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NUMERICAL SIMULATION OF POROUS MEDIUM INTERNAL COMBUSTION ENGINE

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ABSTRACT

Porous media (PM) has interesting advantages compared with free flame combustion due to the higher burning rates, the increased power range, the extension of the lean flammability limits, and the low emissions of pollutants. Future internal combustion (IC) engines should have had minimum emissions level, under possible lowest fuel consumption permitted at all operational conditions. This may be achieved by realization of homogeneous combustion process in engine. In this paper, possibility of using PM in direct injection IC engine, with cylindrical geometry for PM to have homogeneous combustion, is examined. A three-dimensional numerical model for the regenerative engine is presented in this study based on a modified version of the KIVA-3V code that is very popular for engine simulation. Methane as a fuel is injected directly inside hot PM that is assumed mounted in cylinder head. Very lean mixture is formed and volumetric combustion occurs in PM. Mixture formation, pressure, temperature distribution in both phases of PM and in-cylinder fluid with the production of pollutants CO and NO, in the closed part of the cycle is studied.

INTRODUCTION

The most important issue of internal combustion engines that currently exists, is non-homogeneous mixture formation in the combustion chamber, which causes heterogeneous heat release and high temperature gradient in combustion chamber and thus production of pollutants such as NO_x, unburned hydrocarbons, carbon monoxide, soot and suspended particles. To avoid the temperature gradient in IC engines, homogeneous charge compression ignition (HCCI) engines, as an option, have been proposed. Control problems with start of ignition time under variable engine operational conditions and heat release rate, in this type of engines exist. In such engines, lean mixture with high amount of exhaust gas recycled (EGR), should be used that increases amount of carbon monoxide and soot particles, also with increasing the load of engine, NO_x formation and fuel consumption increases. Other important problem is that main oxidation of fuel should be occurred in the expansion process to reduce the maximum temperature and hence, low compression ratio should be used for these engines, while in reality, the compression ratio must be high enough that temperature at the end of compression process, lead to self-ignition of mixture with reasonable delay time. So, this type of engines is suitable for low and medium loads, thereby, it is better in high loads, they work in mode of compression ignition [1-3]. In direct injection engines, generally lack of homogeneous mixture formation and combustion, is unsolved problem yet. On the other hand, various methods are used for reducing pollutants in IC engines, such as high pressure fuel injection, variable valve timing, EGR, or adaptive and fuzzy control methods for setting the air fuel ratio and mixture formation but actually combination of these methods cannot solve air pollution problem under all conditions. Thus, this is a question, what is the method of homogeneous combustion in IC engines under all operational conditions (load and speed)? This target may be possible with volumetric combustion and prevent from formation of three-dimensional flame front that lead to temperature gradient in the combustion chamber. PM as an option is used to achieve homogeneous mixing and combustion in all conditions to reduce combustion temperature and formation of pollutions [3, 4]. Flame stability in PM with lean and rich mixtures, significant reduction in pollutants and increasing combustion efficiency, was proven [5, 6]. The first idea to use PM in the IC engines was proposed by Weclas and Durst in 2001. They investigated the performance of a PM in single-cylinder, air-cooled diesel engine, without using of catalyst. They mounted a Silicon Carbide (SIC) PM in the

cylinder head between the intake and exhaust valves and removed bowl of the piston. Their results showed increasing in engine efficiency, reducing the emissions and noise in comparison with the original engine. The mean cylinder temperature, about 2200 K in normal operational condition without PM, decreases to 1500 K with PM and also, it did not change significantly during combustion process [7, 8]. In 2002, Park and Kaviany simulated the effect of a SIC PM as a regenerative, which was moving near piston in the cylinder of diesel engine. They used thermodynamic two-zone model, single-step reaction for methane/air combustion, Lagrangian model for determination of motion droplets in fuel injection and showed that the maximum cylinder pressure during combustion increases and hence more work is done during a cycle, also engine efficiency increases, but due to high temperature of PM that its temperature is higher that adiabatic flame temperature of methane/air, the production of NO_x is higher than normal operation, while the amount of soot decreases [9]. In 2003, Macek and Plasek simulated IC engine with a PM for homogenization of mixture and combustion stability [10]. In 2006, Hwan, Ou and Chein mounted PM on the piston bowl of single-cylinder diesel engine and observed that power increases and noise decreases because of reduction in maximum value of pressure, also amount of NO_x decreases significantly while, unburned hydrocarbons and soot increase [11]. In 2007 Weclas and Faltermeier investigated the effect of liquid-fuel injection in a PM, as a matrix set of cylinders, with different diameters that were mounted on a flat plate and changed the arrangement of cylinders to find optimum geometry for PM that could produce the best mixture formation [12]. In 2008, MaoZhao and ZhiGuo studied homogeneous combustion in a PM compression ignition. They used KIVA-3V code for their simulation. Methane was injected inside hot PM and combustion occurred. They studied effect of initial temperature of PM and fuel injection time on the mean cylinder temperature and pressure. Their computational results showed that the initial temperature of PM is the key factor to start of combustion [13]. In 2008, MaoZhao and ZhiGuo simulated the injection of liquid fuel in axisymmetry PM engine with KIVA-3V code to investigate the effect of injection pressure and spray cone angle on hot PM in order to study the penetration of droplets in PM [15]. In 2008, Liu, Wu and Maozaho studied combustion and expansion processes. They used the two-zone model, effect of heat transfer from the cylinder wall, the mass exchange between zones and heat transfer from PM was considered to investigate the effect of inlet temperature and pressure and the compression ratio and excess air on the performance of engine and showed that the minimum value for compression ratio and initial temperature of PM on the start of combustion [16]. In 2009, Liu, Wu and Maozaho investigated working process of PM IC engine with ideal thermodynamic model and evaluated the influences of the expansion ratio, initial temperature on the net work that is done and efficiency of the engine[16].

