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INFLUENCE OF VARIABLE COMPRESSION RATIO ON CHARACTERISTICS OF DIESEL ENGINE

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Abstract: *The compression ratio strongly affects the cylinder process and provides an exceptional degree of control over engine performance. At conventional internal combustion engines, the compression ratio is fixed and their performance is therefore a compromise between conflicting requirements.*

One fundamental problem is that drive units in the vehicles must successfully run at variable regimes in different ambient conditions. If a diesel engine has a fixed compression ratio, a minimal value must be chosen that can achieve a reliable self ignition when starting engines in cold start conditions.

In diesel engines, variable compression ratio provides a control of peak cylinder pressure, improves cold start ability, lightens load operation, enabling the multi-fuel capability, increases fuel economy and reduces the exhaust emissions.

This paper contains theoretical and experimental results on the impact automatic variable compression ratios have on the working process parameters of experimental diesel engine. Alternative methods of implementing variable compression ratio are illustrated and critically examined.

Keywords: combustion process, diesel engine, emission, variable compression ratio

INTRODUCTION

It is well known that Diesel engines are among the best candidates to face the future CO_2 limitations thanks to their high thermal efficiency. In modern Diesel engines, the relation between NOx and particle emissions has to be deeply improved maintaining low CO_2 emission [1]. On the other hand, downsizing tendencies lead to increased specific power and torque output [2]. Objectives from 55 to 65 kW/l at 4000 rpm, and from 170 to 200 Nm/l at 1500-2000 rpm are practicable today [2].

As a solution, adequate after-treatments, NOx and particle traps, have been developed. However, there are some concerns about fuel economy, robustness, fuel sulphur sensitivity and costs due to their complex management [3, 4].

Another way could be the reduction of these pollutant emissions directly in the engine, concurrent with after-treatment, using HCCI combustion process. That combustion principle consists of preparing a highly homogeneous air/fuel mixture, in achieving its simultaneous ignition over the whole area of the combustion chamber and in precisely controlling such combustion for the best performances [2].

Figure 1 shows the basic correlation between compression ratio, specific power and cylinder peak pressure, and also the direction in which development is going with lower geometric compression ratio and higher permissible exhaust temperature [5]. Lower compression ratio engine design provides performance with low peak cylinder pressure, and reduces friction, cost and weight. The glow plugs improve engine cold start ability at lower compression ratios. That results in low start time, stable idle operation and improved smoke opacity of diesel engines [1-5].

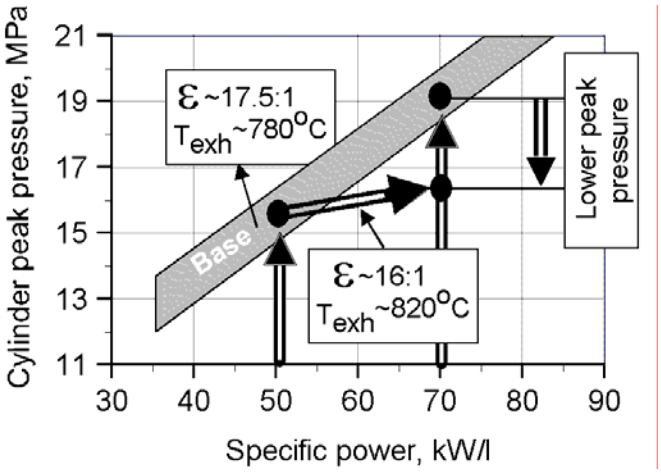

Figure 1. Specific power and cylinder peak pressure [5]

A REALIZATION OF THE AUTOMATIC VARIABLE COMPRESSION RATIO

The existing patents indicate many methods for realization of the variable compression ratio - VCR [1]. This is achieved by altering:

- Volume of an auxiliary combustion chamber
- Distance between the cylinder head and the crankshaft axis of rotation
- Effective length of the connecting rod
- Distance between the piston pin axis and the top face of the piston...

