POLLUTANT EMISSIONS REDUCTION OF TRACTOR DIESEL ENGINES

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Abstract: *In order to meet the European Community standard requirements the TUMOSAN Company, manufacturer of agricultural tractor engines, in cooperation with Istanbul Technical University takes part in the projects that are supported by TUBITAK. The aim of these projects is to develop a new generation, patent protected new tractor engines on the basis of existing production TUMOSAN diesel engines and to comply with 2004/26/EC standard requirements. In these studies, four valves per cylinder, cylinder heads with specially designed intake and exhaust ports and a new fuel-air mixture formation and combustion mechanism are applied, simultaneously are improved performance, fuel consumption, emissions and noise characteristics of these engines. This paper gives information about theoretical and experimental studies carried out to develop these engines.*

Keywords: Tractor diesel engine, combustion mechanism, 4-valves per cylinder, performance, emissions

1. Introduction

TUMOSAN Company is one of the major engine manufacturers in Turkey where are being produced 8 versions of 3- and 4-cylinder water-cooled diesel engines with a total piston displacement range from 2930 cm^3 to 3908 cm^3. These engines are basically used on TUMOSAN branded tractors. R&D, design and application projects are being carried out in cooperation between Istanbul Technical University (ITU) and TUMOSAN Company with the support of TUBITAK (The Scientific and Technical Research Council of Turkey). As a result are attained the exhaust emission limit values of Stage IIIA of the European Community directive 2004/26/EC, which is in force this year in Turkey [1]. The Stage IIIB standard which will be in effect after two years, states that exhaust gas particles should be reduced (*PM*) by 16 times (from 0.4 g/kWh to 0.025 g/kWh), the carbon monoxide (*CO*) emission should be reduced by around 7 times (from 1.3 g/kWh to 0.19 g/kWh) and hydrocarbon plus nitrogen oxide's emissions ($HC+NO_x$) should be reduced by around 28% (from 4.7 g/kWh up to 3.39 g/kWh). Therefore, it is important to look for solutions to the problems and then immediately apply obtained positive results and to continue intensive R&D work for development of non-road vehicle engines for the next couples of years. Otherwise, manufactured diesel engines will lose their competitive power in domestic and foreign markets.

Considering this, TUMOSAN is carrying out projects in cooperation with ITU and with the support of the TUBITAK. These projects aim to develop new generation tractor engines with ultra-low emission based on the existing TUMOSAN diesel engines. Moreover, it is aimed to bring performance, fuel consumption, emission and noise quality of these engines up to the level of European standards with the new fuel-air mixture and combustion mechanism through use of a new cylinder head with 4 valves per cylinder and a special design intake and exhaust ports.

A brief summary of the theoretical and experimental studies related with the development of the diesel engines is given below.

2. Theoretical Studies

Theoretical studies have been carried out using a mathematical model of the actual cycle of a diesel engine which was developed by authors. Vibe equation, which determines the burned fuel fraction or "the combustion law", was used with this model and detailed explanations of this model are given at references [2, 3]. It is theoretically possible to achieve the combustion law wanted for the engine by changing the Vibe equation parameters (m, α_z) among allowable limit. This makes the optimization studies, which are difficult to conduct experimentally, related to the combustion process considerably easy by the use of a computer; thus enabling us to find the required parameters of TUMOSAN tractor engines before design.

As it is mentioned at [1, 3], it is possible to bring the emission standard of TUMOSAN classical diesel engines with 2 valves up to Stage IIIA by carrying out "the optimum combustion law" using the single swirl MR-1 fuel-air mixing formation and combustion mechanism (shortly MR-1 combustion chamber). It is determined with a combustion analysis which is made with the help of experimental indicator diagrams, that the optimum combustion law forms theoretically at m=1.2 and α_z=55 ^{o}CA values of the Vibe equation, and in the case of the mixture igniting close to the TDC (ignition advance θ=~0 ^{o}CA). With all these taken into account, the investigation of a newly 4 valve engine is done with vibe parameters constant at m=1.2 and α_z=55 ^{o}CA, ignition advance θ=~0 ^{o}CA. The main purpose of theoretical analysis is to determine the optimum working parameters keeping the *NO* emissions below the 300 ppm of the newly developing Stage IIIB engine with 4 valves per cylinder with having nearly 15-20% more power (105-110 HP versus 90HP) comparing with Stage IIIA engine with 2 valves per cylinder at the full load regime. Another criterion is not to exceed 12-13 MPa (120-130 bar) of the combustion pressure to conserve the engine block durability. Some of the results about the theoretical analysis are given below.

