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ANALYSIS OF THE THERMAL EFFICIENCY AND CYCLIC VARIATIONS IN A SI ENGINE UNDER LEAN COMBUSTION CONDITIONS

Mehmet Akif CEVİZ*, İlhanVolkan ÖNER*, Ferhat KAYA* and Tevhit KARACALI**
* Department of Mechanical Engineering, Faculty of Engineering, University of Atatürk, Erzurum 25240, Turkey, aceviz@atauni.edu.tr, ivoner@atauni.edu.tr, fkaya@atauni.edu.tr
** Department of Electric and Electronic Engineering, Faculty of Engineering, University of Atatürk, Erzurum 25240, Turkey, tevhit@atauni.edu.tr

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Abstract: There are significantly variations in cylinder pressure traces from cycle to cycle in spark ignition engines. The variations, named cyclic variations, exist even when engine is stable. Cyclic variability must be considered in the design and control of spark ignition engines and are magnified under certain engine operating conditions. In this work, effects of air excess coefficient on cyclic variability, engine performance characteristics and engine emission characteristics were investigated at lean operation conditions. The test engine is a spark ignition engine with electronically controlled fuel injectors. According to the test results, engine air excess coefficient can be increased to improve engine thermal efficiency and to decrease the harmful exhaust emissions under low load conditions without exceeding the limit of cyclic variability.
Keywords: Thermal efficiency, Cyclic variations, SI engines, Exhaust emissions.

FAKİR YANMA ŞARTLARI ALTINDA BUJİ ATEŞLEMELİ BİR MOTORDA TERMAL VERİM VE ÇEVRİMSEL FARKLARIN ANALİZİ

Özet: Buji ateşlemeli motorlarda bir çevrimden diğerine silindir basıncı değişimi incelendiğinde önemli seviyede farklılık olduğu görülür. Çevrimsel farklar olarak adlandırılan bu değişiklikler, motor kararlı halde çalışırken bile vardır. Çevrimsel farklar, buji ateşlemeli motorların dizayn ve kontrol çalışmalarında dikkate alınmalıdır ve belli motor çalışma şartlarında önemli seviyelere ulaşmaktadır. Bu çalışmada fakir karışım şartlarında çevrimsel farklar, motor emisyon karakteristikleri ve motor performans karakteristikleri üzerinde hava fazlalık katsayısının etkileri incelenmiştir. Test motoru, elektronik kontrollü yakıt enjektörleri bulunan buji ateşlemeli bir motordur. Test sonuçlarına göre, düşük yük şartları altında çevrimsel fark sınırı aşılmadan hava fazlalık katsayısı artırılabilir, bu esnada motor termal verimi artırılırken zararlı egzoz emisyonları da azaltılabilir.
Anahtar Kelimeler: Termal verim, Çevrimsel farklar, Buji ateşlemeli motorlar, Egzoz emisyonları.

INTRODUCTION

Among the several techniques that are available for improving the fuel economy and emission characteristics of a spark ignition engine, the lean combustion engine concept can be considered as a viable technique. Using of the lean combustion strategy with a carefully optimized SI engine decreases the harmful exhaust emissions. Another benefit of lean operation is increased thermal efficiency due to the increase in the ratio of specific heats for lean mixtures (Manivannan et al., 2003; Ceviz and Kaymaz, 2005). However, cyclic variations are particularly severe for lean air-fuel mixtures (i.e., when the ratio of air to fuel is greater than that required by chemical stoichiometry). As the mixture leaned, cyclic variations increase.

The existence of cyclic combustion variations in spark ignition internal combustion engines has long been recognized. From the measurements of the pressure-time history of consecutive cycles in the combustion chamber in a SI engine, it can easily be seen that variations from one cycle to another exist. One of the many factors that must be considered in the design and control of spark ignition engine is the cyclic variations.

