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Review and Synthesis of Literature on Single and Multizone Thermodynamic Combustion in a Diesel Engine

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Authors' contributions

This work was carried out in collaboration among all authors. All authors read and approved the final manuscript.

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Review Article

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ABSTRACT

This work is largely devoted to a review of existing works in the literature on single and multi-zone models of thermodynamic combustion in a diesel engine. It is found that numerical simulations of diesel engine operation based on thermodynamic models are of great interest in predicting engine performance and developing new concepts. Also, the advantages and disadvantages of the different models of a multi-zone are given. This study summarises several studies over several years of these thermodynamic models in use by presenting the different results. The limitations of the single-zone model led to the development of the multi-zone thermodynamic combustion model. This work therefore suggests a comprehensive study that takes into account the different models (kinetics, fuel flow, injection, heat transfer, etc.) to better appreciate the performance of the diesel engine. This will give a clear idea of how to

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develop new concepts for single-zone or multi-zone thermodynamic modelling that will help car manufacturers, for example.

Keywords: Modelling; combustion; multi-zone thermodynamic; engine.

1. INTRODUCTION

Simulation of engine operation has become an essential part of internal combustion engine design due to the increase in design parameters and existing alternative fuels. Engine cycles are governed by very complex processes. Indeed, they involve many areas of physics and chemistry such as fluid dynamics, heat transfer, thermodynamics and chemical kinetics to evaluate the combustion, emissions and operating characteristics of an engine. Numerical simulations of diesel engine operation based on thermodynamic models (single-zone, dual-zone or multi-zone) are of great interest for many reasons: they allow performance prediction and provide engineers with more data than experiments to develop new concepts [1-3]. However, numerical simulation of the operation of an internal combustion engine fuelled by biodiesel or diesel remains mixed in the scientific literature. Although the combustion of fuels (diesel or biodiesel) is a simple concept, the influence of parameters such as injection phases and thermodynamic conditions on this type of combustion deserves to be investigated in order to identify the physical locks for its optimisation. The modelling and consideration of the interaction phenomena between the droplets, the reactive zones and the turbulent flow lead to new difficulties and very few modelling studies exist to date. However, we can mention the recent attempts of Borghi [4-6] and Durand et al [7,8]. This work is devoted to a literature review on the thermodynamic single and multi-zone combustion of diesel and biodiesel fuels in a diesel engine combustor and a brief synthesis of the literature in this area. We analyse the influence of the parameters and the thermophysical and physicochemical properties of the fuels used on the engine performance. We recall that the objective is to make a literature review and to highlight the different thermodynamic numerical models used during the combustion of fuels in a diesel engine combustion chamber [9-13]. Finally, the novelty of this study is to summarise the work done on thermodynamic modelling over several years. It is structured in several parts, including a brief description of the diesel engine and the problems associated with its use, the modelling and finally the synthesis.

2. PROBLEMS RELATED TO THE USE IN DIESEL ENGINES

Homogeneous charge compression ignition (HCCI) engines have received increasing attention in recent years due to their advantages in reducing emission levels of greenhouse gases, such as HC, SO2, CO2, NOx [9, 14, 15, 12, 13]. Some regulatory bodies, particularly in developed countries such as Europe, the United States and Japan, have attempted to reduce the emission levels of conventional engines [13, 12].

The technique of HCCI engines involves compressing a homogeneous mixture of air and fuel until auto-ignition occurs towards the end of the compression stroke, followed by a significantly faster combustion process than compression ignition (CI) or spark ignition (SA) combustion [13, 16, 1]. The quality of the air/fuel mixture in HCCI engines is generally lean, auto-ignition occurs in several places and the combustion is then volumetric without noticeable flame propagation [17, 12, 18, 19]. When the homogeneous mixture has reached its chemical activation energy and is fully controlled by chemical kinetics corresponds to the production of combustion [13], rather than by a spark or an injection moment. The chemical kinetics fully control the mixture which makes cold engine starting difficult. For auto-ignition models, the questions that arise mainly concern:

- The kinetic reactions and their number, of which the major reactions involved in the auto-ignition process are not very well known
- The turbulent mixing and the local characteristics of the turbulent aerodynamic field which will favour or not the contribution of premixed gases likely to self-ignite,
- The transition to turbulent combustion,
- The search for and consideration of the extinction-ignition effects that occur in the premixed zones must be elucidated,
- The consideration of the presence of droplets and their evaporation in the flame and the transition to the combustion phase are to be developed,
- etc.

In summary, it can be said that the use of fuels in diesel engines faces numerous technical, economic and environmental difficulties. From a technical point of view, the main problem encountered when using fuels in engines is the formation of deposits and the fouling of certain engine parts. Indeed, the formation of carbon deposits and the strong dispersion of the liquid jets are the problems frequently encountered. This can lead to significant mechanical problems [20, 21, 9]. Other technical problems are often observed, such as clogged filters, feed pump problems, rapid clogging of the furnace, and rapid deterioration of certain engine parts.

