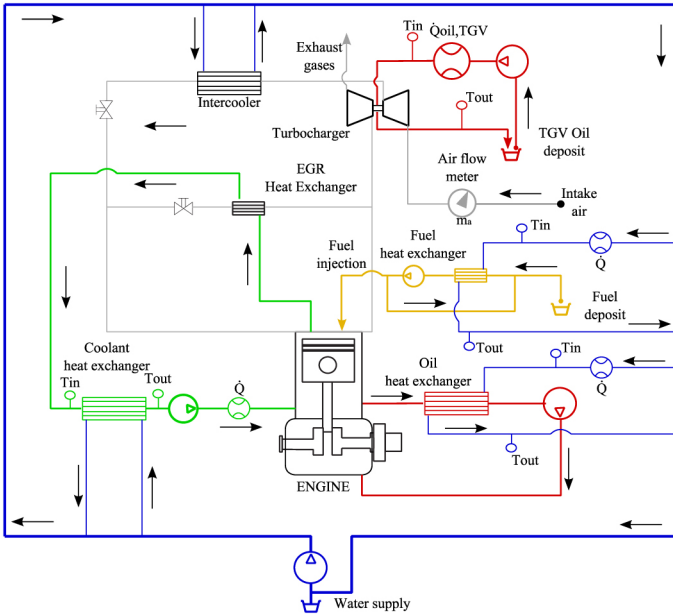


Figure 1. Experimental setup.

Table 1. Main characteristics of the engine.

Displaced volume	1560 cc
Stroke	88.3 mm
Bore	75 mm
Connecting Rod	136.8 mm
Compression ratio	16:1
Number of cylinders	4
Inlet valves	2
Exhaust valves	2
Injection system	Common rail

Table 2. Test cell instrumentation.

Variables to measure	Sensors	Measurement uncertainty
Temperature of liquids (coolant, oil, cooling water at exchangers...)	Thermoresistances (RTD)	0.2 °C
Temperature of gases (inlet and exhaust lines, EGR...)	Thermocouples	1.5 °C
Wall temperatures		
Fuel mass flow	AVL 733S Fuel meter	0.2 %
Air mass flow	Sensiflow DN80	2 %
Turbo oil mass flow	Krohne OptiMass 3050C	0.5 %
Coolant flow	Krohne 4010 Optiflux	0.5 %
Oil exchanger cooling	Isoil MS500	0.5 %
Fuel exchanger cooling	Yoko AdmagAE208MG	0.5 %
Torque	Dynamometer	0.5 Nm
In-cylinder pressure	AVL GH13P	Linearity 0.3 %

A combination of thermocouples and resistance temperature detectors (RTD) was selected (taking into account the expected temperature variation) to measure liquids and gas temperatures. For the sake of accuracy, the original oil circuit was modified so that the oil was cooled with an external water circuit and an oil conditioner was used for controlling the temperature and mass flow of the oil in the turbocharger. Thus, the heat flow to the oil in the block was measured separately from coolant and turbo oil thermal flows.

The in-cylinder pressure was measured at the 4 cylinders by means of AVL GH13P glow-plug piezoelectric transducers with a range between 0 and 250 bar, and a sensitivity of 16 pC/bar. The electrical charge yielded by the piezoelectric transducers is converted into a proportional voltage signal by means of Kistler 5011B charge amplifiers. A crank angle increment of 0.5° was used for the in-cylinder pressure acquisition, which was performed using a Yokogawa DL708E oscillographic recorder with a 16 bits A/D converter module.

The mean variables were acquired at a low sample frequency of 100 Hz using SAMARUC, a home-developed test system that collects the signals of different sensors and controls the electric dynamometer. The exhaust emissions were measured with a Horiba MEXA 7100 DEGR. EGR flow rate estimation was inferred on the basis of the intake and exhaust CO₂ concentrations.

The adjustment of the lumped model, which will be later described, was performed in a second unit of the stated engine (Engine 2), Euro 4 compliant, with similar geometric characteristics to Engine 1. In his case, the metal temperature was measured with 16 thermocouples in the cylinder-head, 23 in the cylinder liner and 2 RTDs in the piston [33].

GLOBAL ENERGY BALANCE IN A DI DIESEL ENGINE

Taking into account all the energy transformations that can take place in a DI Diesel engine, Fig. 2 shows the energy flows that were considered, from two points of view: internal and external analysis.