To minimize the cyclic variations is an important goal for both spark and compression ignition engines. Previous studies have shown that if cyclic variability could have been eliminated, there would be a 10% increase in the power output for the same fuel consumption. Since the pressure rate is uniquely related to the combustion, the pressure variations are caused by variations of the combustion process. When the measured consecutive in-cylinder pressure cycle data is transferred to a chart at the same crank angle especially on the combustion stroke (as shown in figure 2), some deviations can be seen from the mean cycle. The fast combustion cycles causing deviations determine the upper compression ratio for chosen fuel. On the other hand in the slow combustion cycles, there is a risk of uncompleted combustion when the exhaust valve opens, which will result in higher unburned hydrocarbon (UHC) emissions and lower efficiency (Ozdor et al., 1994; Scholl and Russ,1999; Ceviz and Yüksel, 2005).

The cyclic variations are recognized as a limit for the range of lean and highly diluted operating conditions. They are usually attributed to the result of random fluctuations excess air ratio and flow field due to the turbulent nature of the flow in cylinder and limit the range of operating conditions of spark ignition engines.

The cyclic variations are caused by both chemical and physical phenomena. Of these phenomena, the variations in the residual gas fraction, the air-fuel ratio, the fuel composition and the motion of unburned gas in the combustion chamber can be taken into consideration. It is proposed to relate each of the cyclic variation factors to one of the four following sources:

1. Mixture composition,
2. Cyclic cylinder charging,
3. Spark and spark plug,
4. In-cylinder mixture motion.

The factors which belong to the second group influence mostly the main stage of combustion, while all other factors play role at any of the combustion stages (Ozdor et al., 1994).

The objective of the study reported in this paper is to analyze the cyclic variability and emission characteristics of gasoline-fuelled spark ignition engine at lean and low load operating conditions to handle the advantage of lean combustion without exceeding the limit of cyclic variability. Variations in indicated mean effective pressure and in-cylinder maximum pressure, engine thermal efficiency and specific fuel consumption are presented in relation to air excess coefficient. Variations in the CO, CO_2, NO and HC emissions are also discussed.

MATERIALS AND METHODS

A schematic layout of the test setup used is indicated in figure 1. The specifications of the engine are presented in Table 1.

The engine was coupled to a hydraulic dynamometer. The engine tests were performed at the same engine speed (2500 rpm) by loading the engine with a hydraulic dynamometer, at thirteen different air-fuel ratio (0.974, 0.989, 1.087, 1.116, 1.170, 1.199, 1.246, 1.305, 1.339, 1.374, 1.392, 1.420 and 1.453) after running the engine for some time until it reached steady state.

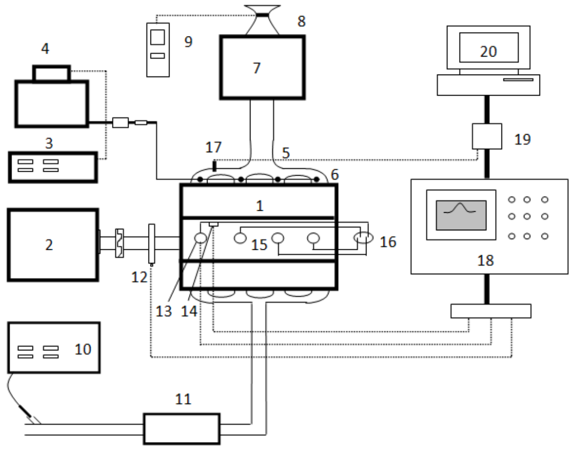

Figure 1. Schematic diagram of experimental apparatus.

1- Engine
2- Hydraulic Dynamometer
3- Fuel Flow Meter Unit
4- Gravimetric Fuel Flow Meter
5- Intake Manifold
6- Fuel Injectors
7- Air Surge Tank
8- Air Flow Meter Probe (Hot Wire)
9- Air Flow Meter
10- Exhaust Gas Analyzer
11- Muffler
12- Shaft Encoder
13- Spark Plug Mounted Piezoelectric Transducer
14- Ignition Pick-Up
15- Spark Plugs
16- Distributor
17- MAP Sensor
18- Charge Amplifier and Oscilloscope
19- Analog to Digital Converter
20- Personal Computer

Table 1. Engine Specifications.

