DEVELOPMENT OF A COMBUSTION CHAMBER FOR OPTIMUM COMBUSTION PROCESS IN DIESEL ENGINES

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Abstract

The main process which is directly related with the improvement of performance, fuel economy and emission characteristics of an internal combustion engine is the combustion process itself. In order to find an optimum combustion process that meets these demands a simple Vibe based theoretical computation model was first established. Using this theoretical model some results are obtained and presented in this study. A construction of a suitable combustion chamber that will realize this process is designed and tested in a single cylinder experimental diesel engine. Experiment results of this novel combustion chamber are presented and compared with those results obtained from engine equipped with the standard combustion chamber. Indicated cylinder pressure and related pressure rise and heat release curves of the novel combustion chamber are compared with those of standard engine combustion chamber. The comparison of both type diesel engines is done at the same original maximum power. This maximum power is achieved by 23% lower maximum combustion pressure, with lower ignition retard, with approximately two times lower NO_x and pressure rise values, thus with less noise emission.

Keywords: *Diesel engine, Combustion process, Combustion chamber, Exhaust emissions.*

INTRODUCTION

A direct injection process is widely used in diesel internal combustion engines, especially in those intended to be equipped in passenger cars. This widespread usage is primarily due to the lower specific fuel consumption of the direct injection (DI) diesel engine compared to indirect injection (IDI) engines. Lower specific fuel consumption means lower carbon dioxide (CO_2) emission, the reduction of which, as a way to preserve natural resources, is one of the main goals of many R&D centers and manufacturers.

The direct injection process is associated with high injection pressure fuel supply, which is realized by using common-rail (CR), pump-nozzle systems and solenoid actuated distributor pumps. The common rail system seems to be more preferred one, because it provides a continuous high pressure up to 2000 bar and pressure storage is not restricted to a crank angle window generated by the cam contour, as it is in the case of using the pump-nozzle system.

Disadvantages of DI diesels are the steep pressure rise (noise) and higher exhaust emissions compared to IDI diesels. In order to achieve mild combustion, preinjection of small quantities of fuel and multi-stage (pulsed) injection are applied. Additionally, in order to have high durability of the CR system, the fuel used must have special properties (cetane number (CN) >60, aromatic hydrocarbon amount <20%, final boiling temperature <350 ^{o}C etc.), which raises its cost. Otherwise, if inappropriate fuels are used malfunctions will occur in the system [1]. This situation decreases the attractiveness of the considerably expensive CR system and brings up finding alternative ways, which are more effective in terms of both performance and emissions.

In this study an alternative way to assemble positive features of DI and IDI diesels is described. This can be done by applying a new mixture formation and combustion mechanism, which is performed by using a new designed combustion chamber (CC). A simple Vibe function based computational model is prepared, which allows the investigation the affect of the fuel burning rate or the ôôcombustion lawöö on the performance, nitrogen oxide (NO) and noise emission variations of the engine. By applying a ñlowò, ñfastò and ñoptimumò fuel burning rates predetermined with this model, the performance behavior of the engine is investigated, and theoretically it is predicted that in order to comply with current emission standards it is sufficient to apply the ñoptimumò combustion law without use of additional emission after-treatment devices. To realize this ñoptimumò combustion law, a novel CC, designated as MR-1 CC, is designed and the theoretical predictions are verified by the indicator diagrams of one cylinder experimental diesel engine equipped with a piston having this novel CC.

THEORETICAL MODEL AND ITS RESULTS

In diesels with pre-chamber CC and MAN-M process, it is possible to reduce NO_x emissions and soot by injecting the fuel at relatively lower pressure (<500 bar) over the walls of the CC. However these types of diesels are not widespread, because their fuel consumptions are higher compared to those of direct injection diesel engines. A new combustion system is intended to be developed in order to combine the positive aspects of both direct injection and MAN-M process diesels. For the development of this system a simplified Vibe-type calculation model is used and in addition to engine performance, the nitrogen oxide (NO) is calculated simultaneously. The basic equations used in this model are as follows [2]:

1) Vibe equation which determines the burned fuel fraction during combustion or ñthe combustion lawò

$$x = 1 - \exp\left[-6.908\left(\frac{\alpha}{\alpha_z}\right)^{m+1}\right]$$

2) First Law of Thermodynamic

$$dU = \xi_d\left(H_u - \Delta H_u\right)gdx - dQ_w - pdV$$

3) Ideal gas equation

$$pdV = GRdT$$

4) Cylinder volume according to crankshaft rotational angle

$$V = \frac{V_h}{(\varepsilon - 1)} \cdot \left(1 + \frac{\varepsilon - 1}{2}\right) \times$$
$$\times \left[\left(1 + \frac{1}{\lambda_b}\right) - \left(\cos\alpha + \frac{1}{\lambda_b}\sqrt{1 - \lambda_b{}^2 \sin^2\alpha}\right)\right]$$

