

## IMPROVEMENT OF EMISSIONS AND PERFORMANCE BY USING OF AIR JET, EXHAUST GAS RE-CIRCULATION AND INSULATION METHODS IN A DIRECT INJECTION DIESEL ENGINE

by

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*This article presents the improvement of operation characteristics and emissions reduction by means of creating an air-cell inside the piston body, exhaust gases re-circulating and insulating combustion chamber in a direct injection diesel engine simultaneously. The engine considered is a caterpillar 3401 which is modeled with an air-cell included as part of the piston geometry. This air-cell demonstrates that air injection in late combustion period can be effective in a significant reduction of soot emission while cold exhaust gas re-circulation can be effective in reduction of  $NO_x$  emission. Also for increasing of performance parameters, combustion chamber with air-cell is insulated. The analyses are carried out at part (75% of full load) and full load conditions at the same engine speed 1600 rpm. The obtained results indicate that creating the air-cell has a slight effect on improvement of performance parameters and it has significantly effect on soot reduction. The air-cell decreases the soot pollutant as a factor of two at both part and full load conditions. Also, the adding 5% of cold exhaust gas re-circulation in inlet air decreases  $NO_x$  by about half and insulating the engine increases the power and indicated mean effective pressure by about 7.7% and 8.5% and decreases the indicated specific fuel consumption by about 7.5% at part load and increases power and indicated specific fuel consumption by 8.5%, and decreases indicated specific fuel consumption by 8% at full load condition, respectively. Using this method, it was possible to control emissions formation and increase performance parameters simultaneously. The predicted results for mean in-cylinder pressure and emissions are compared to the corresponding experimental results and show good agreements.*

Key words: *air-cell, diesel engine, emission, injection, performance*

### Introduction

Due to the growing importance of future emission restrictions and also fuel deficiency, improvement of engine performance and reduction of exhaust emissions have become important issues for automotive manufacturers. In order to improve the engine performance, the combustion process is now being studied in more detail. Also, the recent global environmental regulations for reducing engine emissions, designers' engine have forced to explore new engine concepts and to study the effect of engine parameters on the formation of pollutant emissions with more detail. Simulation of the combustion system by means of computer modeling makes it possible to explore combustion regimes that may be difficult and/or expensive to achieve with experiments [1].

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Flow field models, such as KIVA II code in 1990 [2, 3] and numerical Ricardo Code in 1992 which were employed to investigate an optimum value for gas fuel injector angle in a dual fuel diesel-gas engine to obtain minimum unburned hydrocarbon (UHC) emissions [4], are premiere examples. Wook, *et al.* [5] used STARCD code and particle image velocimetry (PIV) analysis to study in-cylinder flow field of a single cylinder direct injection (DI) diesel engine. It was indicated that high Reynolds  $k-\varepsilon$  turbulent model pretty much satisfies such flow fields.

Lots of experimental researches have been done to decrease soot and  $\text{NO}_x$  emissions simultaneously. Kawazoe *et al.* [6] have significantly reduced exhaust soot emission using an air-plunger pump type air-jet generator into combustion chamber in a DI diesel engine. Nagano *et al.* [7] have studied an air-accumulation type air-jet generator to reduce exhausted  $\text{NO}_x$  and soot emissions. Choi and Foster [8] have utilized a high pressure gaseous injector to control mixing in-cylinder in a DI engine. To investigate the effects of the composition of injected gas, both nitrogen and  $\text{CO}_2$  have been used as the gaseous medium.

Because of opposite behavior of  $\text{NO}_x$  and soot production in combustion process, their simultaneous decreasing needs accurate study on the flow field in combustion chamber, mixture formation and their effects on the combustion process. Computational fluid dynamics (CFD) is one of the most efficient methods for this purpose.

Uludogan *et al.* [9] have explored the use of multiple injectors and split injection to reduce DI diesel engine emissions. They have shown multiple injectors increase engine power density and significantly reduce particulate and  $\text{NO}_x$  emissions. Combining exhaust gas re-circulation (EGR) with multiple injection system results reduced both  $\text{NO}_x$  and soot without negative effects on fuel consumption [10].

Mather and Reitz [11] have shown that the utilizing an air-cell created inside the piston, improves the fuel-air mixing and reduces soot emission from diesel engine without increasing the emission of  $\text{NO}_x$ . Their results indicated that it can be an effective way to reduce particulate emissions by increasing mixing during the late combustion period with an air-jet injected into the main combustion chamber. In this study reduction of particulate by as much as a factor of two was observed with a simultaneous, but slight, reduction in  $\text{NO}_x$ . Kurtz and Foster [12] experimentally have studied the effects of increased in cylinder mixing on emission in diesel engine. Joergl and Becker [13] experimentally have studied the effect of air injection and EGR on the emission in a single cylinder passenger cars and they showed that the emission can be improved by this work.

In order to increase thermal efficiency and improve operation characteristics engine such as power, SFC and MEP, reducing the amount of energy rejection from an internal combustion engine is essentially. In a conventional internal combustion engine, approximately one-third of total fuel input energy is converted to useful work. Since the working gas in a practical engine cycle is not exhausted at ambient temperature, a major part of the energy is lost with the exhaust gases. In addition, another major part of energy input is rejected in the form of heat loss via the cooling system. If the energy normally rejected to the coolant could be recovered on the crankshaft as useful work, then a substantial improvement in fuel economy would result.

The major investigations about the low heat rejection (LHR) engines for achieving high performance engines have been done by Bryzik *et al.* [14-17]. Van Sudhakar [18] has studied the development of the adiabatic diesel engine with the goal of 65% reduction in net in-cylinder heat transfer over a cooled engine and several un-cooled engines with intermediate levels of reduced heat transfer. The engine performance and exhaust emissions are also measured and compared.

