DIESEL ENGINES WITH NEW FUEL-AIR MIXTURE FORMATION AND COMBUSTION MECHANISM

ДИЗЕЛОВИ ДВИГАТЕЛИ С НОВ МЕХАНИЗЪМ НА ГОРИВО-ВЪЗДУШНО СМЕСООБРАЗУВАНЕ И ГОРЕНЕ

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Abstract: *One of the most difficult problems to be resolved during the development of diesel engines is to decrease nitrogen oxides, soot (smoke) and particulates in exhaust emissions, without decreasing performance and efficiency, to limits proposed by emission standards, which will be in force in near future. During last years by cooperation and projects with different engine manufacturers Istanbul Technical University developed new fuel-air formation and combustion mechanism that is used in actual combustion chambers of diesel combustion engines. In this paper are presented some theoretical and test results of diesel engines developed by applying this new mechanism.*

Keywords*: Diesel Engine, Mixture Formation Mechanism, Combustion Chamber (CC), Performance, Emissions*

1. Introduction

Problems related to environmental pollution and global warming has become a major scientific and social issue during the past decade. In this context it is proposed to reduce significantly harmful exhaust emission limits of off-road vehicles besides of those of road vehicles. These off-road vehicles such as tractors and other agriculture type vehicles essentially are equipped with diesel engines. When compared with *spark ignition* (SI) engines, these engines have less fuel consumptions by approximately 25-30%. However, since nitrogen oxides (NO_x) and particle matter (PM) emissions are higher in exhaust gases of diesel engines, intensive R&D works are necessary to adopt these engines to future emission standards.

Table 1 shows exhaust emission limits for diesel engines with power ranging 37 kW $\leq P_e \leq$ 75 kW and related stages. From 01/01/2008 according to 2004/26/EC standard applied in Turkey Stage II has been in force, by which it is required to reduce *nitrogen oxides* (NO_x) by 31% (from 9.2 to 7.0 g/kWh) and particles (PM) by two times (from 0.85 to 0.40 g/kWh) on the condition that the carbon monoxide (CO) emission is reduced by around 30% (from 6.5 to 5.0 g/kWh) while the *hydrocarbon* (HC) emission is kept fixed (1.3 g/kWh). The Stage IIIA standard which will be in force two years after this date, expects to reduce hydrocarbon plus nitrogen oxides emissions nearly two times (from HC+NOx= 1.3 + 7.0 = 8.3 up to 4.7 g/kWh) on the condition that particle emissions are kept fixed (0.4 g/kWh). Two years after the Stage IIIA is being in force, the Stage IIB standard will request to reduce particles (from PM=0.4 to 0.025 g/kWh - 16 times), hydrocarbons (from HC= 1.3 to 0.19 g/kWh – nearly 7 times) and nitrogen oxides plus hydrocarbons by around 28% (from HC+NOx= 4.7 to 3.39 g/kWh). Therefore it has to search solutions to the problems and immediately to apply obtained positive results and to continue intensive R&D works for development of off-road vehicle engines in the next 2-3 years. Otherwise manufactured diesel engines will loss its competitive power in domestic and foreign markets.

At the 6th International Diesel Engines Symposium in Italy [1] it has been envisioned to reduce pollutant emissions by using certain technologies such as multi-stage fuel injection (pre-, main and post-) at 1400-2000 bar of pressure from a 6-8 hole injector nozzles by the Common Rail electronic fuel injection system, employing the open type, volume mixture, ω form classical *combustion chamber*s (CCs), EGR application, application of a catalytic converters that operate by injecting urea to reduce NO_x and employment of special filters to reduce smoke and particles for diesel engines of both road and off-road vehicles, in line with information provided by leading engine manufacturers and R&D centres in Europe. Indisputably, despite these measures solve the problem; they are not easy to apply in terms of time, cost, manufacturing and technical facilities. In addition, applying these measures to off-road vehicle engines in the near future is a big question mark even in developed European countries. Therefore, the situation in Turkey is unfortunately not hopeful. Thus finding solutions to the above posed problems in a near future will be remarkably important to the country.

