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THE EXPERIMENTAL VCR DIESEL ENGINE AND DETERMINATION OF DOUBLE VIBE FUNCTION PARAMETERS

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ABSTRACT - Compression ratio is a design parameter with highest influence on fuel economy, emission and engine characteristics. By application of variable mechanisms, the optimal regime fields are extended to the prime requirements: consumption, power, emission, noise, etc., and/or the possibility of the engine to operate with different fuels is extended.

An experimental Diesel engine with variable compression ratio has been developed at the Faculty of Mechanical Engineering from Kragujevac. Detailed engine tests were performed at the Laboratory for IC engines. Special attention has been given to increase of economy and decrease of exhaust emissions. An optimal field of compression ratio variation has been determined depending on the given objectives: minimal fuel consumption, minimal NO_x emission, minimal particles emission, etc.

Beside experimental research, modeling of operation process of Diesel engine with direct injection has been performed. The basic problem - selection of double Vibe function parameters used for modeling the engine operation process - has been solved after processing the indicator diagrams acquired on experimental engine using different compression ratios. Thus, the influence of the compression ratio on selection of double Vibe function parameters for mathematical model of real operating cycle of Diesel engine with direct injection has been determined.

INTRODUCTION

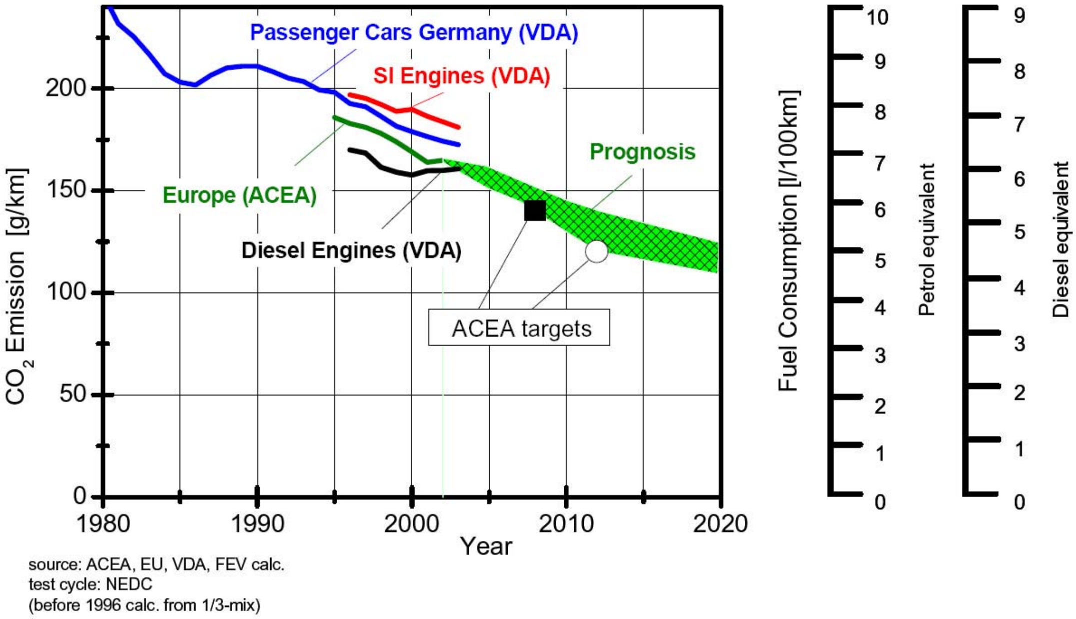
The pressing need to reduce the impact of modern technologies on the environment and human health, becoming even tighter in the perspective of ensuring a sustainable growth to new emerging countries, imposes a continuous development and up-grading of methodologies aimed at a severe reduction of pollutant emissions. Although this scenario involves all the potential pollutant's sources, yet, the problem appears of major concern for automotive emissions, since these contribute to strongly depress the quality of life in big metropolitan areas.

The actual projections on energy demand forecast in fact a ca. 50% growth in fuel request for the short-term period, with fossil fuels still remaining the preferred energy source (1).

Public discussions often give the impression that the CO_2 emission problem in the European Union can be solved by just reducing the CO_2 emissions from passenger-car traffic. It must be kept in mind, however, that the contribution of passenger cars to the global anthropogenic

CO_2 emission level is just about 6%. In the EU and in Germany, passenger car traffic contributes about 20% to the overall CO_2 emissions. Although other human activities generate far more CO_2 than passenger cars, the automotive industry has been working intensively for years on reducing CO_2 emissions by improving the fuel economy of their cars.