Kawamura *et al.* [19] have improved combustion process in a LHR diesel engine using a centrally located pre-combustion chamber in the cylinder head and throat holes drilled in it as radiating to cylinder inner surface. It is realized about 10% improvement in fuel economy compared to conventional DI diesel engine. Cheng *et al.* [20] have compared heat transfer characteristics in insulated and non-insulated diesel engines at the same speeds, loads and air-flow rates. Compared to the baseline engine, the insulated engine had a higher brake specific fuel consumption (BSFC) which was attributed to a slower combustion process.

Rakopoulos *et al.* [21] have simulated combustion process in chamber insulation using a multi-zone model and investigated its effects on the performance and exhaust emissions. It is found that the rising heat insulation leads to a significant increase of the exhaust  $\text{NO}_x$  and a moderate increase of the exhaust soot concentration. The more detail about ceramics characteristics which are utilize LHR engines are found in [22-25].

As can be seen in the relevant literature, a few studies seem to have been performed recently about simultaneously improvement of performance and emission for DI engines. At the present work, intended for a contribution to the open literature, a CFD code has been utilized to numerically investigate the simultaneous effects of creating an air-cell inside the piston, adding EGR to the inlet air and insulating the combustion chamber walls on the combustion process, operation characteristics, and exhaust emissions formation at both part (75% of full load) and full load conditions.

### Model initial and boundary conditions

Calculations are carried out on the closed system from intake valve closure (IVC) at  $147^\circ$  CA bTDC to exhaust valve opening (EVO) at  $100^\circ$  CA aTDC. Figures 1(a) and 1(b) show the numerical grids for the baseline (Caterpillar 3401) and baseline with air-cell created engines, which are designed to model the geometry of the engines. Also as can be seen from the fig. 1(c), grid dependency is based on the in-cylinder pressure and present resolution is found to give adequately grid independent results. The volume clearance and also the compression ratio in both geometries are the same. Considering the symmetry of the model, problem is only solved for a 60 degrees section (six holes injector) and contains a maximum of 21500 cells at  $147^\circ$  CA bTDC and crank angle (CA) resolution is  $0.5^\circ$  CA. The present resolution was found to give adequately grid independent results. Utilizing symmetry geometry has negligible error in the study of real engine parameters (piston

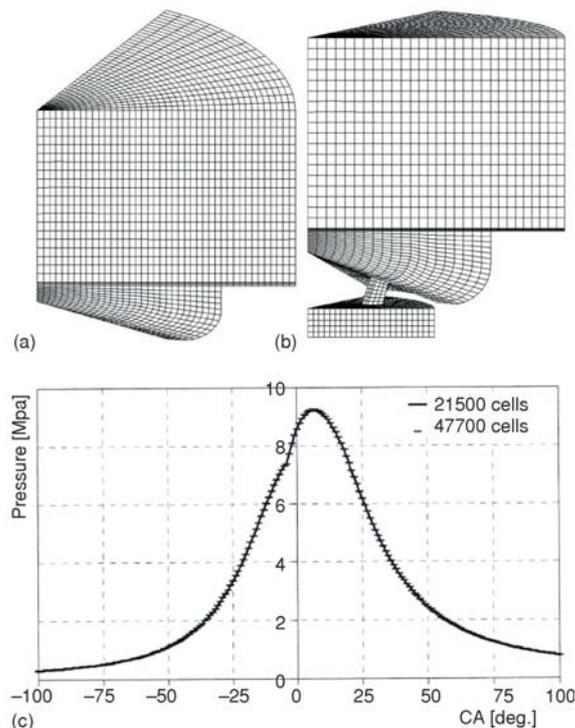


Figure 1. Computational geometries for (a) baseline engine (Caterpillar 3401), (b) air-cell created engine, (c) grid dependency based on the in-cylinder pressure

head, off-centric injector and *etc.*) [26]. Initial pressure in the combustion chamber is set to be 200 kPa and initial temperature is calculated to be 344 K, and swirl ratio is assumed to be on quiescent condition for all cases. This study has been performed at 1600 rpm constant engine speed for two different load (75% of full load and full load) conditions. In the cases with insulated walls, the boundary condition is assumed to be adiabatic wall and in the non-insulated cases, all boundaries temperatures are supposed to be constant throughout the simulation, but allowed to vary with the combustion chamber surface regions.

### Model formulation

The numerical model is carried out for the Caterpillar 3401 DI diesel engine with the specifications on the tab. 1. The governing equations for unsteady, compressible, turbulent reacting multi-component gas mixtures flow and thermal fields were solved from IVC to EVO by the AVL Fire CFD code [27]. The turbulent flows within the combustion chamber are simulated using the RNG turbulence model, modified for variable-density engine flows [28].

At the present study because of high pressure injection system (common rail) standard WAVE break up model [29] is used for the primary and secondary atomization modeling of the spray and resulting droplets. Also drop parcels are injected with characteristic size equal to the nozzle exit diameter (blob injection). The Dukowicz [30] model is applied for treating the heat up and evaporation of the droplets, which is described in. A Stochastic dispersion model is employed to take the effect of interaction between the particles and the turbulent eddies into account by adding a fluctuating velocity to the mean gas velocity [27]. This model assumes that the fluctuating velocity has a randomly Gaussian distribution. The spray/wall interaction model used in this simulation was based on the spray/wall impingement model [27].

The Shell auto-ignition model is used for modeling of the auto ignition [31]. In this generic mechanism, six generic species for hydrocarbon fuel, oxidizer, total radical pool, branching agent, intermediate species, and products were involved. In addition the important stages of auto ignition such as initiation, propagation, branching, and termination were presented by generalized reactions, described in [27, 31].