2. Problem definition and current situation

A R&D, design and application projects sponsored by a government are being carried out from *Istanbul Technical University* (ITU) with domestic engine manufacturers in order to develop their diesel engines.

The aim of these projects are to bring performance, fuel consumption, emission and noise quality of these engines up to the level of European standards, to rise their competitive power in domestic and foreign markets and to increase the added values of the engine manufacturers by developing new models that apply the new fuel-air mixture formation and combustion principle protected by patents.

Classical direct injection diesels have ω type CCs that are located on the crown of their pistons (Figure 1). These types of CCs are widely used in both road and off-road vehicle engines. In addition to these types of CCs, a Common Rail electronically fuel injection system as an advanced engine technology for road vehicle engines is becoming more widespread used. The essential reason for this is the availability to automatically change the fuel injection advance according to load and speed regimes, as well as multi-stage injection into the CC at pressures up to 14000-2000 bar. There has been remarkable success in reducing exhaust emissions of high speed passenger vehicles (n=4000-5000 min^{-1}) by the use of the Common Rail fuel injection system. But in engines with these CCs there are excessive increase of NO_x and noise emissions due to high rates of combustion resulted by the injection of fuel at high pressure (>1000 bar) in order to prevent the soot formation. The obligation of keeping these emissions above the limits proposed by current standards requires using additional systems such as the multi-stage injection (pre+main+post) method, fuel injection after the top dead center (TDC), EGR and catalytic converters. Therefore, performance and efficiency characteristics of the engine are worsened [2, 3]. In addition, 7-8 nozzle holes of electromagnetic injector with diameters of 0.12-0.20 mm, and the special software control unit of the Common Rail System have increased the cost and service need of this system too much, which is not suitable for off-road vehicles.

Therefore, many tractor manufacturing European firms (Perkins, Iveco, John Deere etc.) have applied changes to the mechanical fuel injection system in order to catch up with the Stage II and Stage IIIA emission standards, reduced the injection nozzle holes diameter from 0.30 to 0.20-0.23 mm and increased the

Table 1. Emission limit values of engines with power range 37 kW ≤ P_e < 75 kW

Emission test Standard 2004/26/EC	Date in force Turkey	CO (g/kWh)	HC (g/kWh)	NO_x (g/kWh)	Particles (PM) (g/kWh)
Stage I	01/01/2003	6.5	1.3	9.2	0.85
Stage II	01/01/2008	5.0	1.3	7.0	0.40
			HC + NO_x (g/kWh)		
Stage IIIA	01/01/2010	5.0	4,7		0.40
Stage IIIB	01/01/2012	5.0	0.19 + 3.3 = 3.39		0.025

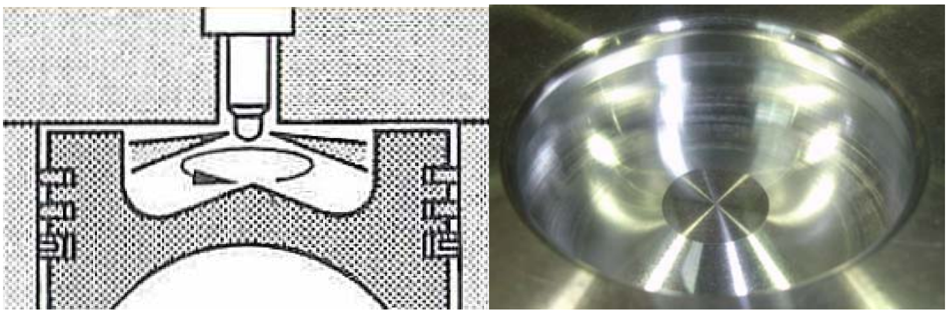

Figure 1. The classical ω type CC of direct injection diesel engine.

number of the holes from 4 to 7 in order to increase the pressure of the fuel injection to the cylinder up to 800 bar, so they succeeded in reduction of soot (smoke) and particles. However, in this case the NO_x emissions reached high values and they are reduced by lowering the fuel injection advance values (even negative values), which resulted in the lower performance and economical characteristics of the engine. These explanations show that the use of the ω type classical CC is not likely to offer any simple and cost-effective solutions to the problem.

