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# DEVELOPMENT AND COMPARISON OF A PHENOMENOLOGICAL PREDICTIVE MODEL OF ONE AND TWO ZONES FOR A SPARK IGNITION ENGINE

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**Abstract.** This paper presents an analysis of phenomenological model with one and two zones for a spark ignition internal combustion engine. The combustion products were modeled depending on the kind of combustion reactions: lean ( $\text{CO}_2$ ,  $\text{O}_2$ , Ar,  $\text{H}_2\text{O}$  and  $\text{N}_2$ ); Rich ( $\text{CO}_2$ ,  $\text{H}_2\text{O}$ , Ar, CO,  $\text{H}_2$  and  $\text{N}_2$ ) and stoichiometric. In the combustion reactions, the oxidizer was considered to be dry air (Ar,  $\text{N}_2$ ,  $\text{O}_2$  and  $\text{CO}_2$ ) or moist air. For the mathematical model accomplishment, all substances were considered ideal gases. The properties of gases of products of combustion and ethanol are modeled from data of literature. For closed and open phases of the cycle the first law of thermodynamics was applied, while for combustion evolution it was utilized Wiebe's equation. The model was developed in Matlab<sup>®</sup> using fifth order Runge-Kutta to solve the differential equations in order to obtain temperature, pressure and trapped mass of the cylinder, and Newton-Raphson method for convergence of mass inside the cylinder. The objective of this work is to present the difference of the results obtained when there is one or two zones in the combustion chamber. The results show that the model with two zones are better than that of a zone, that is, it makes a prediction more consistent compared with values of the literature, and is presented how the speed, relation air-fuel and load influences the results of performance parameters.

**Keywords:** Modeling of SI, predictive phenomenological model, one or two zones, combustion model

## 1. INTRODUCTION

The simulation of engine is a process that plays an important role in the conception and development of such machines. Moreover, process simulation studies mean to estimate the engine performance, reduce fuel consumption and emissions. Modeling these processes in engines contributes to improve the basic understanding of physics and chemistry of the phenomena of interest.

Modeling activities can bring great contributions to engineering in different levels of generality or detail, corresponding to different stages of development of the model. The models can be either simplified as the predictive to obtain a trend or more complex using chemical equilibrium, kinetics and other models for more refined prediction of their performance parameters, depending on the type of analysis it's wanted to perform.

There are many nomenclatures for the mathematical models in SI. In this article the same nomenclature adopted by Heywood (1988), that divides the engine models in three groups: zero-dimensional, quasi-dimensional and multi-dimensional. Zero-dimensional model is based on the thermodynamic laws. The geometry of the engine is mostly unconsidered and it is assumed that the mixture of gases in the cylinder is homogeneous. In quasi-dimensional model, the burning rate is determined by the flame propagation, requiring the modeling of turbulence and the knowledge of the geometry of the cylinder in order to estimate the geometry of flame front. The combustion is divided in two zones: the burnt zone and the unburnt zone. Though the flame propagation technique is mostly used in two-zone models, the burn rate profile can also be used, as will be mentioned below. Both types of model are also denominated Phenomenological models. Multi-dimensional models are based on the kinetics of the fluid in the cylinder and require a higher level of detailing of the geometry engine, as well as a much higher computational capability.

The thermodynamic model used the First Law of Thermodynamics to explain the variations of internal energy, pressure, losses of heat transfer to the walls and release of heat. The release of heat is related to a combustion enthalpy (Ramos, 1989).

In models of one zone the temperature, pressure of the mixture in the cylinders and composition is uniformly assumed throughout the cylinder, the gases are modeled according to an ideal gas mixing behavior (Ramos, 1989). The heat transfer losses are calculated by means of correlations, and the internal energy of the cylinder gases are calculated by means of an empirical correlation dependent on the temperature and composition of the mixture, for combustion products

uses Janaf Thermodynamics data (Chase *et al.*, 1985) and for ethanol utilize the Barin data (Barin, 1993). But the assumptions of uniform pressure, temperature and instantaneous mixing made in one-zone thermodynamic models are clearly unrealistic. These limitations can be eliminated, in part, by dividing the cylinder mixture into two zones, with the burned and unburned gas zones assumed to have homogeneous but different temperatures, the heat transfer between the zones has been neglected (Ramos, 1989).

The purpose of this article is to present a zero-dimensional phenomenological predictive model, for one and two zones in combustion chamber, for spark ignition engine (SI). It will also be presented a analyze the difference of the one and two zone models in the combustion chamber in the prediction of the performance parameters of an engine. The results of the model are compared with that of the literature (Gallo, 1990) for anhydrous ethanol and an analysis the influence the speed, relation AF and load in the performance parameters in the engine for hydrous ethanol, moist air.

## 2. METHODOLOGY

The conception of the predictive model started with choosing the involved substances in the combustion process of the SI, depending on the intended analysis type. The involved substances are considered as ideal gases. Although of reaching considerable elevated pressures inside the engines, the hypothesis of an ideal gas in the thermodynamic modeling of the process is rather reasonable. The obtained results in the work of Zacharias (1967) demonstrate that, for pressures until 100 atm and temperatures until 2500 K, the compressibility factor is unitary. That means that, in such range, the ideal gas hypothesis is valid. Just above these values of pressure and temperature, the maximum error for the enthalpy, for example, is approximately 4%.