3. DESCRIPTION OF INJECTION

In Diesel engines, the combustion process is highly dependent on the injection of fuel into the cylinder. Indeed, the distribution of the fuel in the combustion chamber is not uniform, which results in heterogeneous combustion with some areas very poor in fuel, and others very rich in fuel. It is therefore essential to homogenise the distribution of the injected fuel in order to reduce the differences in richness in the cylinder. In this case, fuel injection is an essential step for combustion optimisation.

According to some studies,[22, 23, 24, 1] the pressure in the combustion chamber can reach 5-10 MPa and the temperature around 1000K at the time of injection. However, these values are a result of the engine compression ratio and boost pressure, for internal combustion engines. As a result, the density of the gaseous medium can be about 15-20 times that of the value in atmospheric conditions [25]. In the case of direct injection engines, the fuel, injected directly into the combustion chamber, is sprayed in the form of fine droplets at different velocities as described in figure 1. These droplets are then fragmented due to the large difference in velocity between these droplets and the surrounding gas [26]. These droplets, which are then subjected to turbulent flow in the chamber, disperse, may collide and coalesce with each other. Some of these drops may impact the engine walls and form a local liquid film. Finally, during all these processes, the high temperature of the gases present in the combustion chamber leads to the evaporation of the droplets, which is necessary for the gaseous fuel-fuel mixture in which combustion will take place. This is a critical aspect, as a more or less good description of these phenomena will help to determine the initial conditions necessary for the calculation of the combustion. It is therefore necessary to have a turbulence model and a two-phase model of equal quality that can correctly represent the steps described above. This is necessary to better understand the turbulent flow properties as well as the concentrations of various components of the gas mixture at the time and location of spark or self-ignition (Fig.1).

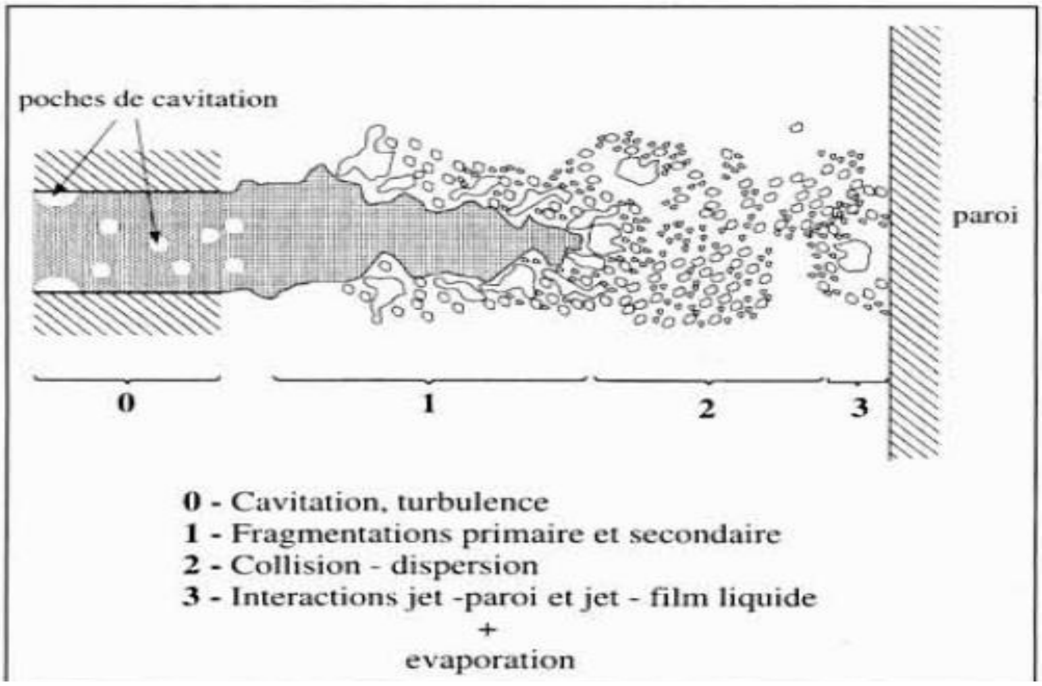

Fig. 1. Schematic description of injection in a diesel engine [26]

4. IGNITION DELAY

The ignition delay is defined as the period between the start of injection and the start of combustion. It represents the first stage of the combustion process in a diesel engine. This period can be expressed in crankshaft degrees or in units of time (milliseconds) [25] and shows that in Diesel engines, the order of magnitude for the ignition delay is about ten crankshaft degrees, which, depending on the operating speed, is 0.2-2 ms. In addition, physical and chemical processes take place in the cylinder before combustion is initiated. Therefore the ignition delay is a very important step for the combustion process [2, 3, 27, 28, 29] as it influences the operation of the engine including pollutant emissions and energy performance.

