Available online at www.sciencedirect.com

ScienceDirect

Procedia Technology 22 (2016) 266 – 273

Procedia Technology

9th International Conference Interdisciplinarity in Engineering, INTER-ENG 2015, 8-9 October 2015, Tirgu-Mures, Romania

Solutions to Improving the Performances and Emission Control in a Poor Mixtures Fueled Automotive Spark Ignition Engine

Edward Rakosi[a,]*, Sorinel Gicu Talif[a], Gheorghe Manolache[a]

[a] "Gheorghe Asachi" Technical University of Iasi, Blvd Dimitrie Mangeron no.43, Iasi, 700050, Romania

Abstract

This paper is a synthesis of some theoretic and experimental researches done during several years, in order to find the most efficient and simple solutions for changing some automotive spark ignition engines that work on homogeneous mixtures to those that function on inhomogeneous poor mixtures, using the gasoline injection. The paper also presents the premises and the ways of achieving at this solution that uses mainly the centrifugal action of the mixture obtained in a cylinder.

Peer-review under responsibility of the "Petru Maior" University of Tirgu Mures, Faculty of Engineering

Keywords: auxiliary combustion chamber; spark ignition engine; automotive; engine efficiency; inhomogeneous poor mixtures; gasoline injection; mixture stratification; noxious emissions.

1. Introduction

Gasoline injection as a fueling method for the spark ignition engines has imposed due to its advantages. Nowadays, gasoline injection outside the cylinders, in the inlet manifold is the most used solution. Direct injection is far less used, especially for automobiles.

An analysis of the processes inside the engine shows the major advantages of this type of injection: cycle heat is more reasonably used and cylinder filling is better. Moreover, mixture formation can be conducted according to the most favorable solutions.

Gasoline direct injection imposes higher pressures than injection into the inlet manifold. Meanwhile, the injection pump should have a long service life while operating at high speeds. As a matter of fact, injection pump service life

* Corresponding author. Tel.: +40-746-118162.
E-mail address: edwardrakosi@yahoo.com

Peer-review under responsibility of the "Petru Maior" University of Tirgu Mures, Faculty of Engineering
doi:10.1016/j.protcy.2016.01.079

was one of the factors that have restricted the use of direct injection. Moreover, the injection system should control with precision the cyclic dose according to the engine's working regime.

Compared to the known advantages of fuel injection, in this case we could also use higher compression ratios, due to the supplementary cooling effect over the charge and cylinder walls of the vaporizing fuel. In the meantime, the same fact led to an increase of the self-ignition delay and to a decrease of the knocking trend. Higher compression ratios lead to a lower quantity of residual burned gases and, on the other hand, to the increase of the engine's efficiency [1, 3, 6].

2. The PMACC solution

Our solution, named PMACC (**P**oor **M**ixtures in **A**uxiliary **C**ombustion **C**hamber), ensure the generation an inhomogeneous poor mixtures. For this purpose, the authors have decided to change the ignition solution, converting a Diesel engine into a spark ignition engine, preserving the auxiliary turbulence chamber and lowering the compression ratio, allowed by the spark ignition principle.
According to this idea, the shape of the PMACC auxiliary combustion chamber was modified, in order to ensure a less intensive motion of the air inside the chamber (Fig. 1).

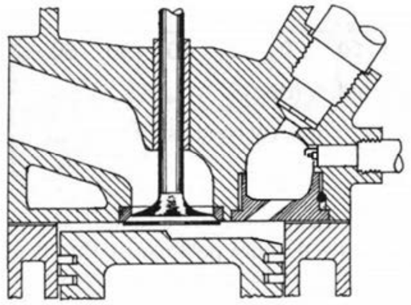


Fig. 1. The **PMACC** combustion chamber shape.

As a result, the gasodynamic losses were reduced and the engine's efficiency has increased. The less intensive motion leads to a more intensive heating process of the charge, which is also due to the higher contact surface of the chamber; the volume of the auxiliary chamber is more that 50% of the cylinder displacement. The lower temperature of the auxiliary chamber, which is due to the increased contact surface with the air-fuel mixture, has allowed the increase of the chamber-cylinder head clearance. The lower compression ratio was achieved by increasing the volume of the both combustion chambers, especially the volume of the main combustion chamber. Based on our previous experience, we have used fuel injection for mixture formation, which has allowed us to maintain some of the components of the base engine. Mixture formation is similar to the one used by the Diesel engine and it takes place mainly in the auxiliary chamber. Due to the fluid motion generated by the PMACC solution, we could use nozzles with only one injection hole and a relatively low injection pressure.