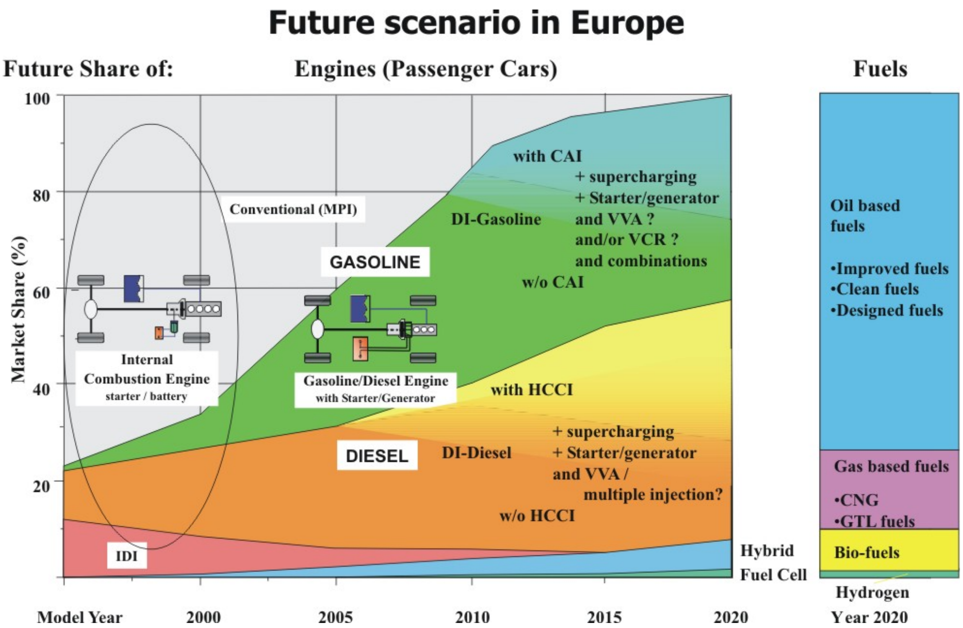
Figure 1: Passenger car fuel consumption and CO_2 emissions – trend and targets



Under the roof of ACEA, the European automotive manufacturers have undertaken to lower the CO_2 emissions of their car fleet by 25% from an average of 186 g/km in 1995 to 140 g/km in 2008 and 120 g/km in 2012 (Figure 1) (2).

This corresponds to an average reduction of fuel consumption from 7.8 l/100 km to 5.9 l/100 km. Since the first official fuel economy measurements in the EU, in 1978, fuel consumption has been cut by more than 50% from 14.1 l/100 km in 1978 to 6.6 l/100 km in 2003.

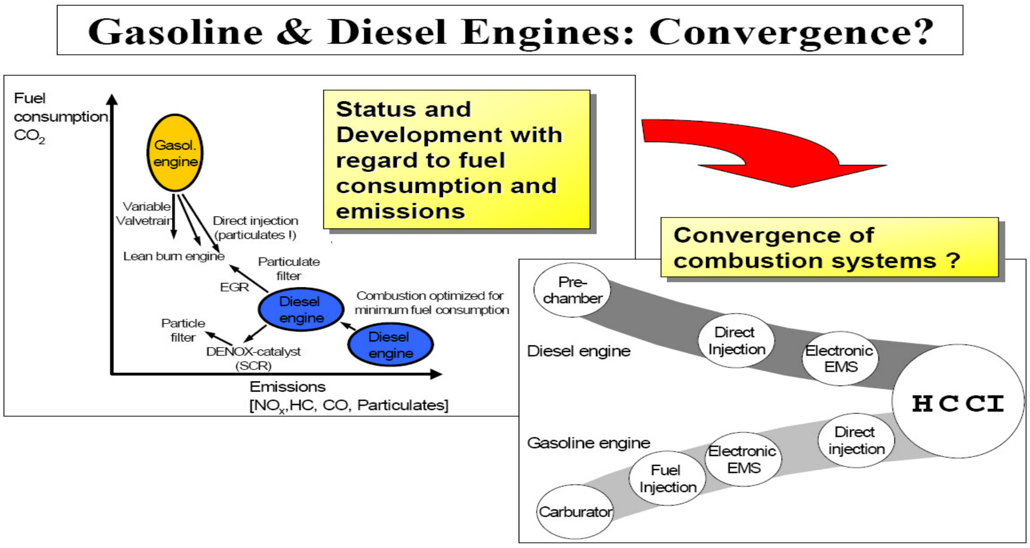
Figure 2: Forecasts regarding passenger-car drive systems



