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Modeling of Combustion and Carbon Oxides Formation in Direct Injection Diesel Engine

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ABSTRACT

When looking at the effects of diesel engine exhaust on the environment, it is important to first look at the composition of the exhaust gases. Over 99.5% of the exhaust gases are a combination of nitrogen, oxygen, carbon dioxide, and water. With the exception of carbon dioxide, which contributes about 5% of the total volume, the diesel engine exhaust consists of elements which are part of the natural atmosphere and are not harmful to the environment. Carbon dioxide emissions are directly related to the efficiency of the combustion unit. The higher efficiency obtained with lower amount of CO_2 emissions. In this study we are interested in the effects of exhaust gas recirculation (EGR) on combustion and emissions direct injection diesel engine. In particular, the effects of carbon dioxide (CO_2), water (H_2O), carbon monoxide (CO), some different quantities of EGR, analysed and quantified numerically. Other parameters tha affect the rate of oxides of carbon in the bowl shape, the Mexican hat and spherical geometry are analyzed in this work. Therefore, A modified version of the computational fluid dynamics (CFD) Code KIVA-3V has been used for modelling combustion process and engine emission, in particular carbon oxides emission and its control. Simulation was carried out by using a two-stroke single-cylinder direct injection diesel engine.

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1. INTRODUCTION

The combustion process used to convert the energy of different fuels to propulsion for current highway vehicles creates pollutants in large enough quantities to be detrimental to the environment and dangerous to human health, particularly in densely populated areas. The species pollutants emitted include carbon monoxide, oxides of nitrogen (NOx), particulate matter (PM), volatile organic compounds (VOCs), and sulfur dioxide. These pollutants are blamed for forming ozone and smog, causing acid rain, and contributing to a number of health issues; especially respiratory problems.

These Vehicles emit 1,460 trillion grams of CO_2 equivalent greenhouse gases per year. Increases in the number of vehicles and the miles travelled per year are responsible for this increase, as well as a general decline in average light-duty vehicle fuel economy. Emissions of pollutants and greenhouse gases from combustion processes depend in large part on the conditions used in combustion and on fuel properties mentioned by Nagle et al. [1].

The transport sector is one of the major contributors to global warming and environmental pollution. Exhaust emission control has become an important factor in the transport sector and industry in recent years. The importance of this dimension will increase further in the years ahead.

2. NOMENCLATURE

In a theoretically perfect combustion, carbon dioxide, water and nitrogen are the products. In reality, the incomplete combustion of diesel fuel results in nitrogen oxides (NOx), carbon monoxide (CO), carbon dioxide (CO_2), water (H_2O) and unburned hydrocarbons (HC).

Therefore, most of the remaining part of the paper will concentrate on the effect of exhaust gas recirculation (EGR) on rate of oxides carbon exhausts, the final part of the paper will look at how carbon oxides emissions are reduced with varying of piston bowl geometry. Both primary and secondary

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parameters, influence to reduce the carbon oxides emissions to meet the current and future diesel engine emission regulations. A modified version of the computational fluid dynamics (CFD) Code KIVA-3V has been used for modeling combustion process and carbon oxides exhausts.

ATDC	[-]	After top dead center
Ai	[-]	The preexponential factor
CO	[-]	Carbone monoxide
CO_2	[-]	Carbone dioxide
EGR	[-]	Exhaust gas recirculation
m_f	[g]	Injected fuel mass
RPM	[-]	Revolution per minute

Special characters

ε	[cm^2/s^3]	Turbulent kinetic energy dissipation rate
k	[cm^2/s^2]	kinetic energy turbulente
ω_{premix}	[-]	Premixed rate
$\omega_{diffusi}$	[-]	Diffusive rate
μ	[-]	Dynamic viscosity
ρ	[kg/m^3]	Density

3. MODELING FRAMEWORK

3. 1. Methods The modeling framework for our studies is based on a version of the Los Alamos code KIVA-3V, Dukowicz, et al. [2], used to simulate gaseous injection, Abraham, et al. [3]. A new methodology was developed to couple detailed chemistry with KIVA-3V, Yakhot, et al. [4], developed at Los Alamos National Laboratories, are capable of modelling Transient, three dimensional, compressible, multiphase flows with chemical reactions by solving the mass, momentum, and energy equations. Modifications to the original code include improvements to the turbulence, heat transfer, spray, and ignition, combustion, and emission submodels.