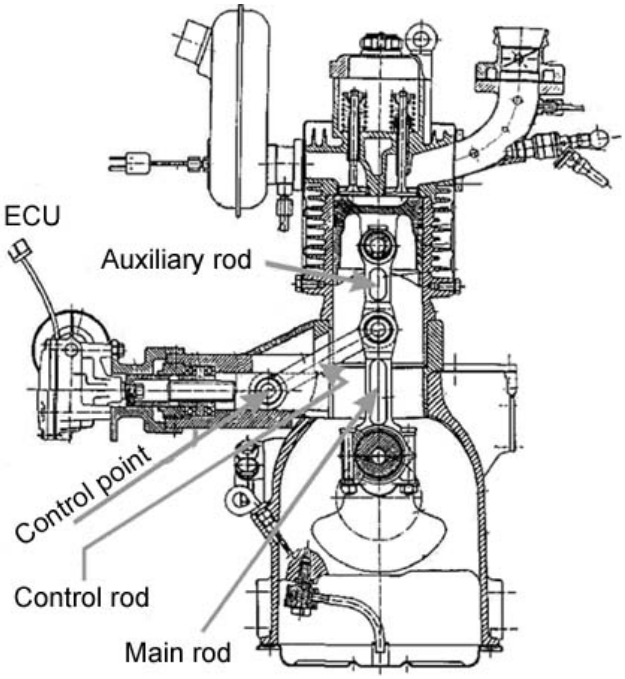

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Figure 2. Engine with automatic variable compression ratio [13,14,15]

The basic principle is applied on an engine with a variable combustion chamber volume, through a secondary piston in the cylinder head – Volvo/ALVAR engine [6]. This design needs supercharging, because the imple-mentation becomes more difficult on the engines with four valves per cylinder technology.

Another possibility, the repositioning of the cylinder or cylinder head, has recently been put into practice by SAAB - SVC engine [7]. The SVC engine consists of an upper part comprising a cylinder head with integrated cylinders, which is known as the mono-head, and a lower part consisting of the engine block, crankshaft and piston. The compression ratio is altered by tilting the upper part of the engine in relation to the lower part. This alters the volume of the combustion chamber with the piston at top dead centre, which also changes the compression ratio. In this group, a new technology trend is currently being developed at FEV: the crankshaft rotates in eccentric bearings. That eccentric rotation results in a change of the vertical position of the bearings and, thus, the top and bottom dead centre of the piston are accordingly shifted [8].

A third possibility is the use of a connecting rod with variable length. FEV realized this mechanism with a split connecting rod and an additional articulated rod [9].

The next construction was classified in the same category as those of engines at which the compression ratio is altered by a piston with adjustable compression height [10]. That piston has a hydraulic, CONTINENTAL, or elastic, FORD, mechanism for the change of compression height. By using this construction at internal combustion engines, a complicated design change of the engine is avoided, promising the automatic variable compression ratio.

Figure 2 shows the cross section of an experimental engine with automatic variable compression ratio, patented in Serbia by the group of constructors [11-15]. A mechanism of this engine consists of the three connecting rods and the compression ratio is continuously varied in the range from 13 to 20, by changing the position of control point.

ANALYSIS OF THE IMPACT OF COMPRESSION RATIO ON COMBUSTION PROCESS

The nominal engine compression ratio is defined as the ratio of the cylinder volume at the beginning of the compression stroke to the volume of the cylinder at the end of the compression stroke, as shown Figure 3 [1, 16]. In fact, by compression before fuel ignition, gas properties in cylinders change with different values of compression ratios, which have an important influence on engine performance.

Diesel engines have high theoretical thermal efficiency, compared to gasoline engines, mainly because of their higher compression ratio. The actual operating processes of Otto and Diesel engines are not realized in their theoretic cycles. Their operation could be better described as approaching a combined constant volume and constant pressure gas cycle, as it is show in Figure 3 [1].

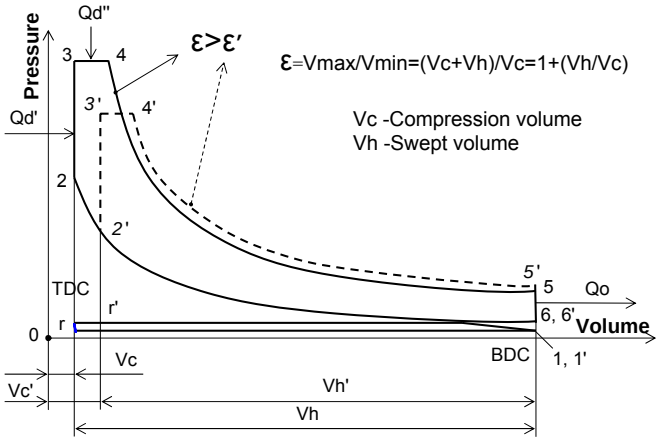

Figure 3. Comparison of closed p-V diagrams at different values of compression ratio [1]