In these equations following symbols are used: *dx* ï the burned fuel fraction or ñThe Combustion Lawò calculated by Vibe function; *m* ï the Vibe factor, which represents the fuel burning rate; $_z$ ï the combustion duration in crankshaft angle degrees (^{o}CA); *dU* ï internal energy of gases (kJ/kg$_{fuel}$); H_u ï the lower heat value of fuel (kJ/kg$_{fuel}$) ; H_u ï the amount of heat loss due to chemically incomplete combustion of fuel (when <1), (kJ/kg$_{fuel}$); *g* ï the fuel amount used in one cycle, (kg/cycle); $_d$ ï coefficient of heat loss due to dissociation of combustion products; dQ_w ï heat loss for cooling (kJ/kg$_{fuel}$); *dV*- change of cylinder volume (m^3); *p*, *G*, *R* ï pressure, mass and gas constant of working gas, (MPa, kg, kJ/kg K); V_h ï stroke volume of cylinder (m^3); ï compression ratio; $_b$ ï the ratio of the radius of crank to the connecting rod length; ï the crankshaft rotation angle, (^{o}CA)

5) Equation for NO formation on the basis of ñExtended Zeldovich Mechanismò

$$\frac{dM_{NO}}{d\alpha} = \frac{2\left(1 - \beta^2\right)}{6n}\left[\frac{R_1\left(R_2 + R_3\right)}{\beta \cdot R_1 + R_2 + R_3}\right] \times$$
$$\times \frac{10^3 \cdot p}{M_\Sigma R T_{local}} \quad \text{kmol/}^o\text{CA}$$

Here, R_1, R_2, R_3 are the formation rates of $O+N_2=NO+N$, $N+O_2=NO+O$ and $N+OH=NO+H$ reactions, respectively; =NO/[NO] ï the ratio of real nitrogen oxide amount to the balance amount; [NO], [N], [OH], [O_2], [O], [N_2], [H] ï the balance concentrations of components, (kmol/kg$_{fuel}$); R = 8.314 is the universal gas constant (kJ/kmol^{o}C); *p*, T_{local}, are the gas pressure (MPa) and local (adiabatic flame temperature at stoichometric conditions) temperature (K), respectively.

Here, it should be noted that the local temperature of combustion products (T_{local}) and equilibrium concentrations of the gas components at this temperature are calculated on the basis of the Zeldovich diffusion combustion mechanism.

It is theoretically possible to achieve the requested combustion law for the engine by changing the Vibe equation parameters (m, $_z$) between specified ranges. Thus, optimization studies in relation to the combustion process, which are difficult to conduct experimentally, can be performed considerably easier by computational analysis.

By using this calculation model, theoretical indicator diagrams of a common-rail equipped engine were determined on the basis of its Rail injection pressures and operation parameters. The indicator diagrams obtained experimentally were compared with these models and it was found that the results were very close to each other. The ñcombustion lawò and Vibe equation parameters for diesel engines operating with different fuel injection pressures were determined, respectively [3]. For example, in the case of the high-rate speed combustion law it was found that Vibe parameters are m = 0.30-0.65 and $_z$ = 45-60 ^{o}CA for direct injection turbo-charged diesels with ñopenò type CCs at rail pressures between 1000 and 1500 bar.

