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New approach to study the heat transfer in internal combustion engines by 3D modelling

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ABSTRACT

The stringent environmental constraints have led the industry to look for new technologies that improve the design of internal combustion engines (ICE), mainly focused on achieving higher thermal efficiency and reducing pollutant emissions. Numerical simulations play a key role for optimizing engine design, but modelling the physical processes such as the combustion and the heat transfer to the walls, represents a challenge due to the complexity of the phenomena involved. Combustion has to be modeled properly, and heat transfer models from gas to wall require coupling with solid wall models (CHT Conjugate Heat Transfer calculations). In this paper, a new approach is presented, focused on optimizing the computing time needed to perform CHT simulations. It is based on the use of the Rate of Heat Release (RoHR) obtained from a CFD calculation to replace the combustion process. The study is performed for a spark ignition (SI) engine. The proposed approach is validated for the heat transfer to the piston wall. The study shows that the CHT-RoHR approach yields very good results in terms of spatially averaged values during the whole engine cycle and allows considerable reduction in computational cost. It is therefore very useful to perform parametric studies.

1. Introduction

Although current research favours alternative and eco-friendly energy sources in order to reduce the environmental damage caused by fossil fuels, the difficulties still encountered to replace internal combustion engines for transport mean that these will probably remain important for the automobile market in the coming years, if only as part of the hybrid solutions. In the last decades, the stringent environmental constraints and concerns about global warming have led the industry to look for new technologies that improve the design of internal combustion engines, mainly focused on achieving higher thermal efficiency and reducing pollutant emissions. Thus, new technologies such as downsizing and turbocharging [1] are currently implemented to improve the performance, and this may lead to extreme conditions within the combustion chamber, which could affect the combustion process.

Combustion involves very complex chemistry and thermodynamics. Additionally other processes such as those due to mechanical friction, blow-by and cooling of the engine may affect the combustion and thus render the computational modelling of combustion in engines even more challenging [2]. In order to improve engine design it is important to understand these phenomena and how they interact with the engine components. This knowledge is essential to optimize engine performance, especially in the early stages of design when the components such as the cylinder head, combustion chamber or piston may still be modified, as well as the various parts that may depend on these.

From the economical point of view it is clearly more interesting to perform computational simulations to check if suggested modifications may improve or not the overall performance of the engine and vehicle. Indeed, experimental tests require prototypes and tend to be expensive. To give but a couple of examples, engine components must be designed to withstand high temperatures, so it is useful to determine the temperature distribution of the fluid exposed surfaces, which is experimentally complicated and intrusive [3,4], if at all possible. Also, it is very important in premixed air-fuel gasoline engines to determine hot spots that may induce early ignition and auto-ignition triggering in irregular combustion [5]. Simulations on the other hand provide inexpensive means for calculating the local gas temperature in the combustion chamber and the heat transfer to the engine walls and provides a much more complete overview of the flow inside the engine. There are however some difficulties, as combustion has to be modeled properly, and heat transfer models from gas to wall require coupling with solid wall models, which increases the difficulty of this type of simulations.

Over the years, researchers have tried to simulate the combustion process in engines in order to predict emissions at different operation conditions without having to perform many expensive experimental

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tests. Due to the complexity of the combustion process, several models have been developed in function of the requirements [6]. However, it is still difficult to obtain accurate results with any model, and development is still ongoing to improve them. Chemistry plays an important role in these models and it is known that by increasing the number of reactions in the model, this becomes more reliable. But it is also a fact that computing time increases dramatically with the number of reactions. Hence, much effort is put into optimizing these models [7,8]. However, when emissions are not the focus of a study, it may not be necessary to model with high precision the combustion. This is the case when studying the heat transfer in an engine, since what is really relevant is the heat released by the combustion process, rather than the chemistry.

It is known that Internal Combustion Engine ICE) performance and fuel efficiency may be improved if heat losses can be reduced [2,9–12]. One way of achieving this is to insulate the engine walls, and this was investigated in the 1980's with little success due to the poor characteristics of ceramic insulation materials used at the time (low heat conduction, but high heat capacity) [13]. However, insulation technology has improved since, and some studies have been recently performed with new insulating materials of low thermal conductivity and low heat capacity [14–16]. In these works, the idea is that by coating the piston surface, the instantaneous temperature on it changes in phase with the gas temperature (temperature swing), and this translates into reduced heat losses.

The proper reduction of heat losses must be based on the detailed knowledge of engine thermal behavior, i.e., on the accurate prediction of material temperatures and heat flows through the engine elements, so as to know which parts may be coated efficiently. However, the description of heat transfer in an ICE is a challenging task, considering the different systems (intake and exhaust ports, coolant circuit, lubricant oil subsystem), the different heat transfer mechanisms (convection, conduction and radiation), and the rapid and unsteady changes that take place inside the cylinder. Considerable experimental and theoretical efforts have been devoted in recent years to overcome these difficulties, since engine efficiency and pollutant emissions are highly affected by combustion chamber wall temperatures. In order to estimate properly these heat flows, it is necessary to combine theoretical studies with the analysis of experimental data. Regarding theoretical approaches, the use of simple lumped models has gained an increasing attention due to their reasonable compromise between computational cost and solution accuracy [17,18]. However, 0D-1D models cannot give detailed information such as temperature distribution on the wall surfaces. It is not possible either to obtain this kind of information experimentally, as the introduction of thermocouples in an engine is delicate and difficult due to the limited access inside the engine. In addition, this technique weakens the components [1,19]. The use of Laser Induced Phosphorescence [20] yields also very limited information, just on one point within the solid, and with an arguable precision.

On the other hand, simulations using three-dimensional Computational Fluid Dynamics (CFD) can provide precise and instantaneous information about the flow within the engine (temperature, pressure, velocity distributions), but require temperature boundary conditions for the engine walls, which are typically assumed as being constant throughout the engine cycle. This will affect the combustion process, thus leading to inaccurate results [21]. Imposing appropriate wall temperature boundary conditions is not an easy task, and may require some iterative process. However, such calculations with given surface temperature are not sufficient when the focus is on analyzing the heat transfer within the engine. Indeed, CFD alone does not permit to calculate the heat fluxes through the engine walls, since they are imposed by the boundary conditions. A solution to solve this issue is to use a decoupled iterative process that requires the separate calculation of the flow on the one hand (CFD), assuming constant wall temperatures, and of the solid on the other hand by some finite element software. The heat flux and gas properties of the CFD are used as boundary conditions for the heat conduction calculation of the solid regions. Then the wall temperatures are updated for the following CFD simulation, and the procedure is repeated until temperature convergence [22]. However, this approach is computationally inefficient due to the use of two different softwares and the difficulty of exchanging the required information between the programs.

The Conjugate Heat Transfer (CHT) method offers a much better alternative. It is a coupled approach that solves simultaneously the fluid and the solid regions in a unique computational iterative process [22]. It therefore yields at the same time all the flow characteristics within the engine, as well as a precise calculation of the heat fluxes through the walls. In addition, other important variables, such as the temperature distribution on any gas exposed surface of the engine, including peak temperatures, the local and instantaneous heat transfer coefficient, and the near wall velocities are also known. Heat losses are adequately identified and this knowledge may help improve engine design. It is also possible to take into account the thermal stresses to optimize the cooling system and the solid materials [23].

