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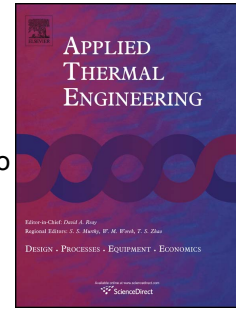
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# Numerical analysis of combustion and transient heat transfer processes in a two stroke SI engine.

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## Abstract.

A zero-dimensional model is presented to simulate the transient processes occurring within a two stroke SI engine. A two zone combustion model, with a spherically expanding flame front originating from the spark location, is applied. The model is numerically solved using the network simulation model which allows coupling the combustion model with a heat transfer model where both radiant and convective heat contributions have been taken into account for the in-cylinder gases. The boundary conditions for this model are the convective heat transferred to the cooling medium. A gas mixture model has been used to obtain the influence of working fluid properties on combustion development.

*Keywords:* Two zone combustion model; Spark ignition engine; Heat transfer; Network simulation model.

## Nomenclature

$a$	Wiebe's efficiency parameter (-)
$A$	heat transfer area (m <sup>2</sup> )
$B$	bore (m)
$E$	internal energy (J)
$h$	specific enthalpy (J kg <sup>-1</sup> )
$l$	connecting rod length (m)

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$L$	stroke (m)
$m$	mass (kg)
$n$	Wiebe's law form factor (-)
$p$	pressure (Pa)
$Q_{LHV}$	lower heating value ( $\text{J kg}^{-1}$ )
$Q$	heat (J)
$r$	ratio of connecting rod length to crank radius ( $r=l/a=2\cdot l/L$ )
$R$	specific gas constant ( $\text{J kg}^{-1} \text{K}^{-1}$ )
$t$	time (s)
$T$	temperature (K)
$\bar{T}_g$	mean gas temperature (K)
$u'$	turbulence intensity ( $\text{m s}^{-1}$ )
$V$	volume ( $\text{m}^3$ )
$W$	work (J)
$x_b$	mass fraction burned (-)

*Greek symbols*

$\alpha$	surface coefficient of heat transfer ( $\text{W m}^{-2} \text{K}^{-1}$ )
$\beta$	radiation adjust parameter (-)
$\delta$	heat transfer wall thickness (m)
$\phi$	relative air fuel ratio (-)
$\varphi$	generic thermodynamic property
$\lambda$	thermal conductivity ( $\text{W m}^{-1} \text{K}^{-1}$ )
$\rho$	density ( $\text{kg m}^{-3}$ )
$\sigma$	Stefan-Boltzman constant ( $\text{W m}^{-2} \text{K}^{-4}$ )
$\theta$	crankshaft angle (deg)
$\theta_0$	outbreak combustion angle (deg)
$\Delta\theta$	combustion duration (deg)

*Mathematics*

$\dot{\phantom{x}}$	time derivative
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*Dimensionless groups*

$Nu$	Nusselt number
$Pr$	Prandtl number

*Re* Reynolds number

*Subscripts*

*0* start of combustion

*a* ambient air

*b* burned gases

*cg* crankcase gases

*clear* clearance (volume)

*comb* combustion (heat addition)

*cond* conductive (heat transfer)

*conv* convective (heat transfer)

*f* fuel

*lub* lubricant

*rad* radiant (heat transfer)

*t* total

*u* unburned gases

*w,b* wall in contact with burned gases

*w,u* wall in contact with unburned gases

*wi* inner engine wall

*wo* outer engine wall

## 1. - Introduction.

Combustion modelling in spark ignition (SI) engines is an established technique for improving engine performance. There are two approaches to combustion modelling: thermodynamic and multidimensional models. The simplest type of thermodynamic model is termed zero-dimensional model and is ideal for general parametric studies. It can be used if the burn rate is known or by the use of semi-empirical relationships to express the burn rate; other thermodynamic models, termed quasi-dimensional models, attempt to simulate actual flame propagation and rely on the prediction of fundamental quantities such as eddy size, turbulence intensity and entrainment velocity. Multi-dimensional models attempt to compute fluid motion by numerically solving the partial-differential equations which represent the basic principles of conservation of mass, momentum, energy and chemical species; these models, owing to their complexity, are impractical for basic engine design considerations [1].

