# RESEARCH OF CYCLE BY CYCLE VARIATON OF SPARK IGNITION ENGINE USING METHANE AS A FUEL

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**ABSTRACT:** *In the article is researched the cycle by cycle variation of SI engine working with methane. On investigation is made analysis what is the effect of the fuel on the value of the maximum pressure into the cylinder from cycle to cycle under different engine operating modes when changing the air–fuel equivalence ratio and the angle of supply of the electric spark. From the collected experimental data on the graphs is shown how is chance the cycle by cycle variation depending from the air–fuel equivalence ratio at three different angles of electric spark delivery.*

**KEY WORDS:** *Internal combustion engine, methane, cycle by cycle variation.*

## 1. INTRODUCTION

Cycle by cycle variation (the value of maximum pressure into the cylinder) depends from several reasons.
- The non-permanent conditions in the engine cylinders for ignition of air-fuel mixture;
- Ignition leaks caused by the ignition system;
- Different quantity air-fuel mixture entering in the cylinder in each cycle.

As a result of increased cycle by cycle variation in the operation of ICE, the specific fuel consumption, the amount of toxic components in the exhaust gases and the noise and vibrations emitted by the engine are increased.

The cycle by cycle variation of engine performance is influenced by the uneven distribution of air-fuel mixture`, depending on the type of fuel (liquid or gaseous) and on the component composition of the fuel. Influence on the homogenization of air-fuel mixture also affects the hydraulic losses in the intake system.

## 2. ANALISYS AND STUDY

### PURPOSE OF THE RESEARCH

Studies have been conducted on SI engine, modified for work with gaseous fuel. The experiments are aimed at investigating the cycle by cycle variation of the engine's working process when changing the regulation parameters, when engine using gas (methane) as a fuel. For each item tested, they are captured using an analog-to-digital converter, a plurality of indicator diagrams are then presented in a text file for easier post-processing. A single saved file contains at least 100 indicator charts. Some of the characteristics of the used fuel the experiments are shown in Tab. 1.

**Table 1.** Table caption

| Parameter | Value | Dimension |
|---|---|---|
| Lower heat value of fuel | 45 670 | [kJ/ kg] |
| Density (measure condition - 20°C; 760 mm Hg) | 0, 00075 | [g/cm$^3$] |
| Molecular mass | 16,77 | [kg/mol] |
| Burning speed of fuel | 33 | [sm/s] |
| Quantity of air needed to burning 1 kg of fuel | 16,20 | [kg] |
| Octane number | 130 | |
| Auto-ignition temperature | 540 | [°C] |
| Mass composition H/ C | 23,5/ 73 | [%] |

## 3.ANALYSYS OF RESULTS

By measuring the pressure variation in the cylinder during engine work, variations in the maximum pressure across the different cycles can be reported. The cycle by cycle variation during the burning process as a results from: the non-uniformity of the mixture entering the cylinder, the different amount of fuel-air mixture entering the cylinder in each cycle, the varying amount of residual gases for the various cycles that affect the combustion of the fuel mixture and misfire. Different values of the maximum cylinder pressure in each cycle are the reason for varying engine power and performance.

Evaluation of cycle by cycle variation of cycles can be given by the coefficient of variation $\mathbf{K_H}$, shows the chance of maximum value of cylinder pressure [1]. It represents the ratio of the standard deviation of the mean maximum pressure relative to the average value of the maximum cylinder pressure.

$$K_{H_{p\max}} = \frac{\overline{\sigma}_{p\max}}{\overline{p}_{\max}} \times 100[\%] \qquad (1)$$

where:

$\overline{\sigma}_{p\max}$ - standard deviation of the maximum pressure;

$\overline{p}_{\max}$ - average value of maximum pressure.

The calculation is made on the basis of the recorded indicator diagrams, by processing a larger number of diagrams (minimum 100). To ensure normal engine performance, the cycle by cycle variation, according to Professor Heywood, should not exceed 10% [1].

