BENEFITS AND CHALLENGES OF VARIABLE COMPRESSION RATIO AT DIESEL ENGINES

by

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The compression ratio strongly affects the working process and provides an exceptional degree of control over engine performance. In 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 operate at variable speeds and loads and 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 the engine in cold start conditions. In diesel engines, variable compression ratio provides control of peak cylinder pressure, improves cold start ability and low load operation, enabling the multi-fuel capability, increase of fuel economy and reduction of emissions. This paper contains both theoretical and experimental investigation of the impact that automatic variable compression ratios has on working process parameters in experimental diesel engine. Alternative methods of implementing variable compression ratio are illustrated and critically examined.

Key words: *diesel engine, efficiency, emission, variable compression ratio, working process*

Introduction

It is well known that diesel engines are one of the best candidates to face the future CO_2 limitations thanks to their high thermal efficiency. In modern diesel engines, the relation between NO_x and particle emissions has to be deeply improved, maintaining low CO_2 emission [1, 2]. On the other hand, downsizing tendencies lead to increased specific power and torque output [3]. Objectives for power (55-65 kW/L at 4000 rpm) and for torque (170-200 Nm/L at 1500-2000 rpm) are possible today [3].

As a solution, adequate after-treatments, NO_x 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 [4, 5].

Another way may be the reduction of these pollutant emissions directly in the engine, concurrent with after-treatment, using homogeneous charge compression ignition (HCCI) com-

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bustion process. That combustion principle consists of preparing a highly homogeneous air/fuel mixture, of achieving its simultaneous ignition in the whole space of the combustion chamber and of precisely controlling such combustion for the best performances. Homogeneous mixing of fuel and air leads to cleaner combustion and lower emissions. In fact, because maximal temperatures are significantly lower than in typical spark ignited engines, NO_x levels are almost negligible. Additionally, the premixed lean mixture does not produce soot [3].

The way to reduce engine exhaust emission and to increase power is premixed combustion realized by multiple pre-injection. When the pre-injection was timed before 40 CAD bTDC the main injection quantity could by increased within the allowed smoke number. Under this condition, the enhancement of power and decrease in combustion noise with respect to the base condition were 6% and 4 dB, respectively. Combustion of the early pre-injection generates a mild heat release in two stages. In-cylinder observations revealed that this two-stage heat release occurred via non-luminous flame combustion. Based on results [6], the two-stage combustion is believed to be a premixed compression ignition consisting of both cool and hot flame combustion. This premixed combustion caused a mild increase in pressure and shortening of the main ignition delay, leading to combustion noise decrease. The premixed combustion also reduces smoke due to the enhanced use of air in cylinder. This means that the quantity of injected fuel could be increased under the limited smoke number condition, which in turn increases the power. These effects are exactly the same as those observed in the well-known "fumigation fuel supply method", in which part of the fuels is injected during the intake stroke [6].

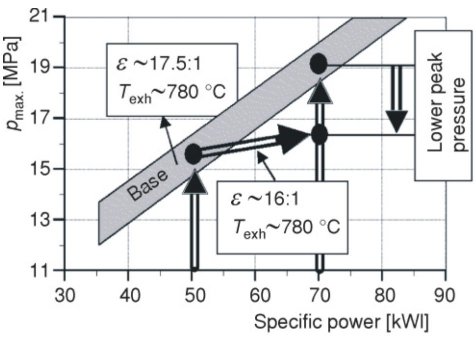

Figure 1. Specific power and maximal cylinder pressure [7]

The possibility to maximize engine power with high pressure-charge is shown in fig. 1. The maximal cylinder pressure (cylinder peak pressure) increases with an increase of engine load (by turbocharged, or *etc.*) and/or compression ratio (CR). Turbocharged diesel engine output is usually constrained by stress levels in critical mechanical components. These maximum stress levels limit the maximum cylinder pressure which can be tolerated under continuous operation, though the thermal loading of critical components can become limiting too. As boost pressure is raised, unless engine design and operating conditions are changed, maximum pressures, friction loses, and thermal loadings will increase almost in proportion. In practice, the CR is often reduced in turbocharged engines (relative to naturally aspirated engines) to maintain peak pressures and thermal loadings at acceptable levels [7].