NOMENCLATURE

BDC	bottom dead center
c_p	specific heat of mixture
c_s	specific heat of solid medium
$D_{\|\|}^{d}$	thermal dispersion coefficient
D_{im}	diffusion coefficient of species i in mixture
h_i	enthalpy of species i
h_v	volumetric heat transfer coefficient of PM
k_g	thermal conductivity of the fluid
k_s	thermal conductivity of solid
P	pressure
Pe	Peclet number
q_r	radiation heat transfer in solid
R	universal constant of gases
TDC	top dead center
T_g	gas temperature
T_s	solid temperature
u	velocity in x direction
v	velocity in y direction
w	velocity in z dorection
W_i	molecular weight of species i
$\overline{W}$	average molecular weight of mixture
X_i	molar fraction of species i
V_i	diffusion velocity of species i
Y_i	mass fraction of species i
ρ_g	mixture density
ε	porosity of porous medium
$\acute{\omega}_i$	rate of reaction i
ρ_s	solid density
σ	viscous stress tensor
μ	dynamic viscosity

Governing Equations

For simulation of PM engine, some assumptions were used for modeling of thePM, include:

(1) There is thermal non-equilibrium between gas and solid phases.

(2) Solid is homogeneous, isotropic, variable property with temperature and has no catalyst effect.

(3) Radiation heat transfer from the solid phase is considered only and the gas phase is transparent.

Regard to the above assumptions, governing equations are:

Continuity equation:

$$\frac{\partial}{\partial t}\left(\rho_g \varepsilon\right) + \nabla.\left(\rho \vec{V} \varepsilon\right) = 0 \qquad (1)$$

ρ_g is mixture density, ε is porosity of porous media and $\vec{V}$ is velocity vector.

Gas phase momentum equation:

$$\frac{\partial(\rho_g \vec{V})}{\partial t} + \nabla.\left(\rho \vec{V}\vec{V}\right) - \nabla P + \nabla \cdot \mu\left[\left(\nabla \vec{V}\right) + \left(\nabla \vec{V}\right)^T\right] - \left(\frac{\Delta P}{\Delta L}\right) \qquad (2)$$

$$\left(\frac{\Delta P}{\Delta L}\right) = \left(\frac{\mu}{\alpha}\vec{V} + c_2 \frac{1}{2}\rho_g \left|\vec{V}\right| \vec{V}\right) \qquad (3)$$

$$\alpha = \frac{d^2}{150}\frac{\epsilon^3}{(1-\epsilon)^2} \quad , \quad c_2 = \frac{3.5}{d}\frac{(1-\epsilon)}{\epsilon^3} \tag{4}$$

Gas phase energy equation:

$$\frac{\partial}{\partial t}\left(\epsilon\rho_g c_p T_g\right) + \nabla.\left(\epsilon\rho_g c_p T_g \vec{V}\right) + \epsilon\sum_i \omega_i h_i W_i = \epsilon\,\nabla.\left((k_g + \rho_g c_g D_{\parallel}^{d})\nabla T_g\right) - h_v\left(T_g - T_s\right) \tag{5}$$

$$D_{\parallel}^{d} = 0.5\,\alpha_g Pe \tag{6}$$

$$h_v = \frac{6\varepsilon}{d^2}\,k_g Nu_v \quad , \quad Nu_v = 2 + 1.1\,Re^{0.6}\,Pr^{0.33} \tag{7}$$

c_p is specific heat of mixture, T_g is gas temperature, Y_i is mass fraction species i, $\acute{\omega}_i$ is rate of reaction i, V_i is diffusion velocity i, h_i is enthalpy of species i, W_i is molecular weight of species i, k_g is thermal conductivity of the fluid, $D_{\parallel}^{d}$ is thermal dispersion coefficient along the length of the porous medium, h_v is volumetric heat transfer coefficient. Correlation (7) was estimated from experimental data by Wakao and Kaguei for heat transfer between packed beds and fluid [17].

Solid phase energy equation:

$$\frac{\partial}{\partial t}\left((1-\varepsilon)\,\rho_s\,c_s\,T_s\right) = \nabla\cdot\left[k_s\,(1-\varepsilon)\nabla T_s\right] + h_v\left(T_g - T_s\right) - \nabla.q_r \tag{8}$$

T_s is solid temperature, ρ_s is solid density, c_s is specific heat of solid medium, k_s is thermal conductivity of solid, q_r is the radiation heat transfer in solid.