As it is a predictive model, the choice of substances was chosen due to the type of combustion reaction, lean mixture ( $CO_2$ ,  $O_2$ , Ar,  $H_2O$  and  $N_2$ ); rich ( $CO_2$ ,  $H_2O$ , Ar, CO,  $H_2$  and  $N_2$ ) and stoichiometric, the fuel is  $C_2O_6H$ . The properties of such species where modeled in presence of the air and combustion gases, according to the values of Janaf (Chase *et al.*, 1985). The ethanol properties where taken from (Barin, 1993).

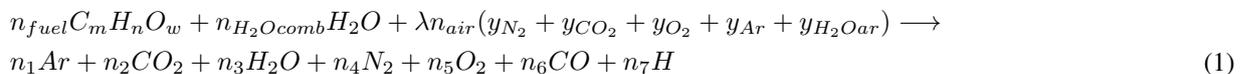
The SI is divided into two phases, such as: closed (compression, combustion and expansion), and open (intake and exhaust). The model is zero dimensional phenomenological, admitting one or two zones inside the combustion chamber. In the combustion phase, the rate of heat release is given by the Wiebe function that determine the fraction of burned mass in function the crank angle, this modeling is most utilize on SI, this function was development in 1967.

The heat transfer model in the closed phase used the heat transfer coefficient of Hohenberg (1980), and in the open phase, two coefficients were used, one the intake and other for the exhaust, taken from Nishiwaki (1979).

The model use fifth order Runge-Kutta method to solve the differential equations of temperature, pressure and mass. The model convergence criterion is the mass inside the cylinder, and uses the Newton Rapshon method. In most of the performed cases there were at most 5 interactions to achieve convergence.

### 2.1 Combustion Reaction Balancing

The composition of dry air is relatively constant, despite the small variations in function of the geographical location and altitude. In this article the composition utilized is Nitrogen ( $N_2$ ) 78.084%, Oxygen ( $O_2$ ) 20.948%, Argon (Ar) 0.934%, and Carbone dioxide ( $CO_2$ ) 0.033% (Lopes *et al.*, 2008). The humidity in the air was obtained by Antoine's Equation (Boublík Tomas, 1973), a new composition the moist air is calculated, depending on the relative humidity. The chemical reaction Eq 1 represents simplified the combustion process for an SI with moist air.



Where:  $n_i$  is the number of moles of the products the reaction,  $i$  are the species of the flue gases,  $y_i$  represents the molar fraction of each specie in moist air.  $n_{air}$  is theoretical stoichiometric air consumption of air and  $n_{fuel}$  is number of moles of fuel, and  $\lambda$  is he coefficient of theoretical air. M, N and W represent the number of atoms the carbon, hydrogen and oxygen respectively.

If  $\lambda = 1$  there is a stoichiometric reaction, complete combustion, with  $n_1$  until  $n_4$  in the exhaust gases. If  $\lambda > 1$ , there is an excess air reaction, it means poor mixture, so, there is  $n_1$  until  $n_5$  in the exhaust gases. When  $\lambda < 1$ , there is a rich mixture, with  $n_1$  until  $n_7$ , but without  $n_5$  that is  $O_2$ , in the exhaust gases. To find the number of moles of each species of the flue gases there is a need for one more equation since there are 7 unknowns and 6 equations in this way this equation comes from the chemical equilibrium that correlates the CO with  $H_2$ , in a simple way: the approximate relation between CO and  $H_2$  is  $n_6/n_7 = 0.47$ .

The determination of the fluid properties is extremely important for the coherent prediction values. The species contained in the air and in the combustion products where modeled using the values of specific heat ( $c_p$ ) at constant pressure given of Janaf tables (Chase *et al.*, 1985). For ethanol, used the data of the cp to Barin's gaseous phase (Barin, 1993).

## 2.2 Closed Phase

The blow-by is a known phenomenon that occurs in engines, so there is no real closed phase. But blow-by can be neglected in engines in good conditions Cró (2014). The closed phase of the thermodynamic cycle contemplates the process of compression, combustion and expansion.

### One Zone Model

In models of single zone it is assumed an instantaneous mixing between the burned and unburnt gases, temperature, pressure and composition of the mixture of the constant gases (Ramos, 1986). The First Law of Thermodynamics for the closed phase is represented in its differential form by Eq. 2;

$$\frac{dU}{d\theta} = \frac{dQ}{d\theta} - \frac{dW}{d\theta} \quad (2)$$

The infinitesimal variation of sensible internal energy of the gases present inside the cylinder for the crank angle,  $dU$ , can be written as a function of the specific heat at constant volume, mass and temperature and is showed in Eq. 3 below.

$$\frac{dU}{d\theta} = mc_v \frac{dT}{d\theta} + mT \frac{dc_v}{d\theta} \quad (3)$$

Where  $m$  is the admitted mass by the cylinder (air + fuel + residues),  $T$  is the temperature of the gases in the cylinder in each crank angle ( $\theta$ ), and  $C_v$  is the specific heat of mixture at constant volume. Different gases coexist inside the cylinder during the combustion process, such as: reactants, products and residual gases. Its respective amounts of internal energy variations are pondered by fraction of burned fuel mass Eq. 4, obtained for the Wiebe's function.

$$c_{vmist} = c_{v_{reag}}(1 - X_b(\theta_2)) + c_{v_{prod}}X_b(\theta_2)(1 - FGR(\theta_1)) + c_{v_{prod}}FGR(\theta_2) \quad (4)$$

Where  $c_{v_{reag}}$  and  $c_{v_{prod}}$  are respectively the reactants and products specific heats at constant volume. Such values are obtained through the sum of the product between the molar fraction and the  $c_v$  of each element in the specific process.