In the last few years, ITU has succeeded to develop new mixture formation and combustion mechanisms and CC geometries which realize these mechanisms by carrying out projects in collaboration with domestic engine manufacturers. The conducted theoretical, experimental and application studies have ascertained that diesel engines with this CC are more economical with more power and multi-fuel capability, and have low noise and exhaust gas emissions [4, 5]. Below is given a brief summary of the theoretical and experimental studies related with the development of diesel engines.

3. *Theoretical research and combustion chamber design*

As known from the *internal combustion* (IC) engines theory, efficiency and emission values of diesel engines depend on the rate of combustion. When the combustion rate is increased, the fuel consumption and soot emissions decrease, on the one hand, and NO_x emissions and in-cylinder pressure increase (rise may exceed the endurance limit of the engine structure >120 bar), on the other. The change form of the burned fuel fraction according to crack rotational speed during combustion, or "the combustion law", is the essential factor that determines the combustion speed. At ITU a mathematical method based on the Vibe function has been developed to analyze its impact on the engine efficiency, combustion pressure, and nitrogen oxide and noise emissions and to get a more convenient combustion law [6]. The Vibe function is used in this method to determine the optimum combustion law in theory.

$$x = 1 - \exp\left[-6{,}908\left(\alpha / \alpha_z\right)^{m+1}\right]$$

x: is the burned fuel fraction,
m: is the Vibe form factor,
α_z: is the theoretical combustion duration in crankshaft angle (°CA) unit.

Using this mathematical model a series of calculations has been made to adjust the optimum combustion law for the 4-cylinder turbo diesel engine (S/D=115/104 mm, compression ratio ε=17 and intake pressure rise ratio p_k/p_o=1.8). The Figure 2 shows the amount of the fuel burnt during combustion process and trace of the pressure at three different values of the combustion curve form factor (m=0.5; 1.5 and 3.0). In all three cases, the injected fuel start of ignition is 5 ° *before the top dead centre* (bTDC), that is, the ignition advance of the fuel is θ = 5 *degree crank angle* (°CA) and the engine speed is *n*=1500 min^{-1}, which corresponds to the maximum torque. As seen in this figure and the Table 2 over it, when the high rate of combustion law is applied with the form factor equal to m=0.5 (this law occurs when high injection pressure (>800 bar) from the 6-8 injector nozzle holes is used in the ω type CC) the mean indicated pressure indicating the engine performance (p_{mi}=1.3 MPa) and the indicated efficiency η_i = 0.52 reach the highest values; the maximum combustion pressure rises, which is the most effective factor for NO formation, engine durability decrease and noise is excessively increased (p_{max}=13.7 MPa, dp/dα=0.66 MPa/°CA). When the slow speed combustion law is applied (m=3) at the beginning of the combustion, a small amount of fuel is burned, therefore the combustion process occurs after 360 °CA, that is, during the expansion process. This combustion character is obtained by post injection in order to decrease soot amount when the Common Rail system is used. In this situation, the maximum pressure reduces by 60% (p_{max}=8.6 MPa), the pressure gradient reduces by 140% (dp/dα=0.28 MPa/°CA) while p_{mi} and η_i reduce by 15%. When the performance and economy factors are not taken into account the limit values proposed by NO emission standards can be met by only such a combustion law. However, the economy factor is indispensable, so that operating of the engine by the combustion law with a form factor of m=1.5 is a more accepted alternative because it operates by the optimum law. As seen in the Figure 2, in this situation NO=750 ppm, maximum combustion pressure is p_{max}=9.9 MPa, and the pressure gradient is low (dp/dα=0.28 MPa/°CA); however, the performance and economy values of the engine have not worsened much (p_{mi}=1.2 against p_{mi}=1.3 MPa, and η_i=0.48 against η_i=0.52 – worsening is nearly 8%). Depending on a number of theoretical analysis's and experimental studies on different engines, a new CC geometry which can realize the optimum combustion law (m=1.2...1.5) has been developed in ITU. Figure 3 shows the cross section and a photograph of the new CC (which is symbolically named as MR-1) designed on the basis of the piston structures and the location of the injector on the cylinder head in experimental diesel engines.

