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A Simulation of Effects of Compression Ratios on the Combustion in Engines Fueled With Biogas with Variable CO_2 Concentrations

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ABSTRACT
The structure of conventional engines is maintained while they are converted to biogas- powered engines so that they can reuse petroleum as fuel when needed as before conversion. In this case the detonation problem should be treated. This paper simulates the effects of compression ratio on the combustion in the engines fueled with biogas containing different CO_2 concentrations. The results show that the compression ratio strongly influences the pressures at the end of combustion processes, but it presents some effect on the final combustion temperatures and on heat release curves. With a given compression ratio of 12, the temperatures and pressures at the end of combustion in engines decrease by 473K and 2 MPa respectively; while, the molar concentration of CO_2 in the fuel increases from 20% to 40%.
Keywords **–** Biogas, Combustion, Compression ratio, Engine, Simulation.

I. Introduction

Biogas is a renewable fuel produced from organic waste. After being filtered from harmful impurities such as H_2S, biogas can be used as fuel for engines like natural gas. The use of biogas as fuel without increasing concentrations of greenhouse gases in the atmosphere contributes to the limitation of global warming [1].

The use of biogas as an alternative fuel for gasoline in electricity generation and in providing power for production and life in rural areas bring practical benefits to the economy and environmental protection. In fact, owing to production requirements, people use many different types of engines with widely varying power ranges, so the conversion of the engines available fueled with biogas is diverse and their effective operation is also various [2].

The world today has seen the manufacture and commercialization of specialized engines using biogas as fuel such as GE Energy Jenbacher, Australia, with a capacity of 330kW to 3MW, or Jinan Diesel Engine Co., Ltd. Chinese manufacture of biogas engines has been used exclusively with a capacity of 150-660kW. The engine designed for biogas is usually much more expensive than gasoline engines using conventional oil. Yanmar Co. (Japan) has commercialized a generator powered by biogas (from December 2007) at the price of 106,000 USD for an engine with a capacity of 25kW [3]. Or at the Shandong Power Machinery Company of China [4] commercialization of biogas-powered generators with the output of 120kW to 500kW costs from 52,000 USD to 137,000 USD, three times more expensive than diesel generators of the same size (compared to the price of the site [5], CUMMINS diesel generators CW 150S, 150 KVA are valued at 16,479 USD each and a diesel engine costs 43,406 USD 600 KVA). Biogas used as fuel for these engines must meet certain conditions such as fuel composition, supply pressure... One important drawback is that the engines only work with biogas and can not work with a liquid fuel.

Some small-sized biogas-powered engines (capacity of about a few kilowatts) made in China (such as Feigue Engine with the capacity of 2kW), imported into our country has a simple structure, operates on the principles of a spark-ignition engine and shows many disadvantages in operation. Because biogas is provided directly to the engine through a simple carburetor and the load is adjusted by changing the mixture of supply to the engine (adjusted for quantity) without the impact of the biogas rate speed, its engine is not stable, especially when it shuts down on sudden increased load. On the other hand, because there is no device to adjust the quantity of biogas to the engine in order that normal working pressure of the fuel supply to the engine can be stabilized. Some small gasoline and biogas engines manufactured in China (such as Huawei engines with the capacity of 3kW HW3500) have similar disadvantages due to the lack of biogas quantity control systems provided by biogas engine load.

Through the research on biogas engine applications above we have discovered that :

- Biogas, a renewable energy source, has been increasingly used all over the world. Biogas used as fuel for generators and vehicles has been applied in developed countries.
- Special motors designed to run on biogas (not fueled with gasoline or diesel) have higher costs than similar-sized diesel engines.
- Small engines running on biogas simply do not

work stably because of the lack of the devices that control the flow rate of biogas.

Figure 1. GATEC20 Kit

- The world's popular biogas-powered engines that operate on the principles of forced ignition engines; dual fuel engines -- biogas / diesel -- are still not popular.
- Universal kits to convert the engines that use liquid fuel to the engines that use biogas as fuel have not been commercialized on the market.

Figure 2. GATEC21 Kit

The biogas engines available on the world market as mentioned above are not suitable for use in Vietnam. Indeed, due to power supply instability, all the production units must be equipped with backup generators. Investment in a generator running on biogas only with a much higher price than diesel generators as shown above is not economical. However, with only one backup generator, one can, on one hand, use biogas to generate electricity for economical reasons and, on the other hand, can still run on petroleum fuel when biogas is lacking and there is loss of grid power. Therefore, a backup engine is required to run on with both biogas and petroleum fuel.