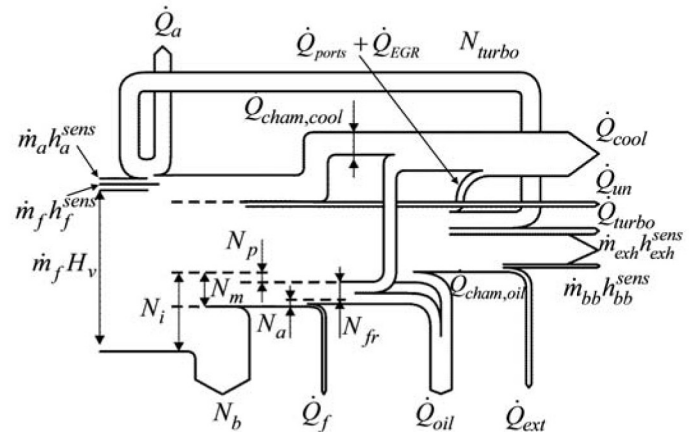


Figure 2. Schema of the energy flows considered.

External GEB

In this analysis the engine is considered as a black box with some energy flow entering and some other leaving the engine. The main characteristic of this approach is the possibility to measure almost all the energy terms.

The energy flows entering to the engine are the sensible enthalpy of air and fuel $\dot{m}_a h_a^{sens}$ and $\dot{m}_f h_f^{sens}$ that depend on inlet temperatures of both fluids, and the chemical energy of the fuel heating value $\dot{m}_f H_v$. The main outlet energy flows are the brake power N_b , the heat flow to the coolant $\dot{Q}_{cool}$, the sensible energy of the exhaust gases $\dot{m}_{exh} h_{exh}^{sens}$, the heat flow removed in the oil exchanger $\dot{Q}_{oil}$, and the heat flow in the intercooler $\dot{Q}_a$. Other outlet terms with lower importance are the HT to the turbo oil $\dot{Q}_{turbo}$, the convective and radiative HT to the ambient from the engine surface $\dot{Q}_{ext}$, the enthalpy flow due to blow-by losses (externally collected) $\dot{m}_{bb} h_{bb}^{sens}$, the energy losses due to uncompleted combustion $\dot{Q}_{un}$, and finally the small term $\dot{Q}_f$ corresponding to the heating of the fuel returning to the tank, which is usually neglected. The electrical system of the engine is connected to an external power supply, so there is no additional energy exchange with the engine.

The stated terms account for the external GEB of the engine and can be measured using the experimental instrumentation described in the previous section. Thus, N_b is computed from engine torque and speed, the enthalpy flows can be obtained from the mass flow of the gases and their temperatures, $\dot{Q}_{cool}$,

$\dot{Q}_{oil}$, $\dot{Q}_{turbo}$, $\dot{Q}_a$ and $\dot{Q}_f$ from the fluid mass flows through the exchangers and the temperature increment, $\dot{Q}_{un}$ can be estimated from the CO and CH emissions in the exhaust, and finally $\dot{Q}_{ext}$ is obtained from the difference of the rest of measured terms. Hence, the thermal balance can be written as:

$$\dot{m}_f H_v = N_b + H_{exh} + \dot{Q}_{cool} + \dot{Q}_{oil} + \dot{Q}_a + \dot{Q}_{turbo} + \dot{Q}_{un} + \dot{Q}_{ext} + \dot{Q}_f + H_{bb} \quad (1)$$

being

$$H_{exh} = \dot{m}_{exh} h_{exh}^{sens} - \dot{m}_a h_a^{sens} - \dot{m}_f h_f^{sens} \quad (2)$$

where h_{exh}^{sens} , h_a^{sens} and h_f^{sens} are the sensible enthalpies of the exhaust gases, inlet air and fuel (calculated at their respective temperatures) and $\dot{m}_{exh} \approx \dot{m}_a + \dot{m}_f$. For the sake of rigour, the blow-by mass $\dot{m}_{bb}$ should be removed from $\dot{m}_{exh}$, however, in all tested operating points the blow-by was lower to 1.5% of $\dot{m}_a$, thus it was neglected in a first approach. Analogously, the term H_{bb} , which accounts for the net enthalpy flow due to the blow-by, was not taken into account, as it was estimated that in all the operating points it is lower to 0.5% of the fuel energy $\dot{m}_f H_v$.