Turbulence is modeled using the renormalized group theory, the k-ε model derived by Hessel et al. [5], and implemented into KIVA for use with engine type flows by Belardini et al. [6]. Convective heat transfer between the cylinder walls and gas is simulated by a temperature wall function model Rutland et al. [7] which is based on the one-dimensional energy conservation equation. It accounts for density variability common in engines and the increase of the turbulent Prandtl number in the boundary layer.

The fuel injection process is modeled by using the blob injection model: the liquid fuel is represented by big drops, having a radius equal to the effective nozzle radius mentioned by Han et al. [8]. A breakup model based on the Kelvin-Helmholtz (H-K) and Reyleigh-Taylor (R-H) instability criteria has been adopted to describe the drop break-up process after injection. The combustion chemistry is modeled with the laminar and turbulent characteristic time combustion model proposed by Ladommatos et al. [10].

For engine applications, the combustion model is activated when a threshold temperature of 1100 °K is reached or a determined amount of combustion products is generated by the ignition process.

3. 2. Spatial Discretization with the ALE Method The spatial discretization employed in KIVA-3v is based on the ALE method. This is a finite volume (FV) method, for arbitrary hexahedrons (or arbitrary four-sided polygons in two-dimensions). The equations f motion are formulated with a moving velocity U, which varies from 0 to u, yielding a continuous range of formulations from completely Eulerian (U = 0) to completely Lagrangian (U = u).

3. 3. The ALE Mesh and Mesh Parameters We will discuss the ALE mesh in the two-dimensional case; the extension to three dimensions follows easily. The two-dimensional ALE mesh is depicted in Figure 1 [11].

In the ALE mesh, a normal cell is defined as the polygon in Figure 1 with the four vertices (i; j); (i +1; j); (i; j+1); (i+1; j+1), the mesh points at which are denoted $x_{i;j}$; $x_{i+1;j}$; $x_{i;j+1}$; $x_{i+1;j+1}$.

3. 4. The Finite Volume Method The spatial discretization is the standard FV discretization, performed by converting volume integrals to surface integrals with the divergence theorem. Once this is done, the integrals are approximated with quadrature rules (i.e., function value at cell center times the cell volume to approximate the Volume integral). Surface integrals over a cell become sums of integrals over cell faces. In two-dimensions, we have of course surface and line integrals in place of volume and surface integrals, respectively

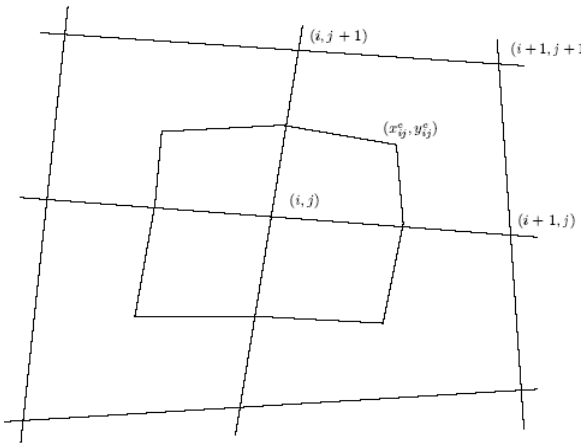

Figure 1. The two-dimensional ALE mesh

4. GOVERNING EQUATIONS AND FLOW MODELS

The equations of motion for the fluid can be solved for both laminar and turbulent flows. The mass, momentum and energy equations for the two forms differentiate primarily in the form of the transport coefficients (i.e. viscosity, thermal conductivity and species diffusivity), which are much larger in the turbulent formulation because of the additional transport created by turbulent fluctuations.

In the turbulent case, the transport coefficients are derived from a turbulent diffusivity that depends on the turbulent kinetic energy and its dissipation rate [12]

4. 1. Continuity Equation for Species m

$$\frac{\partial \rho_k}{\partial t} + div(\rho_k \vec{u} + \vec{j}_k) = M_k \dot{\omega}_k + \dot{\rho}_k^{\,s} \quad (k = 1, ..., N_s) \tag{1}$$

ρ_k^s: mass density of species m,
U : fluid velocity
j_k : diffusion coefficient (Fick's Law)

4. 2. Continuity Equation for Total Density

$$\frac{\partial \rho}{\partial t} + \text{div}\,(\rho \vec{u}) = \dot{\rho}^{\,s} \tag{2}$$

where: $\dot{\rho}^{\,s}$ = source term due to spray,
ρ = total mass density.