To meet the needs for biogas-powered generators in our country, we need to apply a technology to convert gasoline engines, diesel engines used in two biogas fuel/ gas, biogas/ diesel with different capacities. It is necessary to develop conversion kits of biogas fuel/ multipurpose oil.

The center has researched and experimented successfully the reform plan of gasoline-powered engines to run on biogas with the universal kit GATEC-20 (Fig.1) and GATEC-21 (Fig.2) [6] . Once the kit is fitted into the engine, the principles and structure of the original engines have not changed, so the engine can use the gasoline and oil as necessary as before the conversion. It can be said that the converted engines are dual fuel engines powered by biogas / diesel or biogas/ gasoline.

The GATEC technology [7] to overcome the above disadvantages of biogas engines is available on the market. Besides, the universal kit of GATEC allows the conversion of biogas fuel/ liquid fuel for most types of widely-used stationary internal combustion engines (petrol, diesel, a cylinder, multi cylinders, turbocharged, not supercharged,...) and changes a wide range of capacities from a few kW to hundreds of kW.

In terms of economical reasons, The GATEC technology has brought many practical benefits for users:

- It is not necessary to equip with an expensive engine to run on biogas, which has the same capacity with a backup engine. This will save money and avoid the trouble of switching the power system between the two generators.
- It is possible to make use of biogas to generate electricity with unlimited volume. When the biogas source runs out, the generator can be converted to run on gasoline or diesel.
- It does not need to waste on biogas as in the GATEC technology. In this way we can avoid the waste of energy and ensure safety and environmental protection.
- The engines converted to biogas engines can consume an average of 1 cubic meter of biogas to generate 1kWh of electricity. This will save 0.4 litre of gasoline and reduce the emission of 1 kg of CO_2 into the atmosphere.

Figure 3. A forced ignition engine running on biogas was converted from a gasoline engine

As shown above, the GATEC technology can enable us to convert gasoline or diesel engines with a wide range of capacities to biogas-powered engines. Owing to the specific features of biogas fuel supply in different combined measures, the economic value of the engine also changes [8]. Biogas forced ignition engines do not need liquid fuel injection (Fig.3). Whereas, dual fuel engines running on biogas converted from diesel engines need a minimum amount of pilot diesel injection for ignition (Fig.4).

Figure 4. A dual fuel engine powered by diesel/ biogas -- converted from a diesel engine

One of the important requirements for converting a traditional engine to a dual fuel engine -- biogas/ fuel oil -- is that it does not change the components of the engine so that it can re-use liquid fuel when needed. This is related to the treatment of the engine knock when it runs on biogas.

In fact, the fuel of biogas is methane which has high anti-knock properties. Therefore, when this fuel is used for biogas engines converted from gasoline engines, we do not need to apply technical treatment on detonation. However, for a dual fuel engine converted from a diesel engine, anti-knock becomes an important issue which should be handled to ensure the normal operation of the engine.

The anti-knock properties of the engine depends on the compression ratio and fuel composition. In the previous work, we have studied the effects of biogas composition on continuous biogas combustion in an open space [9]. In the following section, we will present the results of a simulation study of the compression ratio effect on the combustion engine that runs on the biogas containing different CH_4 concentrations.

II. Simulation of Biogas Combustion in the Engine Combustion Chamber

In this study, we assume that combustion occurs in the constant volume combustion cap-shaped chamber with ignition at the top (Fig.5). When the combustion occurs in a very short time compared with the operation of the piston stroke, we can see that the volume combustion chamber is constant.

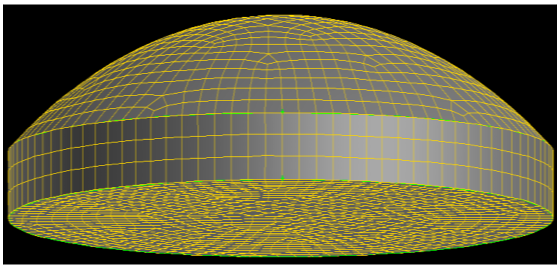


Figure 5. Cap-Shaped Chamber Used in Simulation Study

The compression process in the engine is assumed to be the polytropic compression with the average polytropic index of n = 1.3. If the temperatures and pressure at end of the intake process are Ta and Pa, the temperatures and pressure at the end of the compression process are $T_c = T_a\varepsilon^{n-1}$ and $p_c = p_a\varepsilon^n$.