From the internal combustion engines theory, when the engines with 2 valves per cylinder are converted to the 4 valves per cylinder, engine's performance and economy get better with increasing the volumetric efficiency. According to the fluid dynamic analysis results (details are given at the "3. Design Studies" part), when 2 intake valves are applied to the 4-valves engine, the main reason of the higher volumetric efficiency (η_v) is the decrease of the air flow speed (v_d) into the cylinder from 83.28 to 63.34 m/s with the larger intake cross section area. In this case, volumetric efficiency can be increased from 0.85 to 0.9.

In Figure 1 are shown theoretically calculated graphics of engine parameters versus excess air factor (λ) at different values of volumetric efficiency. As for the calculations, turbo max. pressure ratio p_k/p_o =1.8, compression ratio ε=17, engine speed n=2500 rpm and intercooler outlet or cylinder intake temperature t_{in}=50°C are used, like 2-valves Stage IIIA engine's parameters.

As it is seen from Figure 1, when the air excess factor (λ) increases (leaning the mixture), effective mean pressure (p_{me}) and NO emission decrease, specific fuel consumption (b_e) increase. When the volumetric efficiency is increased from η_v=0.85 to 0.9, engine performance and efficiency increase and on the other hand *NO* emission also increases.

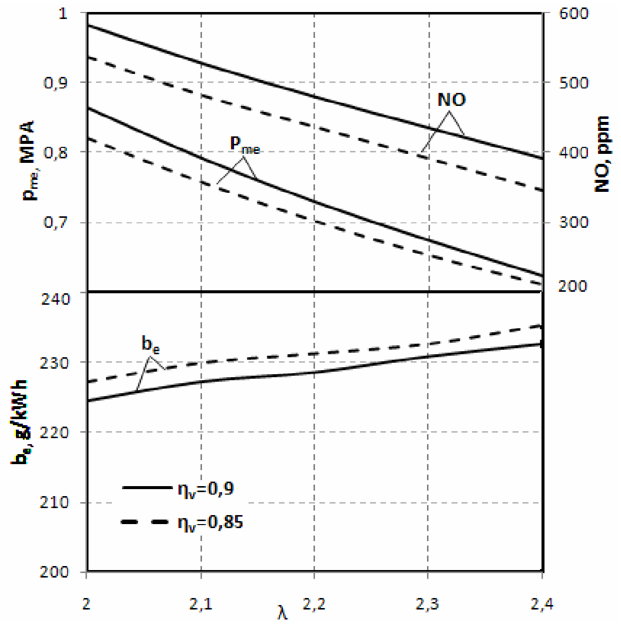

Figure 1. *The change in the engine parameters versus the air excess factor (λ) (p_k/p_o=1.8; ε=17; n=2500 rpm; t_{in}=50 °C).*

For example, at λ=2.2, when effective mean pressure increases from 0.836 to 0.88 MPa, effective power increases (%6.5) from 92 to 98 HP, specific fuel consumption decreases (%1.3) from 231 to 228 g/kWh, *NO* emission increases (%9.6) from 301 to 330 ppm.

As seen from Figure 1, to hold the *NO* emission under 300 ppm, air-fuel mixture should be leaner (like λ=2.3). However, now newly developing engine must keep the intended performance values. Lots of calculations are made to determine the improvement rates of the engine's performance values by increasing the compression ratio at the constant air excess factor λ=2.3. Some of the calculation results are shown at Figure 2.