4. 3. Momentum Equation for the Fluid Mixture

$$\frac{\partial}{\partial t}(\rho \vec{u}) + \text{div}(\rho \vec{u} \otimes \vec{u}) + \text{div}(\bar{\bar{P}}) = \rho \vec{g} + \vec{f}^{\,s} \tag{3}$$

$\vec{f}^{\,s}$: (rate of momentum gain)/(volume) due to spray

g: gravitational acceleration, (cm/s^2)

4. 4. Energy Equation (Specific Internal Energy)

$$\frac{\partial}{\partial t}(\rho E) + \text{div}(\rho \vec{u} E + \vec{J}_q) + \bar{\bar{P}} : \text{grad}(\vec{u}) = \dot{q}^{s} + \dot{q}^{r} \tag{4}$$

where:

E= specific internal energy, exclusive of chemical energy

$Q^{\,c}$ = source term due to chemical heat release

$Q^{\,s}$ = source term due to spray interactions

$\vec{j}$ = heat flux vector and is the sum of contributions due to heat conduction and enthalpy diffusion.

4. 5. Turbulence Models

$$\frac{\partial}{\partial t}(\rho k) + \nabla.(\rho \vec{u} k) = -\frac{2}{3}\rho k \nabla.\vec{u} + \bar{\bar{\sigma}} : \nabla \vec{u} + -\rho\varepsilon + \dot{W} \quad \nabla.\left[\left(\frac{\mu}{Pr_k}\right)\nabla k\right] \tag{5}$$

$$\frac{\partial}{\partial t}(\rho\varepsilon) + \nabla.(\rho \vec{u}\,\varepsilon) = -(\frac{2}{3}C_{\varepsilon 1} - C_{\varepsilon 3})\rho\,\varepsilon \nabla.\vec{u} + \nabla.\left[\left(\frac{\mu}{Pr_\varepsilon}\right)\nabla\varepsilon\right] + \frac{\varepsilon}{k}\left[C_{\varepsilon 1}\bar{\bar{\sigma}} : \nabla\vec{u} - C_{\varepsilon 2}\rho\varepsilon + C_s \dot{W}^{\,s}\right. \tag{6}$$

where:

W^s = source term due to interaction with the spray
$C_{\varepsilon 1}$=1.44, $C_{\varepsilon 2}$=1.92, $C_{\varepsilon 3}$ = -1.0
Cs= empirical constant due to spray, = 1.5
Pr_k = turbulence Prandtl number for k (in general = 1.0)
pr_ε= turbulence Prandtl number, (in general =1.3)

5. APPLYING EGR

Exhaust gas recirculation (EGR) has been used in recent years to reduce the amounts of species emited in light-duty diesel engines and to a lesser extent in heavy-duty diesel engines. Actual EGR utilization in many types' engines is typically less than optimal because of high unburned HC and PM emissions.

Higher EGR rates mean more exhaust gas in the intake air, which usually lead to lower exhaust emissions. Large quantities of EGR must be cooled to keep the volumetric efficiency of the engine at normal levels.

Direct-injection diesel engines have the highest fuel efficiency and the lowest CO_2 emissions of any reciprocating internal combustion engine technology. Reduction of these emissions through clean in-cylinder combustion processes is imperative if vehicles powered by these engines are to be available at a competitive cost. These systems rely on high levels of EGR to increase the ignition delay (which allows greater premixing prior to the onset of combustion) and to reduce the combustion temperature. With high EGR levels, a larger quantity of the in-cylinder air/EGR mixture is needed to burn the fuel, and rapid mixing is required both during the ignition delay period and in the latter stages of combustion (to complete the fuel oxidation process).

Investigated low-temperature combustion regimes; established necessity of high exhaust gas Recirculation (EGR) levels for obtaining low emissions.

6. BOWL SHAPES

Different researches conducted to include that the geometry of piston bowl has an influence on the engine performance, in particular on rate of exhaust emission. Therefore, we like to analyse different geometries of piston. IN KIVA3 [11], the grid generator extended to create more complicated grid for internal combustion engines. In this version a block-structured mesh was adopted, and the program became suitable for modeling geometries containing inlet and outlet ports in the cylinder wall. This development was driven by an interest in crankcase-scavenged 2-stroke engines.