The importance of each term in Eq. (1) vary with the engine and operating conditions, however in CI engines the most important of them use to be in the following ranges: $N_b \sim 25\text{-}40\%$ (excepting at very low load), $\dot{Q}_{cool} \sim 15\text{-}35\%$, $H_{exh} \sim 20\text{-}35\%$ and $\dot{Q}_{oil}$ use to be about one third of $\dot{Q}_{cool}$.

Internal GEB

The described external approach to the GEB is straightforward and it can be performed with the usual engine measurement devices. However when the analysis of the internal GEB is carried out, terms such as walls HT or mechanical losses can be barely measured, due to the experimental difficulty. To deal with the determination of such energy terms, the available experimental measurement must be combined with models of particular processes and systems [33,36].

The internal GEB includes the following terms: N_i (usually ranging 35-50% of the fuel energy) is the indicated power which is split in different terms whose importance depends on the operating conditions; thus, part of it provides the brake power, part the pumping work N_p , and finally, part is lost due to the friction N_{fr} , and to drive the engine auxiliaries N_a (fuel, oil and cooling pumps). The total amount of mechanical losses will be:

$$N_m = N_p + N_a + N_{fr} \quad (3)$$

If a more detailed description of N_a and N_{fr} is considered, they can be expressed as:

$$N_a = N_f + N_{cool} + N_{oil} \quad (4)$$

$$N_{fr} = N_{fr,pis} + N_{fr,bear} + N_{fr,valu} \quad (5)$$

where N_f , N_{cool} and N_{oil} are the power to move the fuel, coolant and oil pumps respectively, and $N_{fr,pis}$, $N_{fr,bear}$ and $N_{fr,valu}$ are the friction losses in the piston assembly, bearings and valve train. The total value of $N_a + N_{fr}$ can be computed from net indicated power ($N_i - N_p$) and brake power, however, obtaining the value of each term included in Eqs. (4) and (5) is not possible through the experimental measurements in the engine. If auxiliaries are characterized out of the engine and specific test rigs are used to measure friction losses [23], their values can be determined, however this is usually not done in practice. Regarding the relative importance of each term of the mechanical losses, a wide variation range can be found in the literature [9,23,37], thus N_b ranges 15-30% of N_m , N_a is assumed to be 15-25% of N_m , and N_{fr} about 45-65% of N_m . The repartition of the terms in N_{fr} is also variable depending on the source: $N_{pis} \sim 40\text{-}75\%$ of N_{fr} , $N_{bear} \sim 20\text{-}40\%$ of N_{fr} and $N_{valu} \sim 7\text{-}30\%$ of N_{fr} . Specific submodels to obtain an estimation of each component will be presented in the next section.

Other two important terms in the internal GEB that must be modeled are the HT flows to the coolant and oil. The main energy contribution to them is the HT from the chamber gases to the coolant $\dot{Q}_{cham,cool}$ and oil $\dot{Q}_{cham,oil}$ and, to a lesser extent the heat flow from ports $\dot{Q}_{ports}$, and EGR, $\dot{Q}_{EGR}$. Finally, friction and auxiliaries power are degraded into heat and then shared out to the coolant, oil and ambient. Thus, the friction between the piston-rings and the cylinder walls, which account for a substantial part of the friction losses, contributes mainly to increase the coolant energy, while the friction in the remaining elements is dissipated in the lubricant oil and the surroundings. Except $\dot{Q}_{EGR}$, that can be directly calculated, the rest of HT terms cannot be measured. Hence, efforts were made for developing accurate chamber HT models, and estimating HT repartition to coolant and oil by means of a lumped conductance model, as detailed in the following section. Finally, for the sake of simplicity, there is also a term that is not explicitly included in Fig. 2, it is the HT from oil to coolant in the block. It cannot be measured and is very difficult to assess by means of modeling, a simple way for its estimation will be later discussed.

Finally, the energy delivered by the turbine to the compressor (throughout the turbo shaft mechanical power and HT) is represented in the schema of Fig. 2 with N_{turbo} . The thermal energy from hot gases to the turbo case and the mechanical friction in the turbocharger shaft are included in the HT to the turbo cooling oil and the convection and radiation to the ambient.