Fuel injection starts during the compression stroke, thus ensuring the vaporization of the fuel; the higher temperature of the auxiliary chamber enhances the vaporization process.

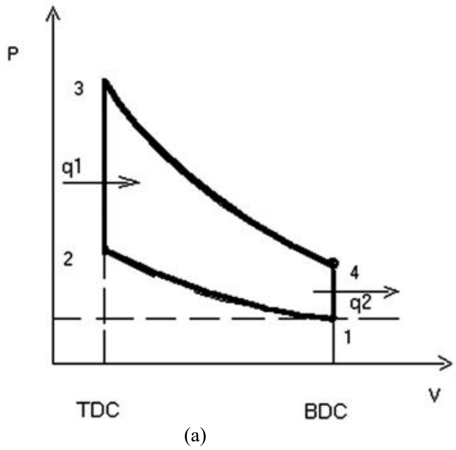
The air pushed by the piston creates the swirl inside the auxiliary chamber. The air-fuel mixture is ignited by a standard spark plug. After ignition, due to the high pressure, the burning mixture is blown out into the main chamber, through the connecting conduit and continues to burn inside the main chamber. The shape of the piston head was imposed by the decrease of the compression ratio and the progressive air feeding of the mixture expelled from the auxiliary chamber. This solution is opposed to the one applied at the *Comet III* and *Comet V* combustion chambers for Diesel engines, where the flame jet is divided in two components, with intensive rotational motion, needed for the completion of the mixture formation. The effect is nevertheless the same – the decrease of the fuel consumption; at the our PMACC solution, this is achieved both by promoting the in-cylinder combustion and by using a reduced volume auxiliary chamber, which leads to the decrease of the energy consumed for the expulsion of the gases. The thermal stress of the piston is also diminished due to the air layer created over that. This solution avoids the droplets deposition upon the cylinder oil layer, thus avoiding the premature wear of the cylinder. The injection system, especially the injection pump, designed following an original solution, allows the use fuels with poor lubricating properties [4, 5, 7].

The PMACC solution has led to the improvement of engine's efficiency due to the increase of the coefficient of admission and compression ratio and to the decrease of thermal losses and exhaust gases temperature.

The engine's rated power has increased and also the maximum torque. In the same time, the fuel consumption has also diminished due to the increase of the excess air coefficient, without affecting the engine's stability.
The hydrocarbons and the monoxide emissions have registered a drop down.

3. Thermodynamic analyses

The thermodynamic analysis of the ideal case has emphasized some particularities and advantages of this solution. It was developed on the basis of the thermodynamic engine cycle for two distinct cases: compression ratio $\varepsilon_{ipm1} = 12$ and compression ratio $\varepsilon_{ipm2} = 14$.



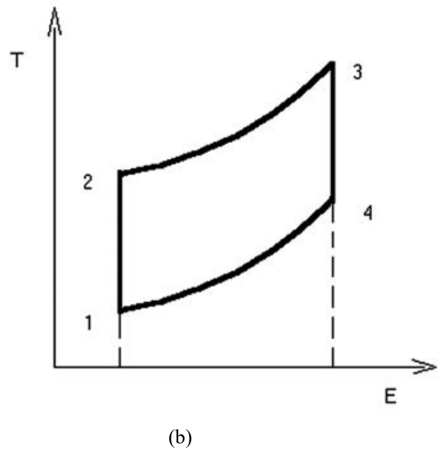


Fig. 2. (a), (b) Spark ignition engine cycle.