The automobile owes its worldwide spreading mainly to the lucky symbiosis between the existence of crude oil - which still can be considered as the least expensive and most uncomplicated energy resource in the world - and the invention of the reciprocating piston engine with its Otto and Diesel variants (3).

Almost from the beginning, engineers had been looking for alternative concepts to replace the Otto and Diesel power plants, and more than once, the era of the combustion-engine had been said to come to an early end. Nevertheless, both drive concepts have prevailed and are sure to do so also in the foreseeable future, Figure 2 (3).

Figure 3: Development IC engines – Otto and Diesel Convergence



Cars will be powered by Otto, Diesel engines until far into this century (2). Development of Otto and Diesel engines leads to symbiosis of their operating processes into a multi-process Otto-Diesel engine (HCCI) that integrates only their good features, Figure 3 (4). The application of engines with automatic variable compression ratio makes this possible (5-12).

ENGINES WITH VARIABLE COMPRESSION RATIO

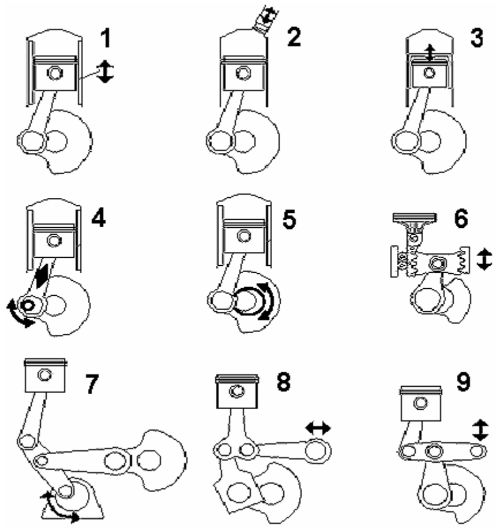
Numerous descriptions of designs of variable compression ratio engines can be found in literature. Some of them are realized in practice, Figure 4 (8).

The engine compression ratio mostly varies automatically with the variation of compression volume. It is achieved by application of engine mechanisms for alteration of:

- *Combustion chamber volume*
- *Distance between the cylinder head and the rotation axis of the engine's crank shaft*
- *Actual length of piston rod*
- *Distance between the axis of piston pin and position of the external dead center…*

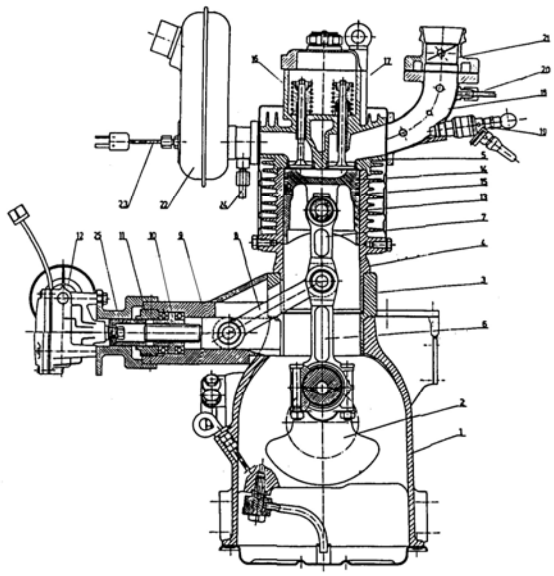
Swedish "SAAB" company has designed a prototype of a gasoline engine with variable compression ratio. At this engine, compression volume is changed through variation of distance between the engine block and the engine head, namely through relative displacement of the cylinder head in relation to the axis of the crank shaft.

Figure 4: Engine mechanisms for alteration of compression ratio (4)



1 Engine block displacement, 2 Auxiliary piston in the cylinder head, 3 Piston with variable compression height, 4 Variable length piston rod (with eccentric crank pin), 5 Eccentric crank pins of the crank shaft, 7—9 Engines with three piston rods.