Chemical species continuity equation:

$$\frac{\partial}{\partial t}\left(\varepsilon\,\rho_g Y_i\right) + \nabla.\left(\varepsilon\,\rho_g Y_i\,\vec{V}\right) + \nabla.\left(\varepsilon\,\rho_g Y_i\,V_i\right) - \varepsilon\,\dot{\omega}_i\,W_i = 0 \tag{9}$$

$$V_i = -\left(D + D_{\parallel m}^{d}\right)\frac{1}{X_i}\,\nabla X_i \tag{10}$$

$$\mathrm{P}e = \frac{\rho_g\,c_p\,|\vec{V}|\,d}{k_g} \quad , \quad D_{\parallel m}^{d} = 0.5\,D_{im}Pe_m \tag{11}$$

X_i is molar fraction of species i, Pe is Peclet number, D_{im} is diffusion coefficient of species i is mixture.

Turbulence model:

The transport equation for κ turbulent kinetic energy:

$$\frac{\partial(\rho k)}{\partial t} + \nabla.\left(\rho\vec{V}\,k\right) = -\frac{2}{3}\,\rho\,\kappa\,\nabla.\vec{V} + \sigma : \nabla\vec{V} + \nabla.\left[\left(\frac{\mu}{Pr_k}\right)\nabla k\right] - \rho\epsilon \tag{12}$$

with a similar one for the dissipation rate, ε:

$$\frac{\partial(\rho\epsilon)}{\partial t} + \nabla.\left(\rho\vec{V}\,\epsilon\right) = -\left(\frac{2}{3}\,c_{\epsilon_1} - c_{\epsilon_3}\right)\rho\,\epsilon\,\nabla.\vec{V} + \nabla.\left[\left(\frac{\mu}{Pr_\varepsilon}\right)\nabla\epsilon\right] + \frac{\epsilon}{k}\left[c_{\epsilon_1}\sigma:\nabla\vec{V} - c_{\epsilon_2}\rho\epsilon\right] \tag{13}$$

The quantities c_{ϵ_1}, c_{ϵ_2} ,c_{ϵ_3}, Pr_k and Pr_ε are constant whose values are determined from experiments and some theoretical considerations. σ is viscous stress tensor [20, 21].

Equation of state:

$$P = \rho_g\,R\,T_g\,/\,\overline{W} \tag{14}$$

R is universal constant of gases, $\overline{W}$ is average molecular weight of mixture, P is pressure inside the combustion chamber and the porous medium.

Combustion model:

Chemical mechanism for oxidation of methane fuel is considered. $\acute{\omega}_i$ chemical production rate:

$$\dot{\omega}_i = \sum_{i=1}^{NR}\left(v_{k,i}^{''} - v_{k,i}^{'}\right)\,R_i \tag{15}$$

$v_{k,i}^{''}$ and $v_{k,i}^{'}$ are stoichiometric coefficients. Combustion process includes ten equations and twelve species. These equations are presented in Table .1. Eq. 1, includes one-step reaction for methane fuel and equation 2-4, are Zeldovich mechanism for NO formation and rate of reaction is computed with Arrhenius method. But for six other equations that reaction rate is very quick relative to last four equations hence, equilibrium reactions are considered. In order to consider effects of turbulence on combustion, Eddy-Dissipation model [18] has been used.

TABLE 1. KINETIC AND EQUILIBRIUM REACTIONS

Number	Equation
1	$CH_4 + 2\,O_2 \rightarrow CO_2 + 2\,H_2O$
2	$O_2 + 2\,N_2 \rightarrow 2\,N + 2\,NO$
3	$2\,O_2 + N_2 \rightarrow 2\,O + 2\,NO$
4	$N_2 + 2\,OH \rightarrow 2\,H + 2\,NO$
5	$H_2 \leftrightarrows 2\,H$
6	$O_2 \leftrightarrows 2\,O$
7	$N_2 \leftrightarrows 2N$
8	$O + H \leftrightarrows O\,H$
9	$O_2 + 2\,H_2O \leftrightarrows 4\,OH$
10	$O_2 + 2\,CO \leftrightarrows 2\,CO_2$

Radiation model:

Due to high temperature of combustion zone, and high temperature of solid, radiation heat transfer is very important. Gas phase radiation in comparison with solid phase, that has a high absorption coefficient, is regardless. Several relations for modeling of radiation heat transfer and obtain the radiation intensity were presented. The heat source term $\nabla.q_r$, due to radiation in solid phase that appears in Eq .8, is calculated from Rosseland [19] method.

$$q_r = -\frac{16}{3}\frac{\sigma\,{T_s}^3}{\beta}\,\nabla T_s \tag{16}$$

σ Boltzmann constant and β is extinction coefficient.

MESH PREPARATION

The engine specifications have been presented in Table. 2. Prior to CFD simulation, computational mesh was generated for the engine. The geometry of a mesh is composed of several numbers of logical blocks that are patched together in a seamless fashion. Patching allows geometry to be created, block by block. Fig.1 shows the grid configuration of the engine at top dead center (TDC).