Zero-dimensional models have been used in a large number of studies in order to study different engine parameters: Descieux and Feidt [2] used a one-zone model to obtain the influence of several engine parameters on the power and efficiency of an air standard diesel engine; Guezennec and Hamama [3] used a two-zone model to extract the heat release rate from pressure data for a SI 6-cylinder engine; Hajireza et al. [4] used a three-zone model in order to study the gas thermodynamic characteristics and their relation to knock in SI engines. A comprehensive assessment of various quasi-dimensional models was presented by Agarwal et al. [5] and Verhelst and Sheppard [6].

The heat release process is, together with combustion, the process most studied in internal combustion (IC) engines. For this reason there exist a large number of studies which treat jointly both processes. Nevertheless, in most cases, to obtain the heat release, wall temperature or heat transfer rate were imposed [3, 4, 7, 8], or wall temperature was obtained as a function of engine operation parameters [9, 10]. Other works [11, 12] are mainly focussed on the piston and do not analyse other engine parts.

This work presents a numerical study of combustion and heat transfer processes in a geometrically simplified two-stroke crank case scavenged spark-ignition engine. Assuming as known the burning law and the ambient temperature, the model allows obtaining the heat transfer characteristics of a 49 cm<sup>3</sup> air-cooled SI engine, including the temperature distribution in the whole engine, from inner cylinder wall to fins wall. The temperature maps obtained allow finding those parts of the engine subjected to high temperatures, which is useful to establish the number and distribution of fins in crankcase and block, in an air refrigerated engine, as well as to design surface coating of inner engine surfaces, including piston, or lubrication system.

Analysis has been made for the transient period comprised between the engine start and the moment when a steady temperature distribution is reached. As a result of this analysis, the engine  $p-\theta$  diagram has been obtained, which contains relevant information about engine run, such as the evolution along the start of the engine (including transient period), which is very important on engine warm up and pollutant emissions, or maximum pressure in cylinder, which alerts in engine knock prevention. The numerical technique used to solve the problem is the Network Simulation Method (NSM) whose efficiency has been established for different non-linear problems, both in the heat transfer domain [13, 14], and other fields [15].

## 2. - Problem statement and governing equations.

The engine simulated in this work is a 49 cm<sup>3</sup> crankcase scavenged two-stroke SI engine. Simulation has been made at full load. Table 1 summarizes the engine's main features and operation conditions.

The problem in this work has been approached from a predictive point of view which requires the consideration of all relevant processes in the form of a combustion model [16]. Wiebe function has been used to obtain mass fraction burned (*MFB*):

$$x_b = 1 - \exp \left[ -a \left( \frac{\theta - \theta_0}{\Delta\theta} \right)^{n+1} \right] \quad (1)$$

where  $a$  (efficiency parameter) and  $n$  (form factor) are constants,  $\theta$  is the crank angle during the combustion,  $\theta_0$  is the crank angle of ignition (25° bTDC) and  $\Delta\theta$  is the duration of the combustion (60°). According to Heywood [17], the curve of *MFB* has been fitted with the values recommended by Sher for small two-stroke cycle SI engines ( $a = 3$  and  $n = 5$ ). Changing either  $a$  or  $n$  will alter the shape of the curve significantly. In the case of comparison to the experimental results these parameters must be adjusted to the engine.

Only two zones, burned and unburned gases, are considered in this work, neglecting the influence of the thermal boundary layer and the flame front thickness. As the flame front is propagating from burned to unburned gas region, the volume of the burned gas increases and consequently the volume of the unburned gas decreases. According to Heywood [18], the density ratio is close to 4 for most SI engine operation conditions:

$$\rho_u / \rho_b = 4 \quad (2)$$

The gases in both zones are assumed to follow the ideal gas law [2-6, 8-10, 18-19]. Combustion is assumed complete (no unburned gases are present in burned zone) and there is neither exhaust gas recirculation (EGR) nor residual gases. The influence of residual gases has been taken into account by assuming a scavenging efficiency of 0.5, calculated applying the perfect mixing model, with a delivery ratio of 0.6 (charging efficiency equal to 0.45). Since the lubrication oil is mixed with the gasoline at a very low mixing ratio, its influence in the combustion process has been also neglected.