Though the CHT approach appears to be the best solution for heat transfer analysis in engines, it suffers from a major drawback. Indeed, compared to a simple CFD calculation where only the fluid region needs to be meshed, for the CHT approach, the solid region has also to be meshed, and with a very fine mesh. This naturally increases computational time. Moreover, the solid regions converge at a much slower rate than the fluid regions, and several engine cycles have to be performed to guarantee the convergence of the thermodynamics properties. The computational times of a CHT calculation are therefore quite high, even using parallel processors with high computing power.

The main purpose of this work is to propose a methodology that allows optimizing the calculation time of the CHT simulation of an internal combustion engine based on the use of the rate of heat release (RoHR) as source term for the energy equation. The idea is to perform a preliminary CFD calculation to obtain the RoHR and consequently use this variable to replace the combustion process in the CHT simulation. The study is performed for a 4-stroke spark ignition (SI) engine. Considering the high computational cost of CHT calculations, it was decided to validate the proposed approach taking into account the heat transfer to the piston wall only. In any case, from a practical point of view, only piston surface and engine head may be coated to reduce heat losses. This paper is structured in 5 sections. In section 2, the methodology proposed is presented in detail. Section 3 is dedicated to the description of the numerical set-up for the calculations considered. Section 4 presents the validation of the proposed approach, as well as a detailed results analysis of its application to the CHT simulation. Finally, section 5 draws the pertinent conclusions about the validity, and possible application of the approach.

2. Methodology

As explained previously, CHT calculations are very time-consuming, in particular because the convergence of the combustion process and the heat transfer through the wall have very different characteristic times. The combustion process simulation itself is already time-consuming because it requires the adjustment of the combustion model, which implies running several full cycles. Since the result expected from CHT calculations is the wall temperature distribution, the combustion process itself is only relevant in terms of heat release. Hence, using the rate of heat release to represent the combustion process may be sufficient for accuracy purposes and would definitely reduce computing time. This needs to be validated, though, via the following methodology. The strategy proposed can be divided in two parts: on the one hand, the validation of the method proposed to simulate the combustion; on the other hand, the application of this method to CHT calculations and verification of its validity.

The method is demonstrated here by applying the CHT only to the piston wall, while the other engine walls remain at constant

Table 1
Engine specifications.

Engine type	4-stroke spark ignited
Number of cylinders [−]	4
Engine speed [rpm]	2500
Torque [Nm]	110
Bore - Stroke [m]	0.0075–0.0093
Compression ratio	12:1
Ignition timing [cad aTDC]	−38
Number of valves [−]	2 intake and 2 exhaust

temperature throughout the calculations. This allows reducing significantly the computing time, as applying CHT to all engine walls would imply adding the whole engine block and its cooling system.

2.1. Validation of RoHR approach for combustion modelling

For this, first it has to be demonstrated that using the rate of heat release as input for the CFD calculation of the full cycle yields equivalent results in terms of pressure trace and gas temperature as performing the full combustion CFD calculation.

Hence, first the full cycle combustion process is calculated with the CFD premixed turbulent combustion G-Equation model for the engine described in Table 1, taking into account only the fluid domain, and considering isothermal wall boundary conditions. In order to ensure the convergence and accuracy of the solution, it is necessary to perform several cycles until adjustment of the pressure trace to that obtained with the 1D model GT-Power for this engine and the chosen operation conditions. This is achieved by tuning the constants a_4, b_1 and b_3 of equation (1) that influence the turbulent flame speed of the G-Equation combustion model used [24]. GT-Power results were used in this case because no experimental data was available at the time of calculation. The same adjustment process would need to be made if an experimental pressure trace is taken as reference.

$$S_t = S_l + u' \left\{ \frac{-a_4 b_3^2}{2b_1} Da + \left[\left(\frac{a_4 b_3^2}{2b_1} Da \right)^2 + a_4 b_3^2 Da \right]^{\frac{1}{2}} \right. \tag{1}$$

where u' is the root mean square of the turbulent fluctuating velocity, S_l is the laminar flame speed, a_4, b_1 and b_3 are modelling constants, and Da is the Damkohler number [25].

The rate of heat release which is calculated in the final adjusted combustion simulation is then used to perform a new full cycle calculation with the combustion model deactivated. This means that instead of calculating the combustion process with the G-equation model, the RoHR is introduced as source term in the energy equation and distributed uniformly on all combustion chamber cells at each time step, so that the combustion process is not calculated, only modeled.

Finally the in-cylinder gas properties of both cases are compared in order to validate that the proposed RoHR approach can be used to substitute the CFD full combustion process calculation.

2.2. Application of RoHR approach to CHT calculations

In the second part of this work, the computational domain considered included the solid walls also, so that not only the heat transfer from gas to wall was considered, but also that within the wall. These CHT calculations were performed for the same operation point as above. The solid region was initialized with the temperature obtained from a 1D lumped model in order to accelerate the solid convergence. Indeed, initializing the simulation with a random temperature in the solid region can increase significantly the number of engine cycles needed.

Though the full cycle combustion calculation including CHT, termed from now on **CHT-Combustion**, was started with the tuning constants for the combustion model as adjusted in section 2.1, it was necessary to run again several cycles to adequately adjust the given pressure trace. Indeed, the temperature of the piston wall changes cycle-to-cycle until it converges and this affects the combustion process. The convergence criteria used for the CHT calculations were the stabilization of the trapped mass for the fluid phase, and the heat flux and wall temperature at the fluid-solid interface surface for the solid phase. This calculation is needed as reference case to compare with the results obtained with the RoHR approach.

In order to simplify the combustion process simulation, the RoHR obtained from the full cycle combustion calculation of part I is used as source term in the energy equation for the CHT calculation, since this approach was validated in the previous section. This type of calculation will be named **CHT-RoHR** from now on.

It is important to clarify that this approach is proposed as an alternative to study the heat transfer to the solid components of an internal combustion engine. Indeed, replacing the combustion process calculation by an equivalent RoHR is not sufficient if the purpose is to reproduce correctly the in-cylinder gas properties. In addition, by not modelling the flame front propagation, it might not catch local heat transfer effects.

3. Numerical setup

3.1. Geometry and engine specifications

For part I the computational domain is formed by the combustion chamber with the intake and exhaust manifolds, while for the calculations of part II the solid piston is also included in the domain. Fig. 1 shows the full engine geometry provided by IFP Energies nouvelles, with the parts integrating the solid and the fluid domain. Table 1 presents the main features of the premixed gasoline engine used for the CFD and CHT calculation. For confidential reasons some characteristics cannot be exposed here.

3.2. Mesh generation

In the frame of this study, the commercial programme CONVERGE was used for all the calculations. CONVERGE generates the mesh at runtime, trimming the cells at the intersecting surfaces, after which the intersection information (surface areas, normal vectors) is reduced before storing for each cell. This process is done at the start of the simulation and whenever the mesh has to be refined. The full geometry of the engine has the primary size of the cells of the base grid.

For all calculation cases the base grid of the computational domain has a cell size of 2 mm. To refine the mesh for better accuracy of the results, the base grid size is reduced in specific regions where the physical processes are more complicated. Fig. 2 shows the full engine mesh with the parts integrating the solid and fluid domain.