TABLE 2. ENGINE SPECIFICATION

Displacement volume	0.981 liter
Bore	102 mm
Stroke	120 mm
Nominal compression ratio	18
Connecting rod length	192 mm
Engine speed	2000 rpm
Mass of injected methane	0.0055 g
Equivalence ratio	0.17
Height of PM	1 cm
Diameter of PM	3 cm
porosity	o.8

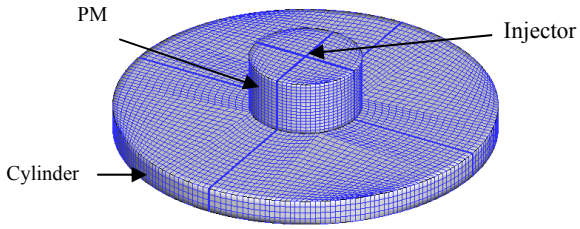

FIGURE 1. COMPUTATIONAL MESH FOR THE CFD CALCULATION AT TDC

INITIAL AND BOUNDARY CONDITIONS

Computation starts at bottom dead center (BDC). Initial charge densities were calculated based on temperature of 300K for fluid flow; and 1200 K for solid phase at BDC. The standard κ-ε turbulence model in the code was used with an initial value of turbulent kinetic energy κ, assumed 10 percent of the total kinetic energy based on the mean piston speed that assumed to be uniform. Radial velocity is initialized assuming a swirl ratio 0.0. The entire boundary wall was taken as zero heat-flux for gas and solid phase in PM and combustion chamber.

COMPUTATIONAL PROCEDURES

The KIVA-3V is an open source code that solves transient three dimensional chemically reactive flows with sprays, was used to solve the in cylinder flow and solid phase calculations. Species, momentum, energy, and turbulence transports equations are solved by finite volume method, which subdivides fluid domain of computation into a number of hexahedron cells. Solutions are marched in a number of cycles or time step, in which the transports equations are solved in three phases namely, phase A, B, and C. Phase A calculates spray droplet collision and oscillation terms and source term due to spray and chemistry while diffusion, viscous, and pressure gradient terms are solved in a coupled and implicit fashion in phase B. The solution procedure is very similar to the SIMPLE scheme, and each individual equation is solved iteratively using conjugate residual method [20, 21, and 22]. In Phase C, the flow field is remapped onto a new computational mesh, which is essentially calculating the convective transport terms. In the case of non moving mesh problem, the mesh is mapped back onto its original location (Phase B calculation where fluid mesh moving with the flow); in the case of moving boundary problem, the mesh is mapped to a new location. Phase C is calculated in a sub cycled and fully explicit fashion.

RESULTS AND DISCAUTION

In Figs. 2a, 2b, injection of methane as a gas fuel, in a hot PM is shown. In these figures mass fraction of methane respectively, after 10 and 20 degree crank angle after injection time (330 degree crank angle), is shown. Injector is in top of the PM in cylinder head and normal to the surface of it. Combustion does not consider in this figure and only injection in a chamber without PM considered (void space in PM location). Diffusion of gas in chamber is completely comprehensive from these figures. After 10 degree crank angle of injection, gas fuel impingement on piston surface and disperse in the cylinder. Figs. 3a, 3b, show mass fraction of methane as the same condition as Figs. 2a, 2b, except that the effect of PM has considered. It is clear that after 10 and 20 degree of injection, gas fuel residues in PM and disperse in it.

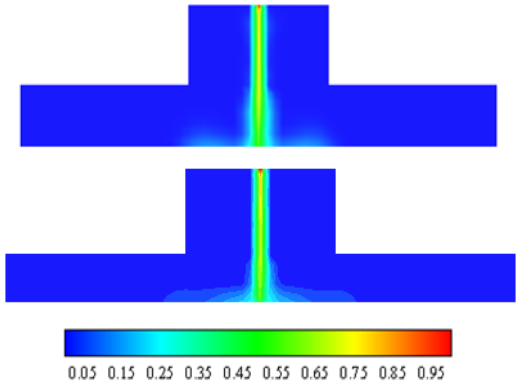

FIGURE 2. MASS FRACTION OF MATHANE IN CHAMBER WITHOUT EFFECT OF PM: TOP FIGURE AFTER 10 DEGREE CRANK ANGLE AND BOTTOM FIGURE AFTER 20 DEGREE CRANK ANGLE OF INJECTION TIME

With comparison between two Figs. 2, 3, the effect of PM in dispersion of gas fuel and mixture formation, is inferred. PM

causes better dispersion of methane in chamber. Also, it can be seen that if we use injection with inclined angle relative to engine axes, we will have better dispersion of gas fuel and can use more capability of PM for mixture formation.

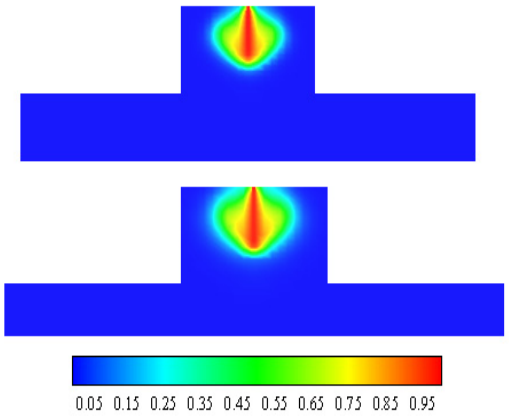

FIGURE 3. MASS FRACTION OF MATHANE IN CHAMBER WITH EFFECT OF PM: TOP FIGURE AFTER 10 DEGREE CRANK ANGLE AND BOTTOM FIGURE AFTER 20 DEGREE CRANK ANGLE OF INJECTION TIME

In Figs. 4, 5, 6, 7, 8, results of combustion for distribution, of oxygen, carbon dioxide, temperature in fluid and solid phase in a PM and in-cylinder, for several crank angle, are seen.