5. HEAT TRANSFER

In an engine, part of the heat released by combustion is transferred by convection-radiation to the cylinder walls. The heat is then transferred by conduction through the engine block and is normally transferred to the coolant. Some of this heat is removed to the atmosphere by radiative and convective transfers between the engine block and the ambient air. Conductive transfers within the fluid or within the engine block and radiative transfers are less important than convective transfers between the gases and the engine block. The heat flow from the gases by convection to the walls of the combustion chamber depends on the temperature gradient at this zone (interface).

It is also important to note that heat transfer through the engine walls is an important phenomenon because of the high temperature that the combustion chamber can reach. This transformation of the chemical energy of the fuel is essential in order to obtain mechanical work on the piston. In the similar cases found for most heat transfers, the modes of heat transfer, mainly convection and gas radiation, between the gas mixture in the combustion chamber and the cylinder wall in the presence of the flame take place according to the different phases of the cycle. Thus:

During diesel engine cycles, the amount of heat transferred through the cylinder walls represents between 25 and 30% of the amount of heat delivered by the fuel[13, 30, 31;.

During the intake phase: the gas velocity is high in the combustion chamber (fresh air intake). The gas mass in the cylinder and the exchange surface is variable. The pressure, temperature and composition of the mixture remain uniform. The heat exchange between the cylinder walls and the fresh air is by convection.

During the compression phase: The gas velocity in the combustion chamber is reduced. The exchange surface is variable as well as the temperature and pressure of the gas. Heat exchange is by convection.

During the combustion and expansion phase: Heat exchange is by convection. This transfer is enhanced by the gas flow velocities and by the radiation from the gas and the flame.

During the exhaust phase: A significant variation in pressure, temperature, chamber wall surface during the exhaust of gases.

Thermodynamic Modelling of the Diesel Combustion Chamber

6. DESCRIPTION

The modelling of the combustion chamber is based on the application of the first principle of thermodynamics of open systems. The system exchanges energy with the outside world, in the form of work and heat, but also mass through valve flows and leakage. The level of complexity chosen for the present model consists of a two-zone thermodynamic model. The choice was made to work on fresh gas (air + fuel vapour) and burnt gas (combustion products). The use of such an approach in the modelling of diesel combustion is explained by the wish to know the evolution of quantities such as the temperature of the burnt gases in view of a phenomenological modelling of the polluting emissions. This approach is mainly used in the case of modelling species such as NOx or thermal CO [13, 19, 12, 27]. For this type of pollutant, the so-called phenomenological models are derived from complex kinetic schemes. The impact of a variable such as temperature is therefore essential for a good estimation of emissions. It should be noted that the introduction of a temperature gradient in the flue gas zone can be considered for a better prediction of the pollutants. The detailed mathematical formulation of the two-zone thermodynamic model, applicable for both analytical and predictive models, consists of a system of 7 algebraic and differential equations. For each zone there is an energy conservation equation and an equation of state, but also an equation for the conservation of mass and total volume.

7. THERMODYNAMIC MODELS

Thermodynamic modelling, also known as 0D modelling, is a combination of several models that are generally designed from empirical or semi-empirical approaches. Its complexity varies according to the model taken into consideration. For example, the use of a simple mathematical model (polynomial models, neural models, etc.) is less complex than a phenomenological model that uses physical phenomena. Thermodynamic models aim to analyse, control and evaluate the performance of complex systems; with a simulation time close to reality, which is not easy to achieve with other types of modelling, hence the great interest of researchers in these 0D models [13,19,12,27,22,23]. In order to proceed with this type of modelling, it is important to first decompose the macro-system under consideration into several sub-systems. The aim of this decomposition is to represent each element of the system as accurately as possible. The sub-systems are then brought together to give the model a body. To complete this model, it is subjected to the identification, calibration and parameterisation phase.

7.1 Single Zone Model

In order to develop a thermodynamic model that predicts temperature as well as cylinder pressure sub-models have been considered. The one-zone model is mainly based on the first principle of thermodynamics applied to the combustion chamber. Thus, the one-zone or zero-dimensional models are among the simplest phenomenological models. They consider that the fuel/air mixture in the cylinder is homogeneous. They are based on the application of the first principle of thermodynamics, which at each engine cycle allows the temperature and pressure in the combustion chamber to be determined. In this model, the parietal heat flows are estimated by empirical correlations. Assuming that the different components, i.e. the fuel, the oxidant (air), the recirculated exhaust gases (EGR) and the burnt gases form a homogeneous mixture, the energy equation verifies the following expression [32]:

$$mC_V\frac{dT}{dt} = \dot{m}_e(h_e - C_VT) - \dot{m}_S(h_S - C_VT) - Q_{paroi} + Q_{comb} - P\frac{dT}{dt} \quad \text{(E-1)}$$

Where ($\dot{m}_e$)˙and($\dot{m}_S$)˙represent the mass flow rates of the fuel mixture at the inlet and outlet,h is the enthalpy of mass, C_V is the mass heat capacity at constant volume, *T* is the temperature and $Q_{comb} = m_{carb}PCI\frac{dX_b}{dt}$ with *PCI* the mass lower heating value of the fuel, X_b the fraction of fuel mixture burnt. Fig. 2 shows the heat balances inside the cylinder and Figure 3 summarises the combustion chamber model.