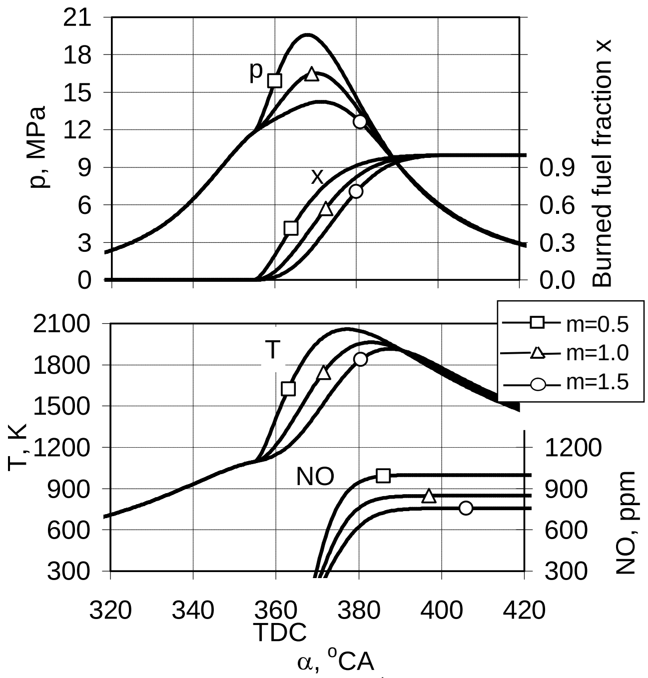


Figure 1: Comparison of indicator diagrams (n = 2000 min^{-1}, start of ignition SOI = 10 °CA, =1.77, p_k/p_0 = 2.73, T_k = 317 K) of the turbo-charged diesel operating at various burning rates (m = 0.5, 1.0, 1.5 and $_z$ = 50°CA)

A series of calculations was performed in order to analyze the effect of Vibe parameters, which essentially represent the fuel burning rate, on the engine performance and emission values by using the above explained calculation model. As an example, Figure 1 shows theoretical curves of the burned fuel fraction (*x*), combustion pressure and temperature (p, T) and nitrogen oxide (NO) versus the crank angle (, °CA) at various burning rates, i.e. with Vibe form factors m = 0.5, 1.0, 1.5 and $_z$ = 50 °CA.

As can be seen from this figure, when the burning rate is high (m = 0.5), the maximum combustion pressure is around p_{Zmax} = 19.5 MPa and the maximum value of the nitrogen oxide is NO = 1050 ppm. However, when burning rate is low (m = 1.5), maximum combustion pressure reaches to p_{Zmax} = 14.5 MPa and nitrogen oxide to NO = 750 ppm. Without doubt, the engine performance is worse in this case.

In case when it is impossible to reduce NO emission to the Euro III and IV standard level by reducing the burning rate, it is necessary to adjust the injection advance of fuel to values lower than the optimum at the expense of worsening engine performance in many situations. In some cases, it is necessary to inject fuel after top dead center (TDC), which is minus advance. In Figure 2 are shown plots of a series of calculation results in order to analyze the effect of the start of ignition (or the fuel injection advance) and the burning rate on the engine performance and the NO emission.

In this figure, the plots of NO, p_{me} and b_e diagrams according to the Vibe parameter *m*, at different start of ignition (SOI) values (-5, 5 and 10 °CA) are shown. As can be seen here, maximum performance values are obtained at the lowest *m* and the highest SOI, i.e. at the highest burning rate. Also nitrogen oxide emissions reach their highest values at these conditions. If we accept a limit value equal to NO = 700 ppm, the maximum limit value that is approximately compatible with Euro III standard, it is impossible to carry NO emissions up to this limit value by any combustion laws which are realized when the SOI is equal to 10 °CA. It is possible to keep the NO emission under the limit value requested by the standard only by operating the engine with a low fuel burning rate if SOI ≤ 5 °CA.

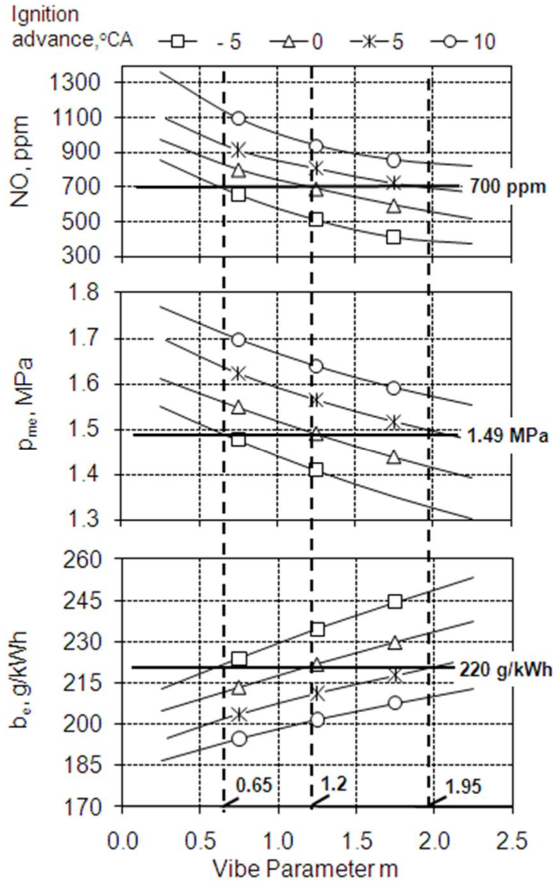


Figure 2: Plots of the nitrogen oxide (NO), the mean effective pressure (p_{me}) and the specific fuel consumption (b_e) vs. the Vibe parameter *m* (burning rate) for various start of ignition (SOI) values.