For all calculations, 2 levels of grid refinement are added near the combustion chamber walls in order to properly model the boundary layer, as this is critical for the heat transfer. In addition, for the cases where the full combustion process is calculated, the region near the spark plug is refined at the start of the ignition to improve the accuracy (see Fig. 2). This refinement is only activated during the ignition process (−39 to −23 cad aTDC). Since the ignition model is deactivated when the RoHR is used as source term to replace the combustion process, the mesh for these calculations is uniformly distributed in the entire combustion chamber region at each time step.

The total number of cells, including both the fluid and solid domains, varies during the simulation, between 1 227 630 cells at bottom dead center (BDC) and 329 021 cells at top dead center (TDC). Furthermore, an adaptive mesh refinement (AMR) algorithm is used to refine the mesh in zones of the combustion chamber where the velocity and temperature gradients are highest. This is advantageous compared to an overall refined mesh, as it allows reducing the computational cost.

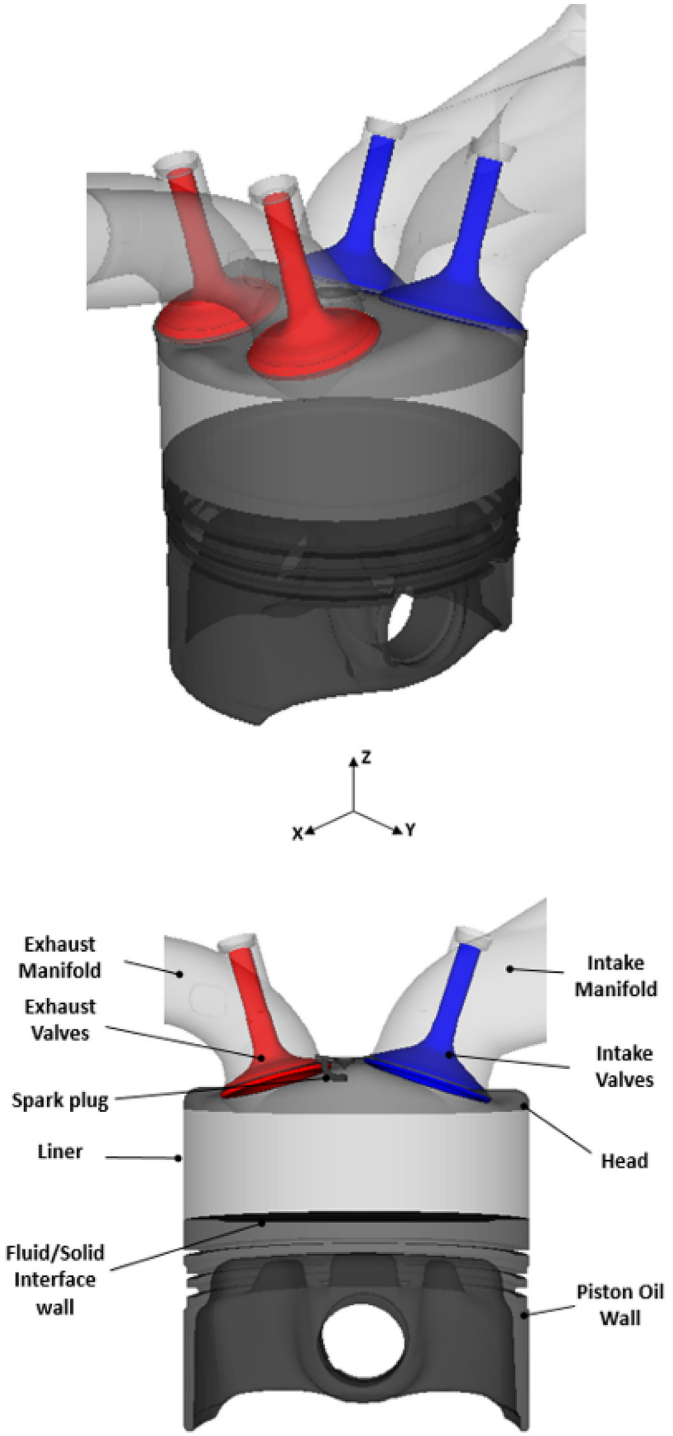


Fig. 1. Calculation domain with the engine parts.

Furthermore, a strategy that consists in relaxing the Courant number from EVO to IVC [26] is used to optimize the calculation time.

3.3. Models and boundary conditions

The combustion process is simulated using the G-Equation combustion model, which tracks the location of the flame front via level set (G is the distance to the flame front) [27]. Its main advantage lies in that it runs faster than a detailed chemistry model, but it can only be used for premixed combustion. The ignition for the G-Equation model is generally achieved through a source term. The RNG (renormalization

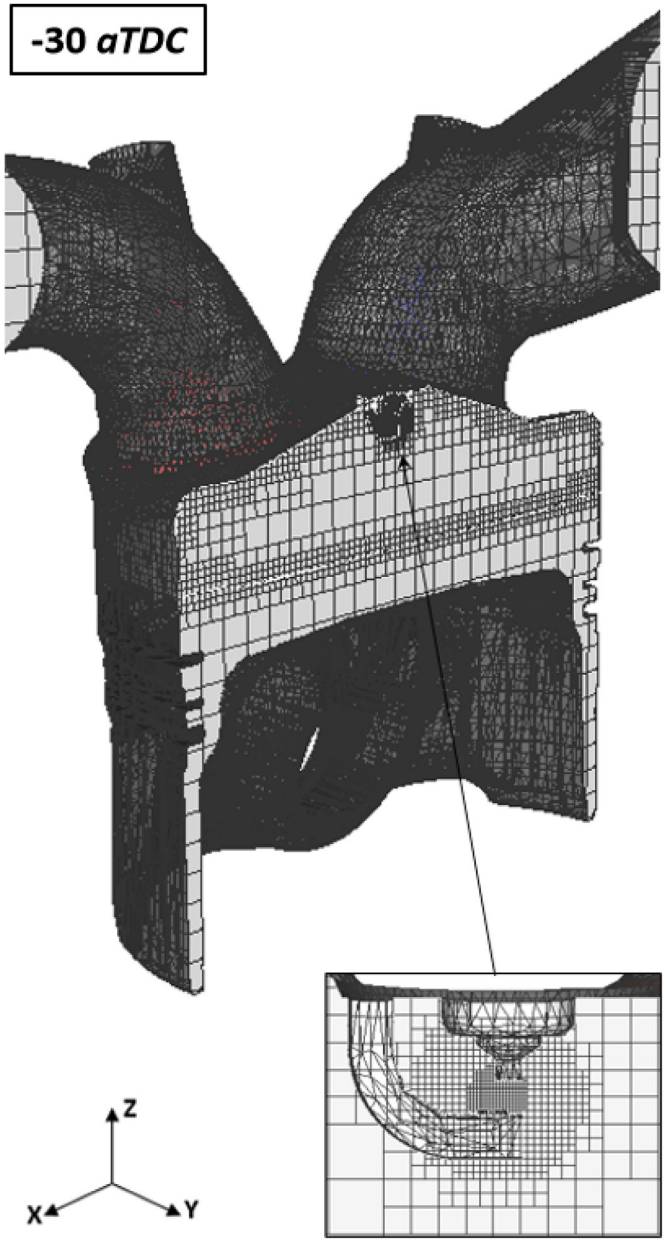


Fig. 2. Mesh charaterization at spark timing.

group) k-ε model is used for turbulence, combined with O'Rourke and Amsden heat transfer wall model [28,29]. This model was chosen on the basis of a preliminary study made with various other models available in CONVERGE, where results were compared to those obtained with a lumped model [17,18]. As is well-known [30,31], SI engines have cyclic dispersion, so that several cycles of the engine were run to ensure that the resulting pressure trace reproduced adequately the given GT-Power data.