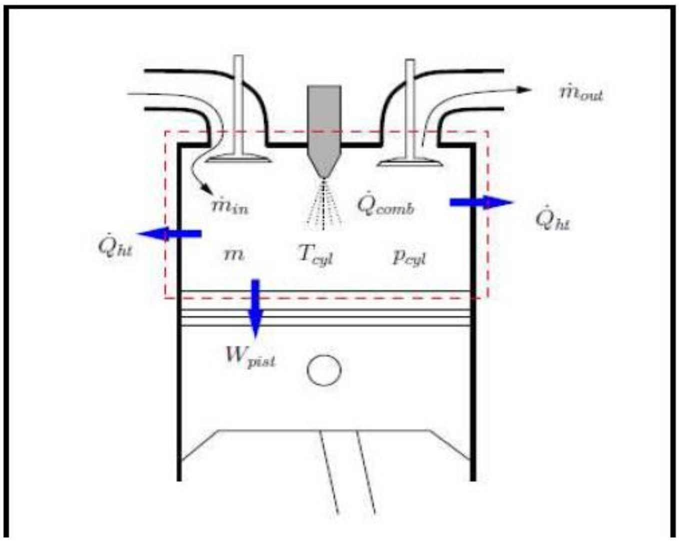

Fig. 2. Heat balance of the single combustion zone model [14]

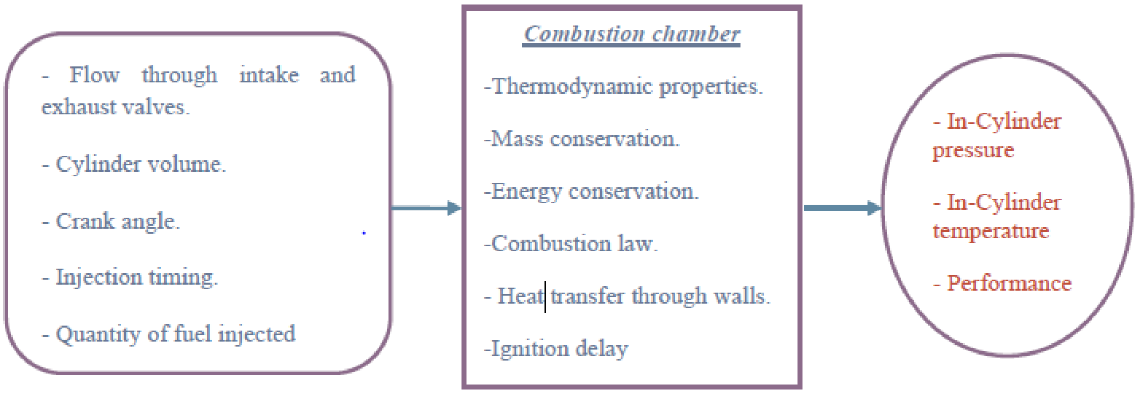

Fig. 3. Single zone model of the combustion chamber [23]

The basic equations considered for the thermodynamic analysis and the energy balance can be written as follows:

❖ Mass balance

$$\frac{dm}{dt} = \sum_i \dot{m}_{i,e} - \sum_i \dot{m}_{i,s} \qquad \text{(E-2)}$$

Where ($\dot{m}_{i,e}$) is the inlet mass flow rate and ($\dot{m}_{i,s}$) is the outlet mass flow rate:

❖ Energy balance - Internal energy [18]

$$\frac{d(m.u)}{dt} = \sum_i (h_e \dot{m}_e)_i - \sum_i (h_S \dot{m}_S)_i + \dot{Q} - \dot{W}_T - P\frac{dV}{dt} \qquad \text{(E-3)}$$

With

u : Internal energy of the system *V* : Volume of the cylinder
m : Mass of the fuel, *h* : enthalpy
$\dot{Q}$: Heat flow, *t* : time
P : pressure

❖ Energy balance – Enthalpy[18]

$$\frac{d(m.h)}{dt} = \sum_i (h_e \dot{m}_e)_i - \sum_i (h_S \dot{m}_S)_i + \dot{Q} - \dot{W}_T - V\frac{dP}{dt} \qquad \text{(E-4)}$$

u : Internal energy of the system
m : Mass of the fuel
Q : Amount of heat
P : pressure
V : Volume of the cylinder
h : enthalpy
t : time

The disadvantages of the single-zone thermodynamic model are related to the fact that this model does not provide data on the gas emissions of the fuels used, nor does it model the temperature gradient existing in the flame and in the near wall, etc.. Hence the need to explore several areas in order to take into account these parameters which are also very important in the design of Diesel engines.