Engine type	Ford MVH-418, fuel injected
Number of cylinders	4
Compression ratio	10:1
Bore (mm)	80.6
Stroke (mm)	88
Displacement volume (dm^3)	1.796
Max. power	93 kW at 6250 rpm
Max. torque	157 Nm at 4500 rpm
Cooling system	Water-cooled

Engine torque was measured with a load cell, and speed was measured using 60 toothed sprocket and magnetic pick-up. In-cylinder pressure measurement was accomplished using KISTLER, 6117BFD17 type quartz pressure transducer connected to a charge amplifier. Crank angle was measured by an optical shaft encoder. The optical shaft encoder was rigidly mounted to the front of the engine and connected to the crank shaft with a flexible coupler. The data from charge amplifier including cylinder pressure and encoder including TDC were collected by National Instruments high speed M series, 6250 model, 1.25 MS/s DAQ Card in a Intel(R) 4 CPU 2.00 GHz personal computer. Data collected using a software package NI-DAQmx obtained from National Instruments Corp. Analysis of combustion parameters and heat release was performed by using software developed in MATLAB for the purpose after obtaining the average pressure signal of 100 consecutive engine cycles. Cylinder pressure data was pegged by assuming the pressure at bottom dead centre after the intake stroke was equal to the mean intake manifold pressure [Lancester et al., 1975]. The exhaust emission measurements were accomplished with an exhaust analyzing system from BOSCH, BEA 270 exhaust analyzer.

Coefficients of variation in indicated mean effective pressure (imep) and in cylinder maximum pressure were used to investigate the cyclic variability. Coefficients of variation in indicated mean effective pressure (imep) is the standard deviation in imep divided by the mean imep (Heywood, 1988) and usually expressed in percent as:

$$COV_{imep} = \frac{\sigma_{imep}}{imep} \times 100 \qquad (1)$$

The indicated mean effective pressure is easy to calculate and provides a measure of the work produced for an engine cycle. imep is defined as:

$$imep = \frac{W_c}{V_d} \qquad (2)$$

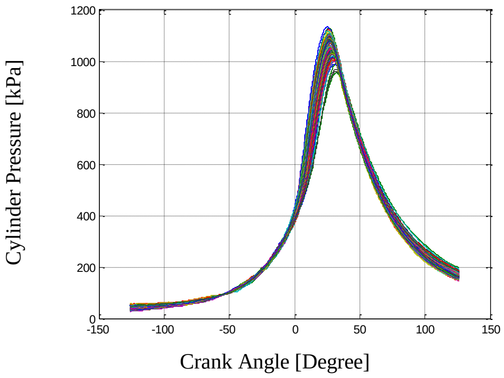

Figure 2. An example of the cyclic variations for 100 consecutive cycles in cylinder pressure.

where V_d is the engine displacement volume and W_c is the work per cycle, defined as:

$$W_c = \oint PdV \qquad (3)$$

The cyclic variations for 100 consecutive cycles in cylinder pressure for compression and combustion strokes, and the average of these pressure data for the 0.989 level of air excess coefficient can be seen from figure 2 and figure 3, respectively. Additionally, figure 4 and figure 5 show the variation of in-cylinder maximum pressure and indicated mean effective pressure during 100 consecutive cycles, respectively.