As a result of increased cycle by cycle variation in the operation of ICE, the specific fuel consumption, the amount of toxic components in the exhaust gases and the noise and vibrations emitted by the engine are increased.

The following figures (Fig. 1, Fig. 2, Fig. 3 and Fig. 4) show the changing of the coefficient of variation of the test engine in function of the Air–fuel equivalence ratio **α** at three different angles of supply of electric spark.

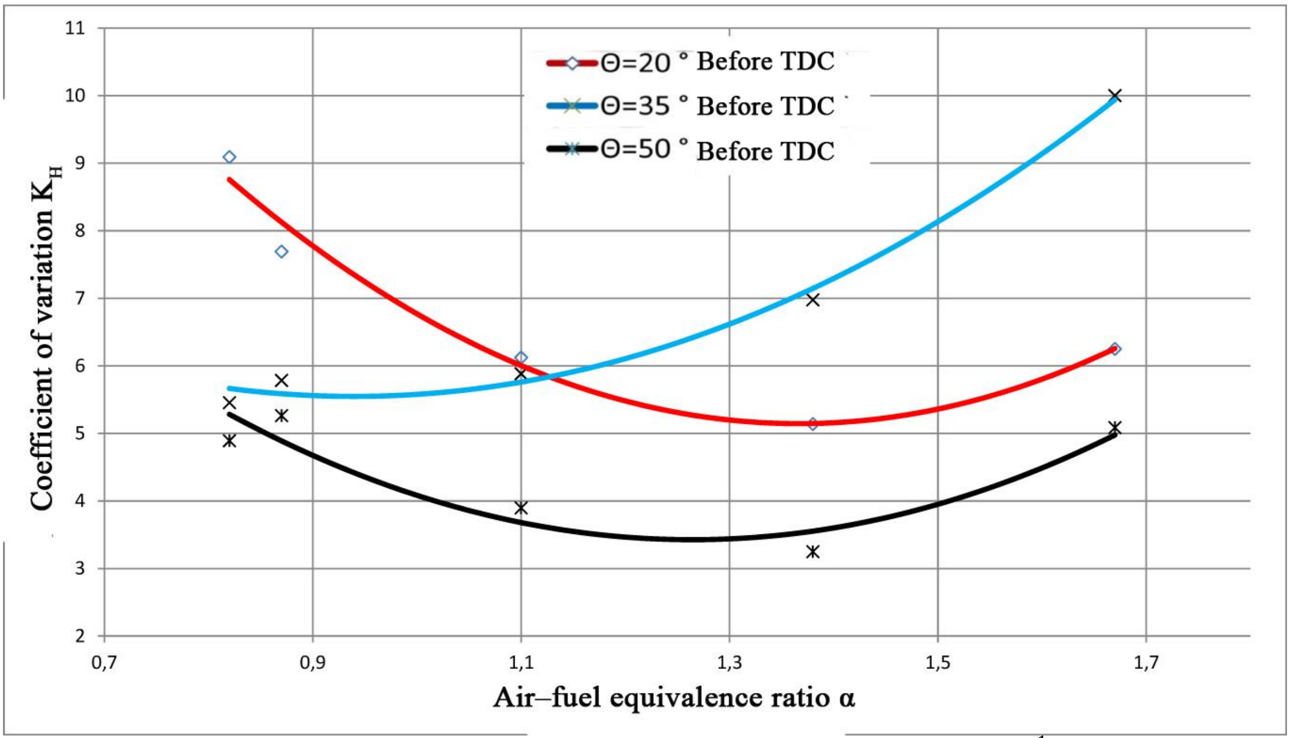


Figure 1. Change of coefficient of variation at rpm 2000 $min^{-1}$

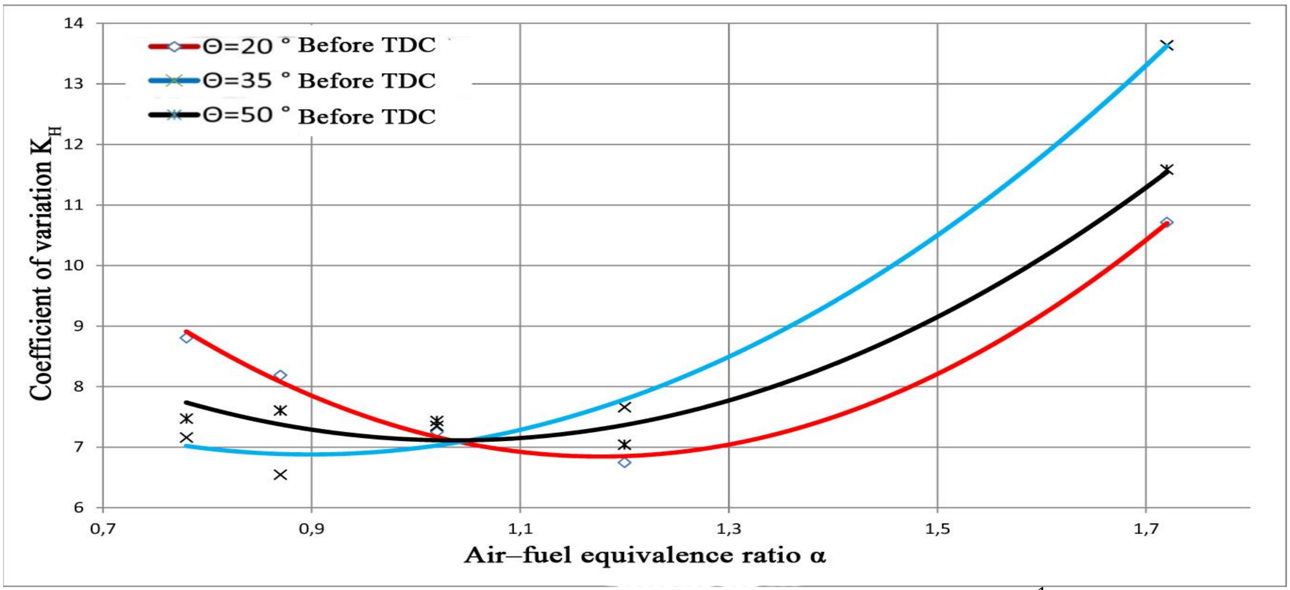


Figure 2. Change of coefficient of variation at rpm 3000 $min^{-1}$

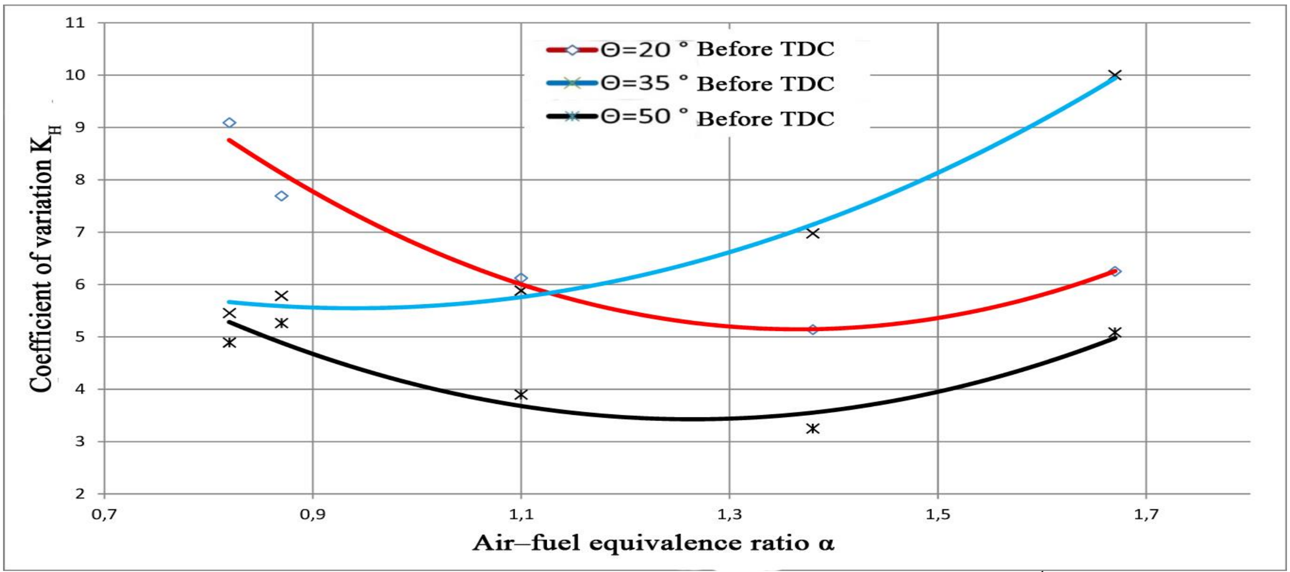


Figure 3. Change of coefficient of variation at rpm 3500 $min^{-1}$

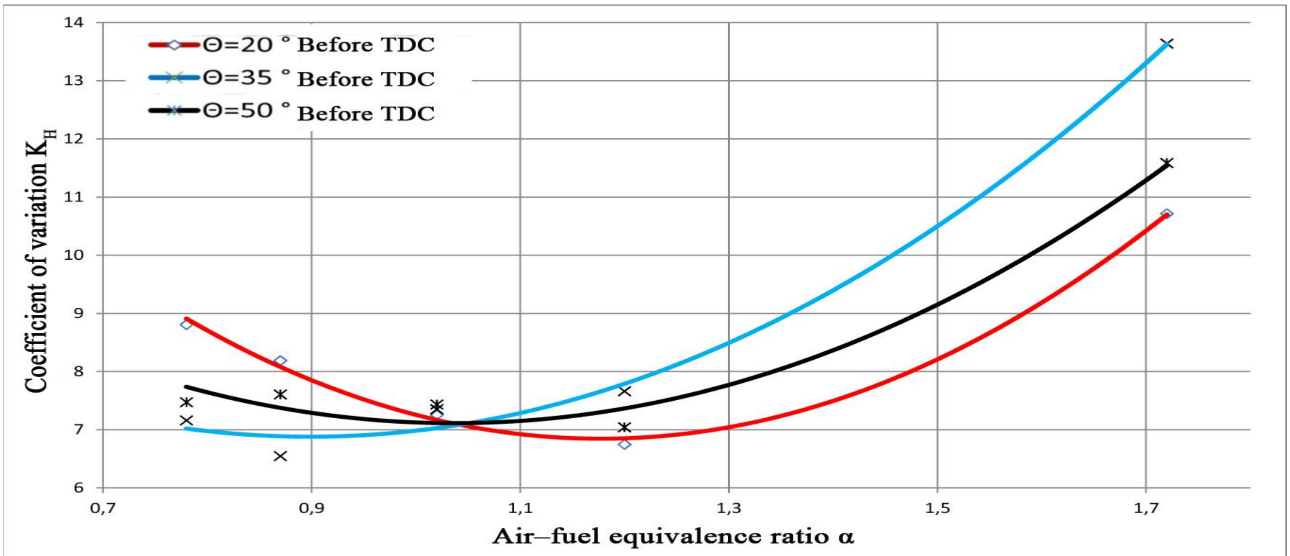


Figure 4. Change of coefficient of variation at rpm 4500 $min^{-1}$

At a lower angle of supply of electric spark than the optimum, the coefficient of variation is higher due to the ineffective combustion process, increased cylinder losses and displacement of combustion along the expansion line. Maximum $K_H$ values are obtained for very poor mixtures due to the non-homogeneity of the air mixture and the more difficult ignition of the engine mixture entering the cylinder.