Table 1. Variable temperatures and pressure at the end of the compression processes in terms of compression ratio (T_a=325K, p_a=0,95 Bar)

□	T_c(K)	p_c(Bar)
9	628	17
10	648	19
11	667	21
12	685	24
13	702	27
14	717	29
15	732	32
16	747	35
17	760	38
18	774	41

Table 1 introduces variations of temperatures and pressure at end of the compression processes versus the compression ratio of the engine when the intake is: T_a=325K, p_a=0,95Bar.

Simulations are carried out by using the FLUENT Software, the model of "partial premixed" combustion in the 3D chamber space. The standard k-□ model is used to describe the turbulent flow in the combustion chamber with coefficients: $c_□$=0.09, c_1=1.44 and c_2=1.92. The speed of layer flame is assumed to be 2m/s.

The initial mixture is unburnt gas, so the "variable progress" c = 0 (Fig. 6). On changing biogas with different CH4 concentrations, we need to adjust *f* to have the same ratio of "mixture" □. The biogas fuel used is symbolized M_xC_y, where *x* is the percentage mol of CH_4 and *y* is the percentage of CO_2 mol. Table 2 shows the value of *f* when □ equals 1.13 with different biogas fuel components.

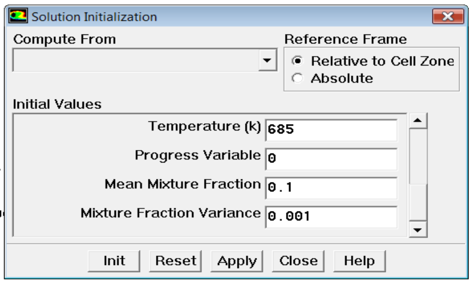
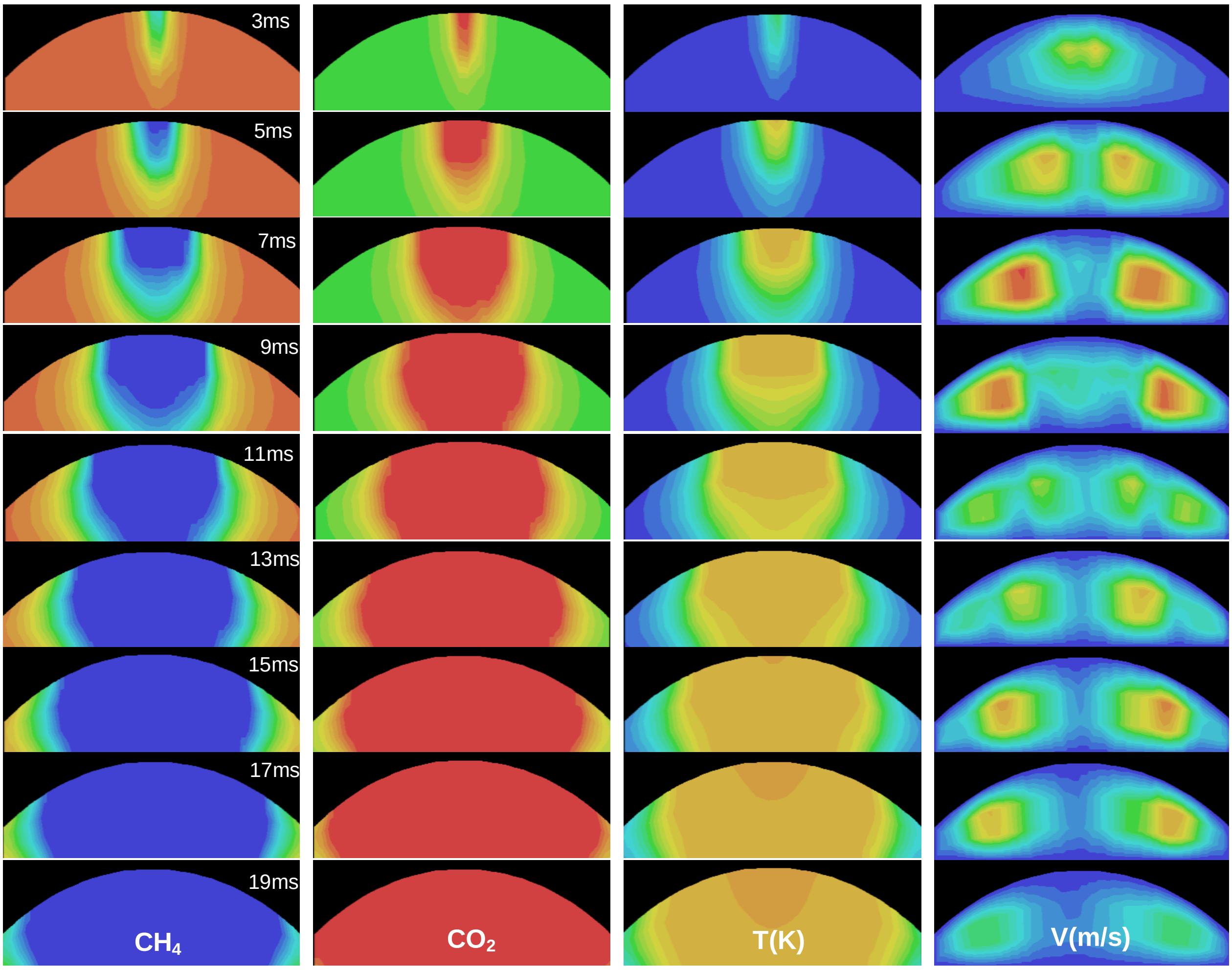