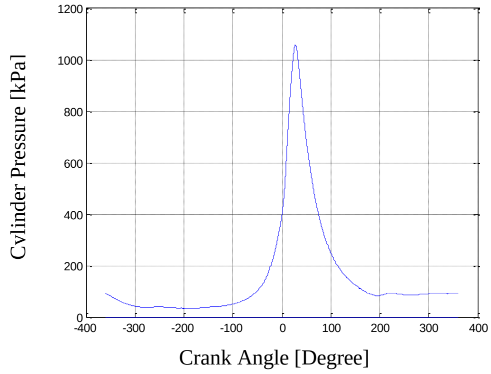

Figure 3. Average of 100 consecutive cycles in cylinder pressure.

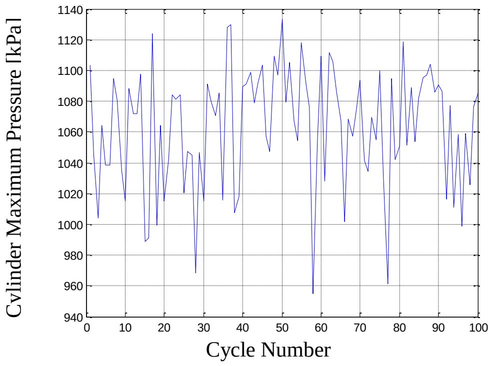

Figure 4. In-cylinder maximum pressure during 100 consecutive cycles.

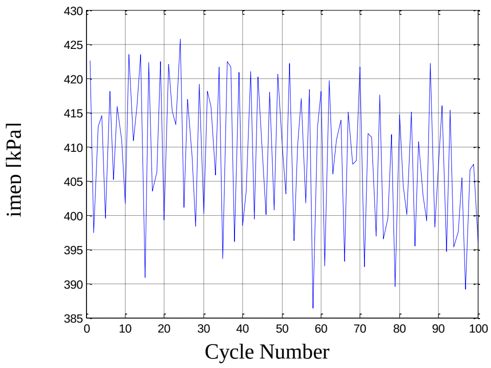

Figure 5. Indicated mean effective pressure during 100 consecutive cycles.

RESULTS AND DISCUSSION

Figure 6 and figure 7 show the effects of the air excess coefficient on the coefficient of variation in indicated mean effective pressure (COV_{imep}), and the coefficient of variations in in-cylinder maximum pressure (COV_{Pmax}), respectively. By using these figures, the combustion stability can be analyzed from near stoichiometric to extra lean combustion conditions since both COV_{imep} and COV_{Pmax} are indicators of cyclic variations. It is apparent from the figure 6 and figure 7 that the increase in the air excess coefficient increased the cyclic variations almost linearly up to 1.30 level of air excess coefficient. After this level of air excess coefficient, the rate of increase in the cyclic variations grew faster.

Figures 8, 9, 10 and 11 show the emission characteristics of the engine. Figure 8 and figure 9 present the variations of the CO and CO_2 emissions with the air excess coefficient, respectively. The most important engine parameter influencing carbon monoxide emissions is the air excess coefficient. All other variables cause second order effects. In lean running engines, flame-fuel interaction with the walls, the oil films, and the deposits become more important on CO emissions. It can be concluded from the figure 8 that for the lean mixtures, CO emission concentrations in the exhaust are low enough after 1.08 level of air excess coefficient. It can be seen from this figure that the CO emissions decreased dramatically down to 1.10 level of air excess coefficient, and continued to decrease as the air excess coefficient increased. The reason of the continuously decrease in the CO emissions when the air excess coefficient increases is the availability of more and more oxygen for oxidation reactions. Figure 9 shows that the CO_2 emissions decreased linearly with the increase in the air excess coefficient. The main effect on the decrease in the CO_2 emission concentration is the continuously increase in the N_2 and O_2 concentration in the exhaust gas components as the air excess coefficient increases.