The piston assembly (piston, piston rings and liner) causes about 60% of total mechanical losses of a reciprocating engine. To decrease fuel consumption, it is necessary to lower this loss. In addition to that, the piston assembly is one of the main places of origin of the oil consumption, and with that, the hydrocarbon and particulate emissions resulting of lubrication oil. The precondition to increase the engine efficiency and to decrease the emissions is a better understanding of the tribological processes at the piston assembly. To characterise this, measurements of the friction force and oil film thickness between piston ring and liner are taken under firing conditions, resulted by Institut für Technische Verbrennung, Beal et al. [13].

The research of John Brevick [16] conducted to include that:

Many variable compression ratio piston designs have been patented and developed to varying degrees over the history of the internal combustion engine. Most designs control the compression ratio throughout the engine cycle and vary the compression ratio on demand (e.g., through controlling the oil volume in an upper versus lower chamber in the piston). A limitation on these designs is that the rate of compression ratio change may not be adequate at times when rapid load changes are demanded on the engine.

A unique feature technology is that the upper piston reacts to cylinder pressure during the power stroke of the engine; during the rest of the engine cycle, the upper piston remains in the high compression position [14].

Also, based on research of Peter van Blarigan, we can conclude that:

* The use of free pistons in internal combustion engines has been investigated for quite some time. In the 1950s, experiments were conducted with free piston engines in automotive applications.

* Several advantages have been noted for free piston IC engines. First, the compression ratio of the engine is variable; this is dependent mainly on the engine's operating conditions (e.g. fuel type, equivalence ratio, temperature, etc.). As a result, the desired compression ratio can be achieved through modification of the operating parameters, as opposed to changes in the Engines hardware [15].

* The combination of the combustion process and the free piston geometry is expected to Result in significant improvements in the engine, thermal efficiency and its exhaust emissions.

6. 1. Mesh Generation The code can generate mesh for a wide variety of piston shapes for CI engine combustion chambers. The mesh generator requires only a simple set of tabular information of the input data. Given the dimensions of the piston geometry, a half cross-sectional view of it is drawn on a graph paper. Based on this, the grid points were defined along the piston boundary, based on the level of resolution available, starting at the bowl axis and ending at cylinder wall.

The next figure shows the top view of a mesh generated for a diesel engine combustion chamber with Mexican hat piston bowl in the piston.

6. 2. Piston Shapes Analyses A different piston shapes used in this work, the two essential geometries are Mexican-Hat and spherical, as presented in Figures 2, 3, and 4.

7. CHEMICAL EQUATIONS

The injected fuel (C_7H_{16}) is oxidized at high temperatures according to Equation (7) [16]. The rate is given by Equation (8).

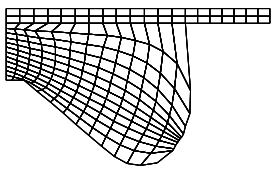

Figure 2. geometry No1 at TDC

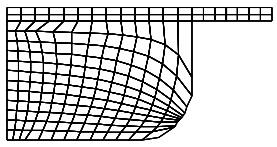

Figure 3. Geometry No 2 at TDC

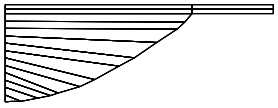

Figure 4. Geometry N°3 at TDC

$$C_nH_m + (n + \frac{m}{4})O_2 \Leftrightarrow nCO_2 + \frac{m}{2}H_2O \quad (7)$$

$$w_1 = \min(w_{premix}, w_{diffusi}) \quad (8)$$

where the premixed and diffusive rates are given by Equations (9, 10).

$$w_{premix} = A_i e^{E_i/T}[C_nH_m]^C[O_2]^d \quad (9)$$

$$w_{dffusi} = C_i \frac{\varepsilon}{k} \times \min\left([C_nH_m], \frac{[O_2]}{s}\right) \quad (10)$$

Here and in the following, the terms in squared parenthesis are molar concentration in mol/cm^3; k and ε are the turbulent kinetic energy and its dissipation rate; *s* is the stochiometric ratio.

Ai and *Ei* are respectively the preexponential factor and the activation temperature for the premixed combustion their values are reported later together with the other constants setting; *Ci* is the proportional factors for the diffusive combustion. This model, derived from Magnussen, is the most diffused in multidimensional diesel combustion computations. To model the other reaction processes going on during combustion the following equilibrium reactions were used, mentioned by Beal et al [13], Table 1.