For the O'Rourke and Amsden model, the wall heat transfer is given by

$$k\frac{dT}{dx_i} = \frac{a_m c_p F(T_f - T_w)}{Pr_m y} n_i \tag{2}$$

where

$$F = \begin{cases} 1.0 & y^+ < 11.05 \\ \dfrac{y^+ Pr_m}{Pr_t} & \\ \dfrac{}{\frac{1}{\kappa}\ln y^+ + B + 11.05\left(\frac{Pr_m}{Pr_t} - 1\right)} & y^+ > 11.05 \end{cases} \quad (3)$$

$$y^+ = \frac{\rho u_t y}{\mu_m} \quad (4)$$

k is the molecular conductivity, κ is the Von Karman constant (0.4187), B is a law-of-the wall constant, in this case of value 5.5, Pr_m is the molecular Prandtl number, Pr_t is the turbulent Prandtl number, T_f is the fluid temperature, T_w is the wall temperature, and u_t is the shear speed.

The CFD calculations without CHT require as boundary conditions the variation of the temperatures of all the gas exposed surfaces (piston, liner, cylinder head) during the engine cycle. These are obtained from a lumped model [17,18,32–34] by iterating between the CFD and the lumped model and tuning the pressure trace with the GT-Power data.

For the CHT calculations the liner and cylinder head temperature boundary conditions are fixed in the same way. The piston wall temperature is calculated with the conjugate heat transfer model of CONVERGE [35]. For this, the required boundary conditions at the inner surface in contact with the gas are the heat capacitance, the thermal conductivity, and the density of the wall material. The thickness of the material is directly given by the meshed geometry of the wall. For the outer wall of the piston, the convection coefficient and temperature of the coolant surrounding the piston are needed as boundary conditions. To accelerate the convergence of the CHT calculations, which are very time-consuming, the final wall temperature of the piston obtained from the lumped model is used as initialization value for the solid region.

Considering that the convergence time for the solid region is greater than for the fluid region, the super-cycling approach of CONVERGE [35] has been used to solve the conjugate heat transfer problem. This method solves both fluid and solid phases using the transient solver. During the first engine cycle it stores periodically the heat transfer coefficient and the near wall temperature at the fluid/solid interface cells in stages at defined time-step intervals. After the 720 CAD (stages number × degree interval), CONVERGE averages the heat transfer coefficient and the temperatures of the solid cells. It uses the new wall temperatures to solve the fluid and solid with the transient solver. The code freezes the fluid solver at the super-cycling time step interval and solves the steady-state solid temperature, then this is updated to recalculate the solid and fluid with the transient solver. The process is repeated until the solid temperature has converged. It has been demonstrated [1] that applying this strategy reduces the engines cycles that need to be simulated to reach convergence of the solid. For this study multiple cycles were run to calculate the evolution of the temperature on the solid surface of the piston, with a time step interval of 60° to store the heat transfer variables.

4. Results and discussion

4.1. Validation of RoHR approach for combustion modelling

As explained in section 2.1, two CFD combustion calculations have been performed to validate the RoHR approach. The reference case is the calculation of the engine cycle using the G-equation model for the full combustion process. Several cycles have to be run to converge the pressure trace given by the 1D model GT power for the given operating point (see Table 1). The second combustion calculation is made with the RoHR approach explained in section 2.2, for the same operation point and several cycles have to be run also for convergence of the solid wall temperature.

Fig. 3 displays the RoHR obtained from the CFD reference combustion calculation and used as source term in the RoHR approach. In Fig. 4 the pressure traces obtained from both calculations are compared. Clearly, there is a very small difference between both traces with

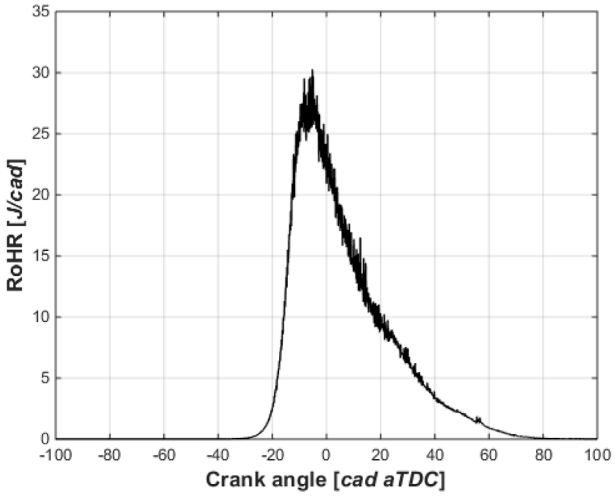


Fig. 3. Rate of heat release obtained with the combustion calculation.

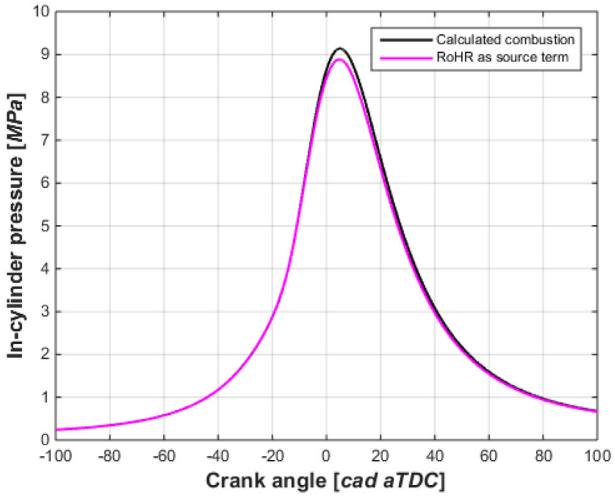


Fig. 4. Pressure trace comparison between both CFD calculations.

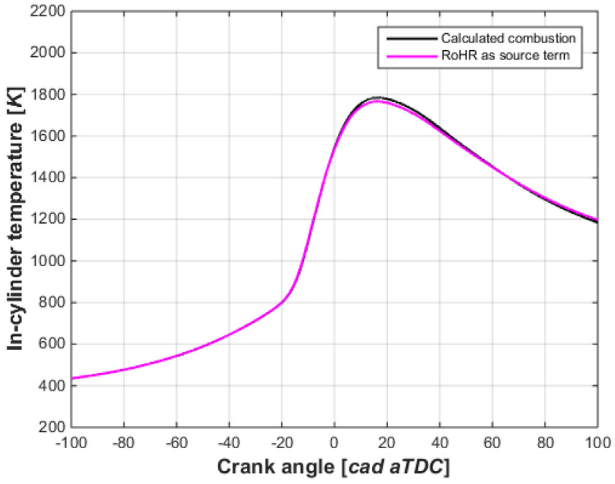


Fig. 5. In-cylinder temperature comparison between both CFD calculations.

a maximum difference of 0.25 MPa at peak pressure. Fig. 5 shows the in-cylinder temperature comparison between both simulations. The gas temperature traces are over-imposed until TDC, after which the temperature for the CFD combustion calculation is slightly higher. These graphs demonstrate that although the RoHR approach does not