At the Diesel engine, the rapid heat release of the pre-mixed phase can be considered as heat addition at constant volume, while the diffusion combustion phase can be considered as heat addition at constant pressure. The theoretical thermal efficiency of this cycle is intermediate between those of the constant volume and constant pressure cycles, depending on how the heat addition is divided:

$$\eta_{th} = 1 - \frac{1}{\varepsilon^{\kappa-1}} \cdot \frac{\alpha \cdot \rho^{\kappa} - 1}{(\alpha - 1) + \kappa \cdot \alpha \cdot (\rho - 1)} \tag{1}$$

where, ε is compression ratio, κ is ratio of specific heat $(\kappa = c_p/c_v)$, α is ratio of pressure rise during the heat incoming at constant volume $(\alpha = p_3/p_2)$, and ρ is coefficient of previous expansion at constant pressure $(\rho = V_4/V_3)$ [13].

A typical compression ratio for Diesel engines ranges from 14 to 24. The higher compression ratio has higher cylinder pressures, as shown in Figure 3 and, therefore, increases friction losses, Figure 4, to lower mechanical efficiency, as shown Figure 5 [1].

The experimental results obtained by using a method of external drive, Figure 4, were used in determination of the values of mechanical efficiency [1], Figure 5, and given by:

$$\eta_m=[P_{me}/(P_{me}+P_{mf})] \quad (2)$$

where, η_m is mechanical efficiency, P_{me} is brake mean effective pressure, and P_{mf} is friction mean effective pressure.

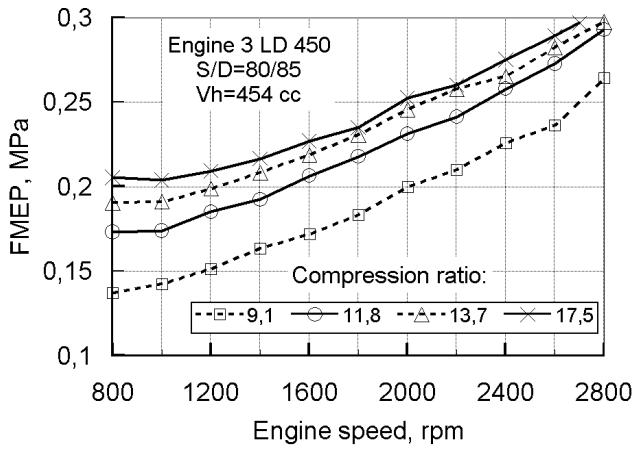

Figure 4. Impact of compression ratio on mechanical losses [1]

The curves of dependence of theoretical and brake thermal efficiency and for indicated and mechanical efficiency, on compression ratio change, were obtained by using the software for engine thermal calculation, Figure 5 [1].

For the needs of the program calculations, the grade of the combustion pressure rise and coefficient of the previous expansion were required to have constant values.

As shown in Figure 5, the engine indicated parameters were improved when the compression ratio was increased gradually from 10 to 22 [1].

A higher compression ratio is necessary for the increase of theoretical thermal efficiency, and at the same time, a diesel fuel ignition lag is shortened, because of the increase of both, pressure and temperature in cylinder at the moment of fuel injection. As a consequence, the operating process is tender, through both, maximal pressure and mechanical load rise, because of the decreased mechanical efficiency.

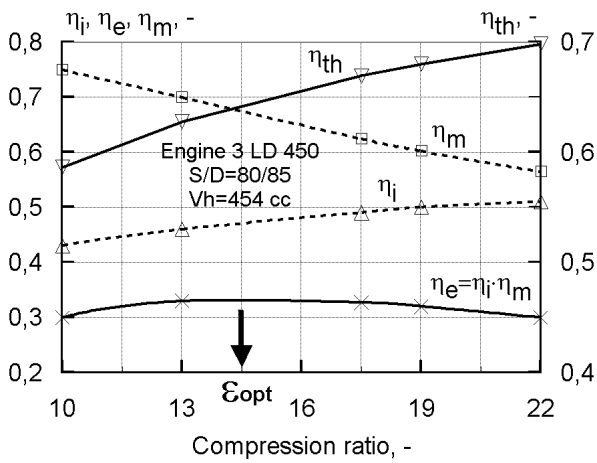

Figure 5. Impact of compression ratio on Diesel engine parameters [1]