Figure 6. Adjustment of average mixture component of f to get an equivalent coefficient of the given mixture ϕ

The input data of each case include pressures and temperatures at the end of compression processes, the mixture components of *f*. The thermodynamic parameters of the fuel mixture with the air-fuel biogas with a different set of thermodynamics of the PrePDF table before performing the calculation.

Table 2. The value of f when □ □ =1.13 corresponding to the biogas fuel with different components

Types of fuel	M60C40	M70C30	M80C20
f	0,156	0,125	0,100



Combustion is started by assumed sparks with the energy Eig = 0.1 J. After ignition, the flame spreads slowly from the spark center to the farthest point of ignition of the combustion chamber. Computation time step is chosen dt = 0.001s. At each calculation, we determine the temperature, average pressure throughout the combustion chamber, the concentration of CH_4 and O_2 and the products of combustion as well as the speed of air movement in the combustion chamber. Fig.7 introduces the results of a combustion mixture of air and biogas engine M70C30 with a compression ratio □ = 12 and equal number of mixed □ = 1.13. The data is recorded on the section apart 2ms axis passing through the combustion chamber. The data include the mass concentration of CH_4, CO_2, the temperature of the mixture and the movement speed of air in the combustion chamber. This result shows that, after catching fire, the flame quickly spreads to the membrane from the ignition center. The concentration of CH_4 in the fire area reduces quickly, while the CO_2 generated increases rapidly. The combustion chamber is divided into two sections: The burnt gas section and the unburnt gas section, which are separated from each other by the surface of the flame. The high-temperature zone of the burnt mixture spreads gradually corresponding to the shift of the flame.

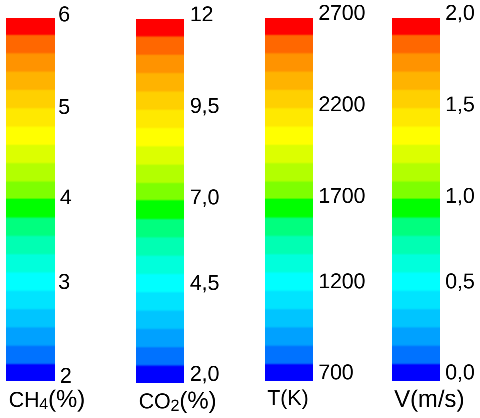


Figure 7. Variations of average concentrations of CH_4, CO_2, temperature and speed of the mixture in the combustion chamber with ε=12 and M70C30 fuel

As temperatures in the combustion zone rise, combustion products expand and compress unburnt gas mixture toward the front, forming a high-speed zone around the fire flame.

III. Results and Discussion

Fig.8 and Fig.9 introduce variable average concentrations of CH_4 and O_2 in the engine combustion chamber corresponding to the engines with compression ratios of 9, 12 and 15. When the engine compression ratios increase, the density of the mixture in the combustion chamber increases accordingly. This makes the speed of combustion (mixture consumption rate) decrease. This represents a change in the slope of the curve of varying concentrations of CH_4 and O_2 over the time.