SUBMODELS DESCRIPTION

Taking into account the considerations made in the previous section, several submodels to assess the terms of the internal GEB are proposed here. The main inputs for these submodels are the in-cylinder pressure and some mean variables (engine speed, coolant, oil, inlet and exhaust temperatures, air, EGR and fuel mass flow...). The in-cylinder pressure allows obtaining the gas thermodynamic conditions in the chamber that feeds the convective and radiative HT models in the chamber, and the filling and emptying model that provides the fluid-dynamic conditions in the ports, and thus the HT flows in these element. The convective and radiative models are linked to a lumped conductance model to calculate the wall temperatures. Finally, in-cylinder pressure feeds the calculation of the instantaneous dynamic forces, necessary for the friction submodels. The auxiliary losses are mainly modelled using some experimental mean variables.

External thermal measurements will be used to validate the models.

Heat Transfer Models

As mentioned, there are two main sources of HT in the engine: the combustion chamber and the gases in the exhaust ports. Both of them will be linked with the lumped conductance model.

Convective Heat Transfer in the Chamber

Heat transfer to the combustion chamber walls in motoring tests (or during the compression stroke and after the end of combustion), is essentially governed by convection, because gas radiation to the walls is negligible in these conditions [29, 30]. When combustion takes place, the gas convection keeps on being the key issue, although radiation from the gas and the soot particles becomes also important.

Many different empirical correlations to estimate the HT coefficient, based on the Nusselt approach, are available in literature. Amongst them, those proposed by Annand [29] or Woschni [31,32] are the best known. Starting from a reference model, authors usually adjust the model constants for their specific engines. In the proposed model, a variation of the Woschni's correlation is used to calculate the HT coefficient:

$$h = CD^{-0.2}p^{0.8}T^{-0.53}v_g^{0.8} \quad (6)$$

where $C = 0.12$, D is the engine bore, p and T are the instantaneous in-cylinder pressure and temperature and v_g is the gas speed in the chamber:

$$v_g = C_{W1}c_m + C_{W2}c_u + C_2 \frac{v_d p_{IVC}}{V_{IVC} T_{IVC}} (p - p_0) \quad (7)$$

being c_m the mean piston speed, c_u the tangential velocity due to the swirl, V_d the displaced volume, p_{IVC} , V_{IVC} , T_{IVC} the pressure, volume and temperature at the intake valve closing, p_0 the motoring pressure assuming polytropic evolution, and C_{W1} , C_{W2} and C_2 constants that must be adjusted. Efforts were done in order to improve the original model, thus, on the one hand the values of the constants have been modified, on the other hand the swirl term is different from that proposed by Woschni. The constant of the combustion term, $C_2 = 0.001$, the instantaneous evolution of the swirl velocity, c_u , (constant in the original Woschni's model) and the ratio $C_{W1}/C_{W2} = 1.7$ were set by means of CFD calculations in two previous works developed in the research group [38, 39]. Hence, the only value that must be adjusted to complete the chamber convective HT model is the value of C_{W1} , from which C_{W2} is obtained assuming the stated ratio between them. A brief description of C_{W1} adjustment is later provided.

Radiative Heat Transfer in the Chamber

There is no agreement regarding the fraction of the HT transferred by radiation: Morel and Keribar [30] obtained values ranging from 4% to 20%, whereas Heywood [9] states that this fraction can be higher than 20%. In any case, the accurate estimation of radiation in the chamber will require the calculation of the soot formed in the spray [40].

Fig. 3 shows a schema of the radiation submodel used to calculate the total radiation power from the soot in the flame, Q_{rad} . A combustion model [41] provides a simplified spatial evolution of the fuel parcels in the spray, along with their temperature and air, fuel and burned products mass fractions. From this information the fuel-to-air equivalence ratio at the lift-off and the residence time of each parcel in the spray is obtained. This information, along with the flame temperature and the air density, allows computing the soot yield (SY), defined locally as the ratio between the mass fraction of fuel transformed into soot and the mass fraction of unburned fuel available at that location [42]. From the SY, and knowing the fuel mass concentration in the spray centerline, the soot mass fraction is calculated, whence the soot volumetric fraction in the centerline is obtained. The soot formation model was adjusted with laser extinction measurements. Once the soot formed is known, the radiation from each parcel in the spray is computed taking into account its temperature, according to the black body radiation, thus obtaining the total amount of radiation. The radiation model was adjusted using the two-color method. A more detailed description of the process, hypothesis and model validation can be found in [43].

Heat Transfer in the Ports

Intake ports HT use to be negligible in comparison with exhaust ports. In both cases, different semiempirical expressions can be found in the literature [44].