The unknown values for the model proposed are temperature, pressure, mass and volume for both the unburned and the burned zones and the mass flow from unburned to

burned zone. Following the traditional two-zone engine modelling approach the analysis is based on applying the mass and energy conservation principles for the two zones, supplemented with the state equation in each zone, the mass fraction burned and the volume and area information (based on the engine geometry and kinematics). Since these equations are applicable at each crank angle, it is very easy to formulate them in a differential form as a function of crank angle:

$$m = m_u + m_b \quad (3)$$

$$m_b = mx_b \quad (4)$$

$$\dot{E}_u = \dot{Q}_{w,u} - \dot{W}_u + \sum \dot{m}_j h_j \quad (5)$$

$$\dot{E}_b = \dot{Q}_{w,b} - \dot{W}_b + \sum \dot{m}_k h_k \quad (6)$$

$$\dot{E}_u = \frac{d}{dt}(mh)_u - \frac{d}{dt}(pV)_u = \dot{m}_u h_u + m_u \dot{h}_u - \dot{p}_u V_u + p_u \dot{V}_u \quad (7)$$

$$\dot{E}_b = \frac{d}{dt}(mh)_b - \frac{d}{dt}(pV)_b = \dot{m}_b h_b + m_b \dot{h}_b - \dot{p}_b V_b + p_b \dot{V}_b \quad (8)$$

$$V_c = V_u + V_b \quad (9)$$

where the instantaneous cylinder total volume depends on the piston position and is calculated by the following equation:

$$V_c = V_{clear} + \frac{\pi B^2}{8} L \left( r + 1 - \cos\theta - \sqrt{r^2 - \sin^2\theta} \right) \quad (10)$$

The areas for heat transfer to the walls from the burned and unburned zones are obtained from equations (1), (2), (9) and (10) assuming that the flame front is perfectly spherical and centred in the plug. The main engine parameters are summarized in Table 1.

Assuming that any thermodynamic property of the system ( $\phi$ ) can be expressed as a function of temperature, pressure, and relative air fuel ratio:

$$\phi = \alpha(T, p, \phi) \Rightarrow \phi = \left( \frac{\partial \alpha}{\partial T} \right) T + \left( \frac{\partial \alpha}{\partial p} \right) p + \left( \frac{\partial \alpha}{\partial \phi} \right) \phi \quad (11)$$

and operating with equations (1) to (9) it can be obtained:

$$\dot{T} = \frac{\dot{Q}_w - \dot{W} + \sum \dot{m}_j h_j - \dot{m} h + p \dot{V} - \phi \left[ m \left( \frac{\partial h}{\partial \phi} \right) \right] - (\rho R T + \rho \dot{R} T) \left[ m \left( \frac{\partial h}{\partial p} \right) - V \right]}{m \left[ \left( \frac{\partial h}{\partial T} \right) + \rho R \left( \frac{\partial h}{\partial p} \right) - R \right]} \quad (12)$$

In order to simplify equation (12) there are some assumptions that can be made for the system analysed:



- Relative air fuel ratio remains constant for the fresh gases during the entire cycle.
- Molecular mass is different for burned and unburned gases, but remains constant in each zone during the entire cycle.
- Since the gases are assumed to follow the ideal gas law, their enthalpy can be obtained as a function of temperature only.
- For both the unburned and burned zones it can be established:

$$\Sigma(m_j h_j)_u = m_u h_u; \quad \Sigma(m_k h_k)_b = m_b h_b \quad (13)$$

$$\dot{W}_u = p_u \dot{V}_u; \quad \dot{W}_b = p_b \dot{V}_b \quad (14)$$

Additionally, the heat addition due to combustion is considered in the burned zone. Its value can be obtained as:

$$\dot{Q}_{comb} = \dot{m}_f Q_{LHV} \quad (15)$$

where  $Q_{LHV}$  is the lower heating value of the fuel (gasoline, assumed to be  $C_{8.26}H_{15.5}$ ).

Taking into account all these assumptions, equation (12) can be considerably simplified, although it must be expressed in different terms for unburned and burned zones. For the unburned gases zone, it can be established:

$$\dot{T}_u \cdot m_u \left[ \left( \frac{\partial h}{\partial T} \right)_u - R_u \right] = \dot{Q}_{w,u} + \dot{p}_u R_u T_u V_u \quad (16)$$

whereas for the burned gases zone it can be expressed as:

$$\dot{T}_b \cdot m_b \left[ \left( \frac{\partial h}{\partial T} \right)_b - R_b \right] = \dot{Q}_{w,b} + \dot{p}_b R_b T_b V_b + \dot{Q}_{comb} \quad (17)$$

The enthalpy variation was obtained assuming that the fresh gases are a homogeneous mixture of air ( $0.79 N_2 + 0.21 O_2$ ) and fuel ( $C_{8.26}H_{15.5}$ ) in stoichiometric ratio and the burned gases are a mixture of  $N_2$ ,  $CO_2$  and  $H_2O$  (complete combustion).