Residual gases coming from previous cycles are present at the beginning of the closed phase inside the cylinder. Its fraction mass is represented by FGR and given by Eq. 5.

$$FGR = m_{res}/m_{total} \quad (5)$$

The proposed model, assumes that the residual gases have the same chemical composition that the combustion products. Such consideration is quite plausible since the amount of unburned fuel at the combustion process is around 1% (Gallo, 1990).

The term relating to the applied work or received work by the system at the crank angle interval is given below Eq. 6.

$$\frac{dW}{d\theta} = P \frac{dV}{d\theta} \quad (6)$$

The heat transfer process in SI is characterized as a tridimensional transient process according to Gallo (1988). In the global models of instantaneous heat transfer in the engine, is accepted quasi-permanent regime, on the supposition that the expression for the heat transmission rate between the operating fluid and the cylinder walls is valid at each instant.

The term that refers to the heat transfer to the system can be interpreted according to Eq. 7.

$$dQ = dQ_f + dQ_w \quad (7)$$

Where  $dQ_f$  is the rate of heat release by the combustion at a determined crank angle can be calculated by as Eq. 8, and  $dQ_w$  is the rate heat transfer between the gases and the cylinder walls Eq. 10.

$$\frac{dQ_f}{d\theta} = \eta_c m_f HC \frac{dX_b}{d\theta} \quad (8)$$

Where  $\eta_c$  is the combustion efficiency,  $m_f$  is the mass of fuel,  $HC$  is the combustion enthalpy and the  $dX_b$  is the fraction of burned mass in cylinder that obtain by Wiebe function. The efficiency of the combustion is calculated according to Eq. 9 and was obtained in function of the theoretical coefficient of air,  $\lambda$ , and the maximum combustion efficiency  $\eta_{cmax}$ . The combustion efficiency  $\eta_c$  is around 90% for conventional SI (Heywood, 1988).

$$\eta_c = \eta_{cmax} - 1.602 + 4.6509\lambda - 2.0746\lambda^2 \quad (9)$$

$$\frac{dQ_p}{d\theta} = \frac{h_p(\theta)A(\theta)(T(\theta) - T_p)}{w} \quad (10)$$

Where  $h_p(\theta)$  is the film coefficient for heat transfer, area  $A(\theta)$ ,  $T(\theta)$  is the gas temperature inside the cylinder,  $T_p$  is the cylinder wall temperature,  $w$  is angular speed of the engine. The wall temperature in a real engine varies some degrees during the cycles, but such variation is neglected in most of cases. There are many models to determine the heat transfer coefficient for an SI. In this work, it was chosen the model proposed by Hohenberg (1980) Eq. 11. A comparative study Gallo (1988) between empiric methods, semi-empiric methods and experimental ones about correlation of heat transfer, concluded that the model of Hohenberg (1980) obtains the best results at the closed phase. The heat transfer coefficient is calculated in function of the volume, pressure and temperature at the crank angle position and at the average velocity of the piston ( $v_p$ ).

$$h_p(\theta) = 130V(\theta)^{-0.006}P(\theta)^{0.8}T(\theta)^{-0.4}(v_p + 1.4)^{0.8} \quad (11)$$

By manipulating the Eq. 2 and Eq. 3 the equation to determine the temperature can be written (Eq. 12) and using an ideal gas equation determine the pressure (Eq. 13) in the Combustion chamber.

$$\frac{dT}{d\theta} = \frac{\delta Q_{conv} - PdV}{m(c_v dT + Tdc_v)} \quad (12)$$

$$\frac{dP}{d\theta} = \frac{P}{T}dT - \frac{1}{V}dV \quad (13)$$

Where  $V$  is the volume of the combustion chamber at each instant which is calculated by the instantaneous volume equation by the kinematics of the engine. The difference of the modeling of one zone for two zones in predictive models is that there are two temperatures: unburned (u) and burnt (b) ( $T_u$  and  $T_b$ ), two volumes ( $V_b$  and  $V_u$ ) and two masses with uniform pressure in the combustion chamber in a detonation free situation.

## Two Zones Model

The two-zone model in more robust models enables a determination of pollutants from chemical equilibrium and  $NO_X$  formation from chemical kinetics with formation mechanisms, for example, extended Zeldovich (Gutiérrez, 2011).

The two-zone model calculates the mass and energy of the unburnt and burnt gases as a function of time, that is in function of crank angle, assuming that infinitely fine flame propagates throughout the cylinder. The flame divides two regions with temperature and homogeneous composition: the regions of unburnt and burned gases. In addition, flue gases are considered in chemical equilibrium in robust model, but in this model the composition is frozen analogous the one zone (Ramos, 1986).

There are many models of two zone as: Heywood (1988), Ramos (1989), Melo *et al.* (2014), Bayraktar and Durgun (2004), Santos (2009), among others. Some make a preposition that has crevices and/or blow by, these two prepositions were disregarded in this work, both to have a very small enthalpy compared to the other plots. The model of this work was based on the Melo *et al.* (2014) and Santos (2009).