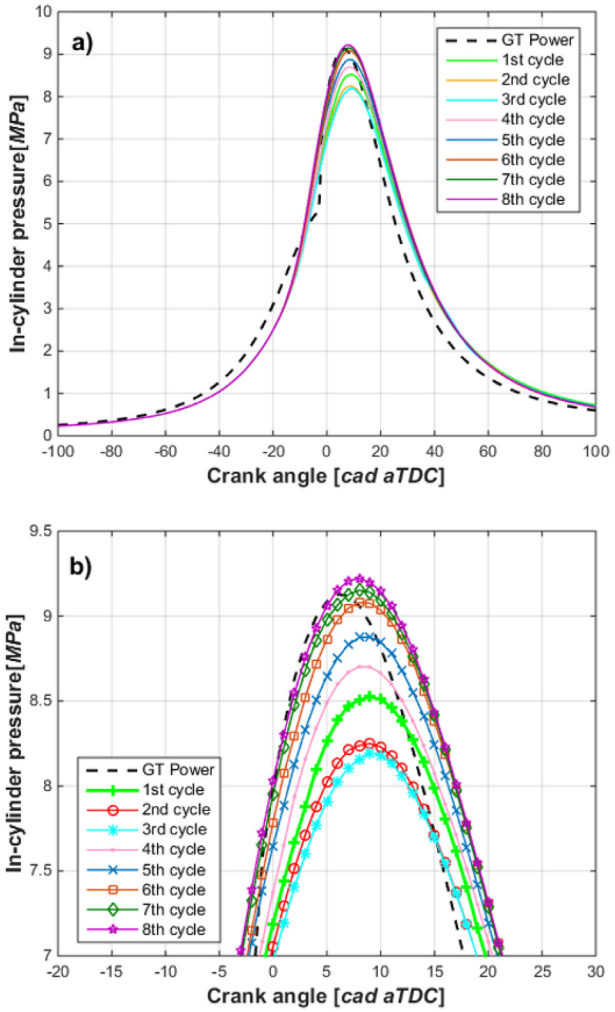


Fig. 6. Combustion calibration with the in-cylinder pressure. a) Closed cycle engine. b) Zoom near TDC.

calculate the combustion process, it yields practically the same evolution of the mean thermodynamic gas properties as the full combustion calculation. From the heat transfer point of view, this means that the RoHR approach may be used for the CHT calculations.

4.2. Results of RoHR approach applied to CHT

Again, the CHT calculation including the full combustion process calculation (CHT-Combustion) is taken as reference here. For the engine combustion calibration eight full cycles had to be run to adjust the pressure trace to the GT-Power trace, as shown in Fig. 6. There are discrepancies between the GT-Power and the CFD simulation pressure traces because the hypotheses and assumptions considered in both models are very different. Since the piston wall temperature variations affect the combustion process and GT Power clearly does not take this into account, the 3D simulation is more accurate in this case. Even though many cycles were needed to adjust adequately the pressure trace, the final mean temperature of the piston surface exposed to the gas is very close to the initial assumption obtained with the lumped model. As a consequence of the flame front propagation, the mean temperature of the piston wall varies during an engine cycle, as shown in Fig. 7. This figure shows in addition the evolution during the 8 cycles of the minimum and maximum temperatures on the piston surface. Note that the minimum and maximum temperatures are not necessarily located in the same cell as the cycles advance. In the first cycle the temperature on the piston surface is not calculated since the software is

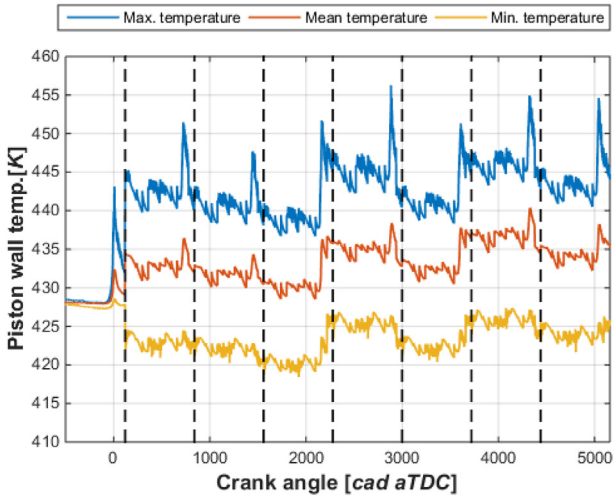


Fig. 7. Temperature on gas exposed surface piston with calculated combustion.

storing the near wall variables as explained in section 3.3. At convergence of the cycles, the observed temperature swings are very similar, as expected, with an average of 424 K for the lowest temperature and 441 K for the highest temperature. The convergence of the gas was verified by the variation of the trapped mass inside the cylinder at the end of the closed cycle. As observed in Fig. 8, there are very small oscillations of the trapped mass due to the cyclic dispersion in gasoline engines, but after the 5th cycle the mass hardly changes.

The CHT-RoHR calculation required fewer cycles to converge both domains, fluid and solid, because the combustion model was deactivated and there was therefore no need for calibration. 4 engine cycles were sufficient, as shown in Fig. 9 that represents the minimum, maximum and mean temperature evolution of the piston wall temperatures during the cycles. Clearly, after the second cycle the temperature follows the same pattern for the next cycles.

On the fluid side, some very small oscillations are observed in the trapped mass, even though the combustion model was deactivated, as displayed in Fig. 10. This means that the oscillations are related to the convergence of the simulation itself. Since the RoHR is imposed, there are no cycle to cycle variations in the pressure trace.

Fig. 11 shows the real RoHR calculated from the full CHT-combustion simulation (RoHRcht) in comparison to that calculated with the CFD-combustion (RoHRcfd) in section 4.1. The latter is the one imposed numerically as source term in the energy equation for the CHT-RoHR simulation. There are some differences, with higher peak values of

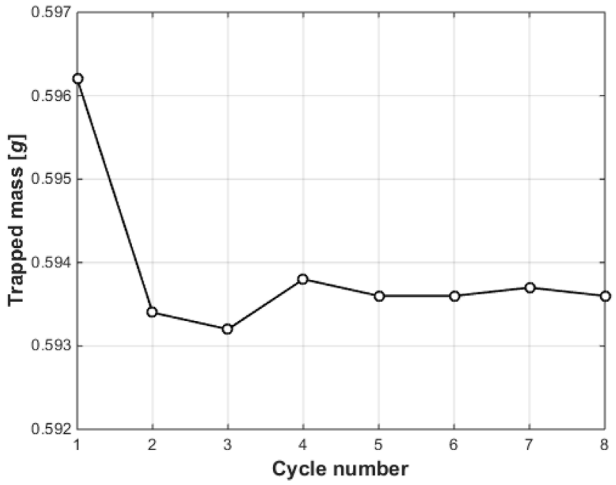


Fig. 8. Trapped mass during the closed cycle with calculated combustion.