The Eddy break-up model (EBU) based on the turbulent mixing is used for modeling of the combustion in the combustion chamber [27] as follows:

**Table 1. Specifications of Caterpillar 3401 DI diesel engine**

Cycle type	Four stroke
Number of cylinders	6
Injection type	DI
Cylinder bore	137.6 mm
Stroke	165.1 mm
Displacement volume	2.44 litre
Compression ratio	15.1 : 1
Engine speed	1600 rpm
Full load injected mass	16.22 g per cycle
75% of full load injected mass	12.165 g per cycle
Injection pressure	90 MPa
Start injection timing	8.5° CA bTDC
Nuzzle diameter at hole center	3 mm
Number of nuzzle holes	6
Nuzzle outer diameter	0.259 mm
Spray cone angle	10°

$$\overline{\rho r'_{fu}} = \frac{C_{fu}}{\tau_R} \bar{\rho} \min \left( \bar{y}_{fu}, \frac{y_{ox}}{S}, \frac{C_{pr} \bar{y}_{pr}}{1+S} \right) \quad (1)$$

This model assumes that in premixed turbulent flames, the reactants (fuel and oxygen) are contained in the same eddies and are separated from eddies containing hot combustion products. The rate of dissipation of these eddies determines the rate of combustion. In other words, chemical reaction occurs fast and the combustion is mixing controlled. The first two terms of the “minimum value of” operator determine whether fuel or oxygen is present in limiting quantity, and the third term is a reaction probability which ensures that the flame is not spread in the absence of hot products [27]. Above equation includes three constant coefficients ( $C_{fu}$ ,  $\tau_R$ ,  $C_{pr}$ ) and  $C_{fu}$  varies from 3 to 25 in diesel engines. An optimum value is selected according to experimental data [9, 32].

$NO_x$  formation is modeled by the Zeldovich mechanism [33] and soot formation is modeled by Kennedy, Hiroyasu and Magnussen mechanism [34, 35].

### Results and discussion

In the present work, firstly the air-cell has been created inside the piston and then the main combustion has been insulated while the initial volume of combustion chamber is constant. The volumetric ratio of air-cell is selected as 16% of volume clearance that which is optimized volume according to reference [11]. Finally the cold EGR effects are investigated by adding 5 and 10 mass fractions to the inlet air. The numerical results of others EGR mass fractions show that improvement in performance and emissions are not achieved for these cases. The operating conditions are 75% of full load and full load at constant speed of 1600 rpm.

Figures 2(a) and 2(b) show the comparison of predicted and experimental in-cylinder pressure for both part and full load conditions for baseline engine. The results presented in this

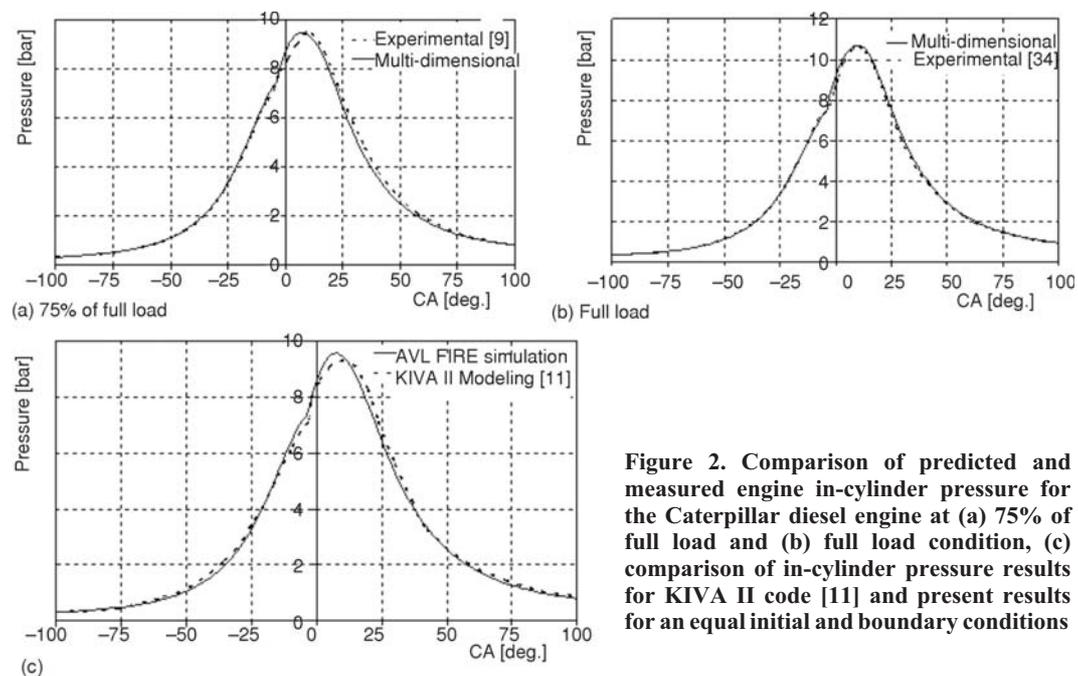


Figure 2. Comparison of predicted and measured engine in-cylinder pressure for the Caterpillar diesel engine at (a) 75% of full load and (b) full load condition, (c) comparison of in-cylinder pressure results for KIVA II code [11] and present results for an equal initial and boundary conditions

figure are global (cylinder averaged) quantities as a function of time (CA). The good agreement between measured and predicted data especially during the compression and expansion strokes with the start of combustion time verifies the result of model. The peak pressures discrepancy between experiment and computation is less than 1%. Figure 2 also shows the quantity of computational and experimental results for start of combustion (SOC) and ignition delay (ID) CA degrees in both cases. The discrepancies of SOC or ID between computation and experiment at part load and full load operation are as small as 1 and 2 CA, respectively. In order to assure the results of the created air-cell geometry, a comparison is performed between the KIVA II code [11] and present results for the in-cylinder pressure results for a same initial and boundary conditions. It is indicated in fig. 2(c) and represents a good agreement.