3.1. Compression ratio $\varepsilon_{ipm1} = 12$

The thermal energy introduced during the engine cycle through constant volume combustion, specific to inhomogeneous poor mixtures is:

$$q_1 = c_V \cdot (T_3 - T_2) \tag{1}$$

and the energy loss is:

$$q_2 = c_V \cdot (T_4 - T_1) \tag{2}$$

where c_V is the specific heat at constant volume.
The efficiency of the engine cycle is:

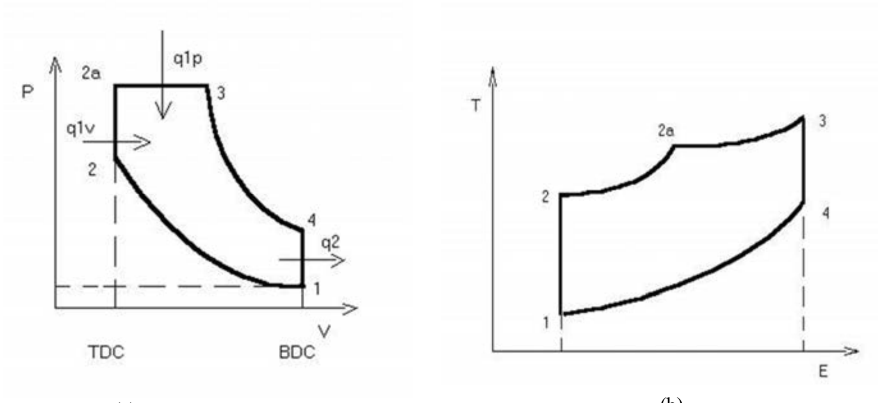
$$\eta_{tv\,ipm1} = \frac{q_1 - q_2}{q_1} = 1 - \frac{c_V \cdot (T_4 - T_1)}{c_V \cdot (T_3 - T_2)} = 1 - \frac{t_4 - t_1}{t_3 - t_2} \tag{3}$$

We get the constant volume cycle efficiency $\eta_{tv\ ipm1} = 56.75\%$.

3.2 Compression ratio $\varepsilon_{ipm2} = 14$

Using the same formulas, we get the following results, $\eta_{tv\ ipm2} = 58.9\%$.
It is obvious that this variant is more advantageous, because engine efficiency increases with nearly 2%. Moreover, comparing this value with the efficiency of a Diesel engine using the theoretical cycle in Fig. 3 a, b, where

combustion takes place partially at constant volume and partially at constant pressure - $\eta_{tv,p}$= 53.3% - we notice an increase of the efficiency with nearly 6%.



(a) (b)

Fig. 3. (a), (b) Diesel engine cycle.

In order to make this comparison, we have assumed that both engines use the same compression ratio, although they are working according to different cycles and are using different fuels. The comparison, presented in Fig. 4 may be done using the temperature – entropy diagram. Assuming that the relative efficiency, η_r (equal with the ratio between the efficiency of the real cycle and the efficiency the theoretical one) and the mechanical efficiency (η_m) are the same for the both engines, we notice that the use of inhomogeneous poor mixtures PMACC solution engine leads us to a higher effective efficiency (η_e) because:

$$\eta_{e,v\,ipm} = \eta_{tv\,ipm} \cdot \eta_r \cdot \eta_m \tag{4}$$

and

$$\eta_{e,Diesel\,fuel} = \eta_{tv\,p} \cdot \eta_r \cdot \eta_m \tag{5}$$

In reality, the difference between the two effective efficiencies is higher because the mechanical efficiency of the inhomogeneous poor mixtures fueled engine is higher than the one of the Diesel engine, which have a higher compression ratio, $\varepsilon_{Diesel} = 20 > \varepsilon_{ipm1,\,2}$ [2].