Equations. 3 to 11 are the same as those used for the two-zone model, but the First Law of Thermodynamics (Eq. 2) is developed for burned and unburnt zone, and with the aid of mass conservation equations Eq. 14 to Eq. 16 and the ideal gas equation Eq. 17 to 19 it was possible to arrive at Eq. 20 and Eq. 21, where represents the derived temperatures burned and unburned for each crank angle respectively.

$$\frac{dm}{d\theta} = \frac{dm_u}{d\theta} + \frac{dm_b}{d\theta} \quad (14)$$

$$\frac{dm_u}{d\theta} = dX_b m(\theta) \quad (15)$$

$$\frac{dm_b}{d\theta} = (1 - dX_b)m(\theta) \quad (16)$$

$$V = V_b + V_u \quad (17)$$

$$PV_u = m_u + R_u + T_u \quad (18)$$

$$PV_b = m_b + R_b + T_b \quad (19)$$

$$\frac{dT_b}{d\theta} = \frac{m_b(c_{p,prod}T_b + dC_{v,prod} + \frac{dh}{dT})}{\delta Q_{comb} + \delta Q_{b,conv}} + V_b dP - m_b \frac{dh}{dP} + dm_b(C_{p,mist}T_u - C_{p,prod}T_b) \quad (20)$$

$$\frac{dT_u}{d\theta} = \frac{m_u(C_{p,mis} + dC_{v,mist}T_u + \frac{dh}{dT})}{\delta Q_{b,conv} + (V_u - m_u \frac{dh}{dP})dP} \quad (21)$$

Applying Eq. 17 in differential form to each zone following the Ideal Gas Law and expanding the terms and isolating  $dP$  we have Eq. 22 which yields the derived pressure:

$$\frac{dP}{d\theta} = \frac{-P}{V}dV - \frac{V_u}{T_u}dT_u + \frac{V_b}{m_b}dm_b + \frac{V_b}{T_b}dT_b \quad (22)$$

Using Eq. 18 and 19 as relations of the ideal gases, substituting the temperature derived of each zone, a pressure can be calculated by Eq. 23.

$$\begin{aligned} \frac{dP}{d\theta} = & \left[ \frac{R_{prod}(V_b \frac{dh}{dP})}{C_{p,prod} + dC_{v,prod}T_b + dhdT \frac{dh}{dT}} + \frac{(Vu - m_u dh dP)R_u}{C_{p,mist} + dC_{v,mist}T_u + \frac{dh}{dT}} - V \right] (PdV - R_u T_u dm_u - R_b T_b dm_b + \\ & \left( \frac{R_u}{\frac{C_{p,mist} + dC_{v,mist}T_u + \frac{dh}{dT}}{\delta Q_{conv}}} \right) - \left( \frac{R_b}{C_{p,prod} + dC_{v,prod}T_b + \frac{dh}{dT}} \delta Q_{comb} \right) - \delta Q_{conv} + dm_b (C_{p,mist}T_u - C_{p,prod}T_b) )^{-1} \end{aligned} \quad (23)$$

In predictive models the composition is fixed, so the  $\frac{dh}{dT}$  and  $\frac{dh}{dP}$  it is zero. In robust models that use chemical equilibrium and / or kinetic chemical this value depends on the derivatives of the composition of each species at each crank angle.

The modeling of SI combustion phenomena is very complex, because it is a three-dimensional transient process, with turbulent and chaotic flow. An important parameter in the combustion modeling is the determination of the fraction of mass burned and the rate of this burnt mass (Alla, 2002).

With the two zones model in SI engines there are two more widespread ways to calculate a fraction of mass burned (FMB), use flame front equations or the Wiebe function (Tinaut *et al.*, 2006). The methodology of the present work assumes a burn velocity profile that has been shown flexible about its fitting with the evaluated engine. The finite duration combustion is modeled by the Wiebe function (Eq. 24).

$$X_b = 1 - \exp \left[ -\log(1 - \eta_{burn}) \left( \frac{\theta - \theta_i}{\Delta\theta_b} \right)^{m+1} \right] \quad (24)$$

Where:  $\Delta\theta_b$  indicates the combustion duration;  $\theta_i$  is the angle where the combustion starts;  $m$  is the curve form factor;  $\log(1 - \eta_{burn})$  is efficiency parameter, the same thing that  $a$ , many author utilizes as parameter  $a$ .

If the burn efficiency is 99,9% (Lanzafame, 1999), where the factor  $a$  equal 6.9. According to Heywood (1988), the values of  $m$  can vary from 1 to 3 and of  $a$  can vary from 3 to 10, but the usual values to the parameters  $a$  and  $m$  are 5 and 2 respectively.

### 2.3 Open Phase

In the open phase (intake and exhaust), the mass flow through valves can be obtained assuming a quasi-permanent process with the hypothesis of isentropic flow, the gases behave as ideal gases, the upstream (before the valve) velocity is neglected, but utilized the correction of the discharge coefficient (Gallo, 1990).