This verification for baseline and baseline with air-cell states demonstrate that multi-dimensional modeling can now be used to gain insight into the combustion process and to provide direction for exploring new engine concepts.

Figure 3 indicates the pressure histories for the baseline engine, insulated baseline, insulated and air-cell connected case with 5% and 10% EGR added to the final modified case at two loads. It is obvious that peak pressure will increase by increasing the engine load. At this type of engine, increasing the load from 75% to the full load is caused to increase the peak pressure by about 15% in all cases (comparison between figs. 3(a) and 3(b)). Also increasing the load had no debatable effect on ignition delay and start of combustion. The peak pressure occurs at 7° CA aTDC at 75% of full load and 10° CA aTDC at full load condition. The insulation, air-cell creating and EGR applying have no considerable effect on peak pressure occurrence point.

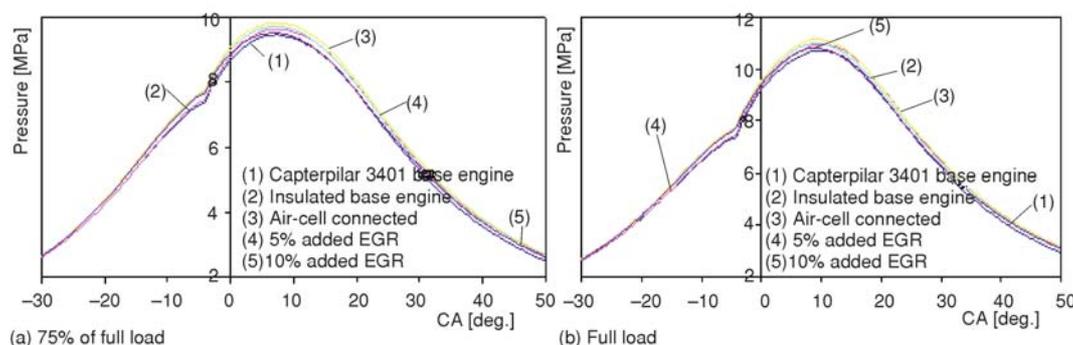


Figure 3. Mean in-cylinder pressure vs. CA for (a) 75% of full load and (b) full load conditions

By insulating the baseline engine because of no heat loss from the cylinder and piston walls, the peak pressure increases by about 1.5% and 2.5% at both part and full loads, respectively, as can be seen in fig. 4. Creating the air-cell under the piston head which connected to the mean combustion chamber with rectangular throat increases the cylinder pressure by about 1.9% and 1.6% at both part and full load conditions, respectively. As it is shown these figures that initial pressure for air cell case was elevated slightly above the initial condition for standard engine. This change in initial condition was required for air cell equipped engine to match the temperature and pressure at the start of injection of standard engine.

Adding EGR to the air inlet because of decreasing air available for combustion in the main chamber also increasing thermal capacity causes the peak pressure decreases. Increasing the EGR mass fraction more decreases the peak pressure.

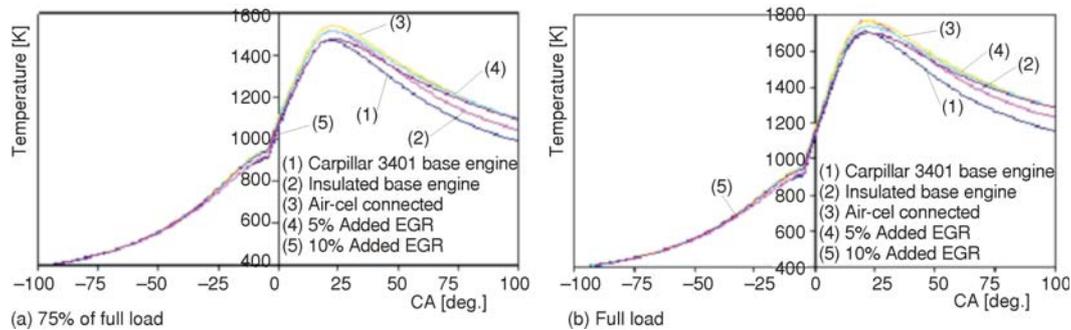


Figure 4. Mean in-cylinder temperature vs. CA for (a) 75% of full load and (b) full load conditions

Figure 4 indicates the mean temperature histories vs. CA for both part and full load conditions. Increasing the engine load from 75% of full load to the full load condition increases the peak temperature by about 250-300 K and exhaust temperature by about 200-250 K. The peak temperatures in all cases occur at 25° CA aTDC.

Insulating the combustion chamber results to reduce the heat loss from the cylinder walls and increases the in-cylinder peak temperature by about 150 K in both part and full load conditions. The maximum exhaust temperature occurs in air-cell cases because of resuming combustion in late combustion period. Due to decreasing the in-cylinder pressure at expansion stage, the reserved air in the air-cell injects to the main chamber and combusts the remaining fuel. Adding EGR to the air inlet decreases the peak temperature.

A match of pressure and temperature behaviors of all cases would imply that the spray vaporization, ignition, and combustion characteristics of standard engine and the air cell equipped engine would be similar [11].