7.2 Multi-Zone Modeling

The shortcomings encountered with single zone models can be partially overcome by the use of multi-zone models. These assume that the combustor is divided into a sufficient number of regions or zones, each with uniform thermodynamic characteristics. The sufficiency of the number of zones is usually determined by a sensitivity analysis, i.e. the minimum number of zones used is defined as the number beyond which the simulation tends to converge. The temperature, composition, volume and all thermodynamic properties of each zone are considered to be uniform within the zone boundaries, but may be different from the thermodynamic properties of the other zones [33-35,13]. This allows the effects of temperature or species stratification in the combustion chamber to be studied, and generally leads to a better estimation of the combustion duration, pressure rise rate and combustion efficiency. Mass and heat transfer in the combustion chamber play a key role in estimating the above operating characteristics and, since the combustion chamber is considered a set of thermodynamically distinct zones, the mass and heat transfer processes between the zones must be taken into account. Figure 4 shows the first law of thermodynamics applied to a multiple zone, which in the general case includes heat and mass transfer. With regard to mass transfer, each zone can be thermodynamically defined as a closed or open thermodynamic system, depending on whether the mass flow between the zone is more or less important. Similarly, heat transfer between zones and to the combustion chamber wall can also be allowed or neglected [1, 2, 33, 11]. However, assumptions about mass and heat transfer can be factors influencing the simulation results significantly; heat transfer affects the temperature stratification inside the combustor, the cooler regions of which are sources of unburned HC and CO. For some authors, the formation of these pollutants is also affected by mass transfer between zones, since any HC or CO moving from a colder to a warmer zone is expected to be (partially) oxidised [31, 36, 37]. The expected rate of combustion, rate of pressure rise, duration of combustion, and peak pressure are also affected by the following factors and the maximum pressure are also affected by the choice of sub-models used to describe heat and mass transfer. Multi-zone models can be divided into stochastic multi-zone models, in which certain properties of the mixture are considered as random variables, and phenomenological multi-zone models, in which probability considerations are excluded (Fig.4 and Fig.5).

$$dV = dV_u + \sum_{i=1}^{n} dV_{b,i} \qquad \text{(E-5)}$$

The mass conservation law, applied to the cylinder content, yields:

$$d(m_f + m_a + m_r) = dm_u + dm_{b,n} = 0 \qquad \text{(E-6)}$$

where the subscripts *f, a, r, u* and *b* refer to the fuel, air, residual gas, and the unburned and burned zones, respectively. Furthermore, the energy conservation equation for the unburned zone and for each burned zone *i* can be written as:

$$-q_u A_u dt + V_u dp = m_u di_u \qquad \text{(E-7)}$$

$$-q_{b,i} A_{b,i} dt + V_{b,i} dp = m_{b,i} di_{b,i} \qquad \text{(E-8)}$$

The terms qb,iAb,i and qu Au express the moduli of the global heat transfer from the considered zone (*u* or *i*) to the adjacent zones and to the combustion chamber walls, while *iu* and *ib,i* are the specific enthalpy of the corresponding zones. Moreover, cylinder pressure p is given by relation:

$$pV = m_u R_u T_u + \sum_{i=1}^{n} m_{b,i} R_{b,i} T_{b,i} \qquad \text{(E-9)}$$

8. SUMMARY OF THERMODYNAMIC COMBUSTION IN A DIESEL ENGINE

Modelling and simulation of the operation of an internal combustion engine has long been an effective tool for assessing the parameters that contribute to its optimal operation and sizing. Various solutions have been used with varying degrees of success. In the past, the technical choices were mainly guided by economic reasons and the desire to achieve the simplest possible combustion using unsophisticated injection methods. The need for high-performance solutions has evolved over the years, necessitating an orientation towards more advanced technical solutions obtained by using, for example, zero-dimensional modelling for combustion. This method aims to express all physical and chemical phenomena in a limited number of equations [30, 38]. Empirical laws have been chosen for combustion modelling in order to reduce the modelling time. The calculations are most often performed in the MATLAB environment, which is capable of predicting the evolution of variables useful for engine control in crankshaft degrees.

The bibliographical synthesis developed in this section highlights the results of experiments and modelling work relating to work carried out on the thermodynamic processes of an internal combustion engine.

C. Barba et al. 2000 [39] modelled the combustion of the main injection by considering a single diffusion combustion phase. For this, the heat release is modelled on a (partial) Weibe phase. This model is suitable for full loads.

Rakopoulos CD, Giakoumis EG, 2005 [40] in their work where the second law analysis is performed for each process of the diesel engine covering its transient operation. Thus, they developed a software package to study the performance of a multi-cylinder, turbocharged, direct injection diesel engine operating under transient load changes from the second law point of view. This model incorporates some important features regarding combustion, heat transfer, mechanical friction and fuel pump operation, while being able to treat each cylinder individually during transient events. Exhaust manifold irreversibilities, including various irreversibilities, are strongly affected, especially when a low heat rejection cylinder wall is chosen. They find that the exhaust manifold irreversibilities are of significant importance apart from those of combustion, especially at high load. Furthermore, they show that the low heat rejection engine leads to a significant decrease in in-cylinder irreversibilities (dominants) which results in an increase in the work recovery potential due to the increase in heat losses. On the other hand, the availability of exhaust gases, although partially recovered during turbocharging, still represents a non-negligible amount to be taken into account for work recovery. The above results best highlight the differentiation between the perspective of the first and second laws of thermodynamics. From the above therefore, the conviction is reinforced that a combined optimisation based on the two laws of thermodynamics is necessary for the correct development and performance improvement of internal combustion engines.