Figure 5: Experimental engine developed at the Faculty of Mechanical Engineering from Kragujevac (5-12)



An experimental engine having three piston rods and automatic variation of compression ratio has been realized at the Laboratory for IC engines of the Faculty of Mechanical Engineering from Kragujevac, Figure 5.

Figure 6: Maximal cylinder pressure without combustion

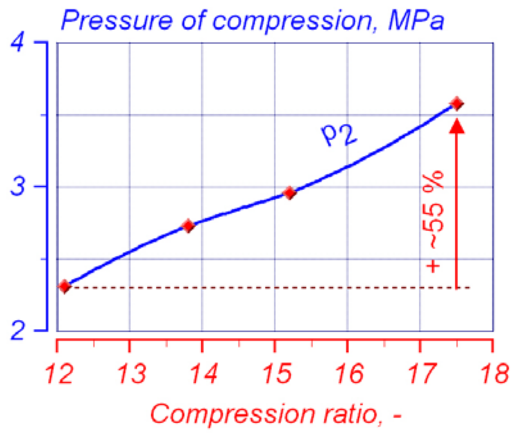
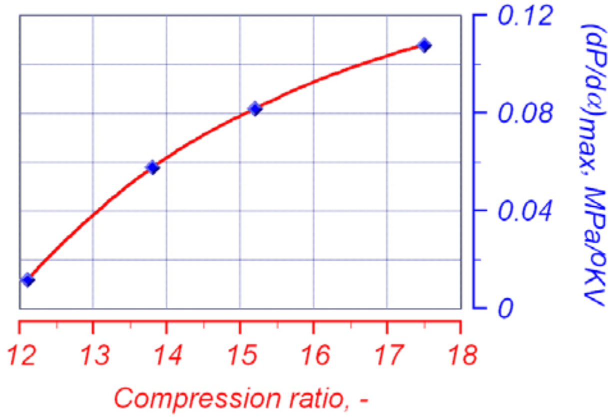


Figure 7: Maximal cylinder pressure-rise without combustion



At Diesel engines, value of compression ratio is determined from conditions for successful cold start. In order to analyze the influence of the compression ratio on operating process parameters during the engine start, corresponding experimental tests were performed and the results are shown in Figures 6 and 7. The starting of the cold engine was performed with *initial engine speed* – using factory built-in electric starter, at the laboratory, during the summer time. It has been determined that the final compression pressure increases with the increase of the engine compression ratio, which is convenient according to the cold start criterion, but it is also transferred to the increase of the maximal pressure-rise in the cylinder.

Dependence between the specific fuel consumption and the compression ratio is presented in Figure 8, for optimal injection and various loads.

Figure 8: The influence of the compression ratio on specific fuel consumption

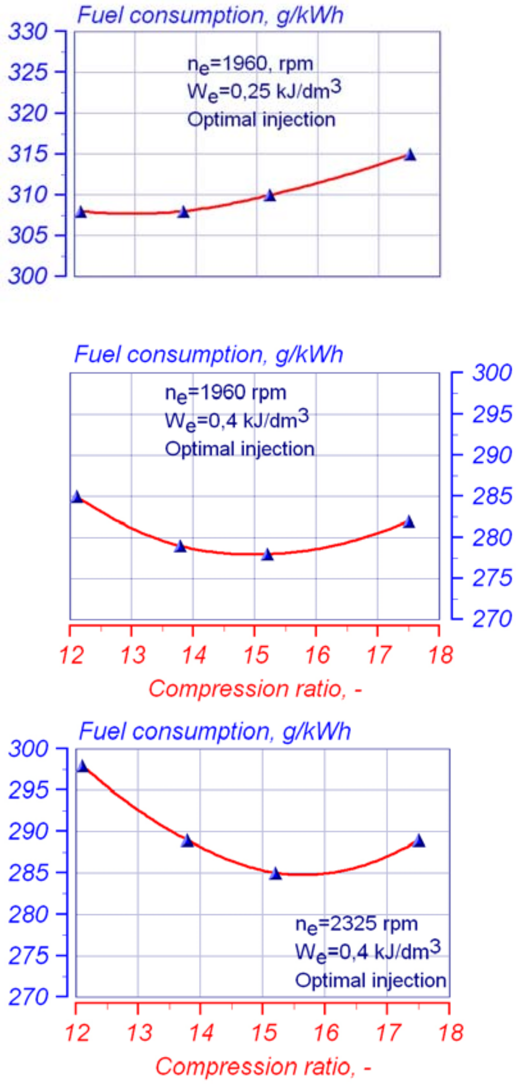
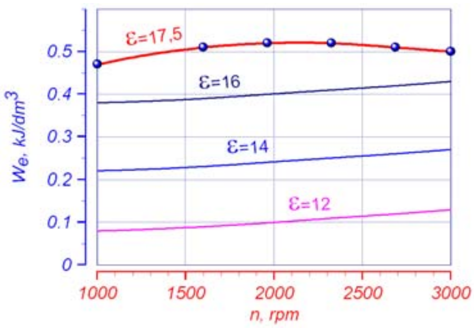