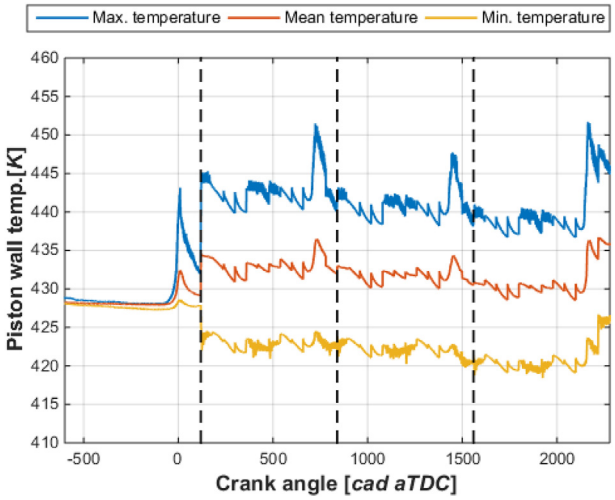


Fig. 9. Temperature on gas exposed surface piston with RoHR as source term.

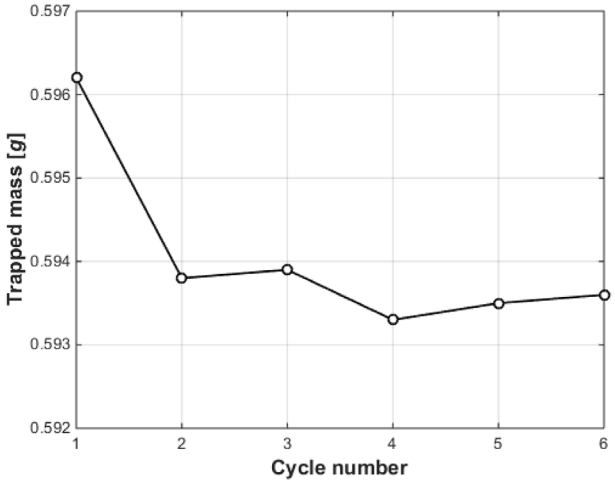


Fig. 10. Trapped mass during the closed cycle with RoHR as source term.

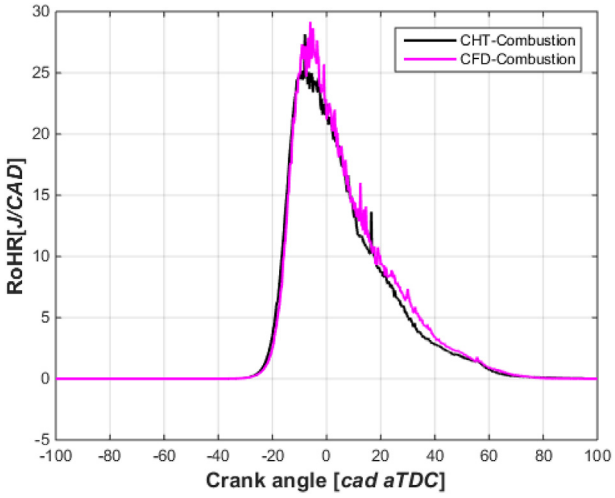


Fig. 11. Rate of heat release comparison between both CHT calculations.

RoHRcht at TDC and partly in the expansion phase in the CHT-RoHR calculation. This difference is due to the less accurate prediction obtained with the CFD-combustion calculation, since wall temperatures are considered as constant.

In the following figures the results of the CHT-Combustion and CHT-RoHR calculations are compared in terms of main thermodynamic properties, for both the gas domain and the solid domain.

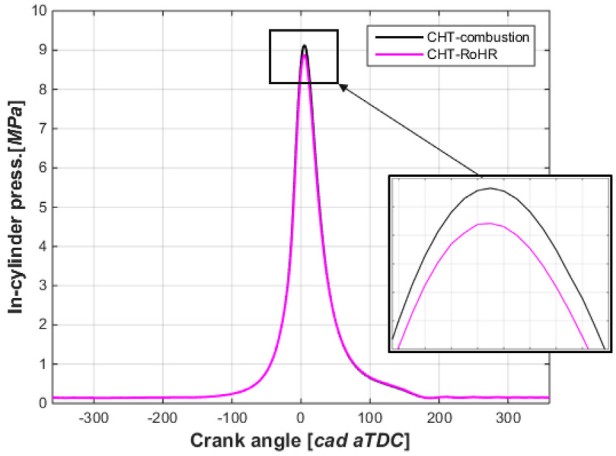


Fig. 12. In-cylinder pressure trace comparison between CHT calculations.

As also observed in the combustion validation case exposed in section 4.1, the maximum pressure peak is lower for the CHT-RoHR (see Fig. 12), with a maximum difference of 0.25 MPa. Looking at the in-cylinder temperature (Fig. 13, during the intake and compression phases the in-cylinder temperature of the CHT-RoHR case is practically the same as for the CHT-Combustion case. The CHT-RoHR curve starts diverging after TDC and the largest difference is observed at the end of the expansion phase and during the exhaust stroke, with a maximum temperature difference of 77 K.

The heat transfer coefficient near the piston wall shown in Fig. 14 has a lower peak value for the CHT-RoHR approach, while the near wall temperature close to TDC is over-predicted, as seen in Fig. 15. Since the heat transfer rate is the product of both variables, there remains to verify what will be the influence on the resulting heat transfer rate. This is seen in Fig. 16: both the CHT-combustion and the CHT-RoHR traces coincide exactly during the complete close cycle, except for the peak values at TDC, where the CHT-RoHR reaches slightly higher values.

Considering that the gas temperature and the heat transfer rate are practically identical during the whole cycle in both calculations, it is hardly surprising to find that the average piston wall temperature evolution obtained with the CHT-RoHR calculation is also almost identical to that of the CHT-Combustion case, with a maximum difference of 1.5 K between the two (see Fig. 17). It is important to remark that the small temperature oscillations observed in this figure are due to

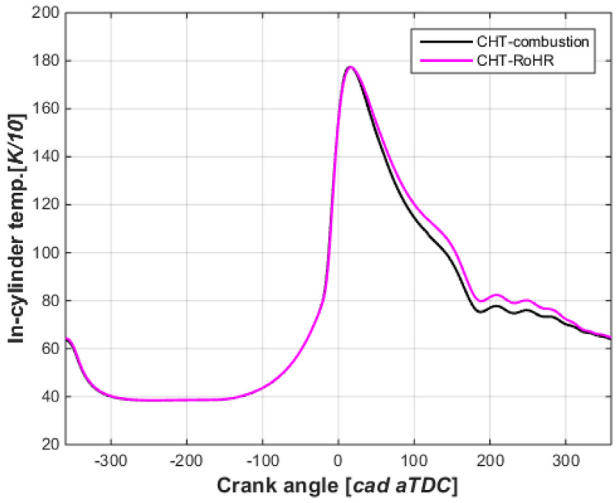


Fig. 13. In-cylinder temperature comparison between CHT calculations.

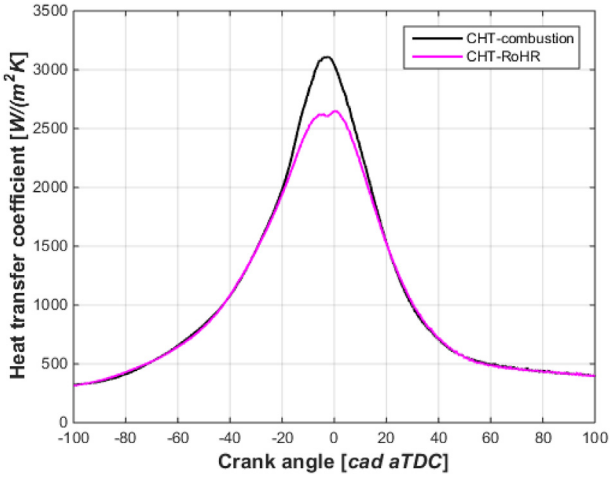


Fig. 14. Heat transfer coefficient near piston wall surface.