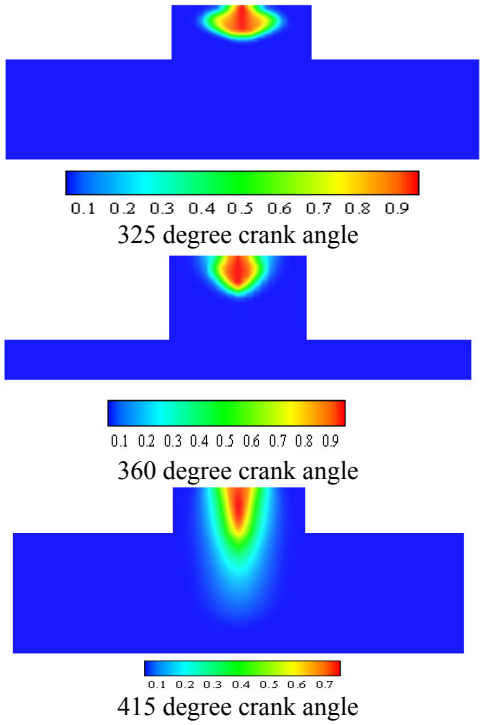

FIGURE 4. MASS FRACTION OF METHANE IN DIFFERENT CRANK ANGLE

Injection time is considered, 300 degree crank angle and duration of injection is 60 degree crank angle, to have more time for diffusion of methane and better mixture formation in order to volumetric combustion. In Fig. 4 mass fraction of methane in different crank angle, with combustion due to effect of PM, from 325 to 415 degree crank angle, is shown. From this figure it is concluded that, all the injected fuel concentrates near the injector during combustion and approximately half of the gas fuel does not consume during combustion because of lack of enough oxygen around the flame in PM (Fig. 5). Therefore, full cylindrical structure for PM, for engine application, is not suitable. We should select structures for PM that could disperse gas fuel in them very well. Also, it is better that injector with such structure of PM be inclined, relative to the engine axes and vertical injector is not a good choice.

In Fig .5 mass fraction of oxygen in different crank angle from 325 to 415 degree crank angle, during combustion is shown. From this figure, it is clear that oxygen near the injected gas fuel consumes rapidly and hence residual methane in PM does not access to enough oxygen to react with it, therefore some part of methane residues in PM.

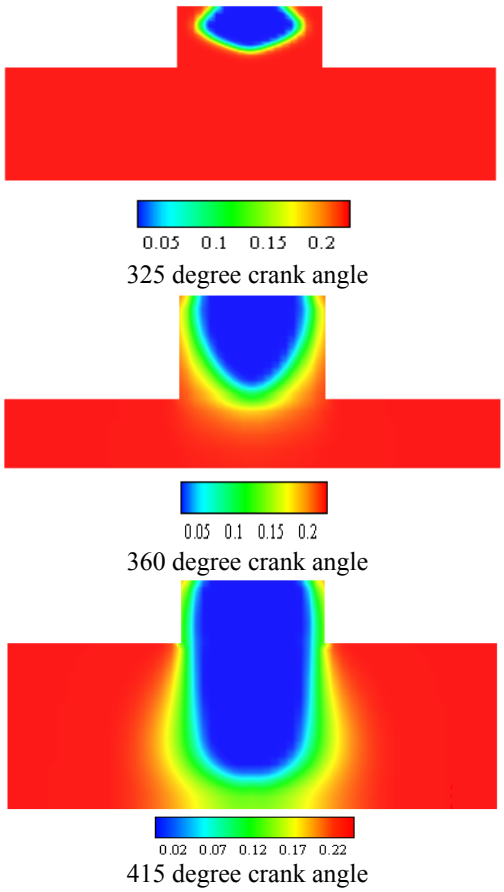

FIGURE 5. MASS FRACTION OF OXYGEN IN DIFFERENT CRANK ANGLE

It is concluded that PM structures with right angle corners, such as full cylindrical geometry for mixture formation, is not a suitable choice and also it is better that clearance volume is deleted, as much as possible.

Fig. 6 shows mass fraction of carbon dioxide in different crank angle during combustion from 325 to 415 degree crank angle. It is seen that reaction occurs around the flame front, this flame front in PM is very thick relative to flame front in normal combustion in engines and carbon dioxide concentrates in a region between injector and flame front.