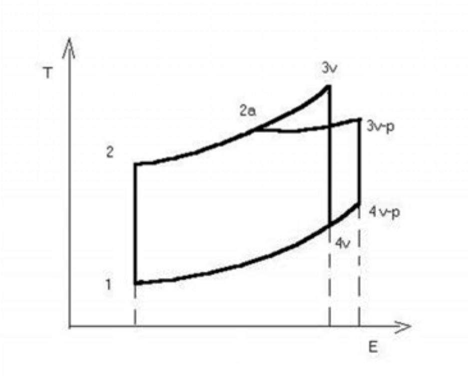
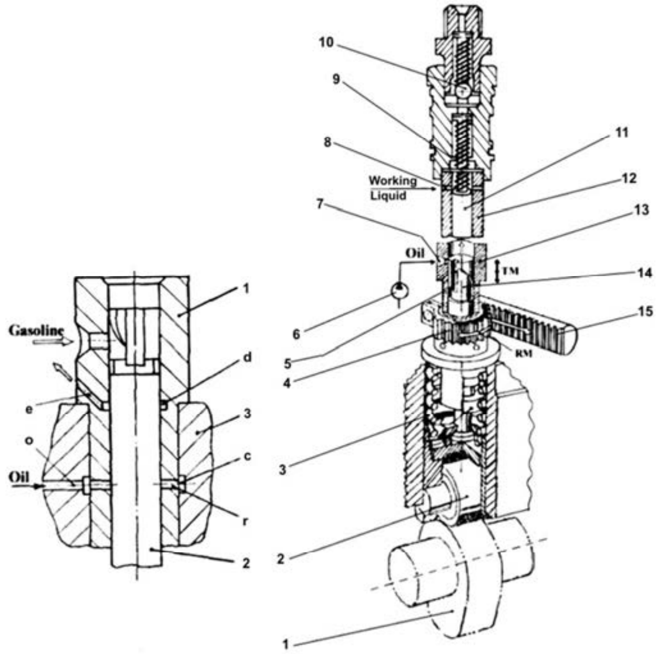


Fig. 4. Thermodynamic comparison.

4. High – pressure injection pump: basic solutions and working principle

To forming the mixture by direct injection of the fuel, some of the restrictions that appeared are connected with the existence of adequate fuel feeding systems, especially some fuel injection pumps that can inject fuels that lack lubrication properties, at high pressure. The injection pumps used the direct injection of gasoline, have limits. Their functional scheme is presented in Fig. 5 a. The oil necessary for the lubrication of the pumping element penetrates between the friction surfaces of the cylinder **1** and the plunger **2**, through the opening **o** that exists in the pump body and then through orifices **r** processed in the cylinder of the element. In order to avoid the dilution of oil with gasoline, the leakage flows between the piston and the cylinder are drained off through the drain channel **d** and opening **e**. Although this solution ensures high opening injection pressures that are necessary for the direct injection of gasoline, the efficient lubrication in the upper part of the pumping element cannot be done because the oil does not reach this space part, and their durability is not satisfactory. To achieving the PMACC procedure, the authors adopted an original solution of a high – pressure injection pump. The principle of these pumps is the separation of the pumping couple from the working liquid. This separation is done by means of a separating element that can be a metallic membrane or a piston. The main functional part of such a pump remains the pumping element, made up of the cylinder **13** and the plunger **14**, as it is shown in the functional scheme in Fig. 5 b. This pumping element is isolated from the gasoline. The gasoline is compressed and carried by means of the separating element **11**, which is of the piston type. The volume between the plunger **14** and the separating element **11** is filled with hydraulic oil, kept at a pressure of 1.2 bar (0.12 MPa) by means of an auxiliary pump **6**. The circulation of the hydraulic oil for its cooling and the drain of the gas that appears are realized with this pump. The hydraulic oil penetrates in the pumping element through the orifice **7**, that can be found in the space **BDC** (**B**ottom **D**ead **C**enter). In this way when the piston reaches in BDC, the opening **7** is completely uncovered. The piston **14**, as any injection pump, has a translation reciprocating movement between **TDC** (**T**op **D**ead **C**enter) and **BDC**, noted as ***TM***. The movement ***TM*** of the piston is transmitted faithfully through the oil to the separating element **11**. Obviously, the compression of the oil can start only when the plunger has completely closed the orifice **7**. The compressing space of the working liquid is above the separating element **11**. The movement of the separating element controls the variation of its volume and it determines the quantity of injected working liquid.



(a) (b)

Fig. 5. (a), (b) Functional schemes for injection pumps.

With this end in view, the changing of the position of the helicon channel **5** in front of the orifice **7** modifies the size of the travel stroke of the plunger. This can be obtained through the rotation of the plunger round its own axis, according to the adjustment movement ***RM***, by means of the toothed sector **4** and the rack bar **15** of the injection pump.