Table 2. Change of engine parameters with different combustion laws

θ °CA	m	α_z, °CA	p_{max}, MPa	$dp/d\alpha$, , MPa/°CA	p_{mi}, MPa	η_i	NO, ppm
5	0.5	50	13.7	0.66	1.3	0.52	1480
5	1.5	50	9.9	0.28	1.2	0.48	750
5	3.0	50	8.6	0.28	1.1	0.45	508

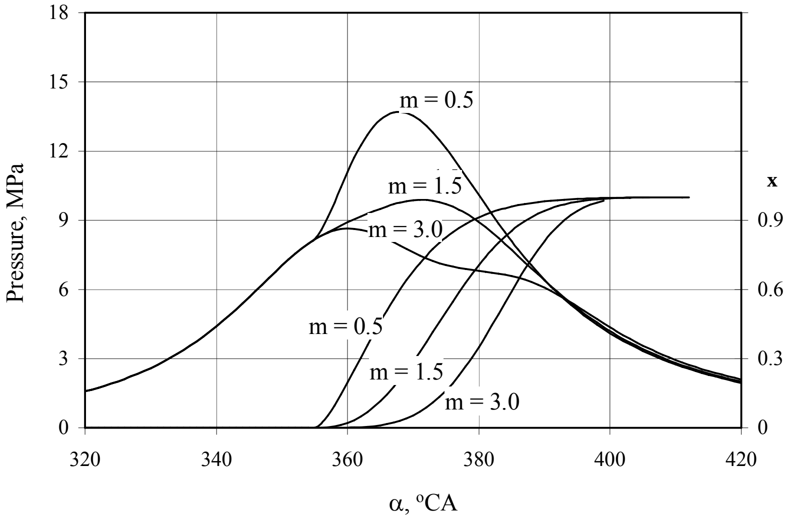

Figure 2. The fraction of the burned fuel and in-cylinder pressure traces.

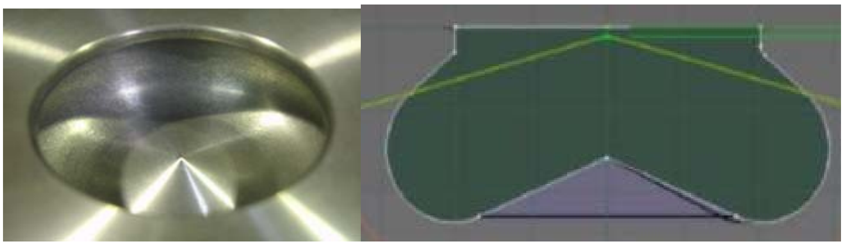
Figure 3. The new MR-1 CC which is used in experimental diesel engines.

In contrast to the classical ω type CC geometry, fuel–air mixture formation in the new chamber occurs after spreading the injected liquid fuel as a film over the CC wall similar as in MAN-M-Process engines [7].

At the end of the compression process, fuel is injected to the walls of the CC, as shown in the Figure 3, at low pressure (<500 bar) at a certain advance angle through maximum 5 nozzle holes. In order to spread injected fuel, it is necessary to increase the surface area of the wall so that the cone angle of the CC and angle of the orientation of the fuel spray to the CC walls is kept within a certain range in order to ensure rapid evaporation of the fuel by the temperature of the walls. In addition, in order to ensure fuel spreading over the wall the rate of the vertical flow of the air must be supplied by choosing correct ratio of piston diameter to bowl diameter between certain amounts, depending on the maximum speed of the engine. Fuel-air mixture formation obtained under these circumstances has the following advantages:

1) The temperature of the CC wall (300-350 °C) is almost twice lower than the end of the compression process air temperature (600-700 °C). However, a large part of the fuel injected per cycle (>%90) can be evaporated easily by spreading over the wall (because the heat transfer coefficient is higher approximately 100 times). The special geometry of the CC mixes the turbulent air formed at the compression process and the evaporating fuel and spurs them to the hottest region – the centre- of the CC so the combustion process ensures efficient combustion which essentially occurs at the centre of the CC.
2) A large part of the fuel is evaporated at a relatively low temperature environment so pyrolisis (change of the hydrocarbon structure of the fuel and its separation into free carbon and hydrogen) process of oil based engine fuels which can occur at temperatures of >400 °C is prevented so that formation of soot and particles(C) is prevented to a great extent.
3) A small part of the injected fuel, which is not spread over the wall, is ignited after a short ignition delay (5% – 10%). Combustion of a large part of the remaining fuel occurs after evaporation and air-mixture processes. Therefore, the pressure rise rate at combustion is moderated to a certain extent and abrupt explosion (such as the rise of combustion pressure when m=0.5 in the Figure 2) is prevented. This is the most important factor which allows reduction of NO_x and noise emissions.

In order to spread the injected liquid fuel as a film over the CC wall, the injection pressure must not exceed 500 bars and the injector holes must not be more than five in order to realize the desired combustion law. In this situation instead of the expensive fuel injection system which is used for direct injection diesels with a lot of holes (7-8 units) and a high injection pressure, an ordinary system is used so that both production and service costs of the engine are reduced.

At the value of m=1.2 which determines the optimum combustion speed in the MR-1 CC, there are performed thermo dynamical calculations of actual cycles of naturally aspirated and turbocharged versions of 4-cylinder diesel engines. The similar computations are done for engines with ω type standard CC, which represents high speed combustion (m=0.5). A part of the calculations results are given in Table 3 and corresponding indicated diagrams are given in Figure 4. When the standard CC is used, the power of naturally aspirated engine is 75 HP, specific fuel consumption is 166 g/HPh, maximum combustion pressure is 92 bar, noise is 92.4 dBA and the nitrogen oxide emissions are 1080 ppm, as are given in the 2nd column of this table. Despite that the power and specific fuel consumption remain the same as in the standard engine, calculated results of the engine with the MR-1 CC are reduced as follows: maximum combustion pressure from 92 to 60 bars (by 34%), the combustion noise from 92.4 to 88.5 dBA (4 dBA) and the NO emissions from 1080 to 588 ppm (by near 2 times).

At the 4th and the 5th columns of the Table 3 are given theoretically obtained results of the turbocharged version of a 4-cylinder diesel engine with MR-1 CC and different compression ratios (ε=17 and 16). As can be seen from the table, by turbo charging power can be increased by around 30% (from 75 to 99-97 HP) without worsening specific fuel consumption, and more importantly without increasing maximum combustion pressure, moreover it can be reduced from 92 to 88 bar (by 4.5%) when the compression ratio is ε=17, and to 83 bar (by 11%) when ε=16; it is also possible to reduce noise from 92.4 to 89 dBA and the NO emission from 1080 to 774 and 669 ppm (almost twice). Thus an important advantage of the MR-1 CC is its ability to prevent excessive increase of combustion pressure, NO and noise emission values in the case of turbocharger application to the standard based engine without additional major construction modifications.

4. *Experimental studies*

A number of adjustment, performance, emission and durability tests have been conducted by assembling a few experimental samples of naturally aspirated and turbocharged versions of 3 and 4 cylinder engines equipped with pistons that have new MR-1 CC and special injectors with nozzle holes of 3 to 5 and diameters of >0.3 mm. In Figure 5 are given photos of these engines coupled to the dynamometer. In order to determine development levels which can be obtained by new pistons through comparison of test results, 6 experiment samples of these engines with standard CC pistons and injector nozzle hole's diameters of 0.23 mm are tested, as well.