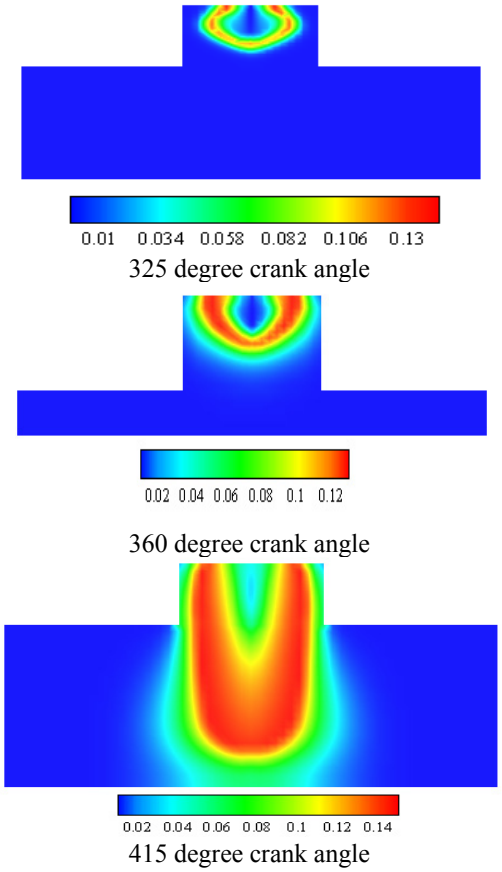

FIGURE 6. MASS FRACTION OF CARBON- DIOXIDE IN DIFFERENT CRANK ANGLE

In Fig .7 temperature distribution in fluid in different crank angle from 325 to 415 degree crank angle is shown. Flame front from its high temperature in PM, is seen. Maximum temperature in TDC is about 2200 K but because of existence of solid phase, some heat release of combustion absorbed by solid phase of PM during combustion and prevent from high temperature gradient in fluid. Fig. 8 shows temperature distribution from 325 to 415 degree crank angle, in solid phase of PM. Near the injector low temperature about, 300 K is seen because fuel is injected in hot PM with the 300 K, then fuel absorbs heat of PM near the injector and become hot in a small distance form injector and reaches to self-ignition temperature in a thick flame front. Figs. 9, 10 show numerical results for average pressure and temperature distribution in combustion chamber via crank angle. Simulation is started from BDC (180 degree crank angle). In motoring condition, no fuel is injected inside PM. The maximum pressure in motoring condition is about 43.5 bar. For four injection times (300, 310, 320 and 330 crank angle degree) with 10 degree duration for injection, the mean cylinder pressure during combustion is shown.

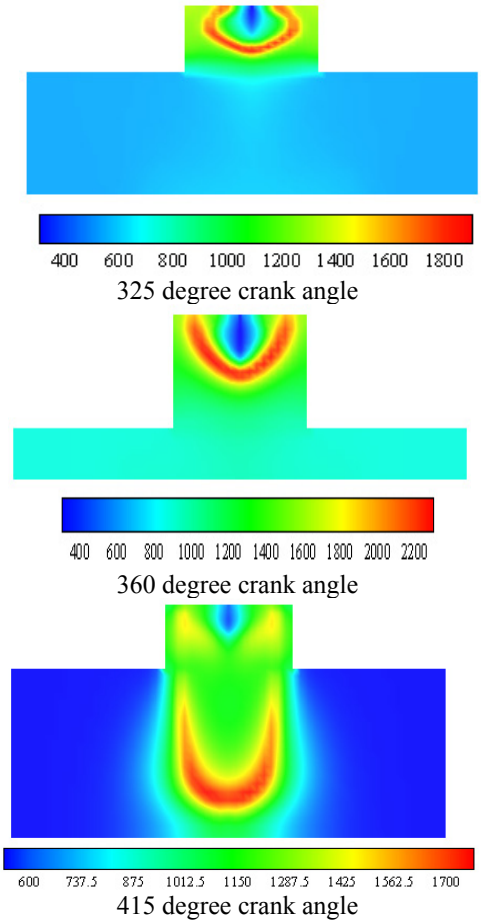

FIGURE 7. TEMPERATURE DISTRIBUTION OF FLUID IN DIFFERENT CRANK ANGLE

It is seen that pressure distribution does not depend on injection time and maximum pressure is about 50 bar in four cases. The maximum temperature in motoring condition is about 840 K. For four injection time (same time as Fig. 9), the mean cylinder temperature is shown. It is seen that maximum temperature does not depend on injection time and maximum temperature is about 950 K for four cases. It should be noticed that with this very low equivalence ratio($\Phi = 0.17$) combustion does not start in conventional engine but with PM technology, it is possible to use very lean mixture in cylinder.

Figs. 11, 12 show results for the distribution of temperature in both phase of PM (solid and fluid) for three different crank

angle injection times (300, 310, 320 degree). Simulation is started at BDC, initial fluid and solid phase temperature respectively, were 300 K and 1200 K. It is seen that temperature in both solid and fluid, does not change with the variation of the time of injection. As expected in the combustion process, gas temperature is higher than the solid temperature and heat is accumulates in solid phase due to high heat capacity of it. It showed that, however the mass of the fuel is very low value but combustion is started in PM.