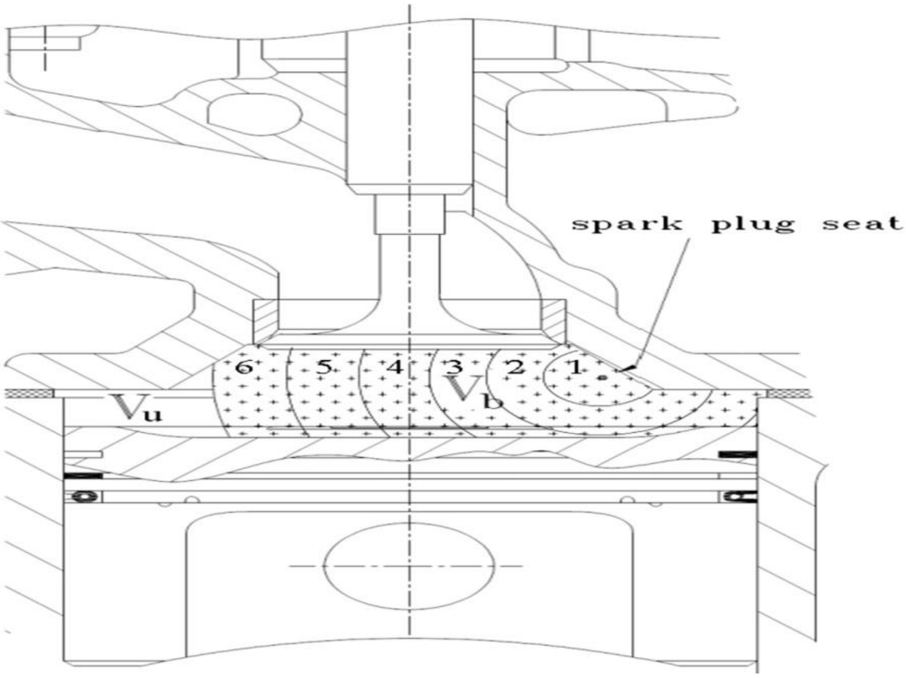

Fig. 4. Burned (*Vb*) and unburned (*Vu*) region volumes in the engine combustion chamber [36]

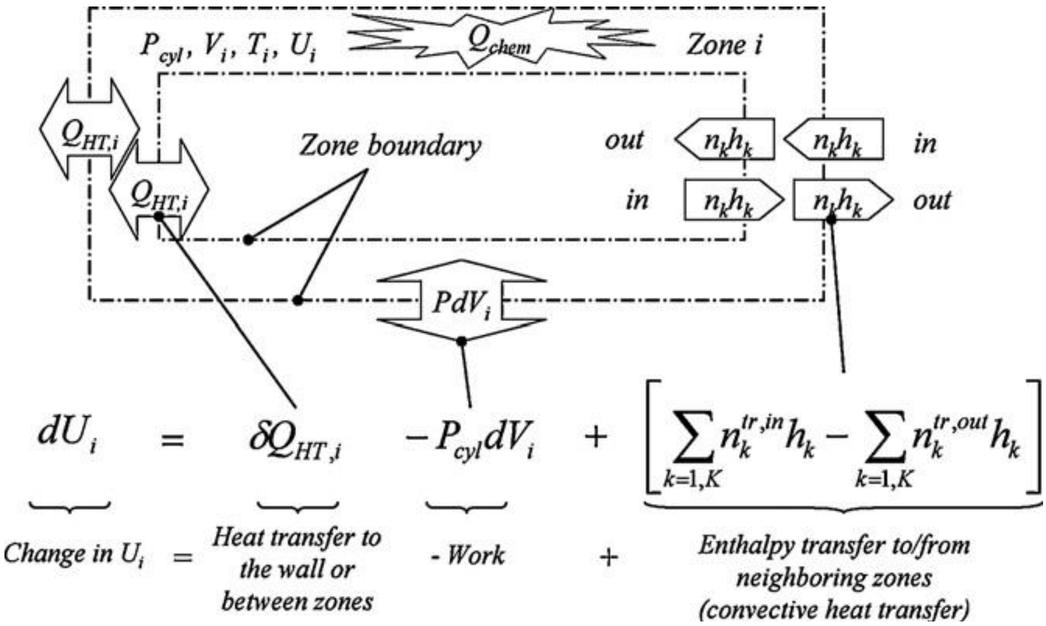

Fig. 5. The First law of thermodynamics applied to a zone of a multi-zone model [28]

H. Yasar et al, [41] model the operation of an HCCI engine by a single zone approach using the double Weibe function and then carry out an experimental study. The simulation results are in good qualitative and quantitative agreement with the experimental results. Thus, it is shown that the fraction of mass burned occurs relatively faster.