The intake and exhaust processes were modeled from the isentropic flow model that corrects the results using the load coefficient  $C_d$  proposed by Kastner and B (1947). Based in tests with different pressure relations between the gases present in the collector and inside the cylinders, and with different valve diameters as well. It was obtained a single curve to the discharge coefficient as a function of dimensionless coefficient  $YD_v(\theta)$ . The mass flow through the valves is generally subsonic flow (the flow velocity is below that the sound velocity) and sonic flow. Therefore, is necessary to verify such flow condition, analyzing the upstream pressure and the downstream pressure as shown in Eq. 25.

$$\frac{P_j}{P_m} < \left( \frac{2}{k+1} \right)^{k/k-1} \quad (25)$$

Where  $P_m$  means the upstream pressure and  $P_j$  downstream is pressure,  $k$  is the relation between the specific heats. The mass flow is obtained through Eq. 26. Such relation is valid for subsonic flows. When the flow reach sonic conditions, the downstream pressure do not influence the mass flow Eq. 27, because the flow is choked.

$$m = \frac{C_d A_j P_j}{\sqrt{RT_m}} \sqrt{2k \left( \frac{k}{k-1} \right) \left[ \left( \frac{P_j}{P_m} \right) - \left( \frac{P_j}{P_m} \right)^{k+1/k} \right]} \quad (26)$$

$$m = \frac{C_d A_j P_j}{\sqrt{RT_m}} \sqrt{2k \left( \frac{k}{k-1} \right)^{(k+1)/k}} \quad (27)$$

It is necessary to test the mass flow to guarantee the flow direction of the mixture, not only in the intake valve, but in exhaust valve as well. Using the downstream pressure and upstream pressure conditions outside the cylinder because the flow tends to go from the higher pressure to the lower. Black-flow phenomena in the exhaust and in some cases in the intake too, that, can result in a reduction of volumetric efficiency.

At intake and exhaust is also applied the First Law, where the kinetic energy and the gravitational potential were neglected Eq. 28.

$$dU = \delta Q - \delta W + (dm_i h_i) - (dm_s h_s) \quad (28)$$

Where  $dm_{e/s}$  is the in/out infinitesimal mass in the cylinder,  $h_{e/s}$  is the enthalpy associated to mass.

Just as in the closed phase, in the open phase is also there are the heat transfer of the mixture with the walls. The most suitable correlations to the heat transfer coefficient for the open phase of the engine cycle where proposed Nishiwaki (1979), intake Eq. 29 and exhaust Eq. 30.

$$h = 84.3D^{-0.193}(v_p P)^{0.807}T^{-0.534} \quad (29)$$

$$h = 679D^{-0.422}(v_p P)^{0.578}T^{-0.199} \quad (30)$$

Where  $D$  is the cylinder diameter,  $P$  is the pressure,  $T$  is the temperature and  $v_p$  is the mean velocity of the piston. The Figure 1 presents the logical the model.

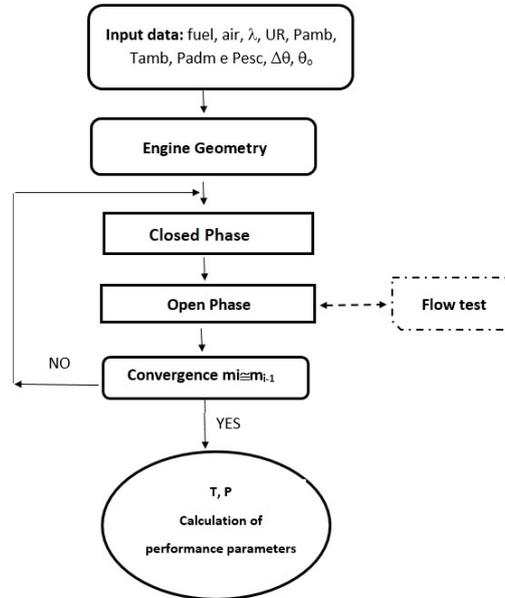


Figure 1. The block diagram of the computational simulation model.

### 3. RESULTS, DISCUSSION AND CONCLUSIONS

The modeling of the thermodynamic properties of the involved species is extremely important for the prediction of the pressure, temperature inside the cylinder and the performance parameters engine.

All values obtained for the involved substances properties are very next near the value of literature. Such values present a slight deviation, the error average value was around 0.0003% and the higher error was around 0.006%.

The thermodynamic models require experimental data to adjust parameters used in the combustion models, heat transfer, pressure, temperature of intake for example, in order to validate the developed model.

In this article a comparative study were performed utilizing the same engine data used by Gallo (1990), for comparison. compared the peaks of pressure and temperature for base case. Other comparison is with the thermal and volumetric efficiency in two speed (3000 and 5000 rpm).

The data for the base case tested and compared with the literature are: bore (80mm), connecting rod (128 mm), stroke (79.5 mm), 5200 rpm, duration combustion 60°, timing spark 340 DPMS, TC=12,  $\lambda = 0.94$ , UR=80%,  $T_{amb}=25^{\circ}C$ ,  $P_{amb} = 101325Pa$ , temperature of intake  $T_{adm} = 288.15K$ , pressure of intake ( $P_{adm} = 860kPa$ ), pressure of exhaust ( $P_{ext} = 1150kPa$ ), anhydrous ethanol.

The Fig. 2 shows a typical curve of an engine pressure vs. crank angle and the Fig. 3 shows the curve pressure vs. volume, and the Fig. 4 curve log P vs. log V, that's is better view the open phase.

