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Modernization of the Tumosan Tractor Diesel Engines by Using New Combustion Chamber

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ABSTRACT: One of the most difficult problems to be resolved in 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 have being more and more stricter. As in 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 occurred as a problem to be solved in future 2-3 years in developments of off-road vehicle's engines. During last years by cooperation and projects with one of the biggest internal combustion engine manufacturers TUMOSAN and other factories Istanbul Technical University developed new combustion mechanisms that are used in actual CCs. In this paper are presented some theoretical and test results obtained during the development of TUMOSAN's diesel engines.

Key words: Diesel engine, combustion chamber (CC), performance, emissions

INTRODUCTION

Today problems related to environmental pollution and global warming are high on the agenda; therefore it is proposed to reduce significantly harmful exhaust emission values of off-road vehicles besides of those of on 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 pollutant 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 emission standards which are becoming stricter.

Since 01/01/2008 according to 2004/26/EC standard applied in Turkey Stage II has been in effect, 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 2,1 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 put into effect two years after this date, expects to reduce hydrocarbon plus nitrogen oxides emissions nearly 2 times (from $HC+NO_x$= 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 put into effect, 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 $HC+NO_x$= 4.7 to 3.39 g/kWh). Therefore it has put on the agenda to search for solutions to the problems and applying obtained positive results immediately by continuing intensive R&D works for development of agriculture type vehicle engines in the next 2-3 years. Otherwise domestically produced tractor diesel engines will lose its competitive power in domestic and foreign markets.

DEFINITION OF THE PROBLEM AND THE CURRENT SITUATION

A R&D, design and application project sponsored by TUBITAK is carried on by the collaboration of TUMOSAN and Istanbul Technical University (ITU) in order to develop TUMOSAN engines.

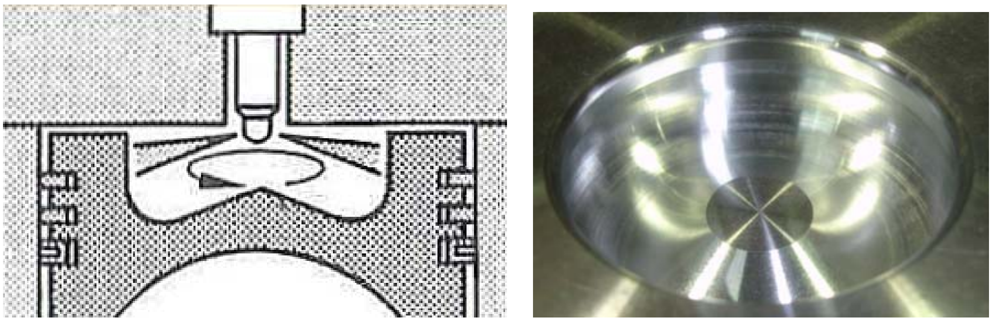

Figure 1. The classical ω type CC of direct injection diesel engine

The aim of this project is to carry performance, fuel consumption, emission and noise quality of these engines up to the level of European standards, to rise competitive power of produced engines in domestic and foreign markets and to increase the added value of TUMOSAN to our country by developing new models using the new fuel-air mixture formation and combustion principle protected by Turkish patents.

TUMOSAN engines belong to the direct injection diesels and the classical ω type CC is located on the piston (Figure 1). This type of CC is widely used in both on road and off-road vehicle engines. But in engines with these CC there are excessive increase of NO_x and noise emissions due to high speed 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 top dead center, EGR and catalytic converter. Therefore, performance and efficiency values of the engine are worsening (Uyehara, 1987; Merola and Vaglieco, 2004).

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, including TUMOSAN. 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 (Mehdiyev et al., 2006; Mehdiyev et al., 2007). Below is given a brief summary of the theoretical and experimental studies related with development of TUMOSAN engines.

THEORETICAL RESEARCH AND DESIGN OF THE CONBUSTION CHAMBER

As shown by the internal combustion (IC) engines theory, efficiency and emission values of diesel engines depend on the speed of combustion. When the combustion speed is increased, the fuel consumption and soot emission decreases, on the one hand, and NO_x emissions and in-cylinder pressure increase might exceed the tolerance limit of the engine structure (>120 bar), on the other. The change in the burned fuel fraction 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 obtain a more suitable combustion law (Mehdiyev et al., 2002).