As shown, when compression ratio (ε) increases from 16 to 19, although engine performance and efficiency values get better, *NO* emission increases considerably. It is seen that *NO* increases faster at ε>18. For that reason and also to improve quality of the engine start on cold weather, compression ratio of newly developing engines is increased by only one unit to ε=18.

The other method to improve the engine's performance values is by increasing the turbo max. pressure ratio (p_k/p_o), so some calculations are made for the different values of it. At Figure 3, effective mean pressure (p_{me}), *NO* emission and specific fuel consumption (b_e) parameter variations depending on the p_k/p_o are shown.

As shown in this figure, with the increase of p_k/p_o, both performance values of engine and *NO* emissions increase considerably. At p_k/p_o=2.2, acceptable results are calculated for the engine performance. Besides that, *NO* emission does not exceed 300 ppm at that turbo maximum pressure ratio value.

Another effective method to improve the engine performance without increase of the *NO* emission is the useof an intercooler to decrease the turbocharger air temperature. At Figure 4, depending on the t_{in}, engine parameters and *NO* emission changes are shown. Depending on the calculation results, turbocharger air temperature should be decreased to around 50-55 °C. So, as a result of the theoretical studies, optimum working parameters of newly developing Stage IIIB engine with 4 valves per cylinder are as these λ=2.3; p_k/p_o=2.2; η_v=0.90; ε=18; n=2500 rpm; t_{in}=50-55 °C.

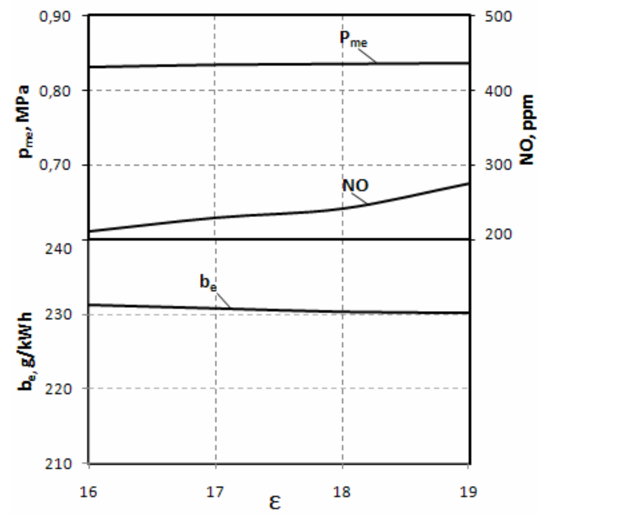

Figure 2. *The change in the engine parameters versus the compression ratio (ε) (λ=2.3; η_v =0.90; n=2500 rpm; p_k/p_o=1.8; t_{in}=50 °C).*

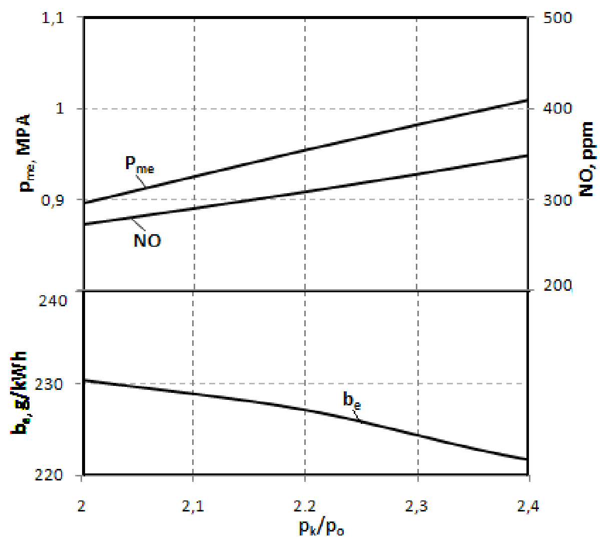

Figure 3. *The change in the engine parameters versus the turbo maximum pressure ratio (p_k/p_o) (λ=2.3; η_v=0.90; ε=18; n=2500 rpm; t_{in}=50 °C)*