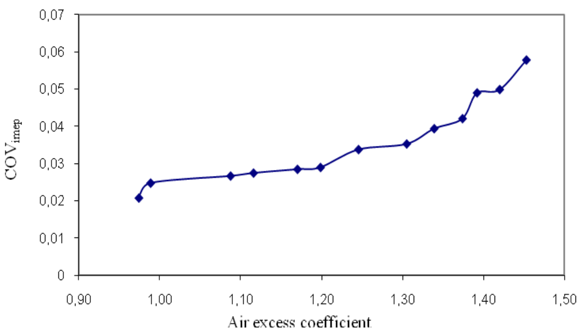

Figure 6. The effect of air excess coefficient on the coefficient of variation of indicated mean effective pressure.

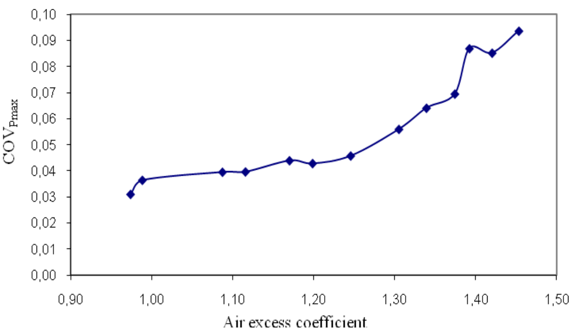

Figure 7. The effect of air excess coefficient on the coefficient of variation of in-cylinder maximum pressure

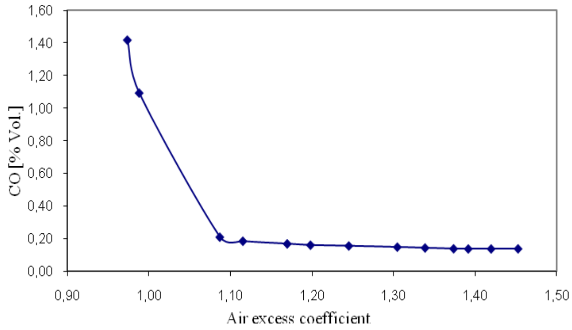

Figure 8. The effect of air excess coefficient on the CO emissions

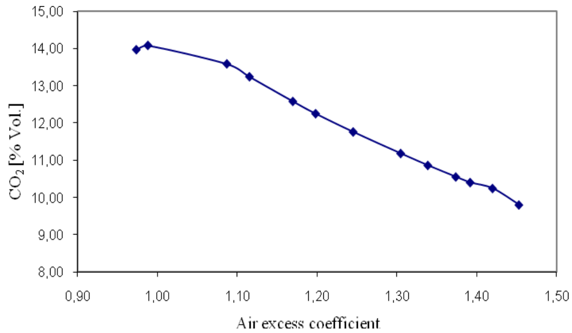

Figure 9. The effect of air excess coefficient on the CO_2 emissions.

Total hydrocarbon emission is a useful measure of combustion efficiency. It is known that HC emission level decrease for moderately lean mixtures because of decreasing fuel concentration and increasing oxygen concentration. These parameters offset the effect of decreasing bulk gas temperatures. As the lean operating limit of the engine is approached, combustion quality deteriorates significantly and HC emissions start to rise again due to the occurrence of occasional partial burning cycles. As it can be seen from figure 10, the HC emission decreased up to 1.25 level of air excess coefficient, and started to increase after this level of air excess coefficient.

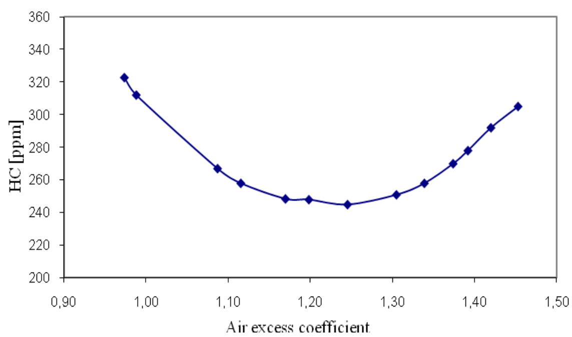

Figure 10. The effect of air excess coefficient on the HC emissions.