Using this mathematical model a series of calculations has been made to adjust the optimum combustion law for the 4-cylinder TUMOSAN turbo diesel engine (S/D=115/104 mm, compression ratio $\varepsilon=17$ 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). As seen in this figure and the Table 1 over it, when the high speed 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, which is the most effective factor for NO formation rise, engine durability decrease and noise is excessively increased (p_{max}=13.7 MPa, $dp/d\alpha$=0.66 MPa/^{o}CA). When the slow speed combustion law is applied (m=3) at the begining of the combustion, a small amount of fuel is burned, therefore the combustion process occurs after 360 ^{o}CA, that is, during the expansion process. In this situation, the maximum pressure reduces by 60% (p_{max}=8.6 MPa), the pressure gradient reduces by 140% ($dp/d\alpha$=0.28 MPa/^{o}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 realized 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 ($dp/d\alpha$=0.28 MPa/^{o}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 analysises 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. The 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 TUMOSAN 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 or the wall guided fuel-air mixing method as in MAN-M-Process engines is applied (Meurer, 1956).

In order to spread the injected liquid fuel as a film over the CC wall, the injection pressure must not exceed 500 *bar*s 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 TUMOSAN 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 the Table 2. 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 *dB(A)* and the nitrogen oxide emissions are 1429 *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 *dB(A)* (4 *dB(A)*) and the *NO* emission from 1429 to 588 *ppm* (by 2.4 times).

At the 4th and 5th columns of the Table 2 are given theoretically obtained results of the turbo version of a 4-cylinder TUMOSAN 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* (4.5%) when the compression ratio is ε=17, and to 83 bar (11%) when ε=16; it is also possible to reduce noise from 92.4 to 89 *dB(A)* 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 turbo charging applied to the standard based engine without additional major construction modifications.

EXPERIMENTAL STUDIES

A number of adjustment, performance, and emission and durability tests have been conducted at the R&D laboratories of the factory together with ITU stuff by assembling a few experimental samples of naturally

Table 1. Change of engine parameters with different combustion laws.

$\theta^o CA$	m	α_z^o 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

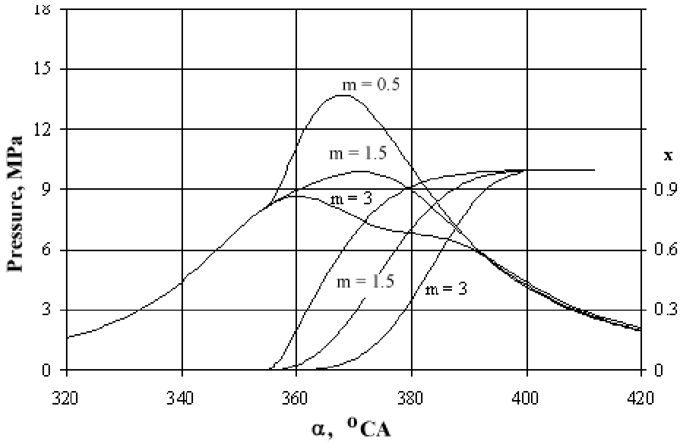

Figure 2. The fraction of the burned fuel and in-cylinder pressure traces.

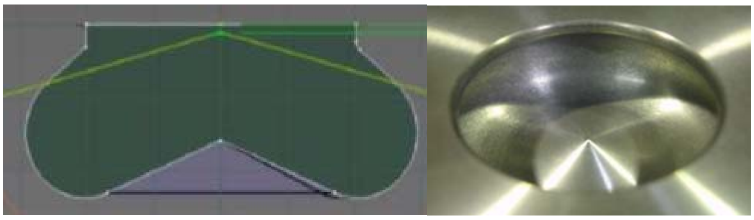

Figure 3. The new MR-1 CC which is used in TUMOSAN engines

Table 2. Calculated parameters of TUMOSAN diesel engines.

Parameters	Naturally aspirated standard CC, ε=17	Naturally aspirated MR-1 CC, $\varepsilon = 17$	Turbo diesel CC-MR-1, $\varepsilon = 17$	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, *dB(A)*	92,4	88,5	89	89
NO, *ppm*	1080	588	774	669

aspirated and turbo version of 3 and 4 cylinder TUMOSAN engines by the pistons with the new MR-1 CC and special injectors with nozzle holes diameter of >0,3 and a number of 3 to 5. 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 a hole diameter of 0.23 mm are tested, as well.

In the Table 3 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*, *Soot* 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 turbo charging has been applied to experimental engines so that it is possible to reduce both *ppm* values of formation of pollutant emissions at combustion by using poor mixtures (λ>1.8-2.0) at full load and *g/kWh* values of exhaust emission by increasing the specific power.