As shown in Figure 2, it is possible to obtain the performance values of the engine equal to p_{me} = 1.49 MPa and b_e = 220 g/kWh by three different combustion laws (m = 0.65, 1.2 and 1.95) on the condition that the limit value, NO = 700 ppm is not exceeded.

1) The combustion law with m = 0.65 parameter represents high fuel burning rate and it can be realized in ñvolumetric mixtureò diesels with common-rail systems, as explained above. In this case, it is sometimes necessary to inject fuel at three (pre+main+post) and even five stages (2pre+main+2posts) in addition to keeping the rail pressure at a high level (>1200 bar) in order to reduce the soot emission.

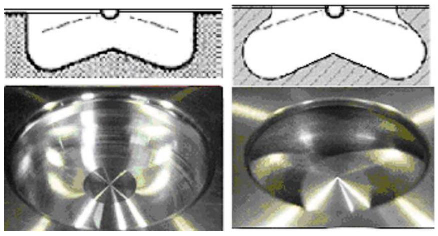
2) m = 1.95 parameter combustion represents low fuel burning rate and it can be used in MAN-M process diesels. In this type of diesels, fuel is injected over the walls of the CC under low injection pressures (<500 bar) through a single or two injector nozzle holes with 95% of it spread over the wall and form a microï thin layer (film) (ñwall guided fuelò mixture) [4]. As found out by experimental studies, direct contact of fuel layer with the wall results in high increase of heat transfer coefficient, and it is possible to occur sufficiently quick evaporation of fuel despite the fact that the wall temperature is almost two times lower than that of the compressed air (400 ^{o}C). When evaporation is carried out in a low temperature environment, fuel protects its natural hydrocarbon structure and is not exposed to a high-temperature pyrolysis [5]. Therefore, unlike in ñvolumetric mixtureò engines, the formation of soot is largely prevented in ñwall guidedò mixture engines before ignition.

Nearly 5% of the fuel droplets which detach from the spray, hence not being spread over the wall, ignite quickly as they are exposed to the hot air in the CCï near its center and it directs the fuel evaporating over the wall to the combustion process without ignition delays, so the noisy burning character of the process is avoided. In this condition, it is possible to keep under control the combustion process in a time period desirable for performance and minimum NO emission according to the rate and amount of injection. As seen here, it is not required to use multi-stage injection under high pressure in order to reduce NO and soot emissions at the same performance values by the ñwall guidedò mixture method. However, as single-nozzle hole injectors are mainly used for engines operating with MAN-M process, the combustion speed with the Vibe parameter m 1.9 is not efficient for engines at n > 2200 rpm engine speeds. Therefore, engines operating with MAN-M process are not preferred for relatively high speed engines.

3) Compared with low speed (m = 1.95) and high speed combustion laws (m = 0.65), combustion law with a parameter m= 1.2 is thought to be the most advantageous. In this case, as shown in Figure 2, possibilities to improve the engine performance values are broader by increasing ignition (or injection) advance without a big increase of NO emission. Moreover, this type of combustion law can be used with no problems at engines operating with n 2200 min^{-1}. Therefore, combustion law with a Vibe parameter m = 1.2-1.5 and $_z$ = 50-60 ^oCA is assessed by us as optimum (appropriate) speed.

COMBUSTION CHAMBER PERFORMING OPTIMUM COMBUSTION LAW

Based on results of a series theoretical investigations carried out and experimental works on various engines, a new CC geometry has been developed in ITU that can perform the optimum combustion law (m = 1.2-1.5, $_z$ = 50-60 ^oCA).



a) b)

Figure 3: Schemes and photographs of standard (a) and MR-1 (b) combustion chambers

Schemes and photographs of the standard and the novel CC are shown in Figure 3. The spray-surface interaction is totally different in both CCs. In the standard CC there is no direct contact between liquid spray and CC wall. On the other hand, the novel combustion chamber, named MR-1, is designed in order to implement the new combustion system. Instead of forming the mixture directly in the CC air volume as is in the case of standard CC, in the MR-1 CC it is formed after spreading the injected fuel over the CC wall and vaporizing it by wallôs heat or in other words ñwall-guidedò fuel-air mixture formation and combustion is applied.