Figure 6 shows the specific fuel consumption (b_e), soot (k) and nitrogen oxides emissions traces of 4 cylinder naturally aspirated engines operated at constant speed regime of n=2500 min^{-1} with standard and MR-1 CCs in relation to load. In both cases the amount of fuel injected per cycle and the static injection advance has been adjusted to obtain the same maximum power 75 HP. As seen from the figure, in all load regimes and partial loads essentially, the MR-1 CC, which realizes effective combustion, ensures lower specific fuel consumption. However, smoke and NO_x emissions have not increased but even, they are reduced.

Table 4. Emission test results of naturally aspirated engines with two type CCs

	CO g/kWh	HC g/kWh	NO_x g/kWh	PM g/kWh	Standard level
Standard CC	4.28	0.26	6.03	0.36	Stage II
New MR-1 CC	3.10	0.93	5.20	0.34	

In Table 4 are given the 8 mode emission test results in accordance to 2004/26/EC standard of naturally aspirated diesel engines equipped with standard and MR-1 CCs. Exhaust gas emission values of engines with both CCs are below the limit values of Stage II standard. However, the MR-1 CC reduces *CO*, *smoke* and NO_x emissions simultaneously further; therefore there are more advantageous possibilities to comply with the Stage IIIA standard. In order to realize these possibilities a turbocharger has been added to experimental engines so that it is possible to reduce both *ppm* values of formation and *g/kWh* values of exhaust emissions of pollutants during combustion by using poor mixtures (λ>1.8-2.0) at full load by increasing the specific power.

Table 3. Calculation results of different type diesel engines.

Parameters	Naturally aspirated standard CC, ε=17	Naturally aspirated MR-1 CC, ε=17	Turbo diesel CC-MR-1, ε=17	Turbo diesel - MR-1 CC, ε=16
Power, *HP*	75	76.5	99.1	97.5
Specific fuel consumption, *g/HPh*	166	169	168	171
Maximum combustion pressure, *bar*	92	60	88	83
Combustion noise, *dBA*	92,4	88,5	89	89
NO, *ppm*	1080	588	774	669

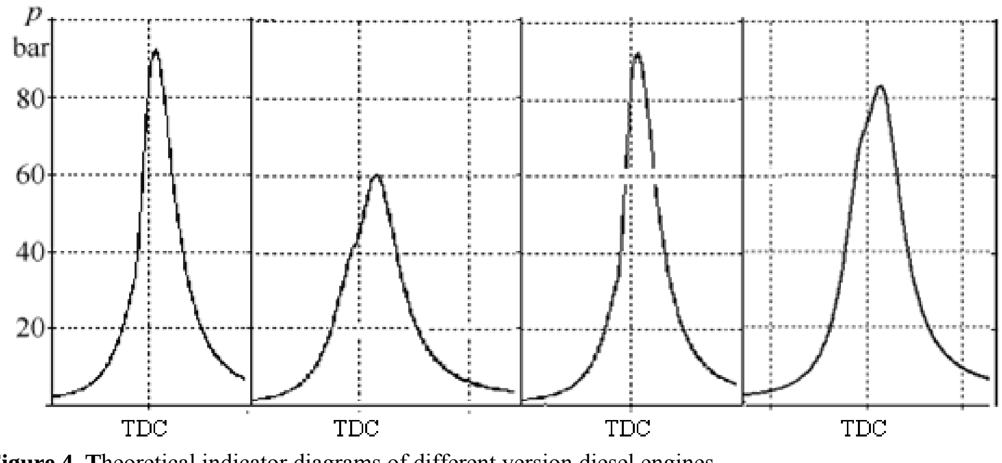

Figure 4. Theoretical indicator diagrams of different version diesel engines.
(Full load, n = 2500 min^{-1})

Figure 5. Photos of 4-cylinder naturally aspirated (on the left) and turbocharged diesel engines.