The standard state enthalpy for species  $j$  is assumed to be temperature dependent and it takes the following form [18]:

$$\frac{h_j}{RT} = a_{j1} + \frac{a_{j2}}{2} T + \frac{a_{j3}}{3} T^2 + \frac{a_{j4}}{4} T^3 + \frac{a_{j5}}{5} T^4 + \frac{a_{j6}}{T} \quad (18)$$

where the constants  $a_{jk}$  for  $N_2$ ,  $O_2$ ,  $CO_2$  and  $H_2O$  are given in [18] in two different temperature ranges, appropriate for unburned and burned mixture property calculations.

Enthalpy for the fuel is also fitted to a polynomial form:

$$h_f = A_{f1}\theta + A_{f2}\frac{\theta^2}{2} + A_{f3}\frac{\theta^3}{3} + A_{f4}\frac{\theta^4}{4} - \frac{A_{f5}}{\theta} + A_{f6} + A_{f8} \quad (19)$$

where  $\theta = T(\text{K})/1000$ ,  $A_{f6}$  is the constant for the datum of zero enthalpy for  $\text{C}_2$ ,  $\text{H}_2$ ,  $\text{O}_2$  and  $\text{N}_2$ , at 298.15 K (for a 0 K datum,  $A_{f8}$  is added to  $A_{f6}$ ) and the values for coefficients  $A_{fj}$  are given in [18].

The convective heat transfer coefficient in both zones is obtained by applying the model proposed by Rao and Bardon [20]:

$$Nu = 0.058 \cdot Re \cdot Pr \quad (20)$$

where the characteristic velocity adopted to obtain Reynolds number is the turbulence intensity,  $u'$ .

To obtain the turbulence intensity, the following model [18] is adopted:

$$u' = u'_0 \left( \frac{\rho}{\rho_0} \right)^{1/2} \quad (21)$$

where the subscript 0 refers to the start of the combustion process. The turbulence intensity at this moment has been assumed equal to about half the mean piston speed.

In the burned gases zone, a radiation term proposed by Annand [18] is added to the convective heat transfer:

$$\dot{Q}_{rad} = \beta \sigma (\bar{T}_g^* - T_{wi}^*) \quad (22)$$

where  $\sigma$  is the Stefan-Boltzman constant,  $\bar{T}_g$  is the mean gas temperature,  $T_{wi}$  is the inner wall temperature and  $\beta$  is an adjust parameter for which a value of 0.6 has been adopted.

Therefore, the heat transfer rate in both unburned and burned zone is obtained as:

$$\dot{Q}_{w,u} = \alpha_u A_{w,u} (T_u - T_{wi}) \quad (23)$$

$$\dot{Q}_{w,b} = \alpha_b A_{w,b} (T_b - T_{wi}) + \beta \sigma (T_b^* - T_{wi}^*) \quad (24)$$

To obtain the inner wall temperature, heat conduction through the cylinder walls is considered:

$$\dot{Q}_{cond} = \frac{\lambda A}{\delta} (T_{wi} - T_{wo}) \quad (25)$$

Finally, to obtain the outer wall temperature, convection between cylinder outer wall and cooling medium is considered:

$$\dot{Q}_{conv,j} = \alpha_j A_{wo,j} (T_{wo,j} - T_j) \quad (26)$$

where the subscript  $j$  refers to the cooling medium,  $T_j$  is the temperature for cooling medium  $j$ ,  $A_{wo,j}$  and  $T_{wo,j}$  are the area and temperature of the engine outer wall in contact with cooling medium  $j$  and  $h_j$  is the convective heat transfer coefficient for cooling medium  $j$ . Three different fluids have been considered:

- Air at a temperature  $T_a = 298.16$  K;  $\alpha_a$  value depends on air velocity and external surface geometry. A constant value of  $66 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$  obtained applying the Colburn correlation [21] for forced convection with an air velocity of 12 m/s was assumed in the entire external engine finned surface. The engine axis has been considered in vertical position (Fig. 2a) and the air flows perpendicular to this vertical axis.
- Crankcase gases at a temperature  $T_{cg} = 333.16$  K;  $\alpha_{cg}$  is equal to  $415 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$  and was calculated applying the correlation proposed by Rao and Bardon [20].
- Lubricating oil, which forms a separating film between piston and cylinder, has been assumed to be at a constant temperature  $T_{lub} = 393.16$  K;  $\alpha_{lub}$  is equal to  $1668 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$  and was calculated applying the correlation proposed by Sieder and Tate [21].