The effective parameters of the engine operating cycle were improved when the compression ratio rises, and strongly depend on the mechanical efficiency that is decreased when the compression ratio rise. However, in view of that fact, it is clear that the brake thermal efficiency depends on both, the rate of increase of indicated efficiency, and the rate of the decrease of mechanical efficiency. It was verified from the above, that the brake thermal efficiency first rises at the beginning, and achieves the maximal value for optimal compression ratio value, and then subsequently declines [1]. The compression ratio value when the brake thermal efficiency achieves the maximal value is the compression ratio's optimal value for presented load regime in engine operation [1].

Another reason for the higher efficiency of Diesel engines is that they compress air instead of an air-fuel mixture. Air has a higher ratio of specific heats than an air-fuel mixture ($\kappa \sim 1.3$), which increases the theoretical thermal efficiency [1, 13].

In reality, diesel engine with variable compression ratio will be able to increase the compression ratio at start-up and lower the power use to get the stable ignition, and lower the compression ratio when full power is wanted in order to be able to burn more fuel and get more power, but still have a reliable ignition.

In exploitation, every fuel has different characteristics, in that they react differently to high pressure and ignite at different temperatures. This means that every fuel has different optimal compression ratio for a given engine and driving situation, i.e. specific fuel quality available limits the optimization of a constant compression ratio engine, even its normal functioning. So far as diesel engines work with different fuel characteristics, they must have variable compression ratio, Figure 6 [12].

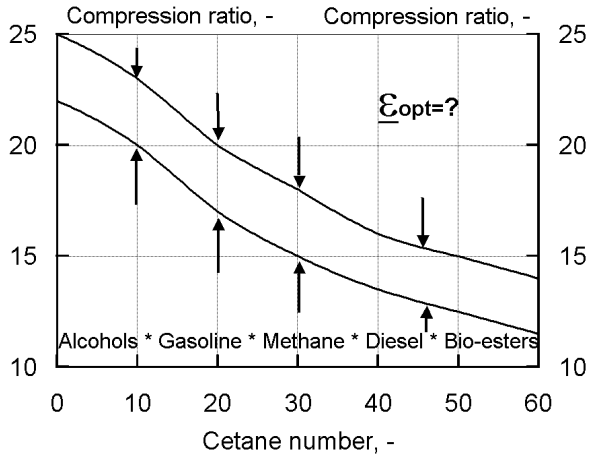

Figure 6. The compression ratio and fuels for diesel engines [12]

OUR EXPERIMENTAL ENGINE AND METHODS

The experiments were carried out at the Engine Laboratory at the University of Kragujevac, with DMB Lombardini single-cylinder diesel engine type "3 LD 450", Figure 7. The four stroke air-cooled engine used in experiments has a conventional mechanical diesel fuel injection system with maximal injection pressure above 20 MPa and with four nozzles injector. The main engine characteristics are [1]:

- intake temperature: around 20 ^{o}C
- displacement: 454 ccm
- stroke/bore: 80 mm/85 mm
- injection advance angle: 18.5 cad (crank angle degree) BTDC
- maximal power: 6 kW at 3000 rpm (DIN 6270)

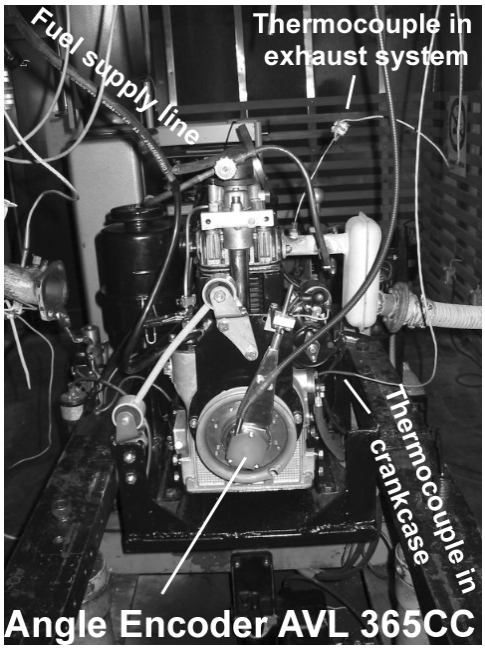

Figure 7. The single-cylinder DMB Lombardini engine used in experiments

Figure 8. Photos of pistons before experiments

The geometric compression ratio value was reduced from 17.5 to 12.1 by exchanging the piston in order to adapt the combustion chamber volume in the piston [1]. That is realized by extending the combustion chamber diameter, Figure 8.