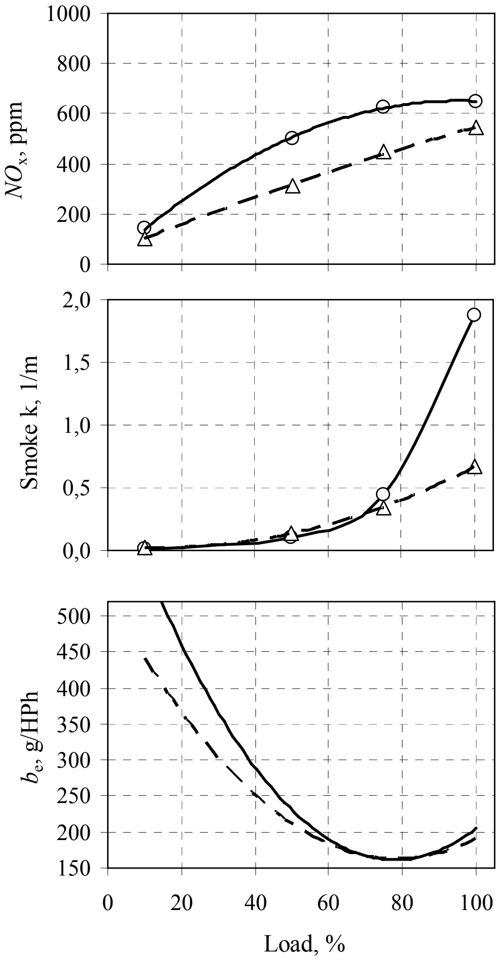

Figure 6. Comparison of load characteristics of 4 cylinders naturally aspirated diesel engines with different CCs
MR-1 CC, 4 nozzle holes injector,
Standard CC, 6 nozzle holes injector.

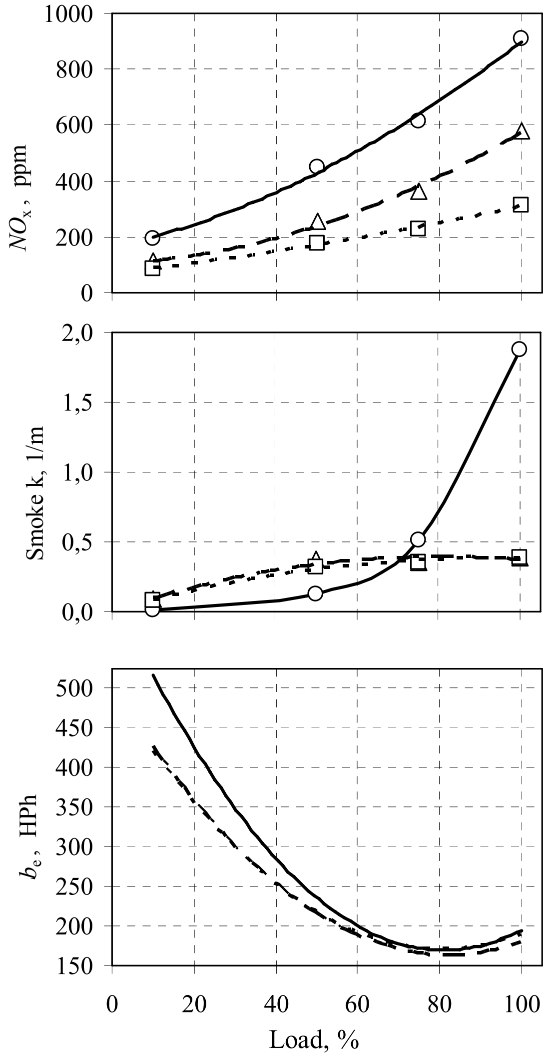

Figure 7. Comparison of load characteristics of 4 cylinder turbocharged diesel engines with different CCs.
Standard CC, 6 nozzle holes injector,
MR-1 CC, 4 nozzle holes injector,
MR-1 CC, Intercooled.