At rpm 2000 $min^{-1}$, angle of supply of electric spark Θ=20° before TDC and air-fuel equivalence ratio α=0,82 , α=1,1 , α=1,67, coefficient of variation have values respectively $K_H$=9,09% , $K_H$= 6,12%, $K_H$=6,25%. At Θ=35° before TDC and α=0,82 , α=1,1 , α=1,67, coefficient of variation have values respectively $K_H$=5,45% , $K_H$=5,88%, $K_H$=10%. At Θ=50° before TDC and α=0,82 , α=1,1 , α=1,67, coefficient of variation have values respectively $K_H$=4,89% , $K_H$=3,89%, $K_H$=5,08%.

At rpm 3000 min-1, angle of supply of electric spark Θ=20° before TDC and air-fuel equivalence ratio α=0,78 , α=1,02 , α=1,72, coefficient of variation have values respectively $K_H$=8,81% , $K_H$= 7,26%, $K_H$=10,7%. At Θ=35° before TDC and α=0,78 , α=1,02 , α=1,72, coefficient of variation have values respectively $K_H$=7,16% , $K_H$=7,35%, $K_H$=13,64%. At Θ=50° before TDC and α=0,78 , α=1,02 , α=1,72, coefficient of variation have values respectively $K_H$=7,47% , $K_H$=7,43%, $K_H$=11,58%.

At rpm 3500 min-1, angle of supply of electric spark Θ=20° before TDC and air-fuel equivalence ratio α=0,78 , α=0,98 , α=1,56, coefficient of variation have values respectively $K_H$=16,13% , $K_H$= 11,46%, $K_H$=15,15%. At Θ=35° before TDC and α=0,78 , α=0,98 , α=1,56, coefficient of variation have values respectively $K_H$=7,69% , $K_H$=8,49%, $K_H$=13,16%. At Θ=50° before TDC and α=0,78 , α=0,98 , α=1,56, coefficient of variation have values respectively $K_H$=5,71% , $K_H$=3,7%, $K_H$=13,8%.

At rpm 4500 min-1, angle of supply of electric spark Θ=20° before TDC and air-fuel equivalence ratio α=0,85 , α=1,05 , α=1,62, coefficient of variation have values respectively $K_H$=18,6% , $K_H$= 17,2%, $K_H$=19,12%. At Θ=35° before TDC and α=0,85 , α=1,05 , α=1,62, coefficient of variation have values respectively $K_H$=14,61% , $K_H$=10,1%, $K_H$=13,3%. At Θ=50° before TDC and α=0,85 , α=1,05 , α=1,62, coefficient of variation have values respectively $K_H$=9,59% , $K_H$=6,94%, $K_H$=10,56%..

Of interest is also the change of the coefficient of variation depending on the crankshaft rotation speed. In higher rpm, the coefficient of variation also has higher values due to insufficient mixing formation time.

## CONCLUSION

Increasing the crankshaft rotation speed also increases the cyclic by cycle variation of the engine expressed in a change in the maximum pressure $p_z$ in to the cylinder, between the individual operating cycles.

At low rotational speeds, the cyclic by cycle variation is about 10% and reaches 32% at n=4500 $min^{-1}$.

At n=2000 $min^{-1}$ → the change of value of maximum pressure in to cylinder is pz = 4,2 ÷ 4,73 MPa; At n=3000 $min^{-1}$ → pz=5 ÷ 5,48 MPa; At n=3500 $min^{-1}$ → pz=4.77 ÷ 5.66 MPa; At n=4500 $min^{-1}$ → pz=3.53 ÷ 4.5 MPa. This is due to a reduction of combustion and mixture formation time.

## REFERENCES

[1] Heywood, J. Internal combustion engines Fundamentals. McGraw Hill, Inc., USA, 1988

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