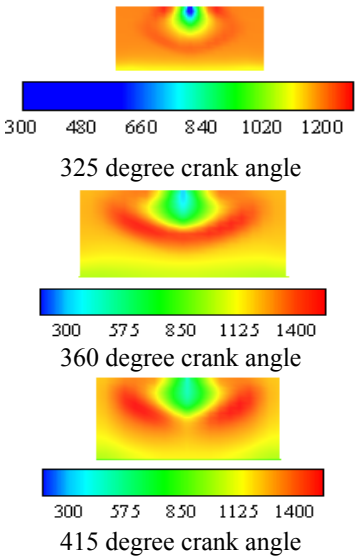

FIGURE 8. TEMPERATURE DISTRIBUTION IN SOLID PHASE OF PM IN DIFFERENT CRANK ANGLE

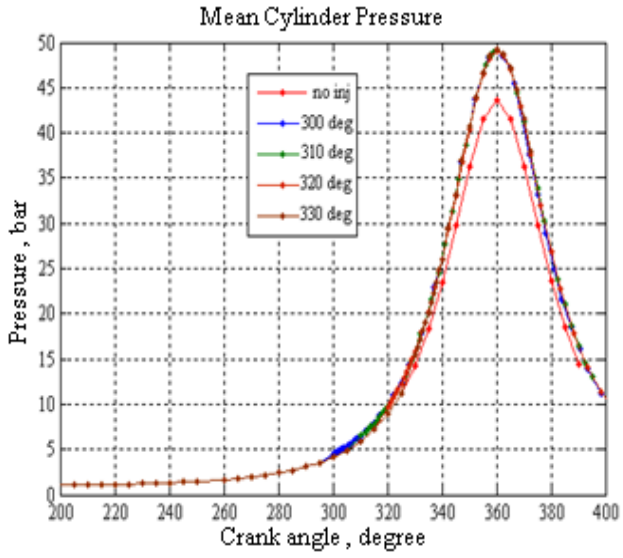

FIGURE 9. MEAN PRESSURE DISTRIBUTION IN CYLINDER VIA CRANK ANGLE

Fig. 13 shows mean mass fraction of methane in combustion chamber for three different injection times (300, 310, 320 degree). with 10 degree crank angle duration for injection, with crank angle. It is inferred from this figure that, some of the fuel is consumed very rapidly after the injection time but due to not enough oxygen in near flame, approximately half of the methane of the fuel residues in cylinder that correspond to Figs. 4 , 5.

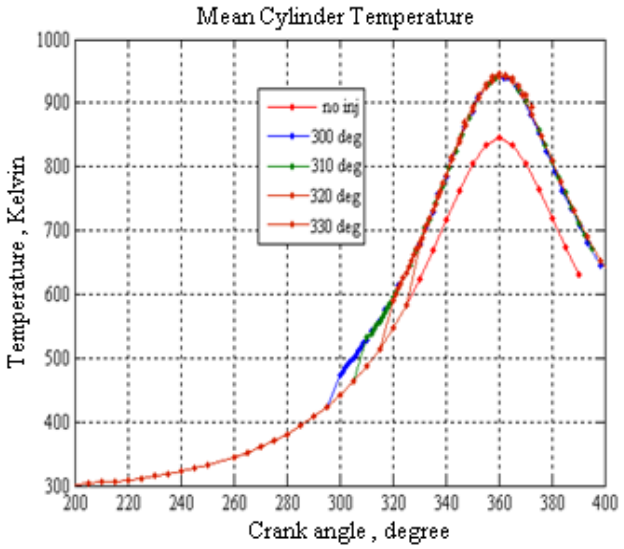

FIGURE 10. FLUID MEAN TEMPERATRE DISTRIBUTION IN (BOTH IN-CYLINDER AND PM) CYLINDER VIA CRANK ANGLE

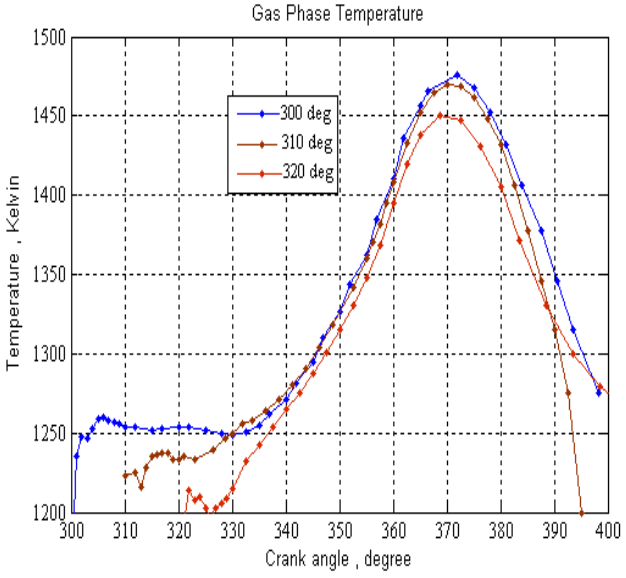

FIGURE 11. MEAN TEMPERATRE DISTRIBUTION IN GAS PHASE OF PM VIA CRANK ANGLE

Figs. 14, 15 show mass fraction of CO and NO via crank angle. Carbon monoxide is produced during combustion and with start of combustion suddenly increases and the CO concentration reaches its highest value and gradually decreases with completeness of combustion and is converted to CO_2 and its value is reduced, but combustion is done in little duration and heat is released in little time that causes to high temperature and CO have not enough time to convert to CO_2. It is concluded that injection time later than 310 degree crank angle, production of CO become very high and late injection should be avoided.