In this pump solution, the pumping element compresses oil that has a greater viscosity than the usual working liquid, the sealing power of the couple made of the cylinder **13** and the piston **14** is improved, and the achieved compressing pressures grow too. The sealing power of the pumping element grows in this case too, due to the possibility of reducing the play between the cylinder and the piston, without altering the quality of the lubrication of the rubbing surfaces. The lubrication of these surfaces is ensured through the oil leaks from the space above. Because the pressure of the oil in this space is equal to the working pressure of the pump, the bigger the working pressure is, the better the lubrication of the rubbing surfaces will be.

The behaviours of the injection process at a camshaft speed of 2000 rpm are shown in Fig. 6, and the dose characteristics of the injection pump, for different positions of the regulation rack bar are shown in Fig. 7.

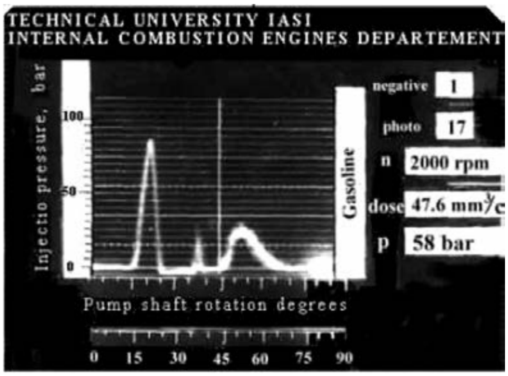


Fig. 6. The behaviours of the injection process.

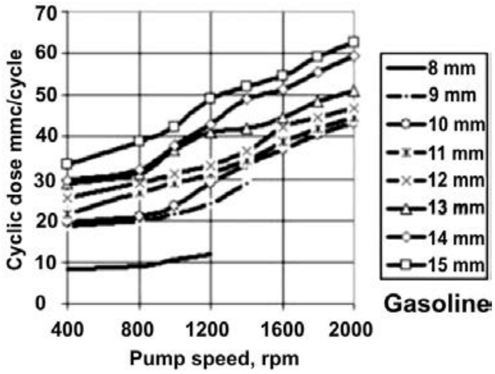


Fig. 7. The dose characteristics of the injection pump.

5. Experimental results. Interpretation

To carry out the tests included in the experimental procedure, a four stroke spark ignition engine, which equipped a passenger car, was used. The engine has four in line cylinders and is water cooled. The engine test bed is equipped with an electric brake (Froude, type EC 38TA), using a P 62 H piezoelectric force transducer. The car tests were performed on the LPS 3000 MAHA chassis dynamometer with the following features: eddy current brake; tractive force: 6 [kN]; measurable engine speed: 0 – 10 000 [rpm]; wheel power: 260 [kW]; speed: 260 [km/h]; measuring accuracy: ± 2%; roller diameter: 318 [mm]; axle load: 2500 kg; PC unit: CPU 1.6 GHz; RAM 256 MB; HDD: 80 GB, OS: MS XP Prof. (Fig. 8). Fig. 9 presents the car test bed and Fig. 10 the engine speed characteristics with various loads.

Fig. 8. The chassis dynamometer.

Fig. 9. The car test bed.

Further to the analysis of these characteristics, one can notice that the performance difference decreases in the same time with the speed rise, perhaps due to the slight increase of pre-ignition based on the reduction of the detonation tendency. Significant but advantageous differences appear in what concerns the rates of the effective specific consumptions obtained for regimes characterized by partial loads. Thus, a slight reduction already appears at 85%, but the consumption rates corresponding to the 70% load are visibly diminished. The lowest rates appear for this load within the 2500 – 3500 rpm range, it being about 267 g/kWh. The minimum consumption rate emphasized is 261 g/kWh and can be obtained for the same load and a 3000 rpm speed. Substantial reductions appear obvious for the 40% and 25% loads. The increase tendency of the burnt gas temperature, depending on speed, with various engine loads is somewhat slowed down by the mixtures which in this case are even poorer.