Figure 9: Selection of the optimal value of the compression ratio for the engine operation with minimal fuel consumption



The increase of the compression ratio results in less intensive increase of the specific fuel consumption at low loads, and then, it more intensively increases for compression ratios above 14, Figure 8. At high loads, the fuel consumption first decreases and reaches the minimal value for compression ratio near 15, in order to start growing again with further increase of compression ratio, Figure 8.

The optimal value of the compression ratio by which the engine has minimal fuel consumption increases with the increase of the load, Figure 9. At full load, minimal fuel consumption is achieved with ε=17.5, while at low load, minimal fuel consumption is achieved with ε=12.

At low loads, the NO_x emission intensively increases with the increase of the compression ratio, while at medium loads, the intensity of the increase is somewhat smaller, Figure 10. At high loads, the NO_x emission firstly decreases and then increases with the increase of the compression ratio, and reaches its minimal value for the compression ratio 15, Figure 10.

From the aspect of minimal NO_x emission, optimal compression ratio at full load has value of 15, Figure 11.

Particles emission is the smallest at medium loads and it increases if the engine is running at low or high loads. At the same time, particles emission increases with the increase of compression ratio at all loads, Figure 12, so optimal compression ratio is 12, Figure 13.

Analysis of universal diagrams shows that the trends of compression ratio variation are almost equal for the same values of fuel consumption and emission. Optimal NO_x emission is achieved by later injection, while the minimal consumption and particle emission are achieved by earlier injection.

Figure 10: Influence of the compression ratio on NO_x emission

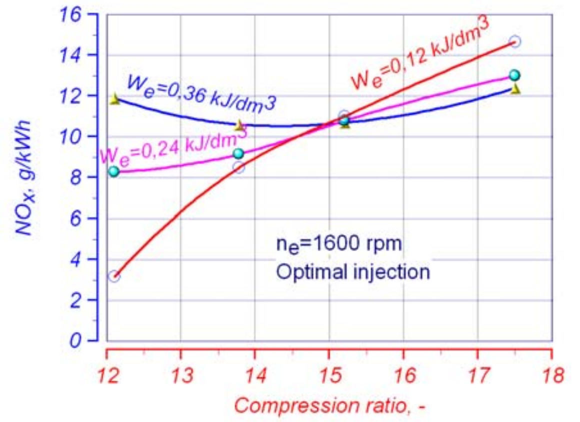


Figure 11: Selection of the optimal value of the compression ratio for engine operating with the minimal NO_x emission