Heat release rate curves are shown in fig. 5 for both part and full load operation conditions. Increasing load to full load mode causes ignition delay decreased to 5° CA from 6° CA with respect to 75% load. Figure 6 indicates that converting the constant temperature boundary condition to the insulated walls, creating the air-cell inside the piston and also adding EGR have no debatable effect on ignition delay. This means that the effects of decreasing oxygen concentration and increasing heat capacity on ignition delay is equal the increased charge temperature when engine is insulated. The value of premixed burning phase slightly increases due to insulating the walls and decreases due to increasing EGR, and air-cell has no sizeable effect on it. At

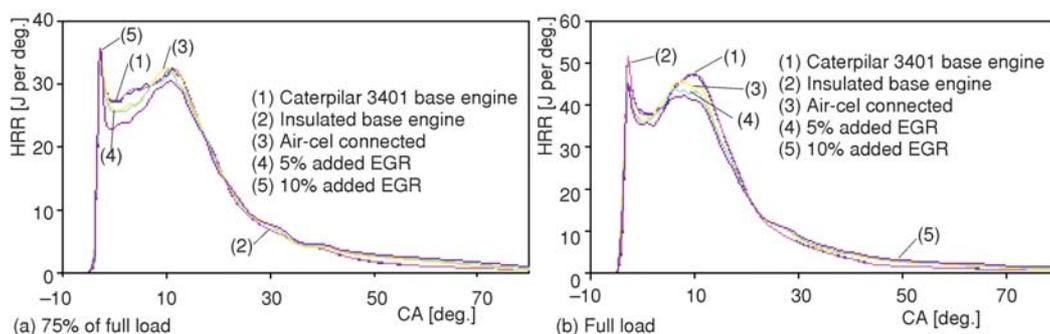
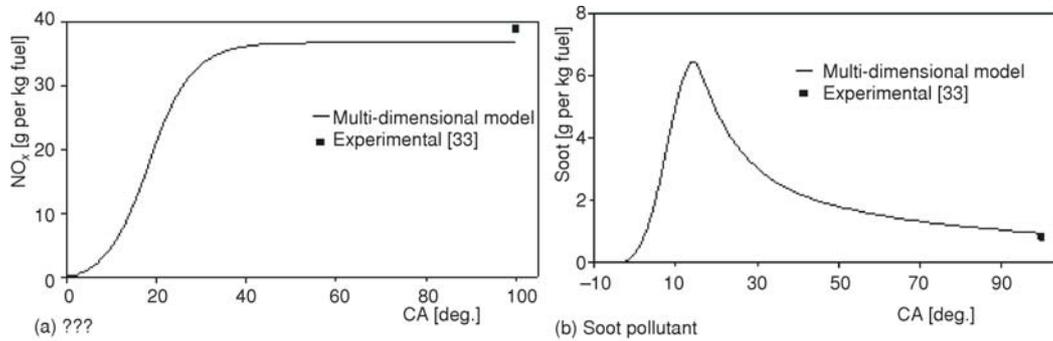


Figure 5. Mean in-cylinder temperature vs. crank angle for (a) 75% of full load and (b) full load



**Figure 6. Comparison of predicted and measured engine pollutants at full load condition of the Caterpillar diesel engine**

the start of diffusion burning phase (near TDC) in air-cell connected cases because of the lack of air in main chamber, burning rate is slow and increasing EGR makes it slower. The causes of fast burning rate from near TDC to 10° CA aTDC (as shown in fig. 5) pertain to high temperature regions indicating significant heat release rate (HRR) within air cell and the decreased resistance to flow into the air cell by changing the turbulence levels at the inlet passage [11].

One feature of heat release rate traces for the air-cell engine is the rise in HRR that can be seen near 10° aTDC at part and full loads. The reason of this rise is that the fuel injection influences the in-cylinder flow field. Eventually, when the spray reaches the air-cell passage, the flow produced by the fuel injection interacts with the flow into the air-cell flow beginning to reverse direction and emerge out of the air-cell at +10° aTDC [11]. But in the late combustion period (after 20° CA aTDC), while the reserved air in air cell injects to the main chamber, combustion continues until almost exhaust valve opening. This combustion has a very vital effect on soot reduction.

Also according to fig. 5, insulation causes that premixed burning fraction increase and a corresponding in the amount of diffusion burning fraction decreases.

### Performance parameters

Table 2 shows the variation of performance parameters including net cycle work, indicated power, mean effective pressure (IMEP) and specific fuel consumption (ISFC) vs. load operation conditions in all cases. Indicated work per cycle was calculated from the cylinder pressure and piston displacement, as follows:

$$W = \int_{\theta_1}^{\theta_2} P dV \quad (2)$$

where  $\theta_1$  and  $\theta_2$  are the start and end of the valve closed period, respectively (*i. e.* IVC = -147° CA bTDC and EVO = 100 CA aTDC). The indicated power per cylinder and indicated mean effective pressure were related to the indicated work per cycle by:

$$P = \frac{WN}{60000 n} \quad (3)$$

$$IMEP = \frac{W}{V_d} \quad (4)$$

where  $n = 2$  is the number of crank revolutions for each power stroke per cylinder,  $N$  [rpm] – the engine speed, and  $V_d$  – the volume displacement. The ISFC was defined as:

$$ISFC = \frac{\dot{m}_f}{P_i} \quad (5)$$

In eq. 2, the work was only integrated as part of the compression and expansion strokes and the pumping work was not taken into account. Therefore, the power and ISFC analyses can only be viewed as being qualitative rather than quantitative in this study.

With the increase of the engine load, net cycle work, indicated power and IMEP increase and ISFC decreases as expected. For both part and full load conditions, the obtained results for studied cases are the same.