At the moment when the piston is near the end of compression stroke, the fuel is injected towards the bowl wall of the MR-1 CC at low pressures (500 bar) through the minimum 3 and the maximum 5 injector nozzle holes (see Fig. 3). In order to increase wallôs surface area for injected fuel to spread and hence to vaporize rapidly by absorbing wallsô heat, the CC bowl conical angle should be kept at certain values and by application of a helical intake port, the turbulence air swirl with a specified angular speed should be formed in the cylinder. Thus, the fuel spread over the low temperature wall (350 ^oC) can be rapidly vaporized without being exposed to the high-temperature pyrolysis. In order to direct vaporized fuel towards the hottest area ï center of CC and ensure rapid ignition and burning by mixing with air available here, the cross section of the CC has a conical shape at its bottom. Additionally, extinguishing of flame front near the low-temperature wall is avoided by this combustion chamber and by swirling mixture motion generated with the helical intake port. Instead of using multi-hole (7-8) injectors with high injection pressures (>800 bar) as in direct injection diesels, the combustion process is accomplished by using of maximum 5 nozzle-hole injectors and low injection pressures (500 bar). Thus incomplete combustion products PM, CO and HC can be reduced simultaneously. Furthermore, since the majority of the fuel injected per cycle (approx. 90%) is vaporized through the CC wall and burned after mixing with air, the pressure rise rate during combustion is decreased to a certain extent and therefore, NO_x and noise emissions are diminished also.

In order to prove experimentally those expectations anticipated theoretically, combustion analysis was carried out by investigating indicator diagrams of an experimental single cylinder engine ANTOR 3 LD 510 (bore × stroke, 85×90 mm, and compression ratio 17:1) [6]. Purpose of the combustion analysis was to observe and evaluate the cylinder pressure and the fuel injection pressure diagrams of an engine equipped with the ñopenò type standard and MR-1 CC, at various speeds and loads. For this purpose, during the engine tests, the indicator pressure measurement system - consisting of Smetec, AVL and Kistler branded various sensor and data acquisition devices - was used to plot the above mentioned curves. From diagrams obtained in this way, following parameters are determined for the combustion process analysis (Figure 4):

1. Dynamic fuel injection advance ï $_{adv}$
2. Fuel injection period ï $_{fuel}$
3. Maximum fuel injection pressure ï $p_{max-fuel}$
4. Ignition delay ï $_i$, $_i$
5. Start of ignition (SOI) ï $_{SOI}$
6. Maximum combustion pressure ï p_{Zmax}

7. Combustion duration up to p_{Zmax} pressure ï pz
8. Combustion duration ï z
9. Mean combustion pressure rise rate ï p/ =(p_{Zmax} ï p_c)/ pz

Indicator diagrams required for combustion analysis are obtained at n =3000 min^{-1} and n = 2500 min^{-1} engine speeds and various loads (100%, 75%, 50%, 25% and 0%, idle), with static fuel injection advances of 30, 25, 20 and 17.5 ^{o}CA.

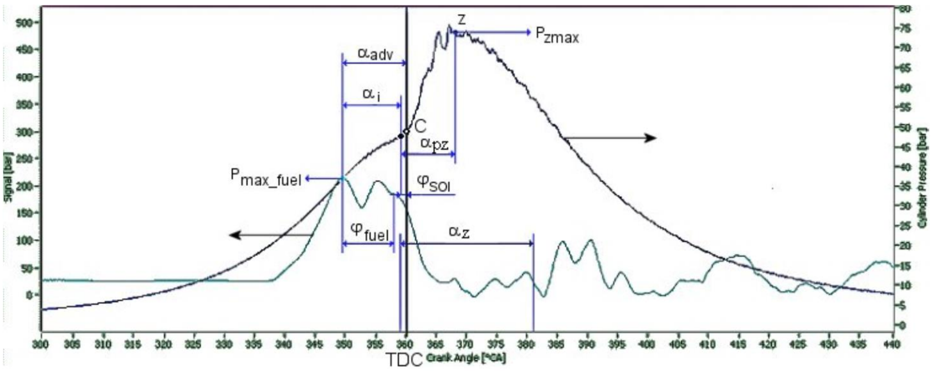
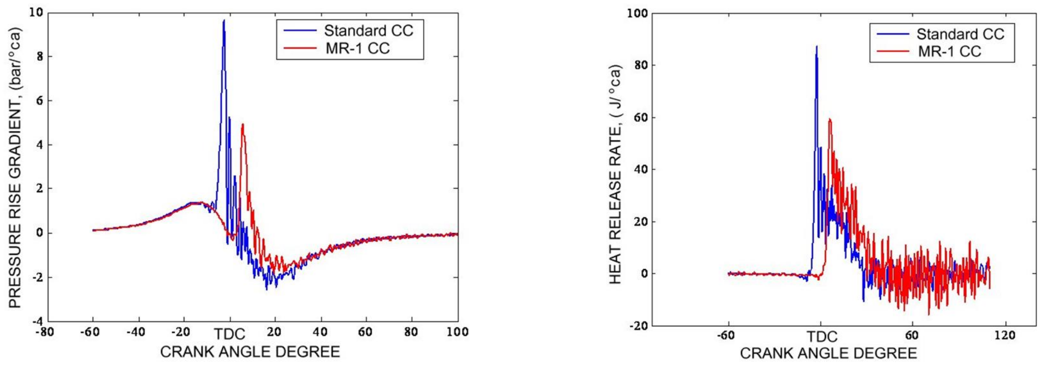