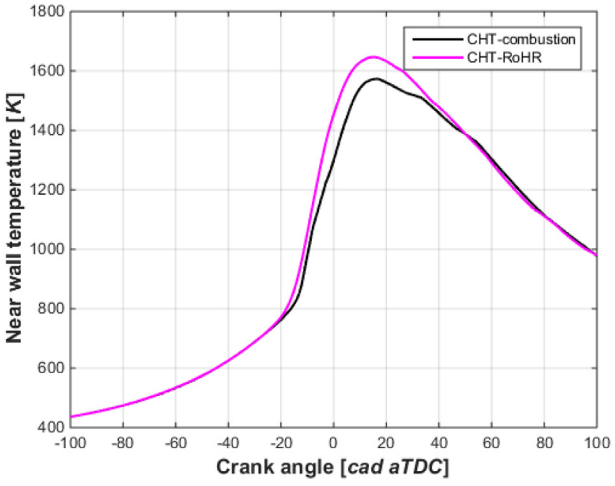


Fig. 15. Near wall temperature on piston.

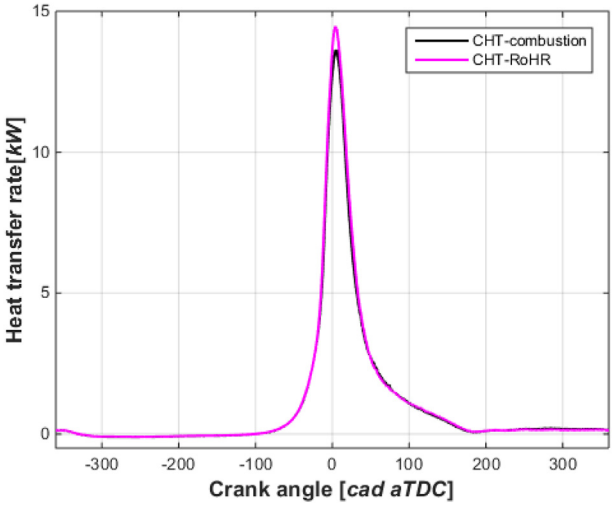


Fig. 16. Heat transfer rate comparison between CHT calculations.

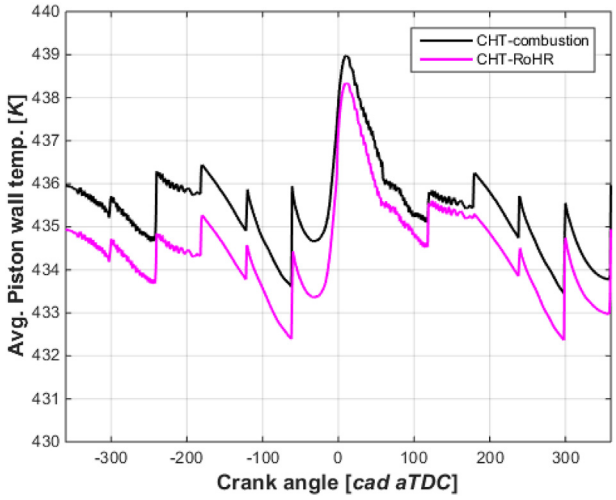
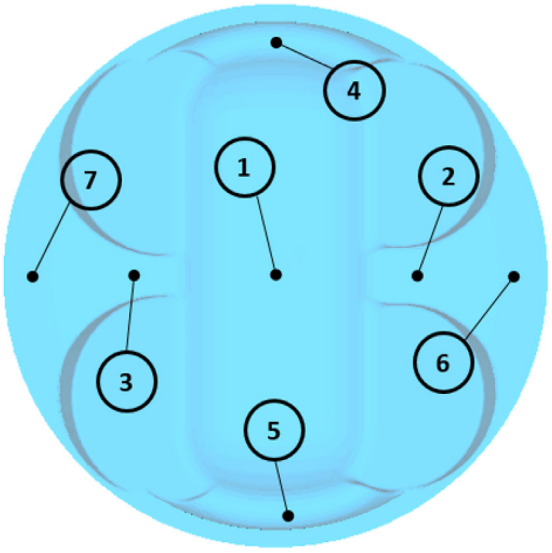


Fig. 17. Average piston wall temperature comparison between CHT calculations.



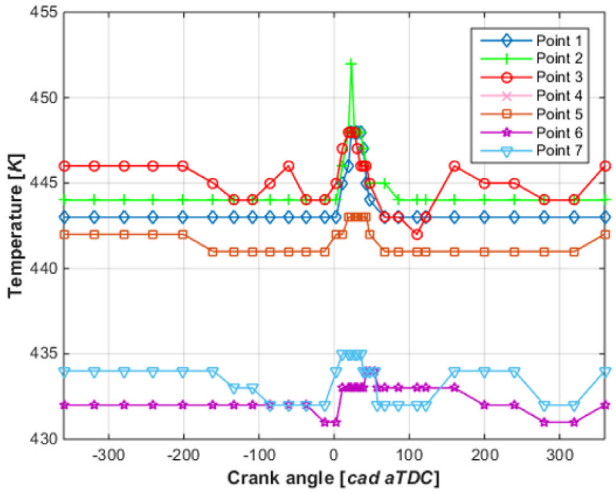


Fig. 18. Temperature evolution of monitor points on the solid piston surface.

the non-uniform distribution of temperature on the piston surface (see Fig. 17), as indeed, the wall temperature is obtained by averaging the temperature of all the cells adjacent to the wall surface.

A more accurate way of looking at the piston wall temperature consists in monitoring the temperature evolution of several points on the piston as shown in Fig. 18. At the points closest to the center, where the piston wall is hottest, the temperature evolves as expected [36] and on the more distant points the temperature changes slightly. However,

the temperature peak occurs at TDC for all the points. The small temperature oscillations observed may be produced by a tumble movement of the gas and the piston geometry. Clearly, all CHT-RoHR averaged variables considered are in phase with those of the CHT-Combustion case, meaning that there is no lag problem, even though the flame front propagation is not modeled in the CHT-RoHR. In order to quantify the differences between both cases the percentage error of the mean values was calculated for each variable during the whole cycle with equation (5).

$$Error = \left(1 - \frac{x_2}{x_1}\right)100 \tag{5}$$

where x_1 is the variable referred to the CHT-combustion model, while x_2 is referred to the CHT-RoHR. The values obtained were −0.21%, −2.11%, 045% y 0.22% for the in-cylinder pressure, the in-cylinder temperature, heat transfer rate on the piston wall and the average piston wall temperature, respectively. This means that during the cycle the error in the gas thermodynamic properties due to the use of the RoHR approach is sufficiently small as not to affect the wall temperature.