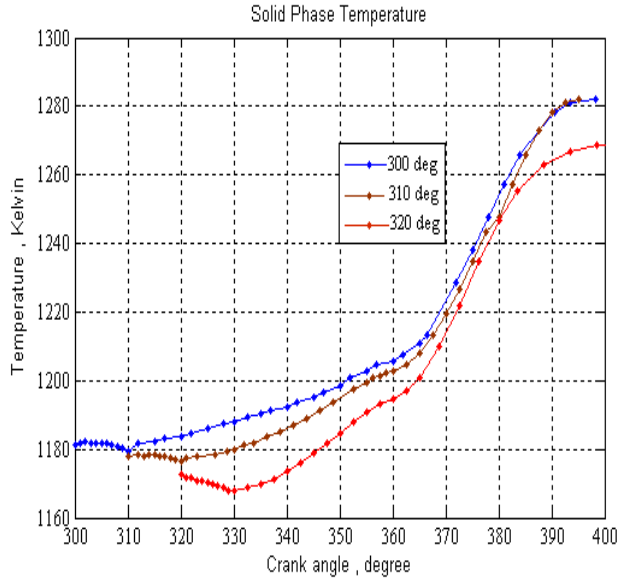

FIGURE 12. MEAN TEMPERATRE DISTRIBUTION IN SOLID OF PM VIA CRANK ANGLE

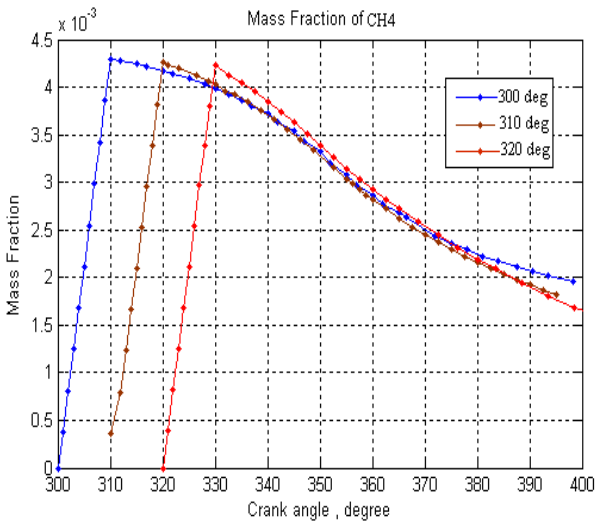

FIGURE 13. MASS FRACTION OF METHANE VIA CRANK ANGLE

As can be seen NO value at the beginning of the simulation is negligible due to low in-cylinder temperature. It seen that approximately after 60 degree crank angle, fluid phase of PM temperature reaches to highest value about 1780 K and decreasing in temperature after this time occurs and hence, no more NO if produced. But due to the fact that the temperature is considerably lower than normal condition of in-cylinder temperature (below 1800K), NO value is lower relative to normal operating condition.

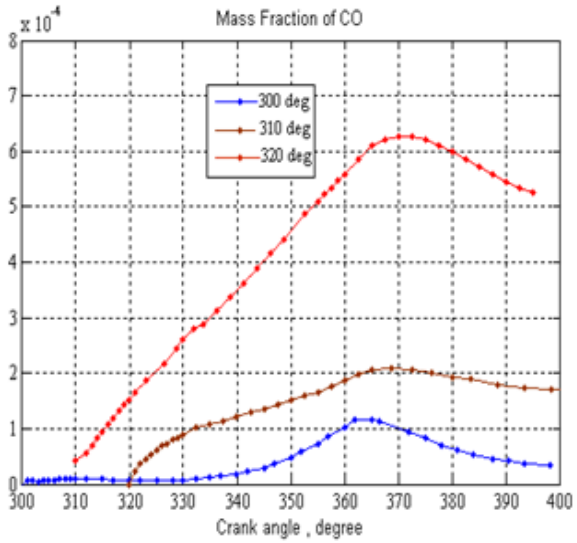

FIGURE 14. MASS FRACTION OF CO VIA CRANK ANGLE

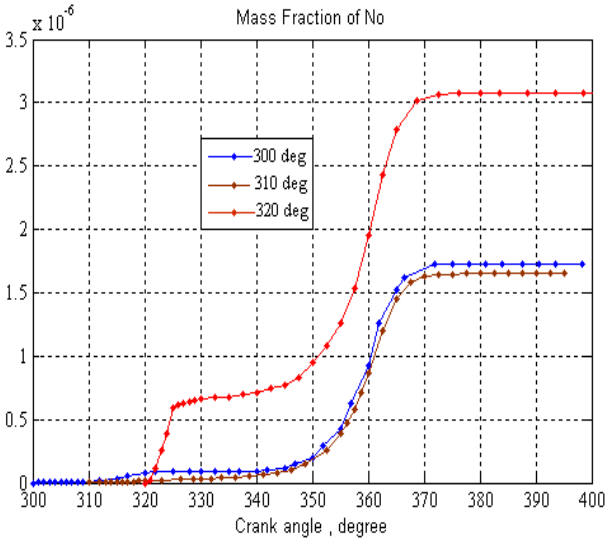

FIGURE 15. MASS FRACTION OF NO VIA CRANK ANGLE

CONCLUSIONS

PM in internal combustion engine with injection of methane in a hot PM was modeled. The contours of mass fraction of methane, oxygen, carbon dioxide, temperature in both phase of PM were shown. Also effect of injection time on in-cylinder mean pressure and temperature and CO and NO concentration as emissions was survived. Following results were obtained:

1) Combustion with very lean mixture become possible with using of PM, whereas in the normal condition with this equivalence ratio combustion ($\Phi = 0.17$) is not possible.

2) Full cylindrical geometries and structures with right angle corner are not suitable choice for PM engine.

3) Some of the fuel dos not participate in reaction because of not enough oxygen.

4) Pressure and temperature history dose nod depend on injection time but late injection can cause increasing in production of CO and NO.

5) After 400 degree crank angle some methane remains in combustion chamber that is mean some fuel is not combusted.

In future publication, we will show and propose better geometry for PM than cylindrical type, and inclined injection relative to cylinder axis and also, squish region will be deleted for using of all capability PM.

6) After approximately 70 degree crank angle, solid phase of PM prevents from increasing if gas phase temperature and thereby, no more NO is produced.

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