8. DETAILS OF ENGINE SIMULATION

Details of the simulated engine configuration along with injection data, baseline operating conditions and modeling choices for this study are shown in Table 2 [17].

9. RESULTS AND DISCUSSION

The Figure 5 shows the evolution of the average temperature in the cylinder with the recirculation of the CO_2. Its value decreases with the increase of the CO_2 percentage The average temperature profiles obtained, essentially reflects the compression process, reaching its peak close to TDC. This peak of temperature is result of compression rate; its value is 1700 °K.

TABLE 1. Equilibrium Reactions

Equilibrium Eq. 1	$CO_2 \leftrightarrow CO + \frac{1}{2}O_2$
Equilibrium Eq. 2	$H_2O \leftrightarrow H_2 + \frac{1}{2}O_2$
Equilibrium Eq. 3	$H_2O \leftrightarrow \frac{1}{2}H_2 + OH$
Equilibrium Eq. 4	$\frac{1}{2}H_2 \leftrightarrow H$
Equilibrium Eq. 5	$\frac{1}{2}O_2 \leftrightarrow O$
Equilibrium Eq. 6	$\frac{1}{2}N_2 \leftrightarrow N$

TABLE 2. Engine Details (Input Values)

Geometric Specifications		
Bore	82.55	mm
Stroke	95.975	mm
TDC Clearance Height	1.8	mm
Connecting Rod Length	162.69	mm
Piston Bowl Shape	Mexican-Hat, spherical	
Injection Data		
Fuel Temperature	350	°K
Injection Duration	7°	CA
Injection Angle	25º	
Operating Conditions		
Speed	2600	rpm
Injection Timing	14°	ATDC
Intake Pressure	1.5	bar
Modeling Choices		
Intake Air Temperature	550	°K
Intake Pressure	1.5 bar / 2	bar
Turbulence Model	k-ε model	
Total mass of fuel injected m_f	0.01	g/cycle
Fuel	N-heptane	

This peak varies with the value of engine compression rate and initial conditions of simulation and the recirculation of CO_2 rate. If all data for exhaust and intake gas compositions were analysed, and a combustion model describing the EGR effect was made, it would be possible to estimate the decrease in combustion temperature.

The timing of the combustion process must be precisely controlled to provide low emissions with optimum fuel efficiency. This timing is determined by the fuel injection timing plus the short time period between the start of fuel injection and the autoignition, called the ignition delay. One of the most important properties of a diesel fuel is its readiness to autoignite at the temperatures and pressures present in the cylinder when the fuel is injected. Under ideal circumstances, all of the carbon in the diesel fuel will burn to carbon dioxide and all of the hydrogen will burn to water vapour. The effects of EGR on diesel combustion have been already studied by some researchers. It was found that ignition delay rose with EGR rate, however, an increase in the initial heat release rate was not observed.

The variation of different species masses in the combustion chamber is function of Crank angle. N-heptane is injected in a square wave injection pulse profile from -14 ATDC to -7 ATDC. After fuel injection ends, the mass of CO_2 in the cylinder is initially constant at a value of quantities recirculated, which is

the mass of CO_2 in the intake air due to the assumption of % EGR.

After combustion starts, CO_2 concentration starts increasing and continues to increase till the end of the calculation, as fuel is being continuously burned. It should be noted that CO_2 is locally assumed to be in equilibrium with CO. Due to this assumption, the rate of increase of CO_2 mass decreases slightly near TDC when the temperatures in the cylinder are locally at their highest values.

Figure 6 shows the CO mass fraction vs O_2 mass fraction; its value has a peak value obtained near TDC and varies with percentage of recirculated CO_2. The results include the maximum peak obtained with 10% for CO_2 EGR and a minimal value with 20% for CO_2.