Lower a constant CR engine design provides performance with low maximal cylinder pressure and reduces friction, costs, and weight. The glow plug and intake heating improve engine cold start ability with lower compression ratios. This results in low start time, stable idle operation, and improved smoke opacity of diesel engines [1-6]. However, a variable compression ratio (VCR) engine offers much more.

The main goals of the paper are to summarize our own research and to obtain some results of influences the CR on the performances of a diesel engine, which may be used in the developing of the new diesel-Otto engine.

Solution of engines with automatic variable compression ratio

Existing patents indicates many methods for realization of the VCR engine [1]. The majority of those mechanisms vary compression ratios by varying combustion chamber volume. 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, and ...

The basic principle is an engine with a variable combustion chamber volume enabled through a secondary piston in the cylinder head (Volvo/Alvar engine) [8]. This design needs supercharging, because implementation becomes more difficult in engines with four valves per cylinder technology.

Another possibility is the repositioning of the cylinder or cylinder head that has recently been put into practice by SAAB (Saab variable compression – SVC engine) [9]. This SVC engine consists of an upper part consisting of a cylinder head with integrated cylinders, which is known as the semi-mono-block, and of a lower part consisting of the engine block, crankshaft, and piston. The CR is varied by pivots at 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 (TDC) which also changes the CR.

In this group, a new technology trend is currently being developed at FEV company: the crankshaft rotates in eccentric bearings. This eccentric rotation results in change of the vertical position of the bearings, and, thus, the TDC and the bottom dead centre (BDC) of the piston are accordingly shifted [10].

A third possibility is a connecting rod with variable length. FEVrealized this mechanism with a split connecting rod and an additional articulated rod [11].

The next design is classified in the same category as those engines where the CR is varied by a piston with adjustable compression height [12]. The piston has a hydraulic (Continental) or an elastic (Ford) mechanism for the change of compression height. By using this construction in internal combustion engines, complicated design change of the engine is avoided, promising automatic variable compression ratio.

Figure 2 shows the cross-section of an experimental VCR engine, patented in Serbia by the group of constructors [13, 14]. A mechanism of this engine consists of three connecting rods and the CR is continuously varied in interval from 13:1 to 20:1 by changing the position of control point. There is an intake system of the Otto version engine in fig. 2, which during the experiment was not been in operational conditions. It is because this study was designed to obtain result that may be used for the development of the new Otto/diesel engine.

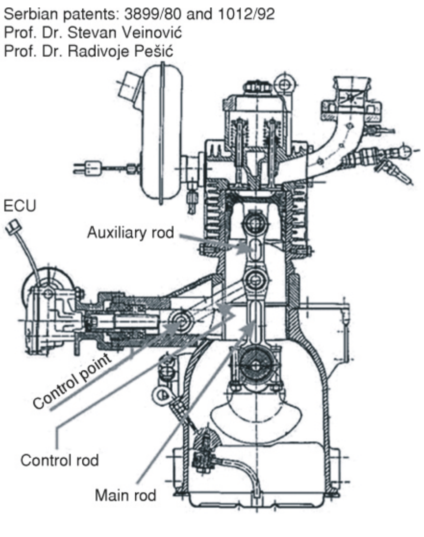

Figure 2. Engine with automatic variable compression ratio [13, 14]

Compression ratio and working process

The engine nominal 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 in fig. 3 [2].

Both the gas pressure and temperature at the end of compression decrease with decreasing CR, which has an important influence on a combustion process and all of engine performance.

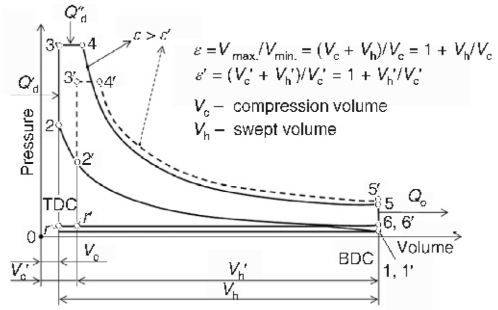

Figure 3. Comparison of ideal cycles for different values of compression ratio [1]

Diesel engines have high theoretical thermal efficiency compared to gasoline engines, mainly because of their higher CR.

In reality, neither Otto nor diesel engines operate according to their ideal cycles. However, the ideal cycle with combined heat input, partly at constant volume and partly at constant pressure, can be used for recognizing the main influences on thermal efficiency. Such a cycle with different compression ratios is shown in fig. 3.