The engine is fitted with two valve cylinder head and direct fuel injection. The intake duct has not been modified, but a swirl number is different because of piston bowl extending, according to compression ratio reduction. The experimental engine was not externally boosted and the exhaust gases were not recirculated.

The engine was tested using diesel fuel D-2 with the following characteristics [1]:

- Cetane number: 52
- Specific density on around 20 ^{o}C: 0.839 g/cm^3
- Kinematics viscosity on around 20 ^{o}C: 3.964 mm^2/s
- Sulphur content: 0.5 %.

The piezoelectric measurement system was realized for indicating the combustion chamber pressure in the cylinder of the experimental engine. The cylinder pressure was measured with an AVL QC32D water-cooled piezoelectric transducer. The signal was amplified with a Kistler 5007 charge amplifier. The signal of pressure in combustion chamber is registered and processed by using the computer and AVL IndiCom, Indicate Software Version 1.2 [1, 16].

The combustion products were analyzed with AVL Dicom 4000 measuring equipment. The smoke was measured separately, with BOSCH method, by using the AVL 409 equipment, and obtained values were transformed to particle emissions [1].

EXPERIMENTAL RESULTS

Figure 9 shows the maximal pressure change inside the cylinder as a function of load and compression ratio. With an increase in major load and compression ratio, under the same injection advance angle, maximal pressure within a cylinder increases; see Figure 9 [1].

This undesirable increase in maximal pressure is followed by a relatively improved decomposition of larger cyclical amount of fuel under higher pressure and engine temperature. As a consequence of improved conditions for combustion process, the entire operation process is improved as well. This may be followed by seeing a decrease in the exhaust gas temperature under the increased compression ratio, see Figure 9.

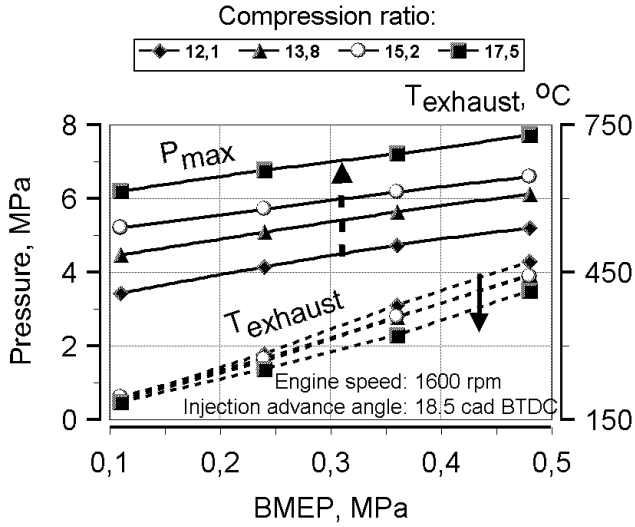

Figure 9. Effects of compression ratio and brake mean effective pressure on engine performance [1]

Under light loads, leaner air-fuel mixture is used in engine operation. This is followed by the decrease of amount of heat generated during the combustion process, and by decreased temperature level of some engine parts. A certain decrease in temperature of the operating engine parts occurs during the compression stroke and the first phase of fuel injection.

In the case of the largest value of the compression ratio (ε=17.5) under all possible loads, the largest temperature occurs inside the engine cylinder, see Figure 10. Large amount of free oxygen under light loads, see Figure 10, in spite of relatively low maximal temperature with respect to full load, leads to formation of the largest amount of NOx. An amount of NOx for that compression ratio, decreases with load increase [1].

In the case of the lowest value of the compression ratio (ε=12.1) under light loads, we have a lowest maximal temperature within the operation cycle, see Figure 10. This leads to formation of the lowest amount of NOx. With an increase in load, temperature increases as well and the amount of free oxygen decreases. Because of that, at the beginning of the process, the amount of produced NOx increases, but when the amount of free oxygen decreases (λ<2, see Figure 10), a decrease in the amount of produced NOx would occur with load increase, see Figure 10 [1].