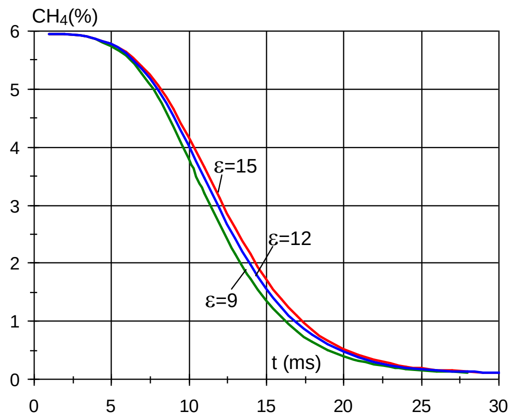


Figure 8. The effect of the compression ratio on the concentration of CH_4 in the combustion chamber (fuel:M80C20)

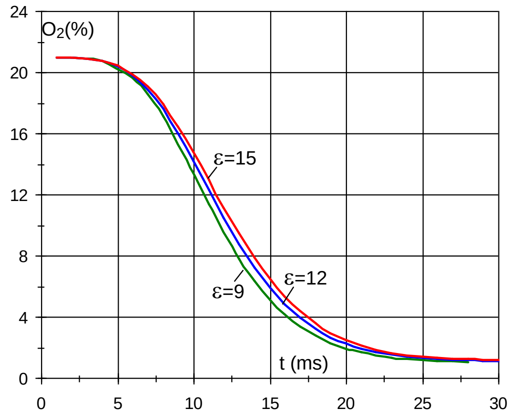


Figure 9. The effect of the compression ratio on the concentration of O_2 in the combustion chamber (fuel:M80C20)

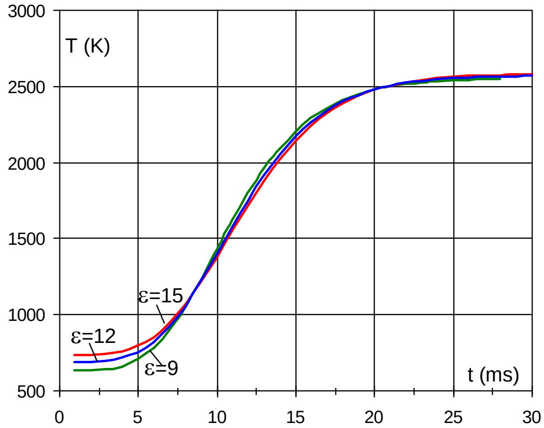


Figure 10. The effect of the compression ratio on the average temperature of the mixture in the combustion chamber (fuel:M80C20)

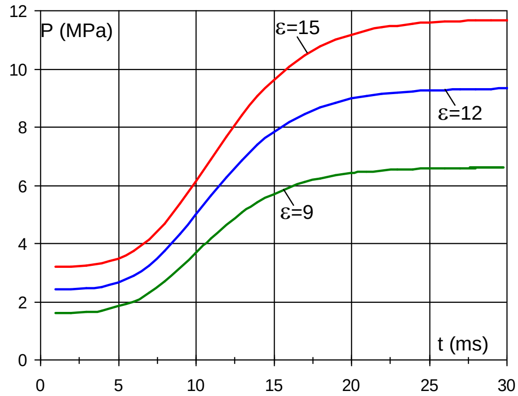


Figure 11. The effect of the compression ratio on the average pressure variations in the combustion chamber (fuel: M80C20)

The change of combustion speed in accordance with the compression ratio leads to change in the speed of heat release, so the contour curves of temperature variation may also be slightly changed with the engine compression ratios (Fig.10). The effect of the compression ratios on the combustion processes of internal combustion engines is clearly shown in the curve of average pressure variations in the combustion chamber. Figure11shows that the maximum pressure at the end of combustion processes changes vehemently in accordance with the compression ratio. According to the equation of the ideal gas state PV = NRT, when the volume V and temperature T are unchanged, the pressure relies on the number of moles of gas n. In other words, it depends on the density of the mixture in the combustion chamber before ignition. Since the temperature at the end of the combustion processes corresponds to the engines of same combustion chamber volumes and different compression ratios slightly change, the pressure at the end of combustion processes is in proportion to engine compression ratios.

The temperature of the mixture before combustion significantly affects the pressure variations in the engine combustion chamber. Fig.12 compares the variable pressure chamber in the engine compression ratio □=9 and □=12 in both the initial temperature Tc = 628K and Tc = 685K. Together with the engine compression ratio, the higher initial temperatures are, the lower the pressures at the end of the combustion become.