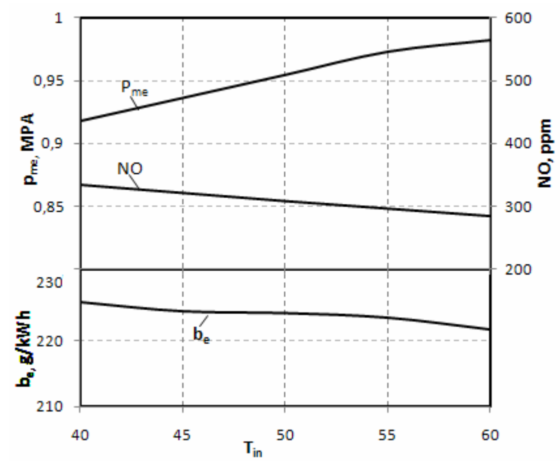

Figure 4. *The change in the engine parameters versus the intercooler output temperature (t_{in}) (λ=2.3; p_k/p_o=2.2; η_v=0.90; ε=18; n=2500 rpm)*

With these values in mind, change of intended theoretical indicator diagram (p), the burned fuel fraction ("the combustion law"-x), mean and local temperatures in the cylinder (T, T_L) and *NO* emission versus crank angle (CA) are shown at Figure 5. As seen, with combustion period duration α_z=55 °CA, maximum combustion pressure is 12 MPa, max. combustion temperature is 1800 K, max. local combustion temperature is 3000 K, max. *NO* emission is 296 ppm. Under these conditions, anticipated values of the engine power is 78 kW (106 HP), maximum torque is 330 Nm at 1500 rpm, specific fuel consumption is 223 g/kWh (164 g/HPh). Combustion pressure's maximum value being lower than 12MPa means that we don't need to make any changes on the engine block and the crank shaft, which allows us to keep the modification cost lower.

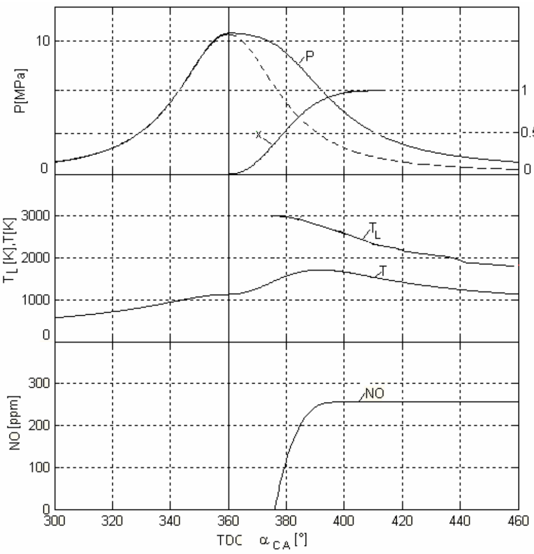

Figure 5. *The indicator diagram of the newly developing Stage IIIB diesel engine*

3. Design Studies

In the design period of the newly developing engine, as a precondition there wouldn't be any changes on the cylinder block and crank shaft of the current engine with 2 valves per cylinder, only cylinder head and current piston would be changed. As a first step, it was planned that a single swirl MR-1 combustion mechanism would be used while designing these parts.

As can be seen from Figure 6, to be able to use the MR-1 combustion mechanism on the engine as mentioned with a fork shape and double intake port looking at the same side must be applied having the same intake area at the cylinder head. Making these ports looking at the same side turns the intake air into a one swirl. As a second step, at the newly developing twin swirl MR-2 combustion mechanism, a double spiral port application facing the opposite sides should be placed. Making these ports facing the opposite side turns the intake air into two swirls with opposite direction and same speed.

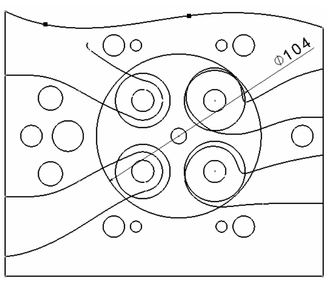

Figure 6. *Schematic locations of the double intake and exhaust ports on the cylinder head*

Spiral intake ports generally start with straight form and go on with the spiral form occurred swirl, finally they finish with a cylindrical port at the combustion chamber. Sucked air goes on straight on its way at first, after it is transferred to the spiral area and swirl motion is carried out in this area, lastly, air passed from the cylindrical port is transferred to the cylinder or combustion chamber.