### 3. – Numerical approach.

The differential equations (16) and (17) under boundary conditions (26) have been solved using the network simulation method (NSM), a powerful numerical method extensively used in a diverse variety of thermal and fluid problems [13-15].

NSM is founded on the classical thermoelectric analogy between thermal and electrical variables. The starting point is a network circuit design, whose equations are formally equivalent to equations (16) and (17). The engine walls have been discretized and a sufficient number of networks are parallel and serially connected to form the engine walls. Heat transfer conditions (23) to (25) and boundary conditions (26) are added by means of special electrical devices. Once the network model is designed, its simulation is carried out through adequate circuit resolution software; PSpice® [22] is the software selected for this study.

The electrical analogy relates the electrical potential to the temperature, while the electrical current is equivalent to the energy flow. According to this analogy, the term

$\dot{T} \cdot m \left[ \left( \frac{\partial h}{\partial T} \right) - R \right]$  for unburned and burned gases in equations (16) and (17) must be implemented by means of a capacitor. Since  $\left( \frac{\partial h}{\partial T} \right)$  depends on unburned (or burned) gases temperature and PSpice does not allow the defining of variable capacitors, equations (16) and (17) must be rewritten in the following form:

$$\dot{T}_u = \frac{\dot{Q}_{w,u}}{m_u \left[ \left( \frac{\partial h}{\partial T} \right)_u - R_u \right]} + \frac{\dot{\rho}_u R_u T_u V_u}{m_u \left[ \left( \frac{\partial h}{\partial T} \right)_u - R_u \right]} \quad (27)$$

$$\dot{T}_b = \frac{\dot{Q}_{w,b}}{m_b \left[ \left( \frac{\partial h}{\partial T} \right)_b - R_b \right]} + \frac{\dot{\rho}_b R_b T_b V_b}{m_b \left[ \left( \frac{\partial h}{\partial T} \right)_b - R_b \right]} + \frac{\dot{Q}_{comb}}{m_b \left[ \left( \frac{\partial h}{\partial T} \right)_b - R_b \right]} \quad (28)$$

In order to avoid convergence problems due to discontinuities during burning zone creation and unburned zone consumption, equations (27) and (28) are solved in an integrated way. Figure 1a schematically shows the network model corresponding to the integrated model for equations (27) and (28). A capacitor C of value 1 represents the temporary variation in gases temperature,  $\dot{T}$ , where  $T$  is an averaged temperature which depends on burned and unburned gases temperature and mass proportion. All the terms on the right of equations (27) and (28), except the heat addition due to combustion, are implemented by means of voltage controlled current generators. In burned gases zone, the heat is transferred through the cylinder walls by means of two different mechanisms and therefore two different voltage controlled current generators are necessary to implement the first term on the right of equations (27) and (28): current generator  $\dot{Q}_{conv}^*$  simulates the convection heat transfer and therefore  $T - T_{wi}$  must be the controlling voltage for this source; current generator  $\dot{Q}_{rad}^*$  simulates the radiation heat transfer and therefore  $T^4 - T_{wi}^4$  must be the controlling voltage for this other source. Additionally, the definition of these current generators takes into account a flame propagation model and the variations in the heat exchange coefficients.  $T$  is the controlling voltage for the second term on the right of equations (27) and (28), whereas the variations in the value of  $\dot{\rho}RV$  are included in the source definition. The heat addition due to combustion only depends on the *MFB* and the lower heating value of the fuel; therefore the last term on the right of equation (28) is implemented by means of a source  $\dot{Q}_{comb}^*$  which varies according to the *MFB*. The denominator  $m \left[ \left( \frac{\partial h}{\partial T} \right) - R \right]$  in all the terms on the right of equations (27) and (28) is implemented by means of a variable voltage

source whose output voltage varies according to a polynomial function of  $T$  in agreement with equations (18) and (19). This source is included in the definition of all the previously described current generators as an additional controlling voltage source.