In diesel engine, the rapid heat release of the pre-mixed phase of combustion can be considered as constant-volume heat input process, while the diffusion-controlled combustion phase may be considered as constant-pressure heat input process. The thermal efficiency of this ideal thermodynamic cycle is between that of the constant volume and of constant pressure cycles, depending on how the heat input is shared:

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

where, ε is the compression ratio, κ – the ratio of specific heat ($\kappa = c_p/c_v$), α – the ratio of pressure rise during heat incoming at constant volume ($\alpha = p_3/p_2$), and ρ – coefficient of previous expansion at constant pressure ($\rho = V_4/V_3$) [14].

A typical CR for diesel engines amounts from 14:1 to 24:1. Higher CR results in higher cylinder pressures, as shown in fig. 3 and, therefore, it increases friction losses, fig. 4, to lower mechanical efficiency, as shown in fig. 5 [2].

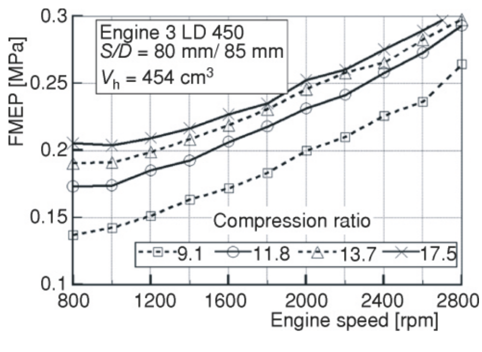

Figure 4. Compression ratio and mechanical losses – FMEP [1]

Engine mechanical losses were determined experimentally using a method of engine external drive [2] and friction mean effective pressure (FMEP) *vs*. engine speed, for different compression ratios is shown in fig. 4. Engine mechanical efficiency is given by:

$$\eta_m = \frac{p_{me}}{p_{me} + p_{mf}} \tag{2}$$

where η_m is the mechanical efficiency, p_{me} – the brake mean effective pressure (BMEP), and p_{mf} – the FMEP.

Influence of the CR on the thermal efficiency of ideal thermodynamic cycle (η_{th}), indicated thermal efficiency (η_i), mechanical efficiency (η_m), and brake thermal efficiency (η_e), was obtained by using the software for engine thermal calculation, fig. 5 [1, 2]. As shown in fig. 5, the indicated thermal efficiency is improved when the CR is increased gradually from 10:1 to 22:1.

Higher CR is necessary for increase of theoretical thermal efficiency, and, at the same time, ignition delay is shortened, because of the increase of both, pressure and temperature in cylinder at the moment of fuel injection. Consequently, the working process is softer.

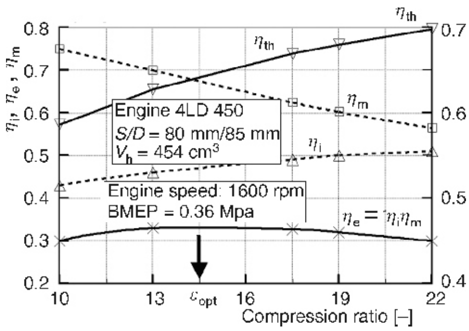

Figure 5. Influence of compression ratio on mechanical, theoretical, indicated, and brake thermal efficiency [2]

The brake thermal efficiency of the engine working cycle is improved when CR rises, and firmly depends on the mechanical efficiency, which decreases when CR rises. However, in view of that fact, it is clear that the brake thermal efficiency depends on both the rate of increase of indicated thermal efficiency and the rate of the decrease of mechanical efficiency. It is verified from the above that the brake thermal efficiency first rises at the beginning, then reaches the maximal value for optimal CR value and then subsequently declines.

The CR value when the brake thermal efficiency reaches the maximal value is the optimal value of CR for this load regime in engine operation.

The key problem is that diesel engines do not run at the same loads. The engine in a truck, for instance, sometimes runs on full power on a steep road or up the hill, and sometimes on idle speed at low-loads. Diesel engines also have to be able to start at any temperature, for example, below zero.

For conventional diesel engine with a constant CR, the CR has to be set so high that a reliable self-ignition can always be achieved even when starting the engine or when running on very low load with little amount of injected fuel into the cylinder.

There is a limit to very high pressures in the cylinder when diesel engine works on full load. Therefore, a high CR also limits the amount of diesel fuel that can be injected at full load.