If the spiral ports are designed properly, swirl motion can be given to intake air without disturbing the efficiency of the flow. Our aim with this study is to design a double spiral intake port without decreasing the flow efficiency or volumetric efficiency by double and single swirl flow.

For better sweeping of the residual gases from the cylinder at the exhaust period and taking place of the injector nozzle at the center of the combustion chamber, two exhaust valves must be chosen. For the less resistance against the gas flow of the exhaust ports, they should be with cylindrical profile as in the classical applications. Double exhaust port having the same outlet area with the fork shape should be in the same symmetry axis as shown at Figure 6. By designing the intake and exhaust ports by this way, we get the combustion period at the intended air condition and increase the valve numbers, and thus increase the volumetric efficiency of the engine. Because of the two intake valves at the cylinder head of the newly developing engine, the total cross-section area of the double intake port increases; therefore, crossing velocity of the air charge to the cylinder decreases. So that reducing the air fluid resistance at the cylinder reduces the pressure loss and then increases the engine volumetric efficiency.

Instead of single intake valve (diameter = 45mm) used at the current engine with two valves, two intake valves with the maximum diameters (diameter=39.3 mm) are placed at the design studies. At the inlet part to the combustion chamber, the original port pass way diameter *($d_{b\ former}$=37 mm)* is 37 mm. Similarly, at the four valves new engine (d= 39.3 mm) that value *(d_{b_new}=30 mm)* is 30 mm.

Cross-section areas of the inlet part of the intake port to the cylinder;

$F_{b\_former} = \pi(d_{former})^2/4 = 1.07*10^{-3}\ m^2$ and

$F_{b\_new} = \pi(d_{new})^2/4 = 0.75*10^{-3}\ m^2$

Conical gap area when the valves are opened;

$F_{s\_former} = F_{b\_former}/1.1 = 0.97*10^{-3}\ m^2$

$F_{s\_new} = F_{b\_new}/1.1 = 0.68*10^{-3}\ m^2$

Inlet velocity of the air charge to the cylinder when the cylinder's mean velocity v_p is known;

$v_d = v_p F_p / i_s F_s$

i_s, intake valve number

$F_p = \pi D^2/4 = 8.49*10^{-3}\ m^2$ *,cylinder cross-section area (diameter(D)=*104 *mm).*

When engine speed *n*=2500 *rpm* and cylinder stroke *S*=115 *mm* values are used, with the equation $v_p = nS/30$, cylinder mean velocity is found as 9.58 *m/s.*

The inlet velocity of the air charge to the cylinder is $v_{d\ former}$*=83.28 m/s* on the original engine with a single inlet valve, the velocity is found as $v_{d\ new}$=63.34 *m/s* on the engine which has 4 valves. As seen, the inlet velocity of the air charge to the cylinder decreases on the engine with double intake valves by 24% from 83.28 to 63.24; because of the increase of the total cross-section area.

With the thermodynamic calculation of the actual cycle it was seen that, reducing the air charge velocity from 83.28 to 63.24 causes an increase in the volumetric efficiency from 0.84 to 0.9; and with the help of the theoretical studies above, it is proven that, this increase helps to improve the engine performance and efficiency.

Intake port's geometry is shaped with an applied fluid dynamics analysis on the Ansys CFX program with the aim of catching the same inlet velocity value (v_d= 63.34 *m/s*) of the air charge sent to the cylinder of the single and twin swirl engine with four valves. Mean velocity was found as 64.56 m/s with the Ansys CFX program and 63.34 m/s with the analytical method at the outlet of the ports last geometry; which made sure that the values matched. The streamlines, the single and twin swirl ports made, as seen fit for the core of the intake ports are shown in Figure 7. This port geometry is originally an engine cylinder head with 4 valves which was designed on the CATIA program. Casting pattern has been shaped considering the port geometries and current connection places with engine block of cylinder head. After determining, all the details and connections of the external lines of the cylinder head of the cooling water line, it was designed from the outside to the inside including the wall thickness for the casting.