Finally, to obtain  $T_{wi}$ , the network models defined for the combustion chamber must be connected to the engine walls. Figure 1b schematically shows how this connection is implemented. As is schematically represented in Figure 2b, the engine inner walls are considered to be divided into eleven different zones, namely *Cells 1* to *Cells 11*. Each “ $j$ ” wall zone is connected to two voltage controlled current generators  $\dot{Q}_{conv,j}$  and  $\dot{Q}_{rad,j}$  that represent the heat transferred from gases to “ $j$ ” zone. In order to assess the piston motion, in *Cells 6* to *Cells 8* zones, the current generators vary not only with gases temperature but with piston position.

In order to simplify the definition of the model’s geometry and reduce the computational cost, a 2-D discretization of the solid body has been built: cylindrical symmetry is assumed for the piston and cylinder geometry, whereas spherical symmetry is assumed for the cylinder head. Figure 2a shows a view of the simplified engine geometry. Annular geometry has been chosen for cylinder fins, and an innovative *spherical fin* has been implemented for the cylinder head. Extended surfaces of the actual engine do not have such geometry, but they have been adjusted to obtain the same exchange surface area as in the real engine, which assures in practical terms the validity of the heat transfer model.

As can be seen in Figure 2b, to discretize the entire engine, a total of 234 unitary networks are serially connected. Figure 3 shows the four types of unitary network used: a) middle symmetrical cell, b) symmetrical cell with convection, c) middle fin cell, and d) end fin cell (with convection on the end surface).

#### 4. – Results and discussion.

In order to analyze the transient period, the simulation runs for a period of 300 seconds which, as can be seen in figures 4 to 6, is enough to assure that a stable temperature distribution is reached for the entire engine. Figures 4 to 6 show the temperature evolution at twelve different points as well as the thermal map obtained for the cylinder body, the cylinder head and the piston six minutes after engine start. As can be seen, for the piston the stable situation is reached in less than one minute, whereas in the case of

the cylinder body and the cylinder head this time increases up to nearly six minutes due to the presence of fins.

The temperature distribution shown in Figure 4 for the cylinder body is in good agreement with that obtained by Šelih et al. [23] or Wu and Chiu [24]. As can be seen in Figure 5, due to its lower thermal conductivity, the temperature in the spark plug is higher than in the rest of the cylinder head. As expected, temperature decreases from the inner wall, in contact with combustion gases, to the fins walls in contact with ambient air. Finally, the temperature distribution shown in Figure 6 for the piston is in good agreement with that presented by Heywood for SI engines [18] and that obtained by previous researchers such as Esfahanian et al. [11] or Morel and Keribar [12].

The in-cylinder pressure obtained is shown in Figure 7 as a function of the crank angle. As can be seen, only slight differences between cold engine and hot engine conditions are found. Once the temperature distribution in the entire engine reaches a stable situation, the peak in-cylinder pressure rises to near 26 bars, which represents only 2 % of increase with respect to the value obtained for cold engine conditions. Similarly, the maximum gases temperature obtained for hot engine conditions is 2101 K whereas for cold engine conditions the maximum temperature obtained is 2061 K.

Finally, Figure 8 shows the overall energy balance in the cylinder as a function of crank angle. According to this figure, around 30.9 % of total energy released during fuel combustion is transformed into useful work (the friction effect has not been taken into account) whereas around 38.6 % of energy added is lost in the exhaust gases enthalpy and the other 30.5 % represents the heat transfer rate to the cooling medium. The values obtained are in good agreement with typical values given in the specialized bibliography [18] and the energy balance (between energy input and energy output) is in near equilibrium, with only a slight deviation, lower than 1 %, attributable to the use of a relatively high data sampling rate. A closer energy balance could be reached by reducing the data sampling rate but, on the other hand, the computational cost would increase. In order to achieve a balance between accuracy and computational cost, the data sampling rate has been fixed at  $1 \cdot 10^{-5}$  seconds.

## 5. – Conclusion.

A zero-dimensional combustion model consisting of two zones, coupled with a heat release model has been numerically solved in this paper. A gas mixture model has been used to obtain the influence of working fluid properties on engine performance. The model allows obtaining the in-cylinder temperature and pressure profile, the engine energy balance or the engine temperature map as well as other derived results. The results obtained are in good agreement with other results obtained by previous researchers, proving the ability of the model proposed in this work.