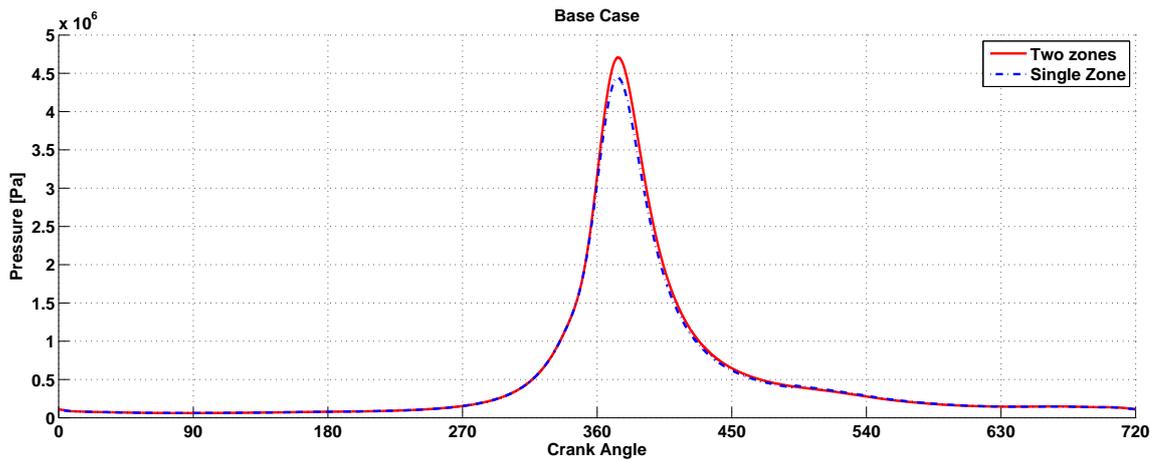


Figure 2. Curve P vs crank angle for base case comparing one zone e two zones.

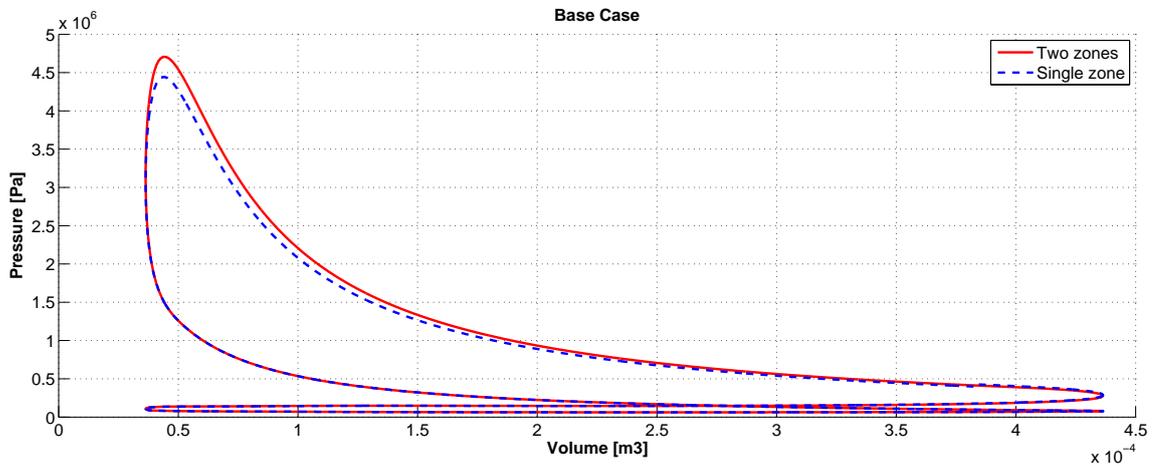


Figure 3. Curve P vs. V for base case comparing one zone e two zones.

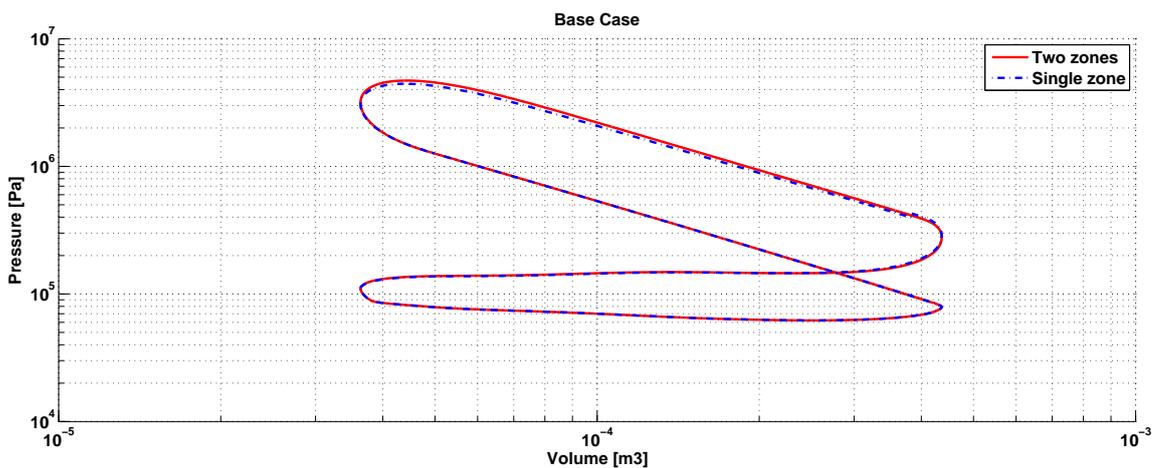


Figure 4. Curve logP vs. logV for base case comparing one zone e two zones.