At Figure 8, the pattern geometry for the cooling water passage and at Figure 9, a top-down cross-section look at the intake and exhaust ports of cylinder head is shown.

Due to the difference of the number of valves of the cylinder head, intake and exhaust ports, the new valve set mechanism driving the four valves and the manifolds fitting the ports are redesigned.

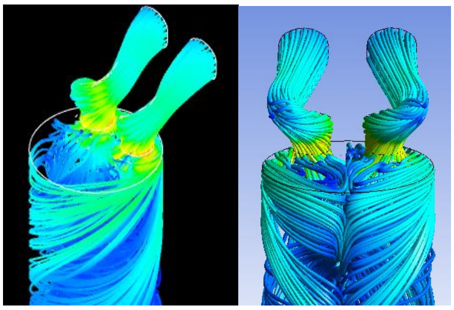

Figure 7. *Air flow lines of the intake ports for the single and twin swirls*

Designs are shown as an assembly at Figure 10.

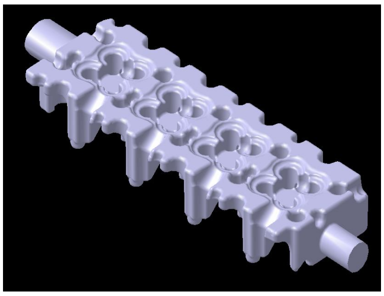

Figure 8. *The pattern geometry of the cooling water passage.*

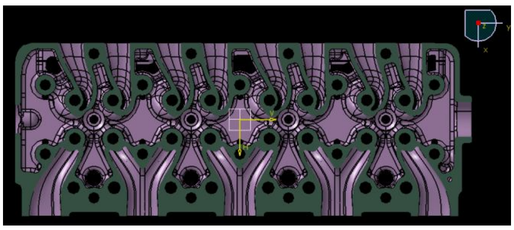

Figure 9. *A top-down cross-section look at the intake and exhaust ports of cylinder head.*

At the first stage, the design of the combustion chamber was formed based on the MR-1 single swirl combustion chamber. With the experimental studies, it was found out that as a disadvantage in the winter and especially when the air temperature is under -5 ^oC, starting time of the engine gets longer and makes it hard to start the engine. After the engine starts, for a short time period (until the engine temperature reaches to its normal values) engine exhaust is thrown out as white smoke which is rich in hydrocarbon compositions (*HC*) or as unburned fuel, which causes air pollution and raises fuel consumption. The cause of this disadvantage is that the temperature of the combustion chamber is inadequate for a while to vaporize the fuel spread over the cold combustion chamber wall. To eliminate these disadvantages with the newly developing four valves engine's combustion chamber, a new geometry which bring together advantageous sides of the "stratified mixture" and "volumetric mixture" is suggested and this combustion chamber is patented as the TR patent number: B.14.1. TPE. 2009/09240,2009-G-265649.

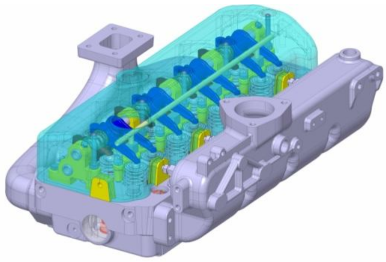

Figure 10. *3D model figure in assembly of manifolds and valve actuation mechanisms with cylinder head (double shaft, with fork shaped valve rockers)*

MR-1/V2 combustion chamber is shown in Figure 11. The special feature of this combustion chamber is that, it consists of two rotational symmetrical volumes which are a main combustion chamber and a pre-combustion chamber. These chambers' symmetry axis is on the same direction; their corners are a rounded truncated cone-like structure and they have a conical protrusion at the bottom. At the start and idle regimes of the engine, injection advance of fuel is set to its maximum values. At this setting, fuel spray is completely injected into the pre-combustion chamber with a definite routing angle according to TDC position of the cylinder creating a fuel-air mixture by the volumetric mixture method and it burns fast without white smoke approximately at 650–800 ^oC air temperature, even in winter.