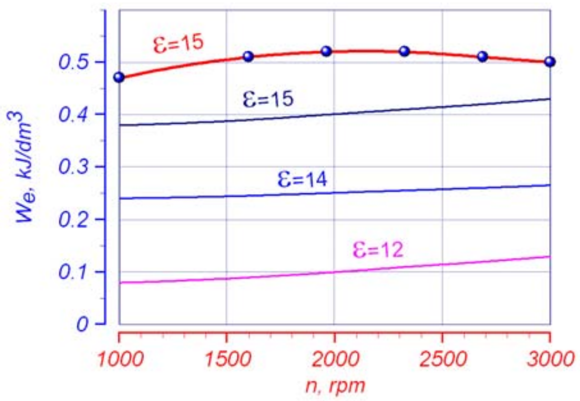


Figure 12: Influence of the compression ratio on particles emission

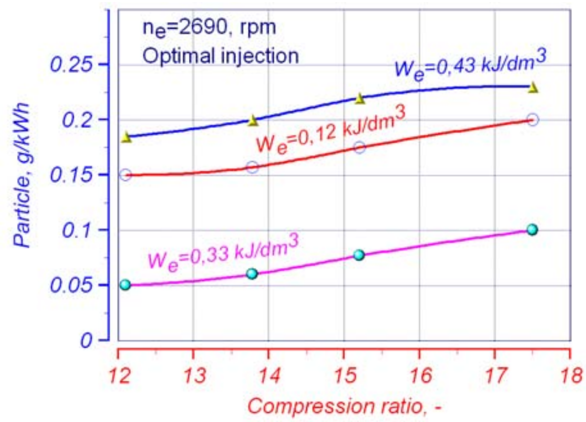
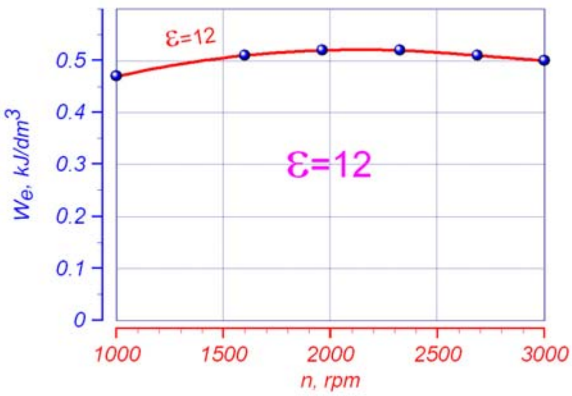


Figure 13: Selection of the optimal value of the compression ratio for engine operating with the minimal particles emission



MODELING OF THE COMBUSTION PROCESS AT DIESEL ENGINES HAVING VARIABLE COMPRESSION RATIOS

At direct injection Diesel engines with distinct explosive combustion and diffusive combustion, a model based on one Vibe function can not describe the characteristic form of the combustion process. In that case, it is necessary to use a model based on superposition of two Vibe functions - one simulating the explosive part (index “1”), the other simulating the diffusive part (index “2”). If operation process of Diesel engine with direct injection and variable compression ratio is modeled, the use of double Vibe function is necessary.

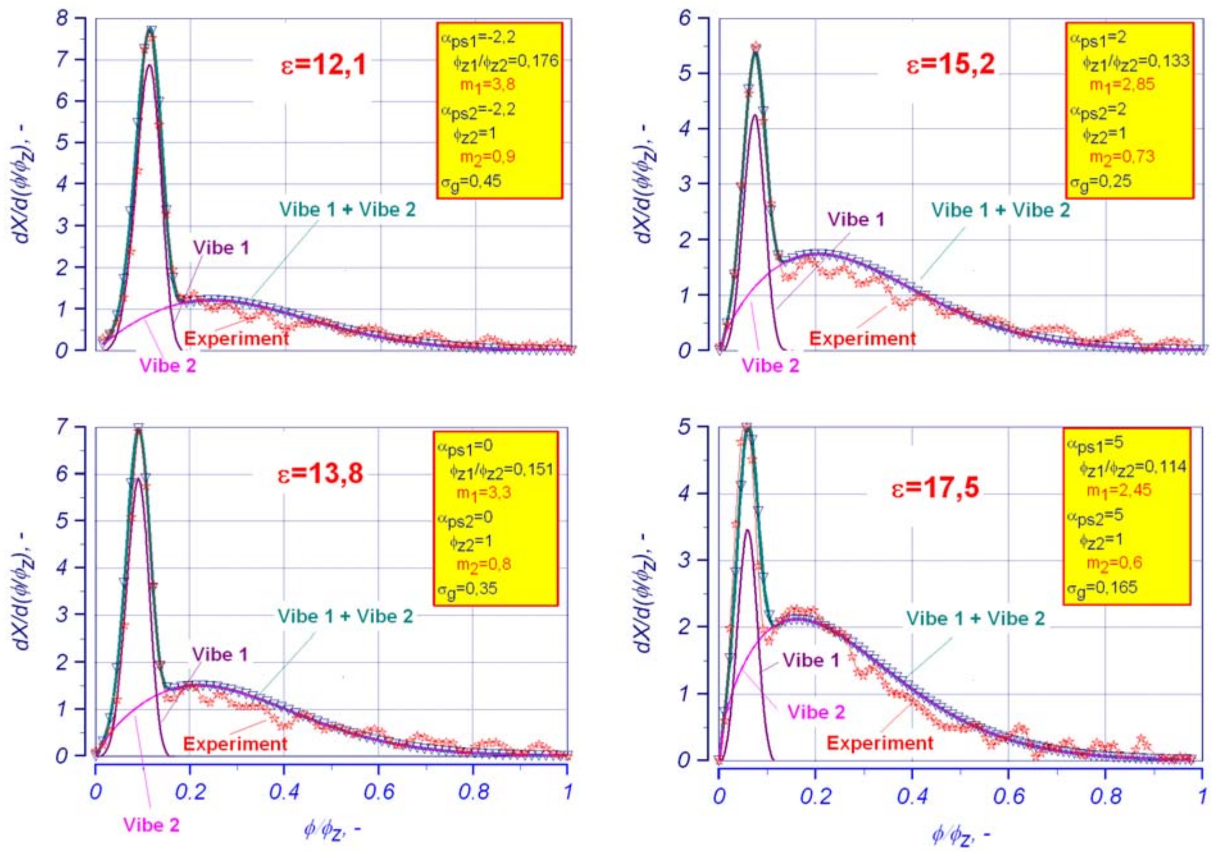
$$x_1 = \sigma_g \left[1 - e^{-C\left(\frac{\phi_1}{\phi_{z1}}\right)^{m_1+1}} \right]; \qquad \frac{dx_1}{d\left(\frac{\phi_1}{\phi_{z1}}\right)} = \sigma_g \cdot C\,(m_1+1)\left(\frac{\phi_1}{\phi_{z1}}\right)^{m_1} \cdot e^{-C\left(\frac{\phi_1}{\phi_{z1}}\right)^{m_1+1}} \tag{1}$$