D. Descieux and M. Feidt, [42] evaluated the thermodynamic performance of a standard air diesel engine with heat transfer and friction term losses using a single zone thermodynamic model of a compression engine. They used a generic methodology to simulate the static response of an internal combustion engine and show the influence of several engine parameters on power and efficiency. In addition, they demonstrate the existence of two optimal engine regimes: one for maximum power and one for maximum efficiency.

N. P. Komninos and C. D. Rakopoulos [35], investigated the formation of CO and unburned oxygenated and unoxygenated hydrocarbon emissions from unburned, oxygenated and unoxygenated pure ethanol and pure isooctane HCCI engines. This is achieved using a multi-zone model, which describes the essential features of HCCI combustion i.e. heat and mass transfer in the combustion chamber, both of which are modelled using phenomenological sub-models. These mechanisms influence the formation of the main pollutants in the HCCI engine, namely unburnt and exhaust gases. Combustion is simulated using chemical kinetics coupled with oxidation mechanisms for isooctane and ethanol. These mechanisms also describe the decomposition of the original fuel into intermediate hydrocarbons and carbon monoxide. A validation of the model for both fuels is given for different load cases.

S. Awad et al, 2013 [3] modelled the combustion in a single zone diesel engine of a biodiesel from animal fat waste. The objective is to predict the cylinder pressure for a better understanding of the characteristic combustion of the different fuels (diesel and biodiesel) tested in the diesel engine and also the engine performance. The combustion of biodiesel was modelled with a triple Wiebe's law. The numerical results are in agreement with the experimental results.

S. Sivalakshmi and T. Balusamy, [43] evaluated the effect of using diethyl ether as an additive to biodiesel on the combustion, engine performance and emission characteristics of a diesel engine. The results indicate that the maximum cylinder pressure and heat release rate are higher for BD5 (5% by volume.) and the potential emissions were higher for BD5 than biodiesel.

A. M. Ashraful et al, [44] showed that analysis of the combustion chamber pressure, temperature and heat release rate of the modified blends highlights interesting features of the combustion mechanism, which are indicative of the performance and emission characteristics of the flue gas. This experiment reveals the potential improvement of palm and Jatropha biodiesel blends with the addition of three promising additives.

H. Karaky et al. [45] used a new semi-physical zero-dimensional NOx emission model for a spark ignition diesel engine using a simulated combustion process. The proposed semi-physical and zero-dimensional NOx prediction model is mainly based on the combustion model coupled with a thermodynamic calculation of the temperature (adiabatic flame temperature) in the burnt gas products of a stoichiometric mixture.

N. Bordet, [13] analysed diesel combustion in internal combustion engines. The objective of his work was to increase efficiency while limiting the associated computation time in order to use simulation as a dedicated tuning tool.

The development of a zero-dimensional model to guide the simulation of the operation of an internal combustion engine and to present the consideration of all the physico-chemical phenomena taking place in the combustion chamber gives the model a significant level of productivity. A new premix combustion model is proposed, allowing detailed modelling of highly diluted fuels and combustions related to early injections. An innovative approach to quantifying the interactions between multi-injection jets is proposed. It is also proposed that after calibration on a limited number of engine tests, the results of the global model are compared with measurements over the entire engine operating range.

A. Rida et al, [24] carried out a modelling and simulation of the thermodynamic cycle of a diesel engine using the neural network method. The single zone combustion model is selected to predict temperature and pressure. The numerical and experimental results are in agreement and show that as the amount of fuel entering the cylinder decreases, the maximum values of temperature, pressure t and power output decrease.

M. Yildiz et al, [46] conducted a comparative study of two models using a single and double Wiebe function on a zero-zone SI engine fuelled with methane and methane-hydrogen mixture. The results show that the double Wiebe function model fits better than the single Wiebe function model.

The thermodynamic analyses are based on reliable models providing robust design criteria. Thermodynamic analysis can be used to estimate gas pressure, temperature, engine power and thermal efficiency. Thermodynamic analysis also allows the optimisation of engine components, weight or volume.

The results also show that the model with the double Wiebe function fits better than the one with the single Wiebe function. In addition, the double Wiebe function leads to a significant improvement in the prediction of the GIMEP software for the SI fuelled by a methane-hydrogen mixture rather than the methane engine model.

B. Menacer and M. Bouchetara, [47] conducted a numerical simulation of the operation of a turbocharged direct injection diesel engine to determine its performance. The results were compared with those obtained using the commercial software GT-Power and showed an improvement in indicated mean effective pressure, mean effective pressure, power, torque and specific fuel consumption.

G Sekhar et al, [48] developed a single zone thermodynamic engine model based on a formulation that the burnt and unburnt gases are homogeneously mixed and the combustion rate is constant. Subsequently, they incorporated heat transfer and a two-zone combustion model that separates the burnt and unburnt gas zone during combustion. The results on engine performance and cylinder pressures and temperatures were almost similar.