With a VCR engine, we could increase the CR at start-up and low power and use it to get stable ignition and lower the CR when full power is wanted in order to be able to burn more fuel and get more power, but still having a reliable ignition. Therefore, the concept of VCR engine is a powerful means for increasing low-load engine thermal efficiency and for making it possible to maximize engine power with high pressure-charge.

Between various diesel fuel properties that could be controlled in order to produce emission benefits, cetane number holds the greatest interest, particularly with regard to NO_x. Cetane number requirements for diesel engines depend on engine design, size, nature of speed and load variations, and on starting and atmospheric conditions. High cetane number fuels enable engine to be started more easily at lower air temperatures, reduce white smoke exhaust, and reduce diesel knock. With a low cetane number fuel, engine knock noise and white smoke can be detected during engine warm-up, especially in severe cold weather. While an engine may appear to operate satisfactorily with low cetane number fuel, after prolonged use, severe mechanical damage (*e. g.* piston erosion) can occur. An increase in natural cetane number can contribute

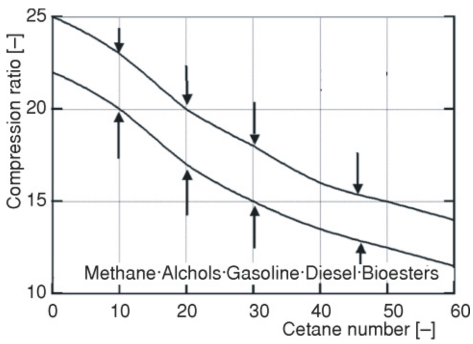

Figure 6. Fuels for Diesel engines and required compression ratio

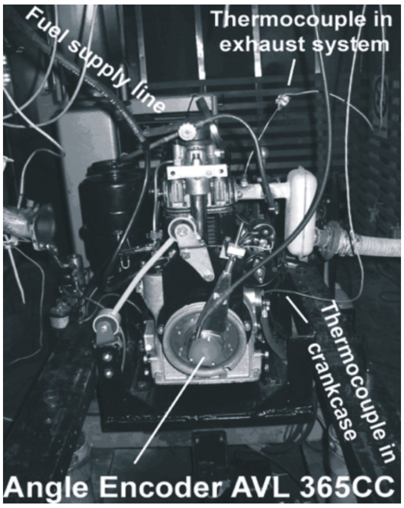

Figure 7. The experimental engine

Figure 8. Pistons before experiments

to reduction of fuel consumption. Only engines with automatic variable compression ratio can use all fuels in normal operation, fig. 6 [14].

Our experimental engine and methods

The experiments were carried out at the Laboratory for IC engines of the University of Kragujevac, on DMB – Lombardini single-cylinder direct injection diesel engine of "3 LD 450" type, fig. 7 [1, 2].

The four stroke air-cooled engine used in the experiments has a conventional mechanical diesel fuel injection system with maximal injection pressure above 20 MPa and with four nozzles injector [2].

The main engine characteristics are [2]:
- intake temperature around 20 °C,
- displacement 454 cm^3,
- stroke/bore 80 mm/85 mm,
- injection timing 18.5 CAD bTDC,
- maximal power 6 kW at 3000 rpm (DIN 6270), and
- specific fuel consumption at maximal power is 262 g/kWh.

The geometric value of CR was varied from 17.5:1 to 12.1:1 by replacing the piston in order to adapt piston chamber volume. This was realized by extending the piston chamber diameter, fig. 8.

The engine has two valves and direct fuel injection. The intake manifold has not been modified, but a swirl number is different because the piston chamber was extending, according to CR reduction. The experimental engine was not externally boosted and the exhaust gases were not recirculated.

During the experiments the engine was operated with diesel fuel D-2 having the following characteristics [2]:
- cetane number (CN) 52,
- specific density at around 20 °C – 0.839 g/cm^3,
- kinematic viscosity at around 20 °C – 3.964 mm^2/s, and
- sulphur content – 0.5%.

The engine is tested on the SCHENK U1-16/2 engine dynamometer over a sequence of European Steady Cycle (ESC) modes, and operated for the prescribed time in each mode. Emissions are measured during each mode of the ESC cycle and averaged over the cycle using a set of weighting factors. The final emission results are expressed in g/kWh.