$$x_2 = (1-\sigma_g)\left[1-e^{-C\left(\frac{\phi_2}{\phi_{z2}}\right)^{m_2+1}}\right]; \qquad \frac{dx_2}{d\left(\frac{\phi_2}{\phi_{z2}}\right)} = (1-\sigma_g)\cdot C\,(m_2+1)\left(\frac{\phi_2}{\phi_{z2}}\right)^{m_1}\cdot e^{-C\left(\frac{\phi_2}{\phi_{z2}}\right)^{m_2+1}} \tag{2}$$

$$x = x_1 + x_2 \quad \text{and} \quad \frac{dx}{d\alpha} = \frac{dx_1}{d\alpha} + \frac{dx_2}{d\alpha} \tag{3}$$

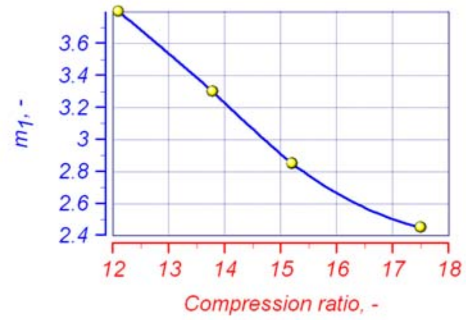
m_1 - the first Vibe function parameter,
m_2 - the second Vibe function parameter,
α_{ps1} - initial combustion angle of the first Vibe function,
α_{ps2} - initial combustion angle of the second Vibe function,
ϕ_1 - angle between initial and current time of the first Vibe function,
ϕ_{z1} - duration angle of the first Vibe function,
ϕ_2 - angle between initial and current time of the second Vibe function,
ϕ_{z2} - duration angle of the second Vibe function and
σ_g - *the share of fuel weight burnt during the first Vibe function* (explosive combustion)

Figure 14: Experimental and theoretical heat release diagrams of Diesel engine
n=1960 rpm, injection angle 18.5 deg. C.A., W_e=0.55 kJ/dm^3

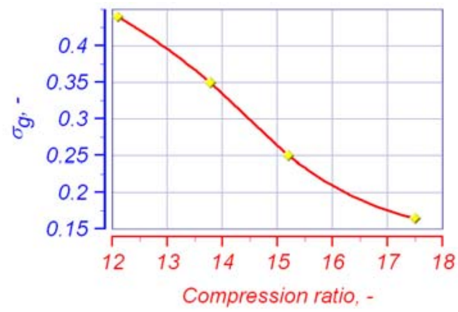


In order to establish the influence of the compression ratio on the parameters of the double Vibe function, corresponding experimental and theoretical investigations of the Diesel engine with variable compression ratio were performed. The research results are shown in Figure 14. As it can be seen, a good match between the model and experiments is achieved.