Figure 4: Indicator and fuel line pressure diagrams and parameters taken from them for combustion analysis

Table 1 Comparison of some characteristics of single cylinder experimental engines equipped with Standard CC and MR-1 CC. (n = 3000 min^{-1})

Load %	CC type	P_e HP	p_{mi} bar	p_{zmax} bar	NO_x ppm	(dp/d) bar/^{o}CA	p_{zmax} place, ^{o}CA	(dp/d)$_{max}$ place, ^{o}CA	HR_{max} J
100	STD	11.3	6.91	75.6	1203	9.1	364	358	684
				-23%	-58%	-50%			
	MR1	11.5	6.94	58.5	510	4.6	370	366	722
50	STD	5.64	3.92	69.1	600	8.3	362	358	346
	MR1	5.74	4.02	54.1	216	4.2	371	367	382

In Table 1 are compared some test results characteristics of the experimental engine with standard and MR-1 combustion chamber at 100% and 50% loads, n = 3000 min^{-1}. As can be seen in this Table, at approximately the same power output characteristic values of an engine equipped with MR-1 combustion chamber are much better than the standard ones. NO_x emissions are reduced by 58% as a result of reductions of pressure rise gradient and absolute maximum pressure.



a) b)

Figure 5. a) Pressure rise gradient and b) Heat release rate diagrams of a single cylinder experimental engine equipped with Standard CC and MR-1 CC at full load (n=3000 min^{-1})

In Figure 5 are given pressure rise gradient and heat release rate diagrams of both engine types. A standard CC equipped engine diagramsô peaks are situated slightly on the left side of TDC, whereas those of MR-1 CC equipped engine are slightly on the right side and have a lower maximum values.

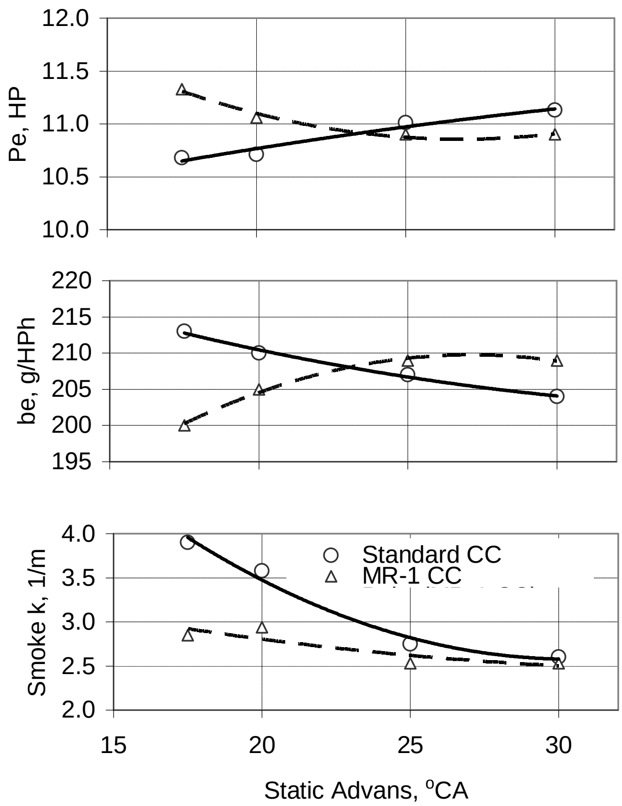


Figure 6: Effective power P_e, specific fuel consumption b_e and smoke number k versus static fuel injection advance (n=3000; Full (100%) load)