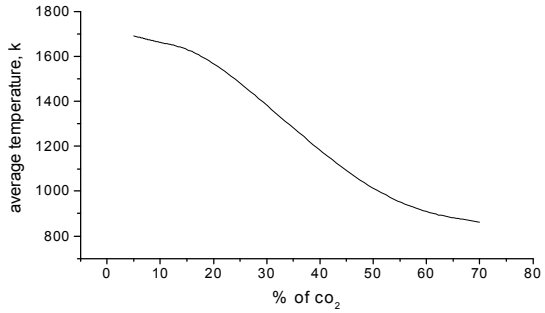

Figure 5. Average Temperature vs CO_2 percentage

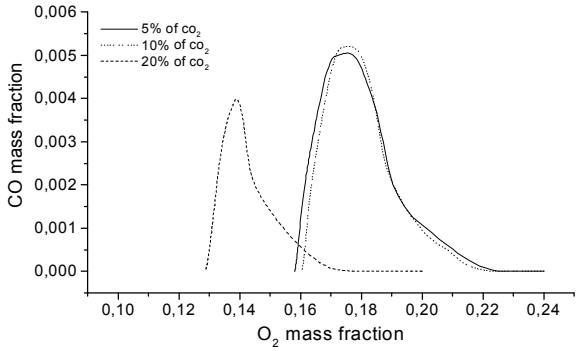

Figure 6. CO mass fraction vs O_2 with CO_2 percentage

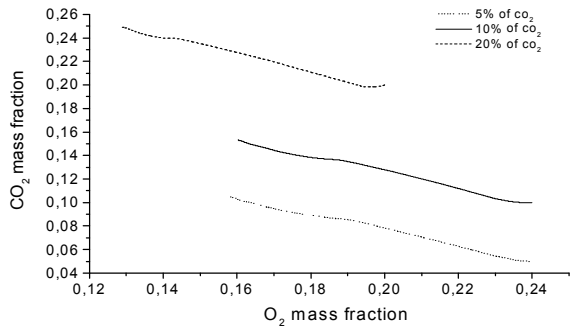

Figure 7. CO_2 mass fraction vs O_2 fraction CO_2 percentage

In Figure 7, only the amount of CO_2 decreases with increase of O_2 concentration. Also, this quantity increases with percentage of CO_2 recycled.

Other parameters analysed are the EGR of H_2O at 5%, 10%, and 20%. Figure 8 illustrates that the CO mass fraction decreases if the amount of H_2O recirculated is increased.

The increase of H_2O rate recirculated gives an increase of CO_2 formed, as illustrated in Figure 9.

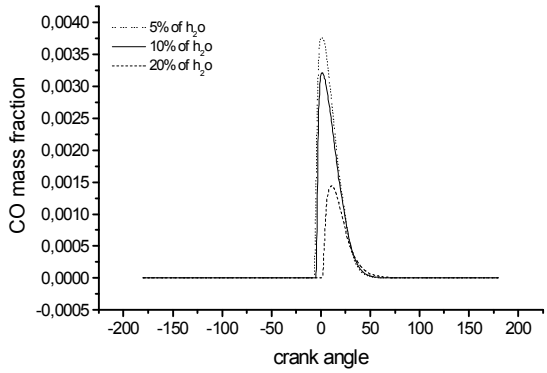

Figure 8. CO mass fraction vs crank angle with H_2O percentage

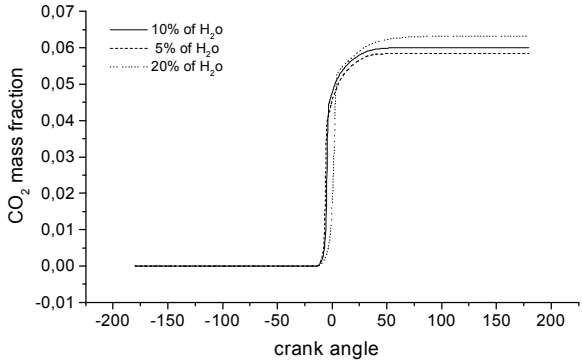

Figure 9. CO_2 mass fraction vs crank angle with H_2O percentage

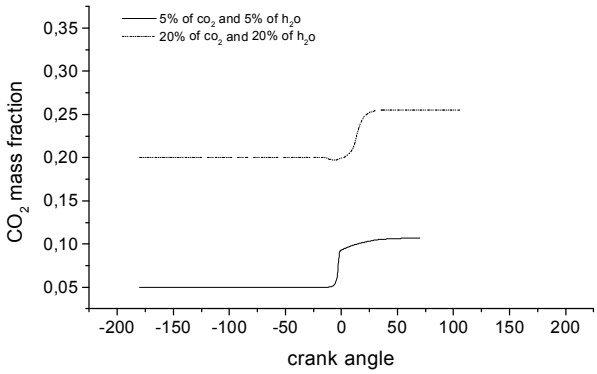

Figure 10. CO_2 mass fraction vs crank angle with H_2O,CO_2 percentage

But, this rate is differing with recirculation of other EGR species. By recirculating 5% of CO_2 and 5% of H_2O, the concentration of CO_2 formed is less than the quantities formed if analysed of 20% of CO_2 and 20% of H_2O.