Insulating the engine and creating the air-cell have positive effect on performance parameters. Insulating the base engine increases the indicated power and IMEP by about 7.7% and 8.5% at part load condition and 8.5% and 8.5% at full load condition, respectively, also decreases the ISFC by about 7.5% and 8% at part and full load, respectively. The effect of air-cell on performance parameters is lesser than insulation (about 3%). As can be seen in tab. 2, adding EGR to the inlet air has negative effect on engine performance.

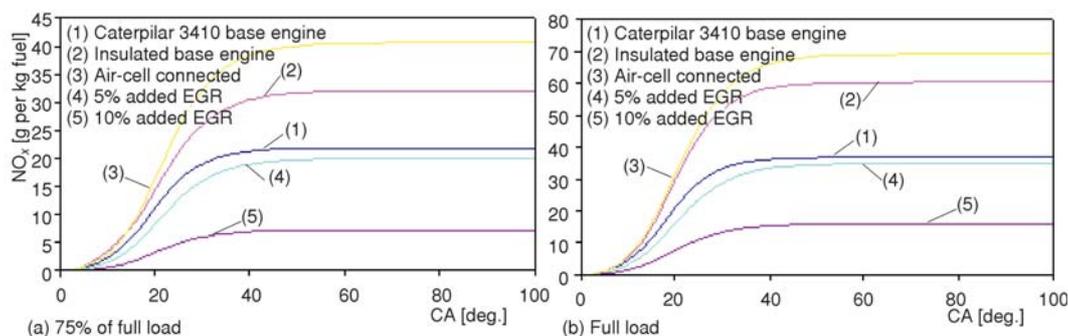
**Table 2. Operation characteristics for all cases at both part and full load conditions**

		Work [kJ cycle]	Indicate power [kW]	$\dot{m}_{fuel}$ [gs <sup>-1</sup> ]	IMEP [bar]	ISFC [gkW <sup>-1</sup> h <sup>-1</sup> ]
Part load condition	Caterpillar 3401 base	2.0	26.64	1.62	8.15	291.92
	Insulated base engine	2.17	28.92	1.62	8.84	269.09
	Air-cell connected	2.27	30.30	1.62	9.25	257.02
	5% added EGR	2.24	29.82	1.62	9.11	261.12
	10% added EGR	2.16	28.80	1.62	8.80	270.17
Full load condition	Caterpillar 3401 base	2.72	36.18	2.16	11.06	215.07
	Insulated base engine	2.94	39.24	2.16	12.00	198.25
	Air-cell connected	3.02	40.32	2.16	12.32	193.08
	5% added EGR	2.97	39.60	2.16	12.10	196.56
	10% added EGR	2.90	38.70	2.16	11.82	201.24

### Emission formation results analysis

In the following section, the productions of NO<sub>x</sub> and soot as main emissions in DI Diesel engines are discussed. In fig. 6 the exhaust NO<sub>x</sub> and soot quantities for the base engine at full load condition are compared and verified with experimental data [32]. There is a good agreement among the results.

In figs. 7(a) and 7(b) are shown the NO<sub>x</sub> formation histories for all cases. As can be seen, increasing the operation load causes to increase the NO<sub>x</sub> emission as factor of two because of higher peak temperature and peak premixed combustion at full load condition (figs. 4 and 5). NO<sub>x</sub> formation starts off at about 14° CA for full load and 15° CA for part load operation state af-



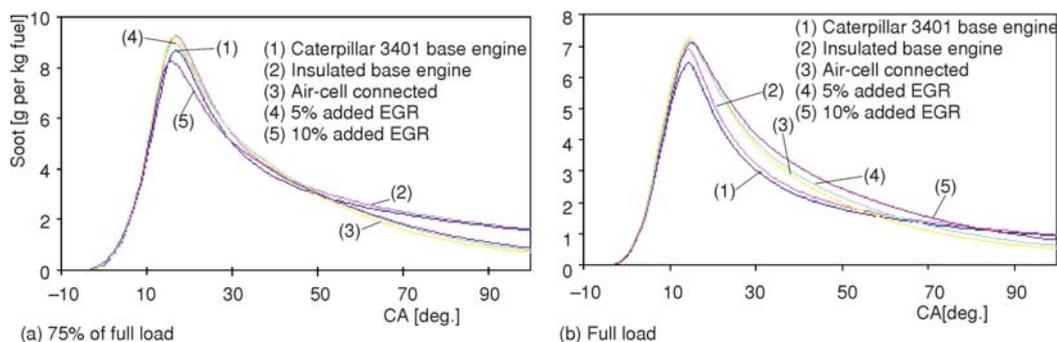
**Figure 7.** Model results for  $\text{NO}_x$  emission for all cases at (a) 75% of full load and (b) full load operation conditions at 1600 rpm

ter the start of injection. The initial increase in the global  $\text{NO}_x$  formation follows the global temperature trend (also on fig. 4).

Insulating the combustion chamber walls causes to decrease the heat loss from the walls and increase the in-cylinder temperature and as a result of that  $\text{NO}_x$  increases. Also creating the air-cell increases  $\text{NO}_x$  emission. Insulating and connecting the air-cell increases the  $\text{NO}_x$  as a factor of two in both part and full load conditions. Adding EGR to the inlet air is a very effective solution in  $\text{NO}_x$  reduction because of decreasing the peak temperature in combustion chamber. As can be seen in fig. 7, 5% of EGR reduces  $\text{NO}_x$  by about half and 10% of EGR reduces this emission by about one fifth.

Figure 8 presents the evolution of the soot formation in the combustion chamber for both load conditions. One could notice that at part load, the exhaust quantity of soot is further than full load by about a factor of two. Soot emission attains its peak at near peak of diffusion phase of combustion and because of the availability of sufficient oxygen; the turbulent soot oxidation diminishes the soot concentrations. The soot is then oxidized again in the leaner regions of the flame, so that most of the soot is already consumed in zones close to the stoichiometric.