In Figure 6, at n = 3000 min^{-1} engine speed and full load (100%) regime, curves of effective power P_e, specific fuel consumption b_e and exhaust gas smoke number k versus static fuel injection advance angles are shown. As understood from this figure, when standard CC is used, best results of the engine from performance and smoke number points of view are obtained at 30 ^{o}CA static advance, however, when MR-1 CC is used, they are obtained at 17.5 ^{o}CA. In order to investigate this feature of MR-1 CC, at tests carried out with MR-1 and standard CC, indicator diagrams obtained at n=3000 min^{-1} engine speed and full load (Figure 7) and at various loads such as 100%, 75%, 50%, 25% and 0% are compared. As can be seen from Figure 6, when engine was operated with MR-1 CC, the optimum dynamic fuel injection advance is $_{adv}$=6 ^{o}CA (static advance 17.5 ^{o}CA) and the ignition delay is $_i$=8.7 ^{o}CA, when the engine was operated with standard CC these values are $_{adv}$=17 ^{o}CA (static advance 30 ^{o}CA) and $_i$=11.2 ^{o}CA, respectively. Although ignition delay is low at operation with MR-1 CC, since dynamic injection advance is also low, combustion process is realized after TDC and the maximum combustion pressure (p_{Zmax}=59 bar) is reached after 10-12 ^{o}CA after TDC. When standard CC is used, combustion pressure reaches its maximum value (p_{Zmax}=77 bar) 5-6 ^{o}CA after TDC and the highest performance is achieved with this adjustment of the engine. Although much lower combustion pressure is formed with MR-1 CC (59 bar against 77 bar, pressure drop 27%) engine power is increased slightly (11.3 HP against 11.1 HP). This is due to combustion process that is realized at a relatively lower speed, in other words at optimum speed when the maximum pressure in MR-1 CC is formed after TDC at most appropriate piston-crank position for the highest torque transfer.

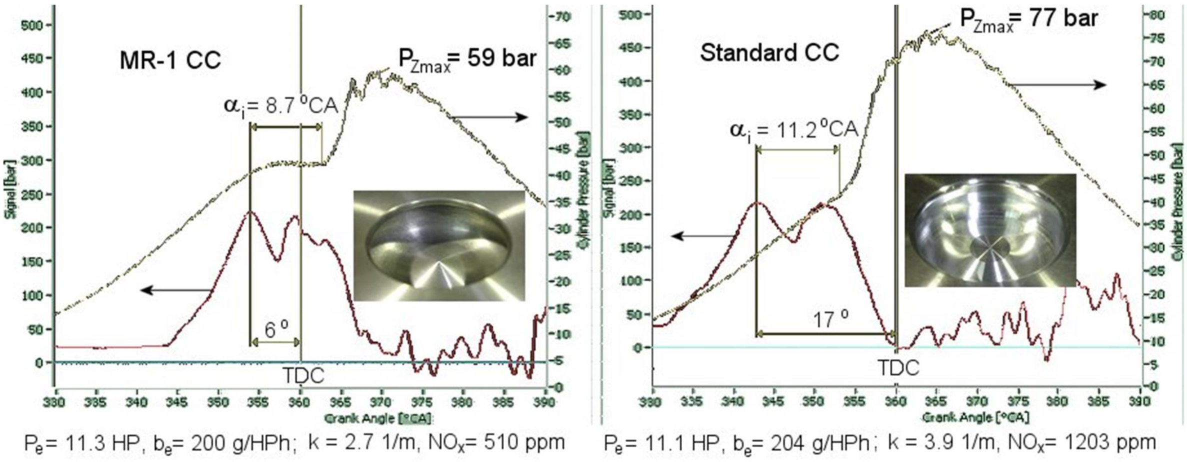


Figure 7: Comparison of indicator diagrams obtained with standard and MR-1 CCs at n=3000 min^{-1} full load

According to crank-piston mechanism theory, maximum value of combustion pressure for formation of maximum moment at crankshaft should be established at 10-12 °CA after TDC [7]. As shown from indicator diagrams in Figure 6, maximum combustion pressure at combustion realized with MR-1 CC at other load regimes also is formed 10-12 °CA after TDC and therefore, engine performance increases and emission values remain low in part load regimes also.