Therefore, the essential compound giving a high rate of CO_2 is the compound containing atoms of carbon. This result is obtained in the Figure 10.

Figure 11 shows the CO mass fraction vs crank angle with H_2O and CO_2 percentage in the intake gas. CO rate decreases when CO_2, H_2O rate increases. Therefore, the simulation in even time with CO_2 and H_2O in the intake gas, it is logical to obtained low concentrations of carbon monoxide formed in ejected gas.

The other parameter affecting CO_2 formation is the geometry of the combustion chamber, as shown in Figure 12 where evolution of CO_2 mass fraction vs crank is plotted. The minimum value of CO_2 obtained when the geometry 2 is used and the maximum value obtained with geometry 3 (spherical).

The rate of CO formed has an inverse evolution with CO_2, as shown in Figure 13. The second geometry gives a maximum peak of CO (a minimum value of CO_2) and with geometry 3 the CO amount is the minimum (maximum CO_2). The same Principle has been demonstrated in previous section.

For this diesel engine model and for this conditions of simulation, (the k-ε model of turbulence is used), the Turbulence Kinetic Energy, TKE (cm^2/s^2), increase during the compression phase and take a maximum value at TDC, the Figure 14 shows the variation of Turbulence Dissipation Rate, EPS (cm^2/s^3),with different parameters. The spherical geometry (geometry 3) gives a high value of turbulence dissipation rate

The recirculation of CO_2 permits to reduced EPS rate successively and increase then to decrease with EGR of H_2O. That verifying with recirculation of H_2O and CO_2 in even time.

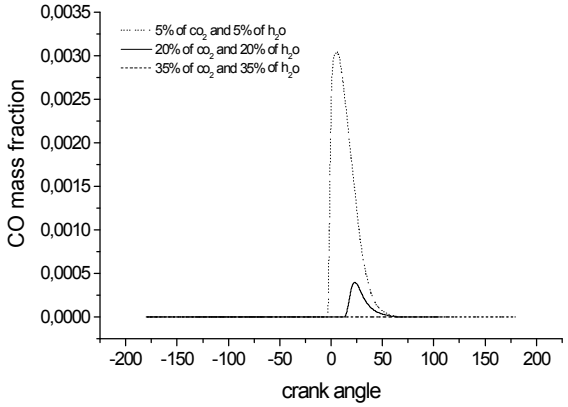

Figure 11. CO mass fraction vs crank angle with H_2O, CO_2 percentage

10. CONCLUSION

The effects of different piston geometries and different rate of constitutes of EGR on pollutants species were analysed numerically, and the following conclusions based on this work are as folows.