When the engine is working under partial load regimes, fuel spray advance decreases a little to meet a middle value. Under this position that the cylinder is in according to the TDC, some of the fuel spray is spread over the bottom part of the pre-combustion chamber while the rest is spread over the main combustion chamber's walls, forming a fuel-air mixture with the stratified method. The fuel injection advance takes a minimum value when the engine is at full load regime, thus causing the cylinder to take a position close to the TDC and the fuel getting spread only over the main combustion chamber's walls, where it is vaporized with the heat of the chamber's walls. Combustion chamber's wall temperature(300 – 400 ^oC) being 2 times lower than that of a compressed air at a 16:1....20:1 ratio, pyrolysis (carbon formation) and the pressure rise during the combustion is halted; which enables the engine to work under the optimum combustion law. In this way, *C* and NO_x emissions improve and a more advantageous work regime can be achieved in performance, efficiency and noise-reduction.

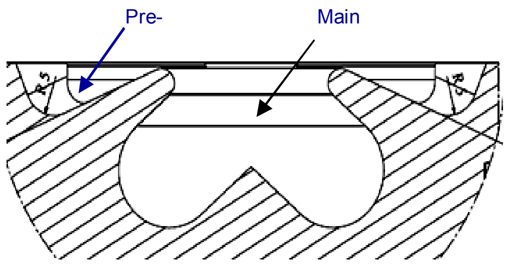

Figure 11. *Cross-section of the MR-1/V2 combustion chamber*

4. Experimental Studies

The first stage of the studies about the MR-1/V2 combustion chamber was made with a Stage IIIA engine which has a MR-1 combustion chamber with 2 valves per cylinder and 90HP. To make the results comparable, cylinders inside the MR-1 combustion chambers were replaced with MR-1/V2 combustion chamber cylinders. First experiments were made with the temperature under -5...-12 ^0C to make sure the engine's starting capability is satisfying. Then; 200 hours of lifespan tests, 8-mod emission and performance tests were done within the TUMOSAN R&D laboratories.

The load characteristics of a 4 cylinder TUMOSAN Stage IIIA engine with 2 valves and different combustion chamber cylinder tests are shown in Figure 12. As shown, the engine displays almost the same amount of effective power (N_e) with both combustion chambers; and also specific fuel consumption (b_e) reaches lower values with the MR-1/V2 combustion chamber with <%50 loads. In other words, the engine works more efficiently. The reason for this is the decrease of the smoke (k) and hydrocarbon (HC) emissions causing an increase in combustion efficiency. At the same time, the decrease in the nitrogen oxide emissions (NO_x) shows that optimum combustion law is achievable with the MR-1/V2 combustion chamber. A 2 valve engine showing NO_x emissions around 250 ppm under full load regime suggests that this combustion chamber can be used with the new generation 4 valve engines to reach the Stage IIIB emission standards.

The 200 hour long lifespan tests of cylinders with the MR-1/V2 combustion chamber returned positive results and field tests are being made on tractors.