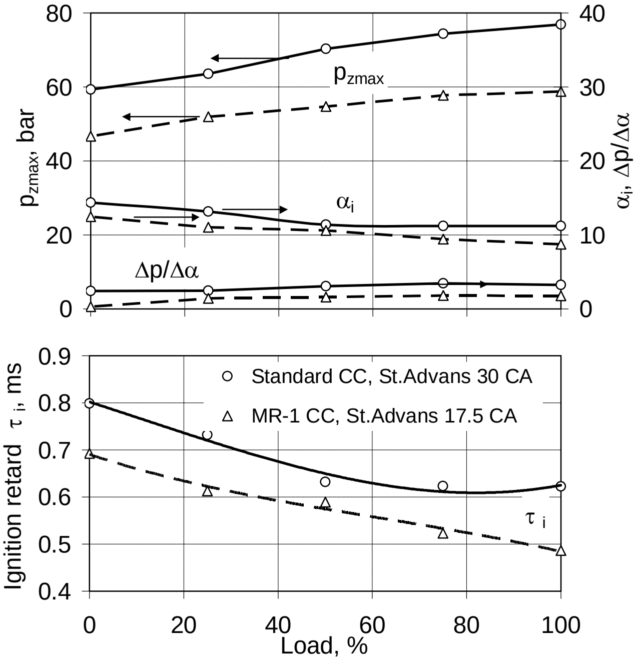


Figure 8: Some combustion process parameters of engines with standard and MR-1 CC vs. load

Variations in mean effective pressure (p_{me}), mechanical, effective and indicated efficiency ($_m$, $_e$ and $_i$) values according to load are shown in Figure 8. According to this figure, when engine operates with MR-1 CC, ignition delay ($_i$ and $_i$), mean combustion pressure rise rate (p/) and maximum combustion pressure (p_{zmax}) are lower than those values obtained with standard CC at all load regimes. However, despite combustion pressures are lower, since more appropriate burning rates are realized with MR-1 CC, engine operates slightly more efficient at all load regimes.

According to test results obtained when engine operates at n=2500 rpm engine speed and different load (100%, 75, 50, 25 and 0) regimes, MR-1 CC realizes optimum combustion speed at 17.5 °CA value of static injection advance. In this situation, maximum combustion pressure is formed 10-12 °CA after TDC. Therefore, engineœs highest performance and lowest emission values are obtained at static fuel injection advance equal to 17.5 °CA as in n=3000 rpm engine speed regime.

Through the combustion analysis, which was carried out with the help of indicator diagrams, it is shown experimentally that MR-1 CC, as claimed above, can achieve optimum combustion process. According to that analysis carried out, majority (>%90) of the fuel injected towards walls of MR-1 CC at a certain advance angle is spread over the wall and rapidly vaporized (when fuel is in contact with CC wall, heat conduction coefficient is approximately hundred times higher). Reduction in ignition delay of fuel ($_i$ and $_i$) indicates that a small portion of injected fuel not spread on the wall (5ï 10%) participates in ignition. Remaining big portion of fuel follows vaporization and mixing with air during combustion process reducing pressure rise rate and sudden burning, thus noise is prevented. Therefore, by reducing mean combustion pressure rise rate (p/), more appropriate combustion process is achieved. In this case, despite the reduction of maximum combustion pressure (at the same time reduction in cylinder temperature), MR-1 CC special geometry mixes turbulent air formed during compression process with vaporized fuel and directs it towards the center of CC where is its the hottest area and performs efficient and low smoke combustion.

With the decrease in maximum combustion pressure (p_{zmax}) and mean combustion pressure rise rate (p/), decrease in NO_x and noise emissions and increase in engine durability is obvious. In this case, during modernization of existing naturally-aspirated engines by turbocharger application, it appears that MR-1 CC shall be useful in order to reduce exhaust emissions and to increase lifetime of the engine. Besides, since this CC completes optimum combustion process with single stage fuel injection through nozzle having number of holes from 3 to 5, diameter >0.3 mm and injection pressure <500 bar, it will allow the common-rail system to be utilized more advantageously, reducing production cost, decreasing service requirement, by using ordinary diesel fuel instead of a special fuel and therefore provides an opportunity to be utilized widely in tractors and heavy duty vehicles as well as in passenger cars.

CONCLUSION

The comparison of experimental test results of a novel combustion chamber with those of a standard one show that the same maximum power is achieved by 23% lower maximum combustion pressure with lower ignition retard and approximately two times lower NO_x and pressure rise values, thus with less noise emission. Thus by optimizing the combustion chamber and combustion process itself it is shown that internal combustion engines have got potentials further to reduce exhaust emissions.

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