M. Baratta et al, 2018 [36] conducted a multi-zone thermodynamic study to determine the cylinder temperature during combustion and to evaluate gas emissions. For the gas analysis, they use two models, Zeldovich model and detailed chemistry model. The model results, based on detailed chemical simulation, were found to be in good agreement with the experimental data for all engine exhaust species, and a better predictive capability than that obtained by simplified reaction and chemical equilibrium methods was demonstrated. The unburned gas fraction, derived from the calculated oxygen concentration and measured at the engine exhaust, was found to be a means of correcting for hydrocarbon emissions. The relative air-to-fuel ratio was changed from 0.8 to 1.53 and the nominal mean effective brake pressure was changed from 0.2 to 1.5 MPa to 1.29 MPa.

E. Neshat et al, 2019 [11] present a new thermodynamic model to simulate the mass transfer phenomenon in diesel engines. The simulation in the diesel engine was done using a multi-zone thermodynamic model coupled with a semi-detailed chemical kinetics mechanism. The heat and mass transfer sub-models are linked to the multi-zone model. Bulk flow and diffusion mass transfer between zones are considered in the mass transfer sub-model. The bulk mass transfer simulates the movement of the fluid between the different zones caused by the piston velocity and the temperature and pressure difference between the zones. Diffusion mass transfer occurs only with the difference in composition of the different zones. The results of their study showed that diesel engine performance and exhaust emissions are accurately predicted using mass transfer submodels. They also point out that the bulk mass transfer mechanism has more significant effects on engine performance and emissions than the diffusion mechanism. They find that by neglecting the mass transfer submodels, the calculated peak cylinder pressure is underestimated and the NOx and soot emissions are overestimated. Thus, the rates of soot oxidation reactions decrease and the rates of significant NOx reactions increase when mass transfer is used.

T. DABILGOU, 2021 [18] showed in his thesis work that the use of biodiesel fuels improves the combustion quality of these biodiesel fuels compared to conventional gasoil with higher temperature and pressure peaks (150 bar for jatropha and 147 bar for gasoil). He emphasised that these higher peaks of biodiesel compared to gasoil are related to its physico-chemical properties, notably its high cetane number (52; 49; 52; 59.5; 52) for neem, rapeseed, cotton and jatropha biodiesels respectively, compared to 42

Table 1. Comparison of zero-dimensional models [1]

Model	Advantages	Disadvantages
1 zone	Air, fuel, recirculated exhaust gas (EGR) and flue gas form a homogeneous mixture in the combustion chamber.	It does not model the temperature gradient in the flame and near wall, nor the aerodynamics of combustion.
MULTIPLE ZONES OF BURNT GASES	Calculations of nitrogen oxide concentrations are improved	-The gradient in the thermal boundary layer is not modelled -Modelling of near-wall combustion attenuation remains a major challenge

for gasoil), the high oxygen content in biodiesel, the speed of combustion and the short auto-ignition time of biodiesel.

P. S. Gautam et al, [10] studied a single zone thermodynamic model to predict combustion characteristics such as cylinder pressure, rate of pressure rise (ROPR), ignition delay and combustion duration and performance characteristics such as brake power (BP), brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE), using fundamental methods. This thermodynamic model uses Viebe correlation, thermodynamic and transfer equations to predict combustion characteristics. Statistical analysis of the predicted data using the regression method and the Z-test showed strong evidence of significant data by this model. The coefficient of determination (R2) for the performance parameters (BSFC and BTE) was greater than 98%, and the Pearson correlation (r) was also greater than 0.5%. The Pearson correlation (r) was also greater than 0.9 for both blends, while the maximum relative error was 7.58% for the MnP15 blend in the BSFC output at 75% load and 6.07% in the BP output in the case of diesel fuel, implying an accurate simulation model for a diesel engine. Table 1 below shows the advantages and disadvantages of some of the models used.

Modelling of near-wall combustion attenuation remains a major challenge although there is an increasing number of publications on engine performance and emissions using biodiesel, especially in the last decades, only fewer people have analysed and reviewed them. The latest publications have been cited to clarify the effect of biodiesel on engine performance without considering the single zone case or their physicochemical or thermophysical properties.

9. CONCLUSION

This study provided a review of the literature on thermodynamic single-zone and multi-zone combustion of fuels in diesel internal combustion engines. Several authors have worked on these different models for several years with mixed results. They have shown that the thermodynamic model has advantages related to its homogeneity of the different gases in the combustion chamber with an assumed uniform temperature and pressure throughout the zone but does not predict the exhaust emissions. However, the multi-zone model takes into account this aspect of emissions but the mitigation of near wall combustion remains a major issue.

The difficulties encountered by the different authors in this literature review for the modelling of combustion in diesel engines are globally

This work therefore suggests that a comprehensive study taking into account kinetic models and intake models, fuel flow models, injection models and globally extend to a quasi-dimensional model of diesel combustion to better appreciate the performance of the diesel engine.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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