As can be seen on fig. 8, insulating the chamber walls have no effect on exhaust soot concentration. Creating the air-cell reserves some of the air at compression stage and due to decreasing the in-cylinder pressure at expansion stage, this air injects to the main combustion chamber. Injecting the low temperature air increases the soot oxidation rate and reduces the ex-



**Figure 8.** Model results for soot emission for all cases at (a) 75% of full load and (b) full load operation conditions

haust soot concentration. Adding EGR to the inlet air decreases the soot oxidation rate and as a result of that exhaust soot increases.

At whole,  $\text{NO}_x$  formation is intensified in areas with equivalence ratio close to 1 and the temperature higher than 2000 K. In addition to this, the area with the equivalence ratio higher than 3 (regions near to spray jet) and the temperature approximately between 1600 K and 2000 K, yields in greater soot concentrations [36]. Under medium and high load conditions, the main cause of the exhaust smoke can be both the fuel adhesion to the chamber wall and stagnation of rich fuel-air mixture.  $\text{NO}_x$  and soot emissions have opposite formation procedure. In any combustion processes, reducing the  $\text{NO}_x$  causes to increase the soot.

At figs. 9 and 10 from left to right are shown the contours of the temperature,  $\text{NO}_x$ , oxygen, vapor mass fractions, soot formation regions at state of base engine, adiabatic base engine, adiabatic air cell connected engine, adiabatic air cell connected engine with 5% and 10% EGR from top to bottom, respectively, in order to study combustion process and emission formation with zonal method. The regions with high temperature and sufficient oxygen mass fraction in the combustion chamber and air-cell are potential  $\text{NO}_x$  producing regions. At ten degrees aTDC when the flow is just starting to reverse and flow out of the air-cell, the  $\text{NO}_x$  contours show two regions of  $\text{NO}_x$ , one in the lip bowl of main combustion chamber and the other in the air-cell at

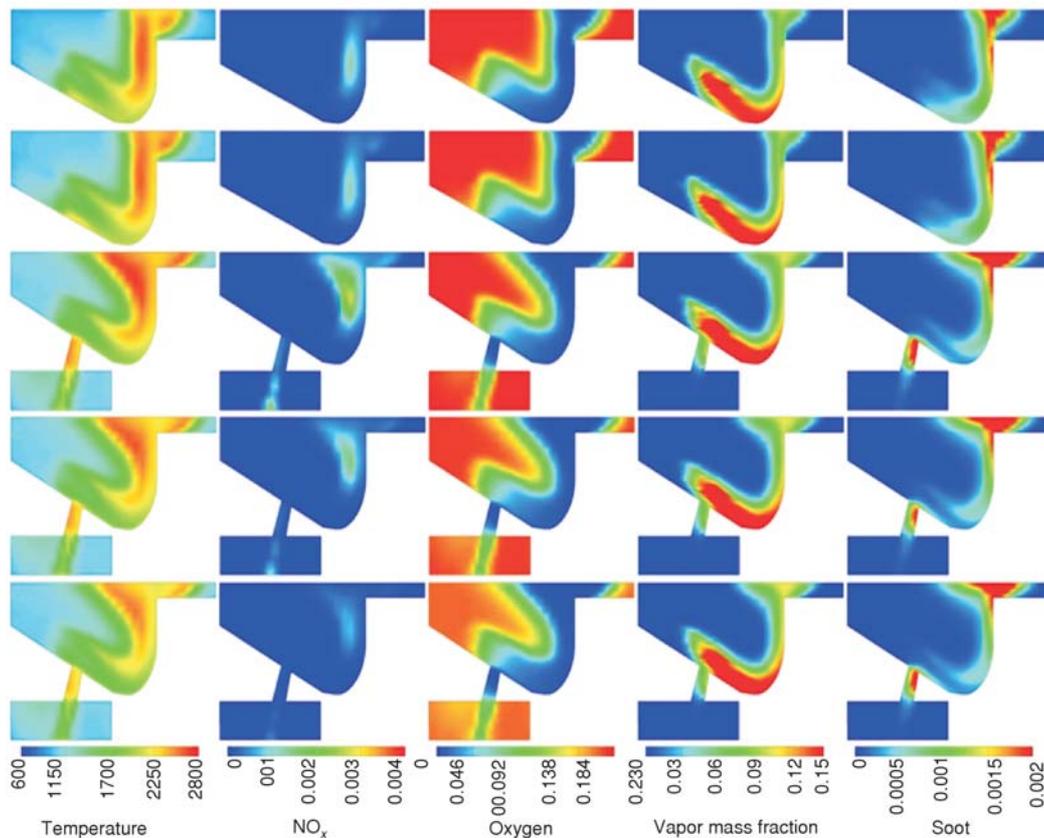
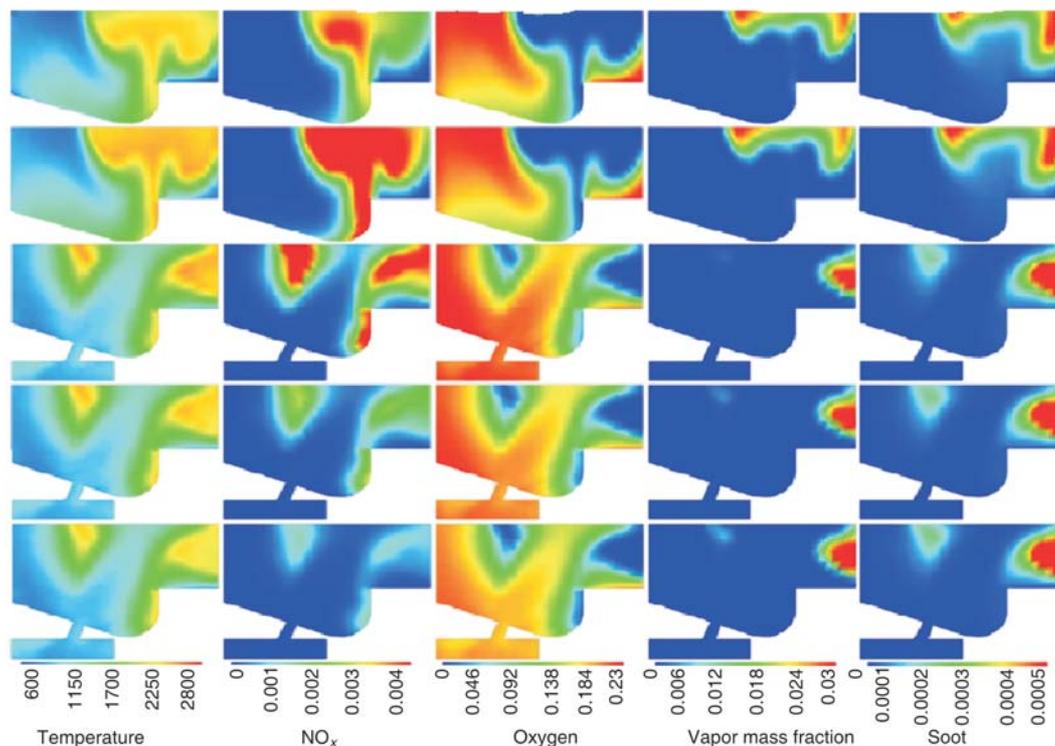