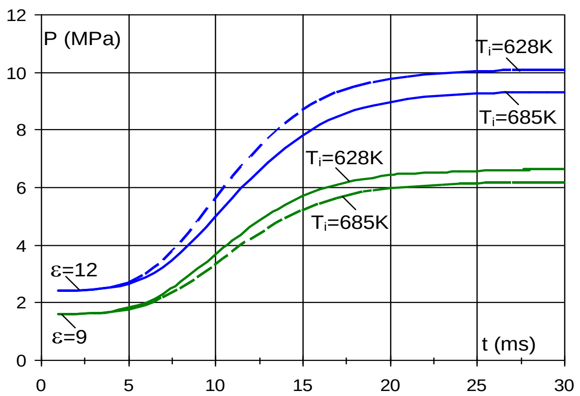


Figure 12. Influence of initial temperature on the variation of pressure in the combustion chamber with different compression ratios (fuel M80C20)

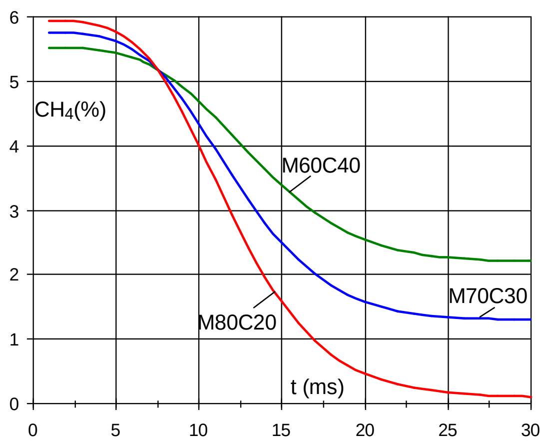


Figure 13. Influences of biogas on average fuel composition pressure variations of the mixture in the combustion chamber (compression ratio □ =12)

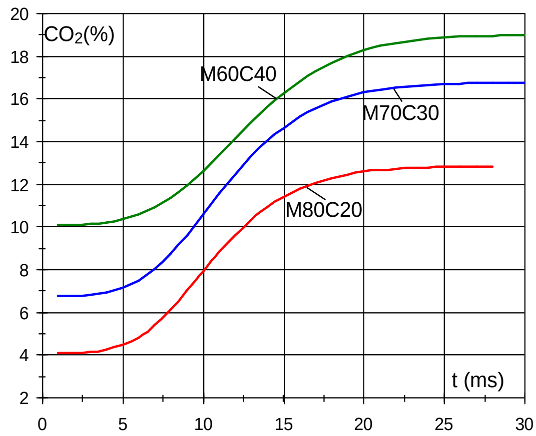


Figure 14. Influences of biogas on average fuel composition pressure variations of the mixture in the combustion chamber (compression ratio □ =12)

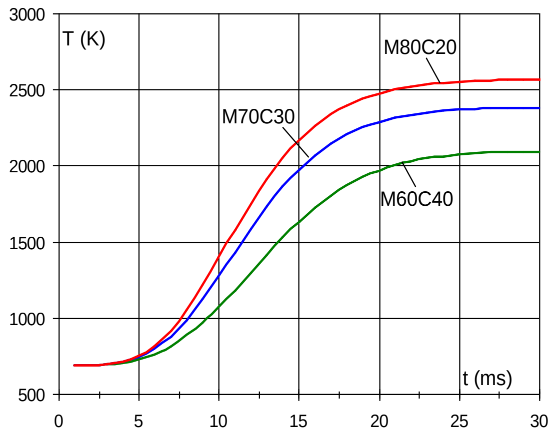


Figure 15. Influences of biogas components on the average temperature variations of the mixture in the combustion chamber (compression ratio □ =12)

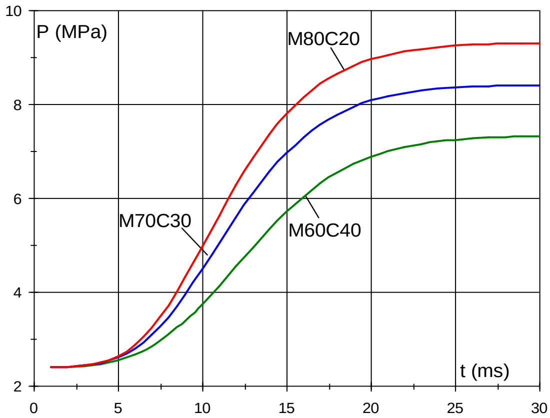


Figure 16. Influences of biogas components on the average fuel composition pressure variations of the mixture in the combustion chamber (compression ratio □ =12)