It is known that the nitric oxides are maximized with mixtures slightly lean of stoichiometric. The increased temperatures favor nitric oxide formation and burned gas temperatures are maximized with mixtures that are slightly rich (Ferguson and Kirkpatrick, 2001). On the other hand, there is little excess oxygen in rich mixtures to dissociate and attach to nitrogen atoms to form nitric oxide. The interplay between these two effects results in maximum nitric oxides occurring in slightly lean mixtures, where there is a slight excess of oxygen atoms to react with the nitrogen atoms. As it can be seen from figure 11, NO emission reached to the maximum value at the 1.10 level of air excess coefficient which was slightly lean of stoichiometric. After this value of air excess coefficient, NO emission was decreased linearly because of the effect of the decrease in the combustion temperatures.

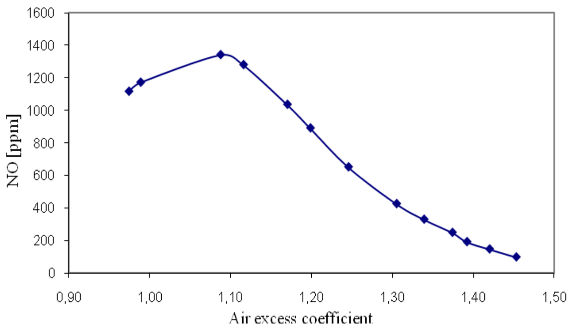

Figure 11. The effect of air excess coefficient on the NO emissions.

Engine thermal efficiency is the ratio of work to the amount of fuel energy that can be released in the combustion process. Another indicator of engine effectiveness is the specific fuel consumption characteristic. The specific fuel consumption is the fuel flow rate per unit power output. It measures how efficiently an engine is using the fuel supplied to produce work. It can be seen from figure 12 and figure 13 that as the air excess coefficient increased, the thermal efficiency increased and the specific fuel consumption decreased up to 1,20 level of air excess coefficient. Combustion of mixtures leaner than stoichiometric produces products at lower temperature, and with less dissociation of the triatomic molecules CO_2 and H_2O. Additionally, it causes an increase in the ratio of specific heats. Thus the fraction of the chemical energy of the fuel which is released as sensible energy near top dead centre is greater. Hence a greater fraction of the fuel's energy is transferred as work to the piston during expansion, and the fraction of fuel's available energy rejected to the exhaust system decreases. It can also be seen from figure 12 and figure 13 that the increase in the thermal efficiency and the decrease in the specific fuel consumption appeared at slightly lean mixtures. While these engine performance characteristics were improved by making the mixture leaner, a reverse affect appeared after especially 1.20 level of air excess coefficient. After this point of operation, the increase in cyclic variations dominated. As mentioned above, cyclic variations caused roughly operation of the engine and uncompleted combustion prior to exhaust valve opening due to the occurrence of occasional partial-burning cycles.

It can be deducted from the figures presenting the variation of in-cylinder pressure on cycle to cycle base (figures 6-7), the variation of engine exhaust emissions (figures 8, 9, 10 and 11) and the variation of the engine fuel conversion efficiency (figures 12-13) that the using of lean burn strategy with a carefully optimized spark ignition engine is beneficial. However, because of the dramatic increase in the cyclic variations and total duration of burning process as the mixture approaches the lean misfire limits.

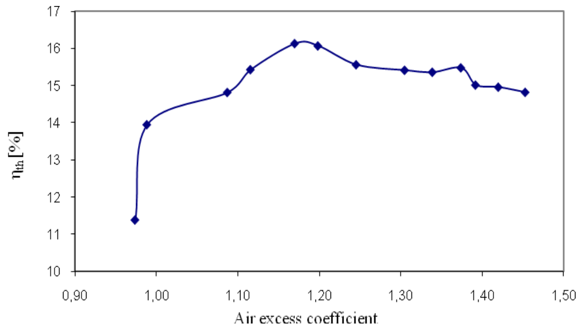

Figure 12. The effect of air excess coefficient on the engine thermal efficiency.