Figure 9. Temperature,  $\text{NO}_x$ , oxygen, vapor mass fractions, and soot contour plots at full load operation condition at  $10^\circ$  CA aTDC and 1600 rpm (for color image see journal web site)



**Figure 10.** Temperature,  $\text{NO}_x$ , oxygen, vapor mass fractions, and soot contour plots at full load operation condition at  $40^\circ$  CA aTDC and 1600 rpm (for color image see journal web site)

the stagnation region under the inlet channel to the air-cell. The two  $\text{NO}_x$  regions may be either the result of two distinct ignition and combustion process, or by combustion gases forced into the air-cell where combustion and  $\text{NO}_x$  production continues.

Heat conduction to the walls had a significant quenching action on the high temperature  $\text{NO}_x$  producing regions inside the air-cell. Subsequent  $\text{NO}_x$  contours in fig. 10 ( $+40^\circ$ ) show that while the air-cell is emanating a jet into the combustion chamber, the region of high  $\text{NO}_x$  omitted inside the air-cell on the cylinder axis side of the passage. Adding EGR to the inlet air decreases the  $\text{NO}_x$  producing in the air cell and adding air cell to base engine and also insulating increase  $\text{NO}_x$  emission.

A demonstration of the effect of the air-cell on the soot contours is seen by overlaying the vapor fuel mass fraction, oxygen mass fraction and temperature contours plots at different cases. During the early part of combustion, soot is forming in the main combustion chamber in high vapor mass fraction regions and gets drawn into the air-cell as the air-cell is charging (don't shown contours). However, the soot that gets drawn into the air-cell is oxidized and negligible soot is present in the air-cell at the time of flow reversal ( $+10^\circ$  aTDC) while soot which is forming in the back of air cell passage mouth, because of lack of oxygen, less mixing is not oxidized. At this time the soot in the main combustion chamber and air cell passage have been swept to the edge of the piston bowl by the flow field produced by the spray. As the air-jet emanating from the air-cell enters the combustion chamber the region of high soot concentration region has moved to the rim of the piston bowl where the reverse squish flow velocity field.

The air-jet emanating from the air-cell has crossed with the flow field of main combustion chamber and stretches the region of high soot concentration (+40° aTDC) and causes augmenting mixing and increasing the rate of soot oxidation. This period of the soot oxidation being highly influenced by the air-jet corresponds to the period of rapid soot oxidation for the air-cell engine and then soot emission reduce to half than the baseline engine. As the simulation continues, the region of high soot production is drawn down the side of the piston bowl under the influence of a recirculation flow zone produced by the air-jet. These figures clearly demonstrate the enhanced soot/air mixing produced by the air-jet emanating from the air-cell. This enhanced mixing is responsible for the factor of two reductions in overall soot production for the air-cell engine that which is seen in fig. 8. Also adding EGR and insulation have no noticeable effect on exhaust soot concentration.

A local soot-NO<sub>x</sub> trade-off is evident in these contour plots where the NO<sub>x</sub> and soot formation occur on opposite sides of the high temperature region [36].

## Conclusions

In this paper, at the first step combustion processes, pollutant formation, and the operation characteristics within Caterpillar 3401 DI Diesel engine were simulated at full and part load operation conditions. The results for calculated pressure, exhaust emissions, and performance parameters were compared with the corresponding experimental data and show good agreement. Such verification between the experimental and computed results gives confidence in the model prediction, and suggests that the model can be used at present works. At the second step of CFD simulation in order to improve of performance and emissions at the modified engine, insulating and EGR applying method are used simultaneously. At the result of this step, it was shown that insulating of the combustion chamber walls improves the operation characteristics, creating the air-cell decreases the exhaust soot concentration and adding EGR to the inlet air reduces the NO<sub>x</sub> emission. Comparing the finally modified case (insulated, air-cell connected and 10% of EGR added to the air inlet) to the base Caterpillar engine indicates that the power and IMEP are increased by about 7% and 6.8%, ISFC is decreased by about 6.4%, and soot and NO<sub>x</sub> emissions are reduced by about 50% and 60%.

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