Exhaust gaseous is analyzed with AVL Dicom 4000 measurement equipment. Particulate matter (PM) emissions are determined idnirectly through the empirical correlation between the measured values of smoke and PM [2]. The smoke was measured using the BOSCH method and the AVL 409 equipment.

The cylinder pressure was measured using an AVL QC32D water-cooled piezoelectric transducer [15]. The signal of pressure in combustion chamber was amplified with a Kistler 5007 charge amplifier and was registered and processed by using the computer and AVL IndiCom Indicate Software Version 1.2 [16].

Experimental results

Cylinder maximal pressure and exhaust gas temperature as the function of engine load BMEP for different compression ratios are shown in fig. 9 [2, 3].

With the increase of CR and engine load, under the same injection timing (18.5 CAD bTDC), maximal cylinder pressure is increasing, fig. 9. This undesirable increase in maximal pressure is followed by a relatively improved atomizing of larger amount of fuel in cylinder under higher pressure and engine temperature. Because of improved conditions for combustion process, the entire working process is improved. Moreover, when the CR is increased, the temperature of exhaust gas is decreased, fig. 9.

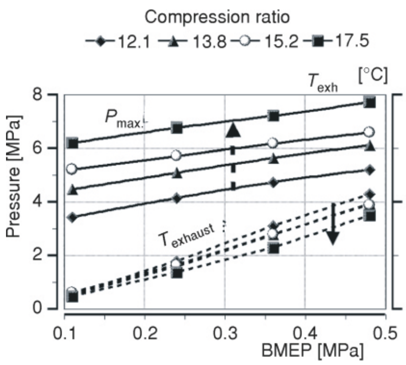

Figure 9. Effects of compression ratio and BMEP on engine performance

Leaner air-fuel mixture is used in engine operation under low-loads. Therefore, the amount of heat released during the combustion process is decreased. A consequence of this is certain decrease in temperature of the engine parts and decrease in cylinder temperature in the first phase of fuel injection.

In the case of the largest value of the CR (ε = 17.5) under all loads, largest temperature occurs inside the engine cylinder, see fig. 10. Large amount of free oxygen under low-loads, see fig. 10, in spite of relatively low maximal temperature with respect to full load, leads to formation of the largest amount of NO_x. Amount of NO_x for that CR decreases with load increase [2].

In the case of the lowest value of the CR (ε = 12.1) under low-loads, we have the lowest maximal temperature within the working cycle, fig. 10. This leads to formation of the lowest amount of NO_x. With an increase in load, temperature increases as well and the amount of free oxygen decreases. Thus, at the beginning of the process, the amount of produced NO_x increases, but, when the amount of free oxygen decreases (λ < 2, fig. 10), a decrease in the amount of produced NO_x would occur with load increase, fig. 10 [2].

Under very low-loads, the degree of emission of PM is somewhat larger, fig. 11. The major reason for this is a relatively low injection pressure of the small amount of fuel that does not atomize so well. As the amount of fuel increases with a load increase, this effect is attenuated and a certain decrease in PM emission occurs, so that, under large loads, it would begin to increase again. Emission of PM increases under all loads with increase in the CR, fig. 11. The

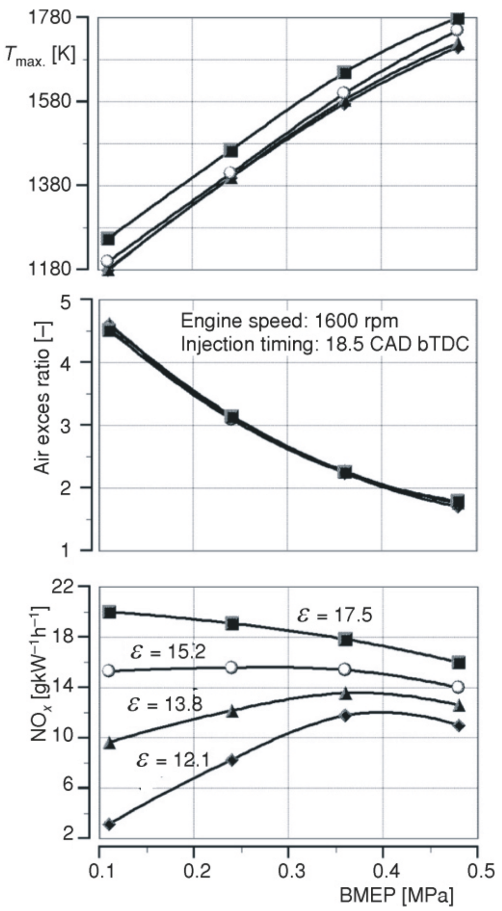

Figure 10. Effects of compression ratio and BMEP on engine performance and NO_x emission