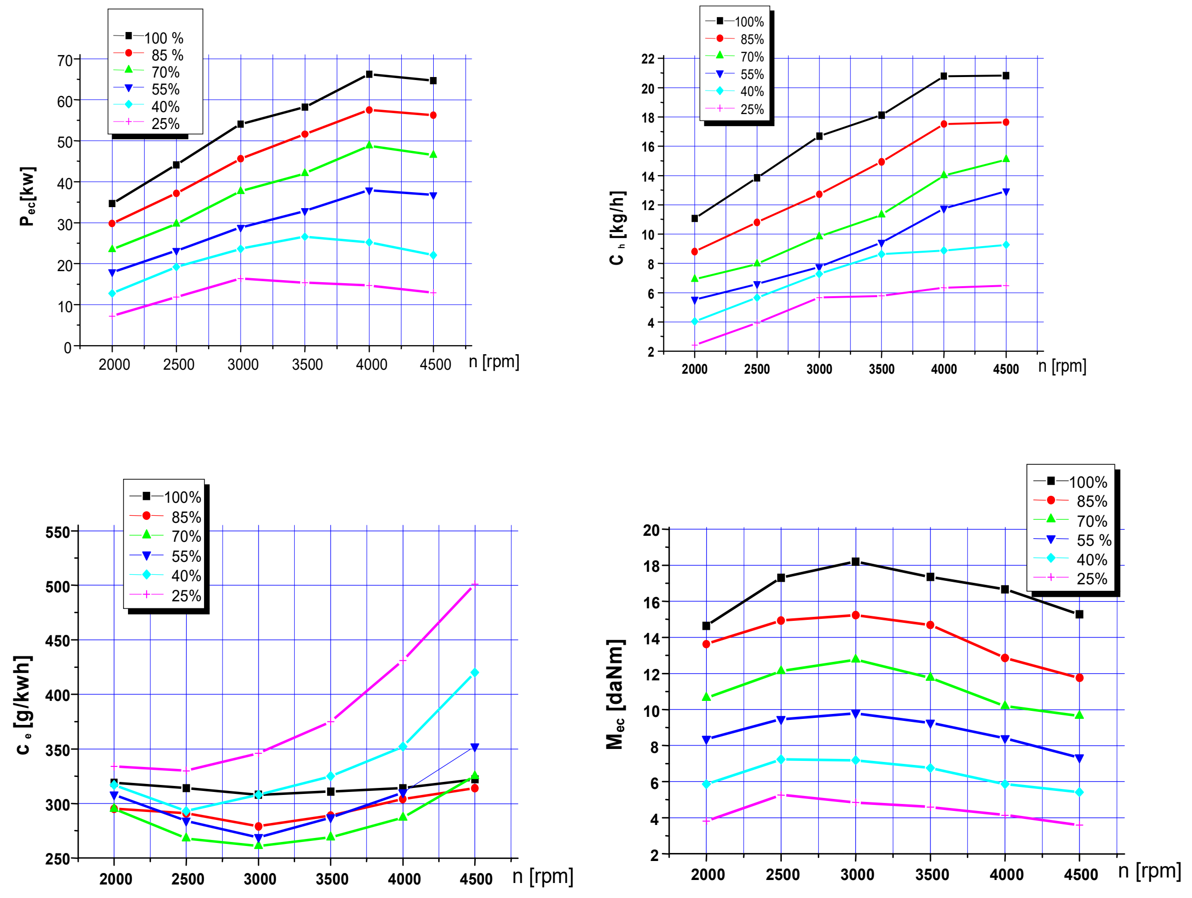
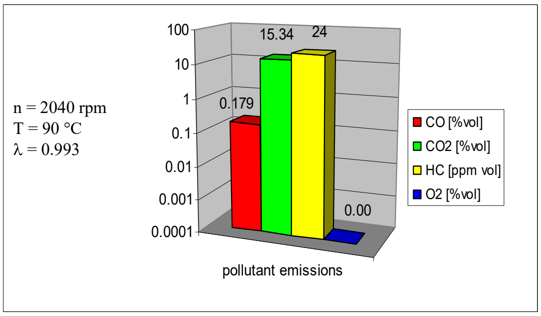
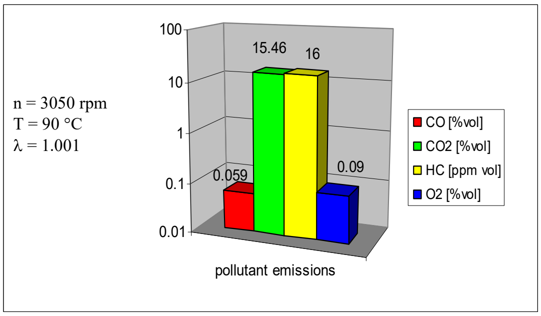


Fig. 10. Speed characteristics.