Figure 7 shows changes of specific fuel consumption (b_e), smoke (k) and nitrogen oxides (NO_x) emissions of the turbo charged 4 cylinder experimental diesel engines at constant speed regime of $n=2500\ min^{-1}$ with different CCs. These experimental results for each engine are obtained by adjusting amounts of injected fuel per cycle and injection advance, so that the pressure rise to be $p_k/p_o=1.8$ and the maximum power to be around 90 HP. For the turbocharged engine with the MR-1 CC and equipped with intercooler, which can reduce the air temperature at the compressor exit up to 50 ^oC. By this intercooler the concentration of the air entering the cylinder or air flow of the engine is increased so that the maximum power of 90 HP at a full load regime is obtained by using a leaner fuel-air mixture. As can be seen in the Figure 7, despite the fact that the power is increased by 20% using turbocharger (90 HP against 75), NO_x and Soot emissions have not increased but even they decreased to a certain extent when the engine has MR-1 CCs. When an intercooler is used, both leaning of the mixture and reduction of the local combustion temperatures result in further reduction of NO_x. Unlike this, when a standard CC, which realizes the high speed combustion law, is used, increasing the engine power and combustion pressure rise rate result in increase of NO_x by around 40 % (from 648 to 908 ppm). In addition, even though high speed combustion is an advantage in thermo dynamical point of view for specific fuel consumption (b_e) (or for indicated efficiency) overload over crank-piston mechanism components increases mechanical losses of the engine, thus effective fuel consumption is not reduced, in contrary it is increased compared to the consumption of the engine with MR-1 CC that realizes the effective combustion law. Thus, when the engine performance is improved by using a turbocharger it is experimentally proven that the MR-1 CC is advantageous, as mentioned in the theoretical explanations above.

Table 5. Emission test results of turbocharged diesel engines

	CO g/kWh	HC g/kWh	NO_x g/kWh	PM g/kWh	Standard level
Standard CC	3.01	0.43	8.10	0.30	Stage I.
MR-1 CC	1.30	0.58	5.05	0.29	Stage II.
MR-1 CC + Intercooler	2.02	0.77	3.79	0.33	Stage IIIA

The Table 5 presents the 8 mode emission test results by the 2004/26/EC standard of each three type 4-cylinder turbocharged diesel engines. As can be seen in this table, when the engine equipped with a standard CC is turbo charged, NO_x emissions rise (from 6.03 to 8.10 g/kWh), so the emission quality of the engine reduces to the level of Stage I. However, when the MR-1 CC is used, pollutant emissions are reduced, thus the Stage II level is preserved and even the Stage IIIA is complied with when an intercooler is used.

5. Conclusions

1) Problems related to environmental pollution and global warming has become a major scientific and social issue during the past decade. Therefore it is proposed to reduce significantly harmful exhaust emission values of not only road vehicles but also off-road vehicles such as tractors and other agriculture type vehicles. As in the developed European countries, by taking in force the 2004/26/EC standard in Turkey the dense research and development activities and practical usage of their positive results is required during near 2-3 years in order to develop off-road vehicle's engines to this standard level. Otherwise domestically produced tractor diesel engines will loss its competitive power in domestic and foreign markets.
2) In this contex many European tractor manufacturers (Perkins, Iveco, John Deere etc.) have applied changes to the mechanical fuel injection system in order to comply with the Stage II and Stage IIIA emission standards, by reducing the injection hole diameter from 0,30 to 0,20-0,23 mm and increasing the number of the nozzle holes from 4 to 7 in order to increase fuel pressure injection up to 800 bar, so they have succeeded in reduction of soot (smoke) and particles. However, in this case the NO_x emission reached high values and it became possible to reduce NO_x by reducing the fuel injection advance to values lower than the optimum (even negative values), which resulted in the performance and economical values decrease of the engine.
3) In the last few years, ITU has succeeded to develop new combustion mechanisms and CC geometries which realize these mechanisms by carrying out projects in collaboration with domestic engine manufacturers. By using the MR-1 CC, where optimum combustion speed is performed enables all versions of the experimental engines to have more power, less fuel consumption, low noise and exhaust gas emissions.
4) By a number of R&D studies three versions of diesel engine group, 4 cylinder naturally aspirated, turbocharged and intercooled, have been developed by using the MR-1 CC and a special injectors with 4 nozzle holes. These engines have granted with the Stage II and Stage IIIA approval certificates according to the 2004/26/EC standard.

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