Under very light loads, the degree of emission of particles is somewhat larger, see Figure 11. The major reason for this is a relatively low injection pressure of the small amount of fuel that does not decompose so well. This leads to the existence of the large leaks of fuel. As the cyclical amount of fuel increases with a load increase, this effect is attenuated and a certain decrease in the particle emission occurs, so that under large loads, it would begin to increase again. Emission of particles increases under all possible loads with the increase in the compression ratio, see Figure 11 [1].

Attenuated fuel splashing under light loads leads to the increase in the emission of CO which is significantly decreased under the increased compression ratio, Figure 11. On the other hand, the emission of CO reduces because of the improved quality of fuel splashing, which improves with an increase in the amount of injected fuel - load. The mixture is depleted in this case, though coefficient of excess air decreases. It is a similar case with a change in the amount of HC, Figure 11 [1].

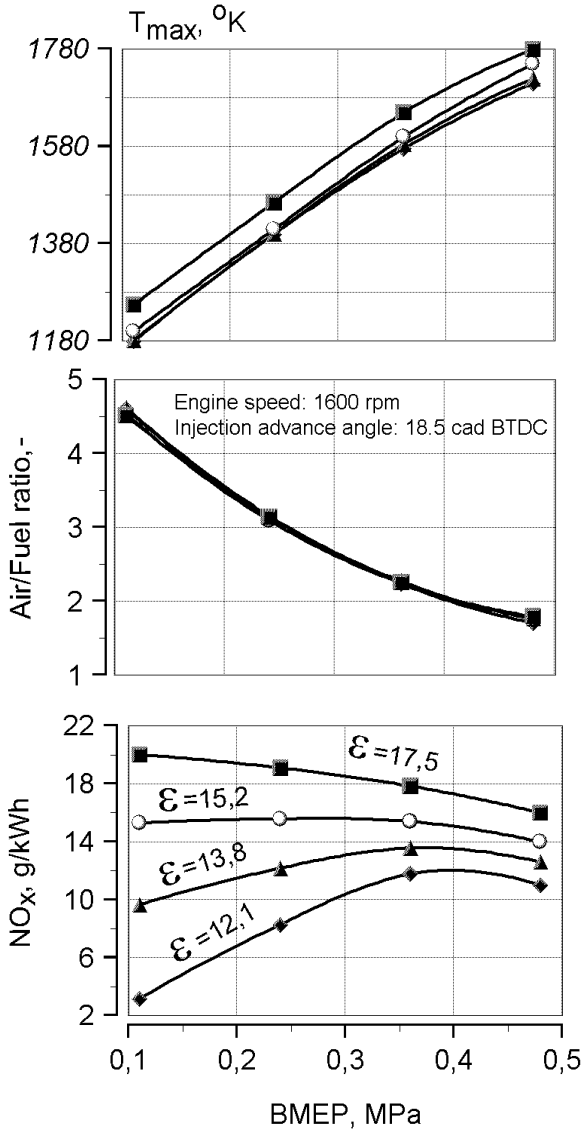

Figure 10. Effects of compression ratio and brake mean effective pressure on engine performance and NOx emission [1]