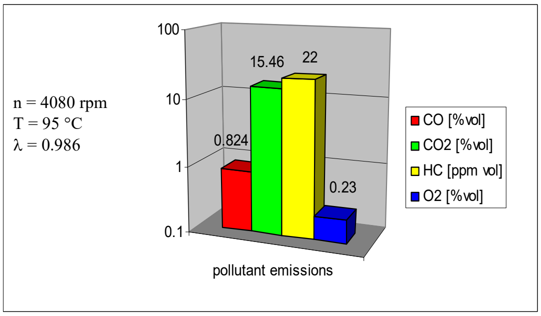
The comparative variations of the carbon monoxide (CO), carbon dioxide (CO_2), hydrocarbon (HC), oxygen (O_2) and air coefficient (λ) for various engine speeds, are shown in Fig. 11 (a), (b), (c), (d). The obtained average reductions are of 22%. The improvement results from the fact that for this gasoline injection solution, there is no need for a quite reach mixture in order to provide ignition and burning stability. Globally, the obtained reductions can be explained, basically, by using poorer mixtures, with rates of the air excess coefficient bigger to 1.



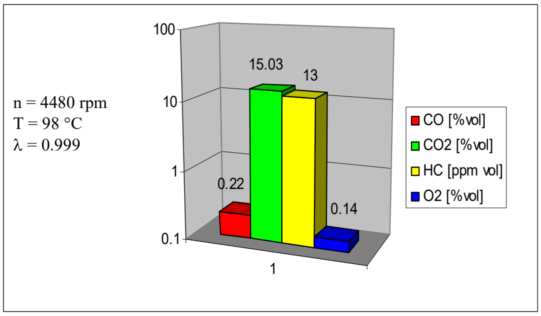
(a)



(b)



(c)



(d)

Fig. 11. The variations of pollutant emissions, (a) n=2040 [rpm]; (b) n=3050 [rpm]; (c) n=4080 [rpm]; (d) n=4480 [rpm].

6. Conclusions

• The PMACC solution has led to the improvement of engine's efficiency due to the increase of the coefficient of admission and compression ratio and to the decrease of thermal losses and exhaust gases temperature.

• The engine's rated power and the maximum torque have increased.

• The fuel consumption has diminished due to the increase of the excess air coefficient λ up to 1, without affecting the engine's stability.

• The carbon monoxide (CO) and the hydrocarbon (HC) emissions have registered a drop down.

• For this type of pump by means of the separating element the direct contact between the working liquid and the elements of the pump, that are friction couples and need lubrication, is avoided.

• The wear that appears inside the injection pump is manifest only at the level of the separating elements, that are relatively simple pieces, cheaper ones, that can be easily replaced.

References

[1] Heisler H, Advanced Engine Technology, SAE International, 1995.
[2] Heywood BJ. Internal Combustion Engine Fundamentals, McGraw – Hill Series in Mechanical Engineering, Library of Congress Cataloging-in-Publication Data, 1988.
[3] Finsterwalder G, Kuepper H. The Deutz Stratified Charge Process, Inst. Mech. E. Conf. on Stratified Charge Automotive Engines, 9-15, London, 1980.
[4] Gruden D, Markovac U, Lorcher H. Development of the Porsche SKS Engine, Inst. Mech. E. Conf. on Stratified Charge Engines, 21-27, London, 1976.
[5] Gruden D, Brachert Th.F. Wurster, W., Porsche One, Six and Eight Cylinder Stratified Charge Engine with Divided Combustion Chamber (SKS-Engine), Inst. Mech. E. Conference on Stratified Charge Automotive Engines, 55-62, London, 1980.
[6] Pischinger F, Kreykenbohm B, Adams W. Experimental and Theoretical Investigations on a Stratified Charge Engine with Prechamber Injection, Inst. Mech. E. Conference on Stratified Charge Automotive Engines, 1-6, London, 1976.
[7] Pischinger F, Adams W. Influence of Intake Swirl on the Characteristics of a Stratified Charge Engine with Prechamber Injection, Inst. Mech. E. Conference on Stratified Charge Automotive Engines, 1-8, London, 1980.