The peak pressure of literature, Gallo (1990), is 50 bar for the base case, to two zones model is 47.06 bar, a 5.88% deviation comparing with literature, and a one zone this value is 44.39 bar a 11.22% deviation. Another important factor to be compared is the temperature profile in each model, which can be observed in the Fig. 5. In the literature the peak of temperature is 2300K, for the two zones model this value is 2370K one deviation of 3.0% and comparing with one zone model the value is 2187, the deviation is 4.9%. Other parameters for compared are the flow mass intake and exhaust thought the valves Figs. 7 and 8. All the cases show that the division the chamber combustion there aren't influence.

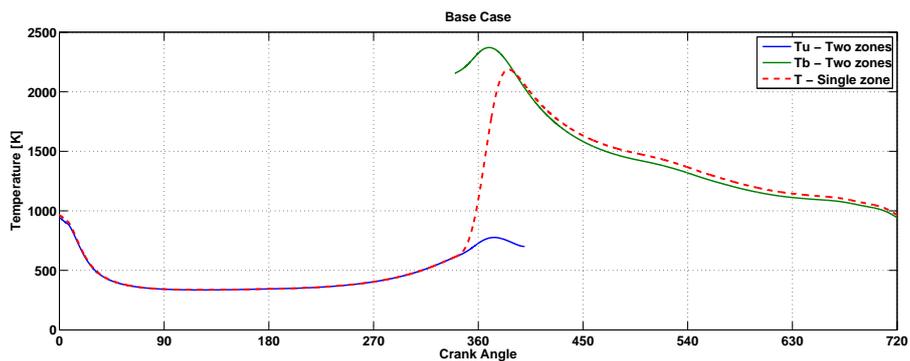


Figure 5. Curve temperature vs. crank angle for base case comparing the two models.

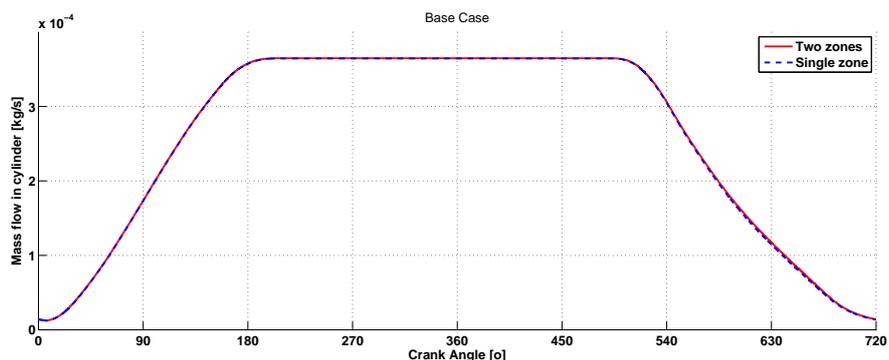


Figure 6. Curve the flow mass thought the exhaust valve vs. crank angle for base case comparing one zone e two zones.

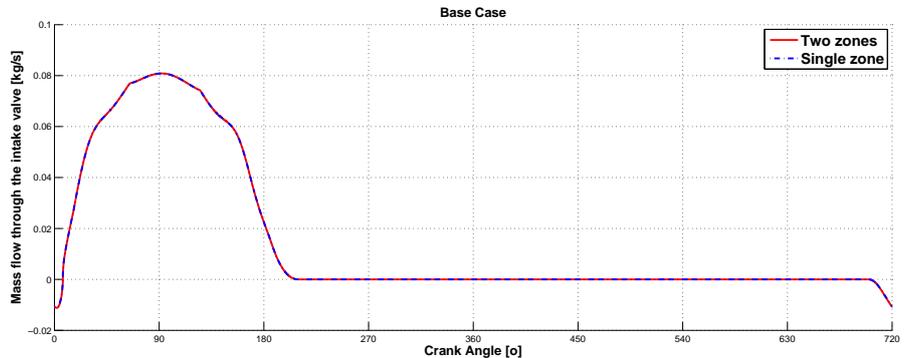


Figure 7. Curve the flow mass thought the intake valve vs. crank for base case comparing the two models.

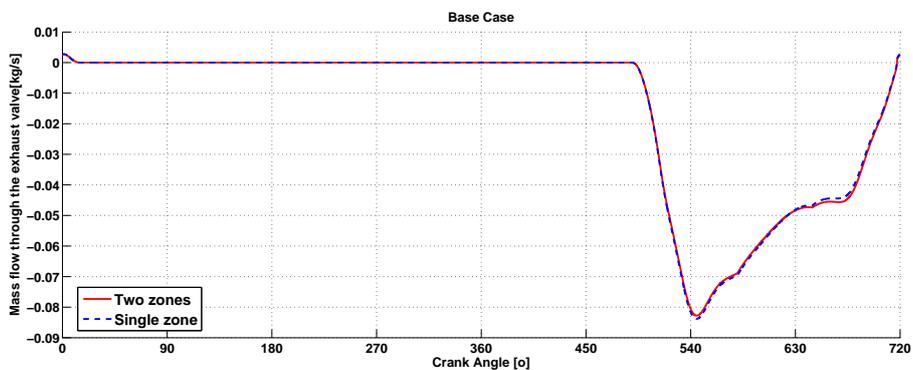


Figure 8. Curve the flow mass thought the exhaust valve vs. crank for base case comparing one zone e two zones.