To further validate the RoHR approach, the total heat transfer to each wall of the combustion chamber is quantified for both CHT calculations, and presented in Fig. 19. The total heat transferred through all the walls of the combustion chamber is slightly over-predicted (3.7%) by the CHT-RoHR approach. Considering the heat transfer through each wall separately, the CHT-RoHR calculation always yields slightly higher values. The largest difference is observed for the liner (4.4%), then the piston wall (3.8%), and the cylinder head (about 3.2%). As in both calculations the liner and cylinder head are assumed isothermal, the small difference in heat transfer is due to the fact that the near wall temperatures are different in both calculations. For the piston instead, the wall temperature is calculated by the code, and since the temperature is not constant on this surface, it also changes the heat transfer through the isothermal walls. This explains the differences between the two approaches. It is worth noting that the total heat transfer through the piston represents 37% of the total heat transfer through the in-cylinder walls.

In Fig. 19 the total heat transfer through the various walls obtained from the CFD calculations is also shown for comparison. It is important to remark that in this case the differences between the CHT and CFD calculations are relatively small because the material considered is aluminum, which has a high thermal conductivity. For the piston the CHT-Combustion calculation yields a higher difference (6.1%) than the CHT-RoHR (2.5%) when compared with the simple CFD-Combustion. If the heat transfer through all walls is considered, this values are 4% and 0.7% respectively. Since the CHT-Combustion simulation is the most precise of all three cases, it clearly shows that depending on the precision needed, it may be unavoidable to perform full CHT calculations.

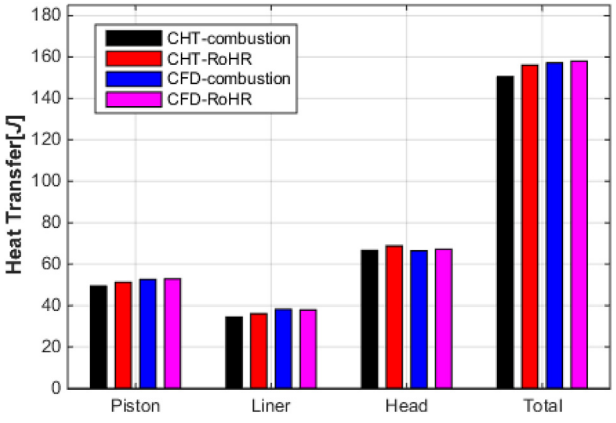


Fig. 19. Total heat transferred to the combustion chamber walls.

Table 2
Engine features.

	Case 1	Case 2	Error [%]
Work cycle [J]	388.73	382.14	1.70
Indicated efficiency [%]	43.94	43.19	1.71

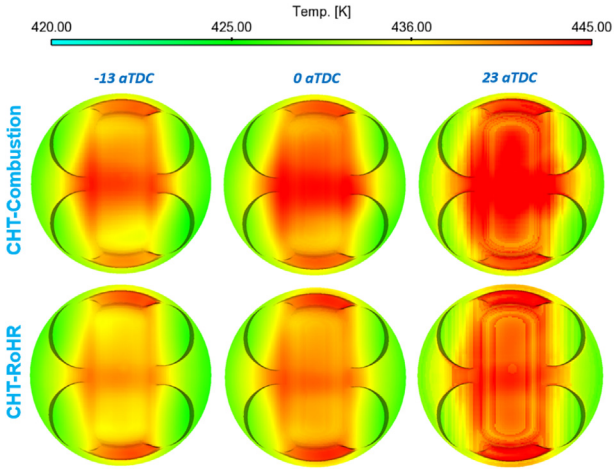


Fig. 20. Temperature distribution on gas exposed surface on solid piston.

Nonetheless, the RoHR approach may be useful if one considers calculating the heat transfer through all the walls of the engine. It is also worth noting that for other materials such as newly developed coating materials to insulate some engine parts, the differences observed would be larger [14–16].

The total work cycle and the indicated efficiency are compared in Table 2. The CHT-RoHR calculation under-estimates the work cycle and the indicated efficiency by less than 2%.

The RoHR approach is also interesting for thermal stress analysis of engine walls, since it allows knowing the spatial distribution of wall temperature, information not provided by the standard CFD calculations (isothermal walls). To complete this study, Fig. 20 shows the temperature distribution on the piston surface exposed to the gas, at three different stages of the engine closed cycle. Noteworthy is that the spatial distribution of the piston temperature is fairly well predicted by the RoHR approach. However, in agreement with Fig. 17, the piston wall is slightly cooler during the closed cycle in the CHT-RoHR case. Also due to the flame front propagation in the CHT-Combustion case, there are hotter regions on the piston that cannot be appreciated with the RoHR approach, especially at the piston center. It is also very interesting to note that the temperature distribution on the piston wall surface is not uniform, contrary to the usual assumption made in CFD calculations of the combustion process. The outer region is about 15° cooler than the center of the piston, due to the effect of the tumble motion in the engine, and this may affect the combustion process in the chamber.

Another important question is the computing cost of the CHT-RoHR versus the CHT-Combustion simulations. The calculations were carried out on a scientific computer cluster where the cores and memory are distributed in several processors. Thus it is difficult to quantify exactly the gain in computing time achieved by using the RoHR approach, since there were no means of controlling on which processors the calculations were performed. However, from previous experience with many CFD calculations, the difference in processors speed is not very significant. Considering this, it may be induced that a reduction in the calculation time of between 38 and 50% by engine cycle has been gained, which is clearly significant. Depending on the accuracy required for the total heat transfer prediction, the cost reduction offered by the RoHR

approach should be taken into account.

5. Conclusions

A new approach is presented that allows optimizing the calculation time of the CHT simulation of an internal combustion engine based on the use of the rate of heat release (RoHR) as source term for the energy equation to replace the full combustion process. The study is performed for a 4-stroke SI engine and considering the heat transfer to the piston wall only.

The approach has been validated by comparing first two CFD calculations, the reference case where the full combustion is solved, and the CFD calculation where the combustion process has been replaced by the rate of heat release. Then, the RoHR approach has been applied to a CHT calculation and compared to a CHT calculation where the full combustion process has been solved (reference case).

One of the main advantages of using the CHT-RoHR approach over the CHT-combustion is that it is not necessary to run many engine cycles just to calibrate the model as is required with the combustion model. Indeed, for the latter, a minimum of eight cycles were needed to guarantee the convergence of the whole domain, while with the RoHR approach three engine cycles were sufficient. This means that the RoHR approach is significantly less costly in computing time, about 38%–50% faster.

Moreover, it has been shown that with the CHT-RoHR approach the in-cylinder gas properties are accurately reproduced when compared to the CHT + combustion solution. Also, the instantaneous average temperature on the piston surface shows a maximum under-prediction of less than 1% throughout the engine cycle. The total heat transfer to the walls is over-predicted by 3.7% with the RoHR approach, and considering each wall separately, the largest difference is observed on the piston liner (4.4%).

In summary, the CHT-RoHR approach yields very good results in terms of spatially averaged values during the whole engine cycle and allows considerable reduction in computational cost. It is therefore very useful to perform parametric studies in cases where it is important to consider the heat transfer within the engine.

However, it has to be noted that if the exact temperature distribution on the piston surface needs to be known, then the RoHR approach may not be accurate enough, especially when it is important to evaluate the hot spots that could induce engine knocking.

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Appendix A. Supplementary data

Supplementary data to this article can be found online at https://doi.org/10.1016/j.ijthermalsci.2019.01.006.

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