Approximately six minutes have been found as the duration of the transient period from start up in the whole engine, although the piston reaches the regime conditions in less than one minute. Referring to temperature, the most critical point of the engine has been found in the spark plug and its vicinity, more even than in piston head.

Using the model proposed in this work it is relatively simple to analyze, in further work, the influence of different operation parameters like cooling medium conditions, engine speed, combustion characteristics (start of combustion, total combustion duration, or Wiebe function adjust parameters). Finally, using a mesh model more adjusted to the engine geometry, the computational cost will increase, but it will be possible to analyze the influence of fin geometry on the engine temperature distribution.

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Figure 1. Schematic representation of the engine network model: 1.a. integrated combustion model corresponding to equations (27) and (28); 1.b. connection between the combustion model and the engine walls.

Figure 2. Engine solid model (a) and mesh (b). Eleven different wall zones have been considered in order to assess the burned gases zone motion. Three different convective boundary conditions exist: ambient air at 298.16 K, crankcase gases at 333.16 K, and oil film at 398.16 K.

Figure 3. Unitary networks used in the engine network model; a) middle symmetrical cell, b) symmetrical cell with convection, c) middle fin cell, and d) end fin cell (with convection on the end surface).

Figure 4. Cylinder body temperature map at  $t = 300$  sec and temperature evolution at fin end (6), fin base (7) and cylinder body inner wall (8)

Figure 5. Cylinder head temperature map at  $t = 300$  sec and temperature evolution at fin end (1), fin base (2), cylinder head inner wall (3), plug outer wall (4) and plug inner wall (5).

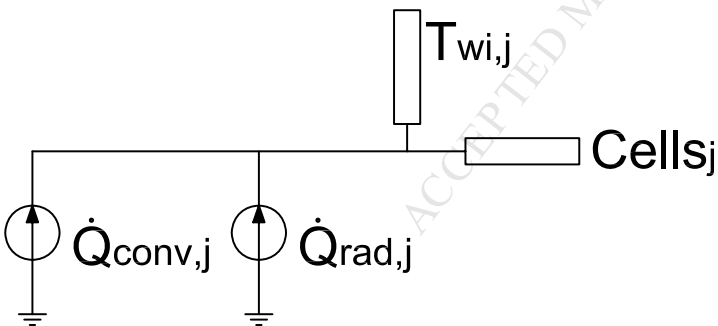
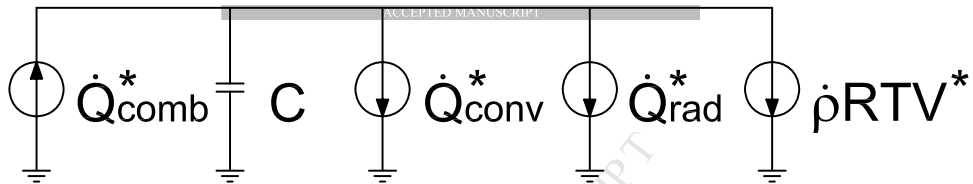
Figure 6. Piston temperature map at  $t = 300$  sec and temperature evolution at piston base (9), piston middle point (10), piston crown outer edge (11) and piston crown center (12)

Figure 7. In-cylinder pressure and temperature curves for  $t = 0$  and  $t = 300$  sec.

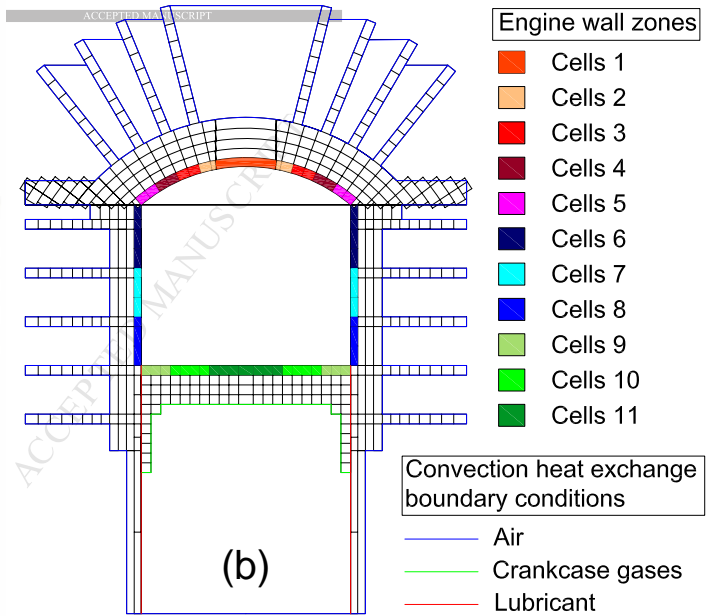
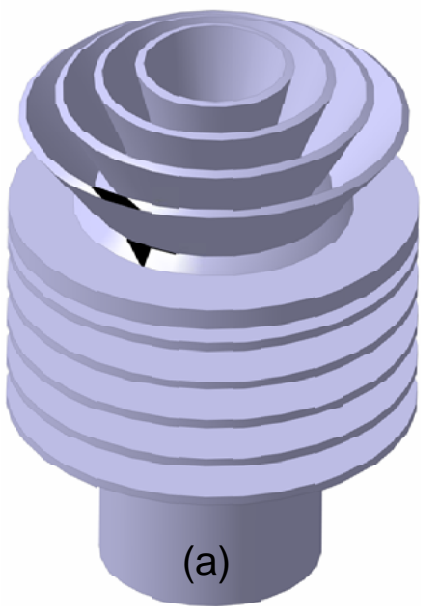
Figure 8. Overall energy balance in the cylinder as a function of crank angle.