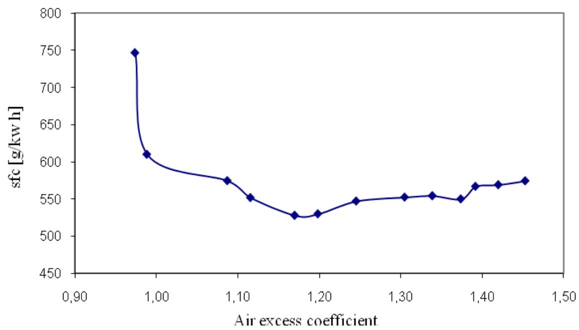

Figure 13. The effect of air excess coefficient on the specific fuel consumption**.**

CONCLUSIONS

From the study given in this paper, the followings can be deduced:

1. The engine can be operated at low load conditions about 1.20 level of air excess coefficient without exceeding the limit of cyclic variability.

2. Increase in the air excess coefficient leads significant reduction in exhaust emissions by about 85% and 23% for CO and HC emissions, respectively, at 1.20 level of air excess coefficient experiments compared to

stoichiometric conditions. At this operating conditions, the thermal efficiency increases by about 30%.

3. As the air excess coefficient exceeded from about 1.25 level, increase in the cyclic variability grew faster, and specific fuel consumption and HC emissions started to increase.

4. It can be concluded from all the experimental results, under low load conditions, lean burn combustion spark ignition engines offer some important advantages when the engine was carefully designed and air-fuel mixture formation strategy was optimized.

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M. Akif CEVİZ was born in Erzurum, Turkey in 1976. He received the B.Sc. degree from the Department of Mechanical Engineering, Ataturk University, Erzurum, Turkey, in 1997 and, the M.Sc. and Ph.D. degree from the same university, in 2000 and 2004, respectively.
From 1998 to 2004, he was a Research Assistant at the Department of Mechanical Engineering, Ataturk University. From 2004 to 2008, he was an Assistant Professor. Since 2008, he has been employed as an Associate Professor at the same department of Ataturk University. His research interest involves internal combustion engines, combustion modeling and alternative fuels.

İlhan Volkan ÖNER was born in Ordu, Turkey in 1979. First, middle and high school education was completed in different schools in Gölcük/Kocaeli and Ordu. He received the B.Sc. degree from the Department of Mechanical Engineering, Ataturk University, Erzurum, Turkey, in 2000 and M.Sc. degree in Energy Science of Mechanical Engineering from Graduate School of Natural and Applied Sciences the Ataturk University, Erzurum, Turkey, in 2009. Ph.D. education began in 2009 continues in the same section. Since 2003, he has been working as a mechanical engineer in the Ministry of National Defense. His areas of interest are internal combustion engines and alternative fuels.

Ferhat KAYA was born in Kars, Turkey in 1974. He received the B.Sc. degree from the Department of Mechanical Engineering, Ataturk University, Erzurum, Turkey, in 1997 and his M.Sc. education began in Energy Science of Mechanical Engineering from Graduate School of Natural and Applied Sciences the Ataturk University, Erzurum, Turkey, in 2007. His areas of interest are internal combustion engines.

Tevhit KARACALI was born in Erzurum, Turkey in 1969. He received the B.Sc. degree from the Department of Electrical and Electronics Engineering, Gazi University, Ankara, Turkey, in 1990 and, the M.Sc. and Ph.D. degree in semiconductor physics from the Ataturk University, Erzurum, Turkey, in 1998 and 2003, respectively.
From 1997 to 2004, he was a Research Assistant at the Department of Electrical and Electronics, Ataturk University. Since 2004, he has been employed as an Assistant Professor in Department of Electrical and Electronics Engineering at the Ataturk University. His research interest involves porous silicon based devices and applications, Scientific instrumentation and measurement.