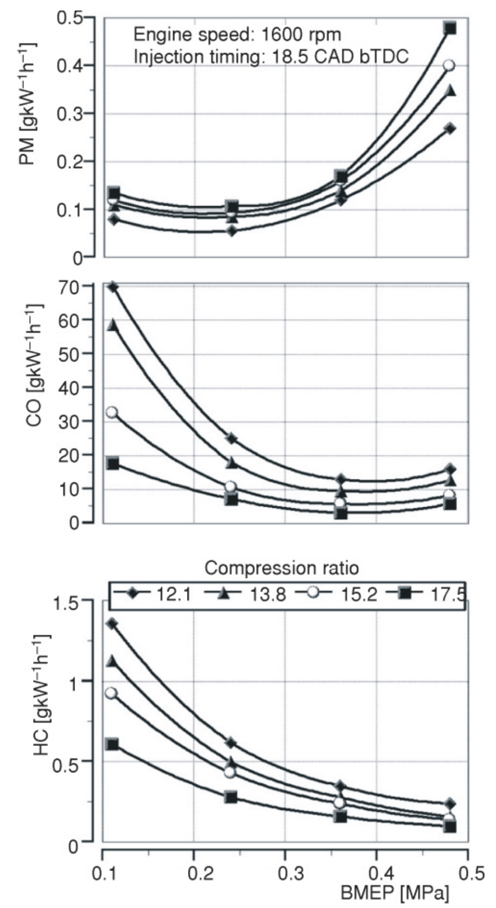

Figure 11. Effects of compression ratio and BMEP on engine exhaust emissions

combustion chamber volume increases if the CR decreases. Thus, the amount of air in the cylinder increases, and it is the cause of decreasing of PM emission when the CR decreases [17].

Poor fuel atomizing under low-loads leads to increase in the emission of CO, which is significantly decreased under the increased CR, fig. 11. On the other hand, the emission of CO is reduced under the improved quality of fuel atomizing, which improves with the increase in the amount of injected fuel *i. e.* load [18, 19]. A similar case is with a change of the amount of HC, fig. 11.

With decrease in CR and injection timing, the brake specific fuel consumption (BSFC) decreases at the beginning, but at later stage it shows a tendency to increase, fig. 12. The combustion process is responsible for increasing of the BSFC. This is obvious in the case of a combined application of low CR and shortened injection timing, when the delay of ignition becomes longer, and, because of that, the combustion process is prolonged to expansion stroke. This is followed by a decrease in maximal pressure and temperature within a cylinder, while temperature during an expansion process shows a tendency to increase [20]. Because of that, the losses become greater and this may be followed by an increase in the BSFC.

Figure 13 is obtained when the values for BSFC (fig. 12) recalculated into brake thermal efficiency. In fig. 13, also are additionally shown values for the maximal cylinder pressure in the ideal and real cycles. In addition to the experimental results, fig. 13 shows the results of theoretical analysis from the chapter *Compression ratio and working process* (fig. 5).

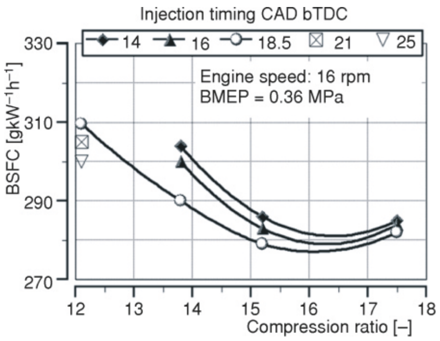

Figure 12. Effects of compression ratio and injection timing on fuel consumption

If the results of theoretical and experimental research are compared, it may be concluded the following:

– the maximum of the break thermal efficiency is greater in theoretical analysis than those obtained by experiment,
– positions of maximum values for the break thermal efficiency, obtained by experiment, are at the higher CR than this obtained by theoretical analysis, and
– maximal cylinder pressure has a tendency to decrease with decreasing CR and injection timing; maximal pressure, in the real process, declines faster with decreasing the injection timing.