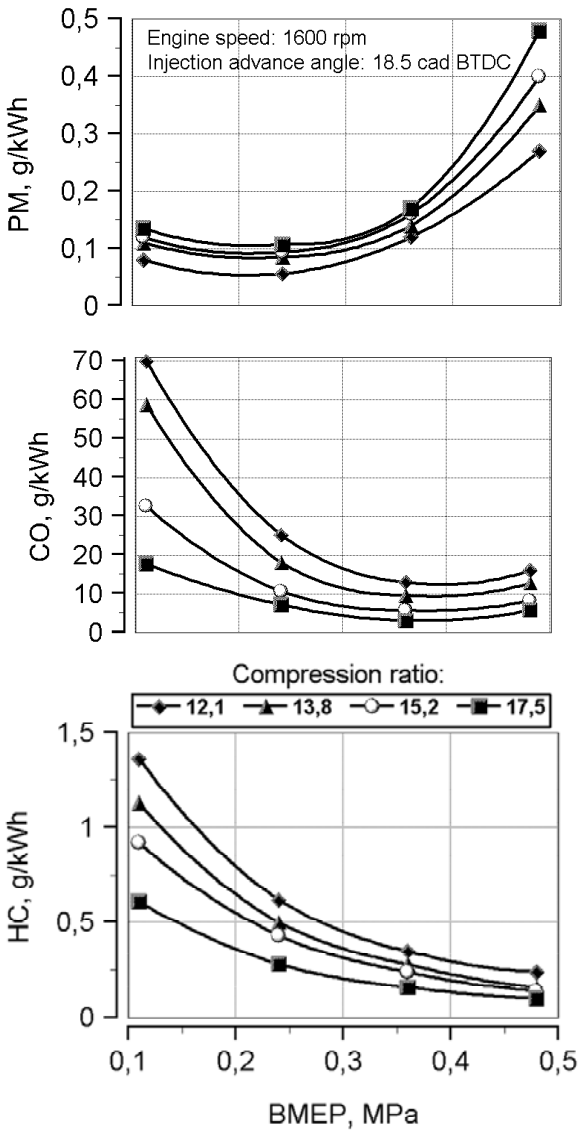

Figure 11. Effects of compression ratio and brake mean effective pressure on engine exhaust emissions [1]

With a decrease in compression ratio and injection advance angle, the brake specific fuel consumption decreases at the beginning, and it has a minimal value when the compression ratio is about 16:1. But, at later stages, it shows a tendency to increase, see Figure 12. The combustion process is responsible for attenuated fuel economy. This is obviously in the case of a combined application of low compression ratio and retarded fuel injection, when the induction period becomes longer, and, because of that, the combustion process was prolonged on expansion stroke. This is followed with a decrease in maximal temperature within a cylinder, while maximal pressure and temperature during an expansion process shows a tendency to increase. As a consequence, the losses become greater and that can be followed by looking at a decrease in the fuel economy [1].

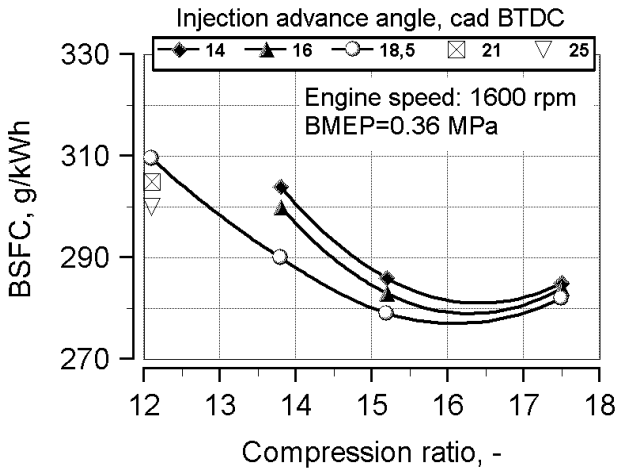

Figure 12. Effects of compression ratio and injection advance angle on brake specific fuel consumption [1]

CONCLUSIONS

1. Summarily, internal combustion engines must satisfy very strong legal regulations in addition to promising cost, and, at a same time, better performances and better economy with respect to classic engines.

2. Application of high compression ratio increases the product of both theoretical thermal efficiency and indicated efficiency, but the brake thermal efficiency depends of both indicated and mechanical efficiency.

3. The motoring loss revealed a mechanical efficiency advantage of the diesel engine concept with lower compression ratio.

4. The maximal pressure within a cylinder increases with an increase in engine load and compression ratio. The peak cylinder pressure increase can be strongly controlled with automatic compression ratio change.

5. For all experimental engine load ranges and constant injection advance angles, the NOx and particle emissions are lowest for compression ratio minimal value 12.1.

6. The specific HC and CO emissions are the lowest for compression ratio maximal value 17.5.

7. At partial loads and low engine speed, the lowest brake specific fuel consumption is achieved when the compression ratio value is about 16, for all values of the injection advance angle.

ABBREVIATIONS

BMEP - Brake Mean Effective Pressure
BTDC - Before Top Dead Centre
HCCI - Homogeneous Charge Compression Ignition
SVC - Saab Variable Compression
VCR - Variable Compression Ratio

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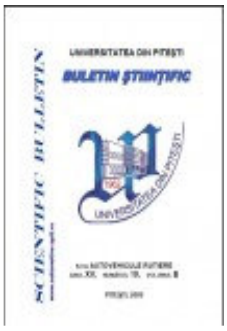

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