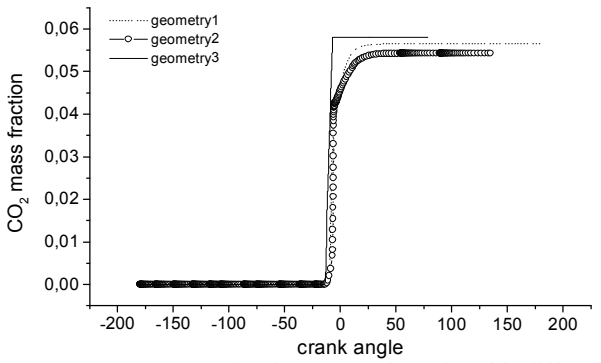

Figure 12. CO_2 mass fraction vs crank angle with different geometries

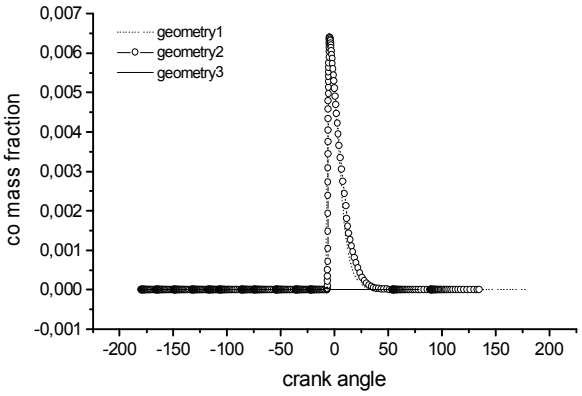

Figure 13. CO mass fraction vs crank angle with different geometries

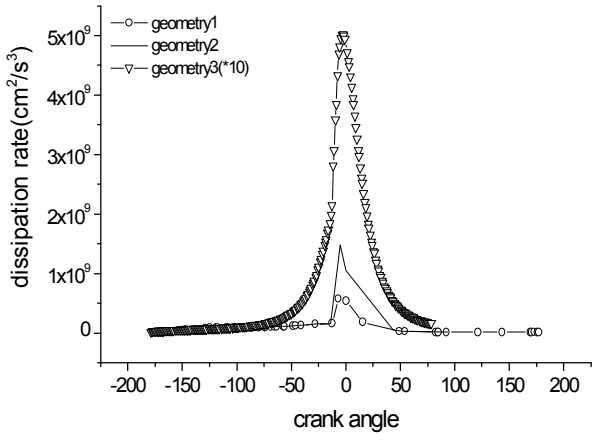

Figure 14. Dissipation rate vs crank angle with different geometries

First, CO_2 production is directly influenced with the recycled quantities of CO_2 or H_2O, The rate of carbon dioxide formed increased with EGR of CO_2 and H_2O successively.

Second, the analyses and simulation with CO_2 and H_2O even time occur more and more increase for CO_2 quantities. Therefore, the EGR method for reduced pollutants species increases the amount of formed carbon dioxide which is considered a necessary pollutant in diesel combustion.

Third, the other parameter which has an effect on the formation of CO_2 is the bowl geometry. The results obtained for three geometries iindicate that the maximum value is obtained with spherical geometry.

Fourth, to show the effect of recirculation of gas ejected by combustion on formation of carbon monoxide (CO), the results demonstrate that we can ameliorate the performance of engine diesel by minimising the amount of carbon monoxide what is verifying by using of CO_2 as principal gas of EGR or H_2O as principal specie of EGR or both the CO_2 and the H_2O principles constituent of EGR. Emission of CO is generally an indication of incomplete oxidation of fuel, and it is very dependent on the combustion temperature. Also, the spherical geometry gives a minimum amount of CO.

Finally, it can be concluded that the strategy of high EGR rates or bowl shapes are responsible for reducing CO emissions at the cost of a significant increase in fuel consumption and CO_2 emissions..

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Modeling of Combustion and Carbon Oxides Formation in Direct Injection Diesel Engine

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چکیده

برای بررسی تاثر گاز خروجی موتور دیزل بر روی محیط بسیار مهم است که در ابتدا ترکیبات گاز خروجی را بررسی کنیم. بیشتر از ۹۹,۵٪ از گازهای خروجی، ترکیبی از نیتروژن، اکسیژن، دی اکسید کربن و آب است. به غیر از دی اکسید کربن، که حدود ۵ ٪ از حجم کل را تشکیل می‌دهد، گاز خروجی موتور دیزل شامل عناصری است که قسمتی از جو زمین را تشکیل می‌دهند و برای محیط زیست هیچ اثر مضری ندارند. انتشار دی‌اکسید کربن مستقیماً به راندمان واحد احتراق وابسته می‌باشد. هر چه راندمان واحد احتراق بیشتر باشد مقدار دی‌اکسید کربن منتشر شده کمتر است. در این تحقیق، تاثیر بازچرخش گاز خروجی (EGR) بر احتراق و انتشار دی‌اکسید کربن را بر موتور دیزل با تزریق مستقیم بررسی کرده‌ایم. به ویژه اثر دی‌اکسید کربن، آب، منواکسید کربن و برخی از کمیت‌های مختلف (EGR) به صورت عددی مورد آنالیز و بررسی کیفی قرار گرفته است. پارامتر دیگری که بر سرعت دی‌اکسید کربن موثر است، شکل کاسه‌ای، کلاه مکزیکی و هندسه کروی است که در این کار مورد بررسی قرار گرفت. بنابراین یک نوع اصلاح شده از نرم افزار دینامیک محاسباتی سیالات (CFD) با کد KIVA-3V برای مدل سازی فرایند احتراف و خروجی موتور به‌خصوص انتشار دی‌اکسید کربن و کنترل آن استفاده شده است. شبیه سازی به وسیله یک موتور دیزل تزریق مستقیم دوزمانه تک سیلندر انجام شده است.

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