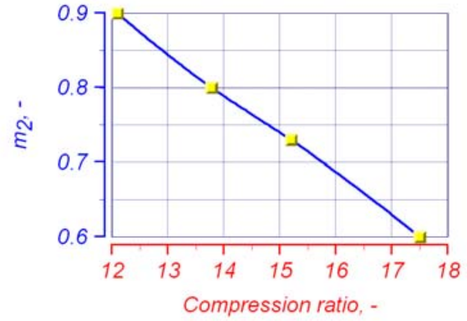
Figure 15: Model parameters variations as functions of the compression ratio n=1960 rpm, Injection angle 18.5 deg. C.A., W_e=0.55 kJ/dm^3



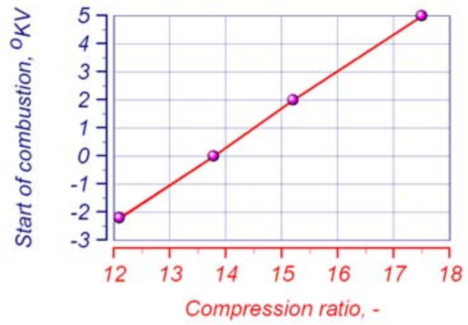
a)



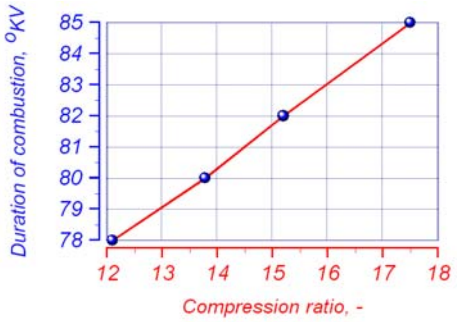
d)



b)



e)



c)

Variation of the identified combustion process parameters is presented in Figure 15 as function of the compression ratio. The increase in compression ratio at constant engine speed and load results in significant decrease of the Vibe function coefficients, Figures 15 a and b.

The tests have shown that duration angle of the first explosive part of the combustion is nearly constant, Φ_{z1}=9 deg. C.A., and this value remained the same during modeling. The total duration angle of combustion, $\Phi_{z2}=\Phi_z$, increases with the increase of compression ratio, Figure 15 c. It is the consequence of the larger amount of fuel combusting with diffusive combustion, as shown in figure 14.

The amount of the fuel burnt at the first stage, σ_g , significantly decreases with the increase of the compression ratio by constant amount of the injected fuel and constant engine speed, Figure 15 d. It is also connected to a shorter period of ignition delay at higher compression ratios.

Initial combustion angle α_{ps} (by unchanged injection advance angle α_{pu}=18,5 deg. C.A.), intensively increases with the increase of the compression ratio, due to shorter period of ignition delay, which is a consequence of higher maximal temperatures at the moment of fuel injection, Figure 15 e.

All tests were performed with experimental engine and conventional diesel fuel injection system: fuel supply pump-pipe-fuel injector. Injection pressure was 200 bar. Experimental engine with direct injection (DI) had two valves and inclined fuel injector having 4 nozzles.

CONCLUSIONS

Value of optimal compression ratio at which the engine runs with minimal fuel consumption increases with the increase of load. At full load, the fuel consumption is the smallest at maximal compression ratio of 17.5, while at low loads, minimal fuel consumption is achieved for compression ratio 12.

From the aspect of minimal NO_x emission, optimal compression ratio at full load has a value of 15. Particles emission is the smallest for medium loads and increases if the engine runs at low or high loads. At the same time, particles emission increases with the increase of the compression ratio, so the optimal compression ratio is 12.

A good coincidence between the model and experimental combustion law of the characteristic shape developed precisely at direct injection Diesel engines is achieved by modeling with double Vibe function.

Initial combustion angle α_{ps} intensively increases with the increase of the compression ratio due to shorter period of ignition delay, which is a consequence of higher temperatures at the moment of fuel injection.

This investigation has shown that the duration angle of the explosive combustion part is nearly constant and this value remained the same during modeling. Total combustion duration angle increases with the increase of the compression ratio, because the combustion is of higher quality and equally divided through all stages.

The amount of the fuel burnt at the first stage significantly decreases with the increase of the compression ratio, by constant amount of injected fuel and constant engine speed. This is also connected to shorter duration of ignition delay at higher compression ratios.

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