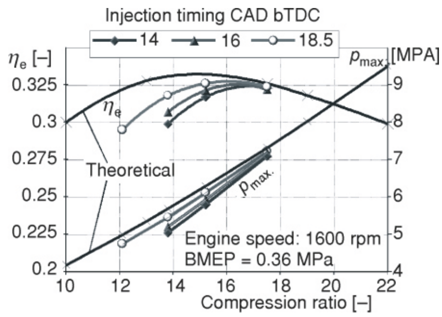

Figure 13. Effects of compression ratio and injection timing on brake thermal efficiency and maximal pressure

The brake thermal efficiency in ideal cycle is calculated with the ideal combustion assumption. Combustion process, in the real cycle, highly depends on cylinder pressure and temperature at moment of fuel injection. When the CR decreases both the cylinder pressure and temperature at moment of fuel injection will decrease, also. Thus, the delay of ignition becomes longer, and, because of that, the combustion process is prolonged to expansion stroke. This is followed by a decrease of both maximal pressure and temperature within a cylinder, while temperature during an expansion process shows a tendency to increase. Because of that, the losses become greater and the brake thermal efficiency decrease faster than one in ideal cycle, which in turn increased optimal CR. In addition, the maximal value of the break thermal efficiency in reality is reduced in comparison with theoretical.

The multiple injection strategies and high exhaust gas recirculation offer the potential to improve the compromise between engine emissions, noise, and fuel economy at lower CR diesel engine. The thermodynamic conditions effects on the combustion process could be achieved via pre-injection. Therefore, the combustion process will be close to optimal, reduction of the brake thermal efficiency will be less, and the optimal value of the CR will be closer to the theoretical. This is possible with an advanced common rail fuel system, which performs as many as five injections per stroke.

Conclusions

A VCR engine offers the potential to increase combustion efficiency and decrease emissions under varying load and speed conditions.

The VCR engine is suitable to use in turbocharged diesel engine because: the VCR concept is beneficial only at part-load where efficiency of the diesel engine is higher than that of the gasoline engine, and the diesel engine has better multi-fuel capability.

High CR increases theoretical thermal efficiency, but decreases mechanical efficiency. The maximal pressure within a cylinder, and mechanical loses, increases with an increase of both engine load and CR.

In experimental engine load range and constant fuel injection timing, the NO_x and particle emissions have the lowest value for minimal value of CR, which is 12.1:1. The specific HC and CO emissions have the lowest value for maximal value of CR which is 17.5:1. At partial load and at low engine speed, the lowest fuel consumption is achieved when the CR value is about 16, for all injection timing.

Nomenclature

c – specific heat, [$Jkg^{-1}K^{-1}$]
D – bore, [mm]
p – pressure, [Pa]
p_m – mean effective pressure [$kJdm^{-3}$], [MPa]
Q – heat, [J]
r – residual gas
S – stroke, [mm]
T – temperature, [°C], [K]
V – volume, [m^3]

Greek letters

α – ratio of pressure rise during the heat incoming at constant volume, [–]
κ – ratio of specific heat, [–]
η – thermal efficiency, [–]
η_m – mechanical efficiency, [–]
η_{th} – thermal efficiency of ideal thermodynamic cycle, [–]
ε – compression ratio, [–]
λ – air excess ratio, [–]
ρ – coefficient of previous expansion, [–]

Acronims

BDC – bottom dead centre
BMEP – brake mean effective pressure, [$kJdm^{-3}$], [MPa]
BSFC – brake specific fuel consumption
bTDC – before top dead centre
CAD – crank angle, [°]
CN – cetane number
CR – compresion ratio
DI – direct injection
ECU – electronic control unit
FMEP – friction mean effective pressure, [$kJdm^{-3}$], [MPa]
HCCI – homogeneous charge compression ignition
PM – particulate matter
SVC – SAAB variable compression
TDC – top dead centre
VCR – variable compression ratio

Superscript

’ – released at constant volume
’’ – released at constant pressure

Subscript

2 – end of compression
3 – end of heat release at constant pressure
4 – end of heat release at constant volume
d – released into cylinder
e – break
exh – exhaust gaseous
f – friction
i – indicate
m – mechanical
o – taken from cylinder
opt – optimal
p – constant pressure
th – theoretical
v – constant volume

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