The same compression ratio engines, biogas fuel components strongly influence the combustion of the engine. Fig.13 and Fig.14 introduces variations of the average concentrations of CH_4 and

CO_2 in the combustion chamber with fuel types such as M60C40 and M70C30 and M80C20. For rich biogas, CH_4 burns almost completely at the end of the combustion processes. When the CO_2 components in the fuel increase, CH_4 concentration remaining in the mixture also increases. Although the amount of fuel burnt is reduced, the amount of CO_2 in the mixture is initially high for poor fuel and CO_2 in the mixture significantly increases. This shows that the concentration of CO_2 in a strong influence on combustion processes, thus affecting the heat release in the combustion chamber.

Fig.15 and Fig.16 introduces the effect of biogas on fuel composition variations in the temperatures and pressures in the engine combustion chamber with a compression ratio of □ = 12 and the equivalent number of mixture □ = 1.13. As for poor biogas, CO_2 in the fuel component hinders the exposure of oxygen and CH_4 as bad combustion processes resulting in the amount of unburned CH_4 in emissions increased. Consequently, the temperatures and pressures at the end of the combustion processes decline. Fig.17 shows that the average maximum temperatures of the mixture at the end of the combustion processes decrease from 2565K down to 2092K (down by 473K) when the moles of CO_2 in biogas increase from 20% to 40%. Corresponding to the increase in the concentration of moles of CO_2, the maximum pressure at the end of the combustion mixture decreases from 9.3 MPa to 7.3 MPa (lower 2MPa) (Fig.18).

Simulation results show that the concentration of CO_2 in the biogas fuel has a strong influence on the combustion engine, especially the speed of heat release. This affects the anti-knock properties of the mixture. Therefore, when converting a conventional engine to use biogas without changing the compression ratio of the engine, we can adjust the concentration of CO_2 in the fuel to control the phenomenon of detonation.

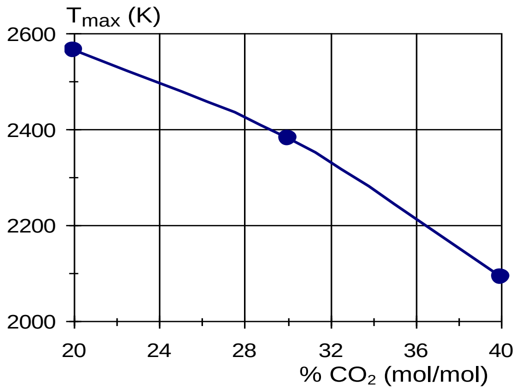


Figure 17. The effect of CO_2 component in biogas fuel on temperatures at the end of combustion processes □ □ =12, □ =1.13)

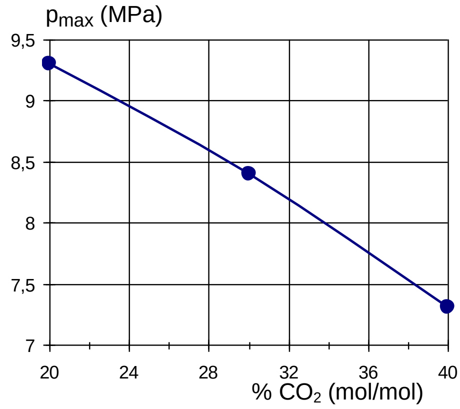


Figure 18. The effect of CO_2 component in biogas fuel on pressures at the end of combustion processes□ □ □ =12, □ =1.13)

1. Conclusion

The results of this study enable us to draw the following conclusions:

1. When the compression ratio engine powered by biogas ranges from 9 to 15, the end of the combustion pressure increases in proportion to the compression ratio, but the heat release curve at the end of the combustion temperatures in this case is less affected by the compression ratio.
2. With the same engine compression ratio given, the biogas fuel components affect the speed of combustion, temperatures as well as pressure at the end of the combustion process. The concentration of CH_4 at end of the combustion processes increases the concentration of CO_2 in the fuel.

The CO_2 components in the biogas fuel are adjustable to control the detonation in the conversion of conventional engines of high compression ratios to the engines fueled with biogas.

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