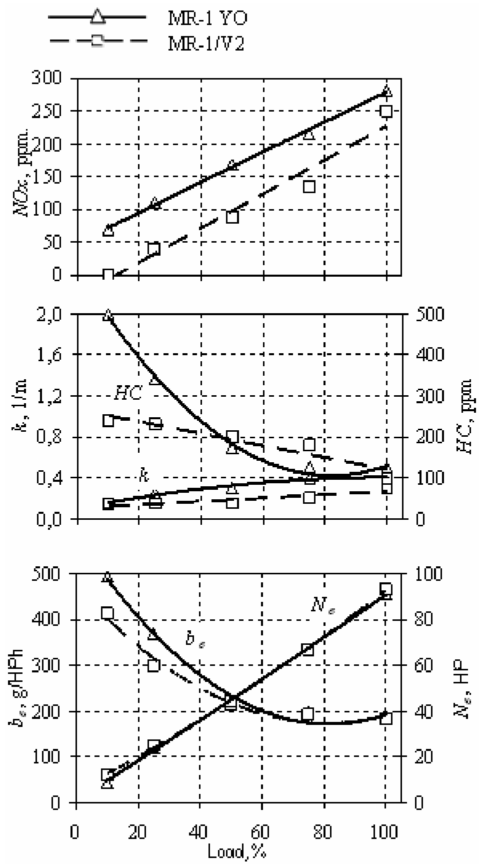

Figure 12. *The load characteristics of a 4-cylinder TUMOSAN Stage IIIA engine with 2 valves with different combustion chambers.* (p_k/p_o =1.88, t_{in}=490 ^oC, ε= 18, n=2500 *rpm)*

Conclusions

1) One of the most important problems today is environmental pollution and as part of university-industry cooperation, ITU and TUMOSAN conduct R&D studies supported by TUBITAK with the aim to reach the Stage IIIB exhaust emission standard which is planned to become effective in 2012. New generation TUMOSAN tractor engines are planned to be made with the use of 4 valves per cylinder, a cylinder head with special design intake and exhaust duct, new fuel-air mixture formation and combustion mechanisms.

2) Optimum working parameters of a new generation 4 valves Stage IIIB engine using a mathematical model of the engines actual cycle, developed by authors, are as follows: excess air factor λ=2.3, turbo pressure air rate p_k/p_o=2.2, volumetric efficiency η_v=0.90, compression rate ε=18 and outflow temperature of the air from the intercooler t_{in}=50-55 ^{o}C. When these values are used under n=2500 rpm speed regime, the engine's power is expected to be 78 kW (106 HP), under 1500 rpm its maximum torque is 330 Nm and specific fuel consumption is 223 g/kWh (164 g/HPh). During a α_z=55 ^{o}CA combustion process, maximum theoretical combustion pressure reaches 12 MPa (120 bar) and *NO* emission reaches 296 ppm. Combustion pressure's maximum value being lower than 12MPa means that we don't need to make any changes on the engine block and the crank shaft, which allows us to keep the cost lower.

3) As a disadvantage, it is determined that when MR-1 combustion chamber is used with the existing 90 HP Stage IIIA TUMOSAN engine, in the winter and especially when the air temperature is under -5 ^{o}C, starting time of the engine gets longer and makes it hard to start the engine. After the engine starts, for a short time period (until the engine temperature reaches to its normal values) engine exhaust is thrown out as white smoke which is rich in hydrocarbon compositions (HC) or as unburned fuel, which causes air pollution and raises the fuel consumption. To avoid this MR-1 combustion chamber's disadvantage; the new 4 valve engine's combustion chamber has a new 2nd Version MR-1 combustion chamber geometry which combines the advantages of "stratified mixture" and "volumetric mixture".

4) As the first step of the experimental studies, by experimenting on 2 valves Stage IIIA engine with a MR-1/V2 combustion chamber installed, it is determined that important improvements on efficiency and exhaust gas emissions will be obtained with this combustion chamber and engine.

Acknowledgment

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References

[1]. Mehdiyev R., Öğün K., Babaoğlu O., Tuncer H., Gürbüz T., Arslan H., Kutlar A., (2008) **Developing TUMOSAN Tractor Engines To Reach Stage III Emission Standarts,** OTEKON'08 4th Automotive Technologies Congree (Bursa, 2008) book, page 613 – 621.

[2]. Mehdiyev, R., İsmailov, A., Ergeneman, M., Çalık, A.T., Şan, D. ve Yıldırım, M., (2002), **Calculating And Decreasing *NO* Emissions In Diesel Engines**, OTEKON'02 1. 4th Automotive Technologies Congress (Bursa, 2002) book, page 205-210.

[3]. Mehdiyev R., Derbentli, T. Ogun K., Babaoglu O., Arslan H., Kutlar A. (2009), **Development Of A Turbo Diesel Engine By A New Combustion Process For Heavy Duty Vehicles And Tractors,** SAE Technical Paper Series 2009-24-0046, p.9.