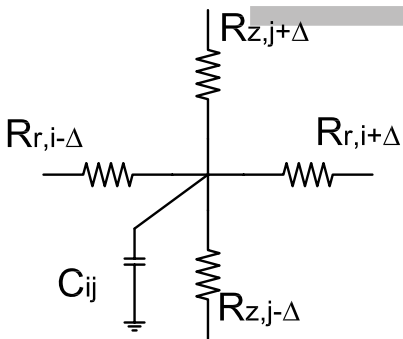
Engine speed	6000 rpm
Bore ( $B$ )	4.3 cm.
Stroke ( $L$ )	3.374 cm.
Clearance volume ( $V_{clear}$ )	6 cm <sup>3</sup>
Compression ratio ( $\varepsilon$ )	9.167
Connecting rod length ( $l$ )	6.248 cm
Ratio of connecting rod length to crank radius ( $r=l/a=2\cdot l/L$ )	3.704
Delivery ratio ( $\zeta$ )	0.6
Scavenging efficiency ( $\zeta_{sc}$ )	0.5
Charging efficiency ( $\zeta_{ch}$ )	0.45

Table 1. Engine parameters.

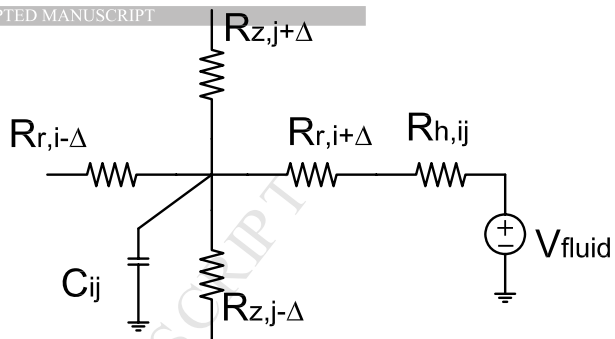


$$T_{wi} = \frac{\sum_j T_{wi,j} \cdot A_j}{A_t}$$

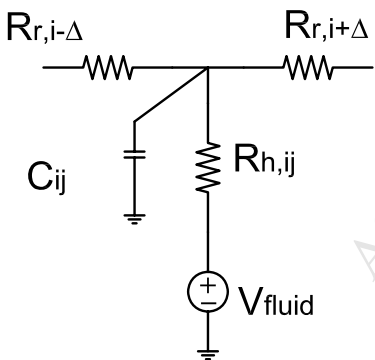




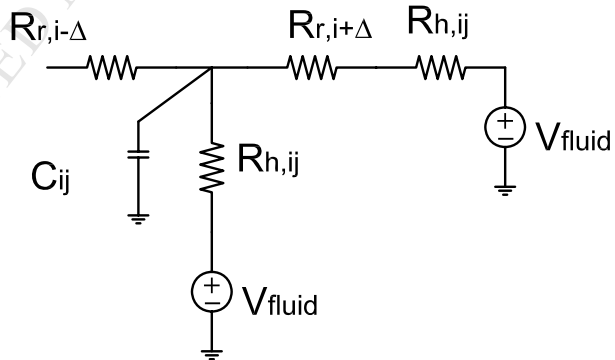
(a)



(b)



(c)



(d)