Table 3. Emission test results of TUMOSAN naturally aspirated engines

Emissions, g/kWh	CO	HC	NO_x	Soot (PM)	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	Stage II

The Figure 4 shows changes of specific fuel consumption (b_e), soot (k) and nitrogen oxides (NO_x) emissions of the turbo charged 4 cylinder TUMOSAN 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 turbo charged engine with the MR-1 CC with intercooler, which can reduce the air temperature at the compressor exit up to 50 ^oC is used. By doing so, 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 4, despite the fact that the power is increased by 20% by turbo-charging (90 *HP* against 75), NO_x and *Soot* emissions have not increased but even decreased to a certain extent when the engine works with the MR-1 CC. 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 that realizes the high speed combustion law is used, with the increase of the engine power that causes excessive rising of the combustion pressure result in increase of NO_x by around 40 % (from 648 to 908 *ppm*). In addition, even high speed combustion is advantageous 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 so effective fuel consumption is not reduced, in contrary it is increased compared to the consumption of the engine with

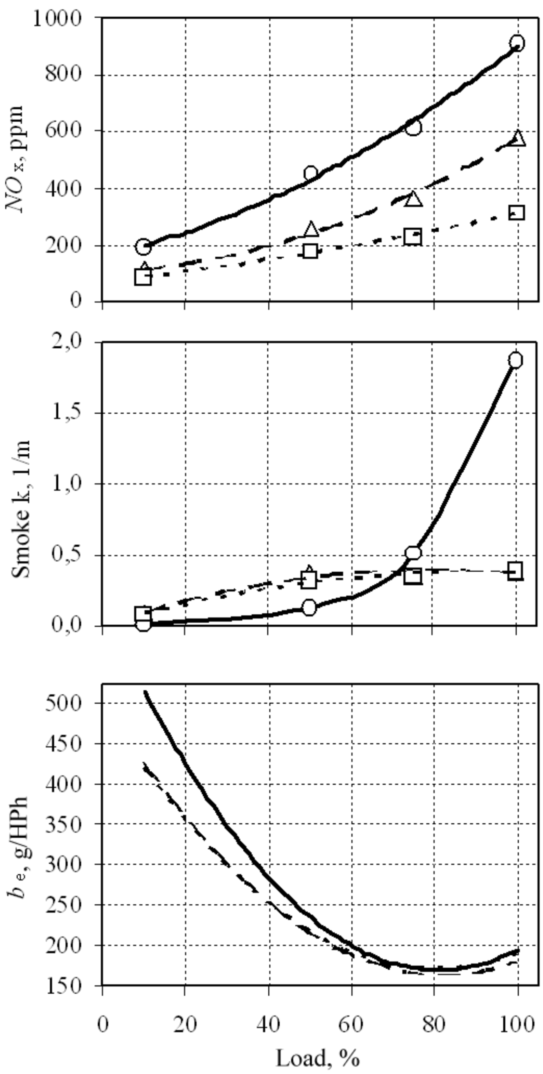

Figure 4. Comparison of load characteristics of TUMOSAN 4 cylinder turbocharged diesel engines with different CCs.

Standard CC, 6 injector nozzle holes,
MR-1 CC, 4 injector nozzle holes,
MR-1 CC, Intercooled.

MR-1 CC that realizes the optimum combustion law. Thus, when the engine performance is improved by turbo-charging it is experimentally proven that the MR-1 CC is advantageous, as mentioned in the theoretical explanations above. The Table 4 presents the 8 mode emission test results by the 2004/26/EC standard of each three 4-cylinder TÜMOSAN turbo diesel engines. As seen there, 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.

Table 4. Emission test results of TÜMOSAN turbo diesel engines

Emissions, g/kWh	CO	HC	NO_x	Soot (PM)	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

CONCLUSIONS

1) 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, including TUMOSAN Company. The MR-1 CC geometry has been used at the first studies on the development of TUMOSAN diesel engines. This CC, where optimum combustion speed is performed enables all versions of the engine at the factory to have more power, less fuel consumption, low noise and exhaust gas emissions.
2) By a number of R&D studies three versions of TUMOSAN engine group, 4 cylinder naturally aspirated, turbo charged and intercooled, have been developed by using the MR-1 CC and a special injectors with 4 nozzle holes. According the 2004/26/EC standard they obtained the Stage II and Stage IIIA approval certificates

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