Two parameters of performance were also compared the thermal and volumetric efficiency, the wanted are presented in the Tab. 1.

Table 1. Comparison of the results of the predictive model with those of the literature, for E100

Parameters	Speed [RPM]	Values of Literature (Gallo, 1990)	Author results	Deviations [%]
$\eta_{ti}$ [%] one zone	3000	41	35.81	10.47
$\eta_{ti}$ [%] two zones			37.92	5.20
$\eta_v$ [%] one zone		74	81.50	10.13
$\eta_v$ [%] two zones			81.24	9.78
$\eta_{ti}$ [%] one zone	5000	40	34.32	16.29
$\eta_{ti}$ [%] two zones			36.49	11.00
$\eta_v$ [%] one zone		70	75.93	8.47
$\eta_v$ [%] two zones			75.98	8.54

Although the deviations found between the developed models and the literature model, the two-zone model presents better results, the parameter that presented a greater deviation was the volumetric efficiency, this can be due to the modeling of the valve control. These models are predictive and show the influence and tendency of the results, to improve the models will adjust it with experimental data.

After the comparison between literature and two zones model, realized other analysis with the model, is study the influence: of the speed (1500, 3000 and 5000 rpm), of the relation air fuel ( $\lambda$  varying of 0.8; 1.0; 1.2) and of the load in performance parameters of engine. The condition are: with hydrous ethanol, with 5% of water (E95h), relative humidity of 60%, and combustion duration is 40°, the other parameters are the same the base case. The results are in Tab. 2.

Table 2. Results of the influence the speed, load and relation AF in the performance parameters for E95h and RH=60%.

$\lambda = 1, 0$ , full load				
Speed [rpm]	Power [kW]	$\eta_t$ [%]	$\eta_v$ [%]	sfc [g/kWh]
1500	4.56	38.15	81.67	362.93
3500	10.83	38.46	81.46	360.04
5000	14.07	37.31	76	371.09
3500 rpm, full load				
$\lambda$	Power [kW]	$\eta_t$ [%]	$\eta_v$ [%]	sfc [g/kWh]
0.8	11.34	32.55	81.67	425.38
1	10.83	33.46	81.14	360.04
1.2	9.41	39.78	81.14	348.03
1500rpm, $\lambda = 1$				
Load [%]	Power [kW]	$\eta_{t0}$ [%]	$\eta_v$ [%]	sfc [g/kWh]
100	4.56	38.15	81.67	362.93
38.9	1.77	32.26	40.51	429.22

The peaks of pressure and temperature also influenced by these parameters, chose the temperature a case for to show this result, because this parameter show the higher difference between a single zone and two zones, Fig. 9. The results show the same influence that other studies.

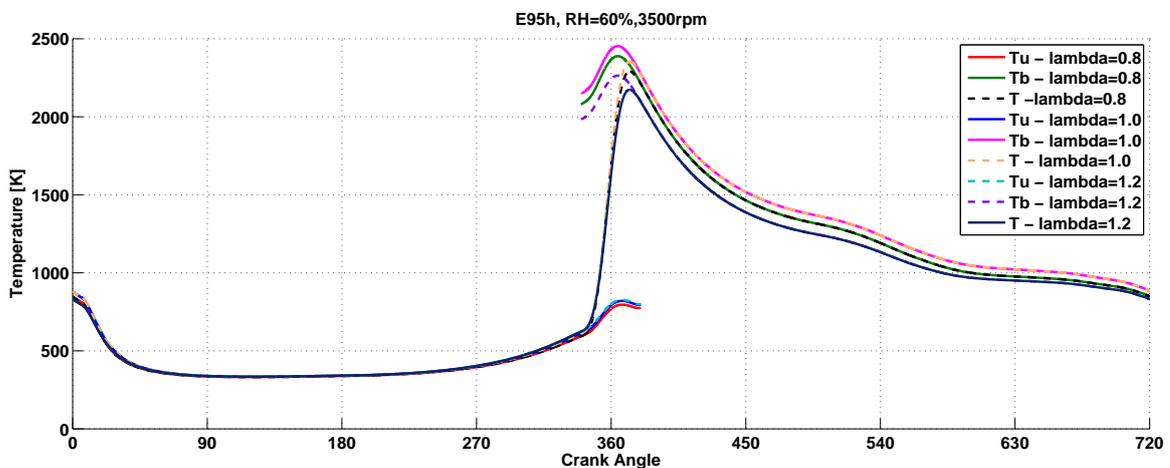


Figure 9. Temperature vs. Crank angle for E95h, full load and RH=60% varying relation AF.

The results show that the higher speed higher the thermal efficiency and the consumption of fuel, the influence the

higher relation air fuel higher is power and thermal efficiency but lower consumption fuel and the load influence all parameter proportionally.

Of this results concluded that the predictive model of two zones is better than a zone, for the prediction of performance parameters and to obtain the typical motor curves presented. The model will be refined with experimental data in order to make it more accurate.

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