Isı Bilimi ve Tekniği Dergisi, 37, 2, 129-138, 2017
J. of Thermal Science and Technology

ISSN 1300-3615

INVESTIGATION OF AN SI-CAI ENGINE FUELLED WITH METHANE-HYDROGEN MIXTURES FOR DIFFERENT EXHAUST VALVE LIFTS

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(Geliş Tarihi: 22.04.2016, Kabul Tarihi: 06.04.2017)

Abstract: In this study, a spark-assisted controlled auto-ignition (SI-CAI) engine with different Methane-Hydrogen blends was numerically and experimentally investigated under different excess air ratio and valve lift value conditions. Experimental results were used to validate the numerical study. GT-Power simulation tool was used for the numerical studies. The valve lifts were created ranging from 3.0 to 5.0 mm with 0.5 increments. The excess air ratio (λ) values were considered as 1.0, 1.1, 1.2, 1.3 and 1.4. Besides, Methane-Hydrogen blends were constituted as 100% Methane (100M), 90% Methane-10% Hydrogen (90M10H), 80% Methane-20% Hydrogen (80M20H) and 70% Methane-30% Hydrogen (70M30H) by volume. Results revealed that the peak pressure values increase when the valve lift increases. The pressure and temperature values tend to reduce with the increasing of λ values. Increasing the volume fraction of Hydrogen in Methane–Hydrogen blend contributes to pressure development earlier. As a conclusion increasing of the volume fraction of Hydrogen in the Methane-Hydrogen blend causes a reduction in the indicated thermal efficiency and mean effective pressure, and a lower specific fuel consumption.
Keywords: Methane-Hydrogen blends, GT-Power, valve lift, excess air ratio, cylinder pressure.

FARKLI EGZOZ VALF YÜKSEKLİKLERİ İÇİN BİR SI-CAI MOTORDA METAN-HİDROJEN KARIŞIMLARININ İNCELENMESİ

Özet: Bu çalışmada, farklı metan-hidrojen karışımlarının buji ateşlemeli kontrollü kendi kendine tutuşmalı bir motorda (SI-CAI) farklı hava fazlalık katsayısı ve valf yükseklik değerlerinde sayısal ve deneysel inceleme yapılmıştır. Deneysel sonuçlar sayısal çalışmanın doğruluğu için kullanılmıştır. Sayısal çalışma için GT-Power simülasyon programı kullanılmıştır. Valf yükseklikleri 0.5 mm artış değeri ile 3.0-5.0 mm arasında ele alınmıştır. Hava fazlalık katsayısı (λ) değerleri 1.0, 1.1, 1.2, 1.3 ve 1,4 olarak dikkate alınmıştır. Bununla birlikte, metan-hidrojen karışımları hacimsel olarak %100 Metan (100M), %90 Metan-%10 Hidrojen (90M10H), %80 Metan-%20 Hidrojen (80M20H) ve %70 Metan-%30 Hidrojen (70M30H) olacak şekilde incelenmiştir. Sonuçlar valf yüksekliklerinin artması ile maksimum basınç değerlerinin arttığını göstermiştir. Hava fazlalık katsayısının artması ile basınç ve sıcaklık değerlerinde azalma eğilimi görülmüştür. Metan-hidrojen karışımlarında hidrojenin hacimsel oranının artırılması ile basınç gelişimlerinin erken gerçekleştiği görülmüştür. Sonuç olarak, metan hidrojen karışımlarındaki hidrojenin hacimsel oranının artışı indike ısıl verimde ve ortalama efektif basınçta azalmaya ve daha düşük özgül yakıt tüketimine neden olmuştur.
Anahtar Kelimeler: Metan-hidrojen karışımları, GT-Power, valf yüksekliği, hava fazlalık katsayısı, silindir basıncı.

ABBREVIATIONS

$BSCO_2$	brake specific carbon dioxide
BSCO	brake specific carbon monoxide
BSHC	brake specific hydrocarbon
BSFC	brake specific fuel consumption
CA	crankshaft angles
CAI	controlled auto ignition
CFD	computational fluid dynamics
CH_4	methane
CI	compression ignition
CO_2	carbon dioxide
CO	carbon monoxide
EGR	exhaust gas recirculation
GDI	gasoline direct injection
HC	hydrocarbon
HCCI	homogeneous charged compression ignition
HRR	heat release rate
IC	internal combustion
LPG	Liquid petroleum gas
NO	nitric oxide
NVO	negative valve overlap
ppm	part per million
SI	spark ignition
SI-CAI	spark assisted controlled auto ignition
VVL	various valve lifts
λ	excess air ratio

INTRODUCTION

Concerns about the increase in emissions emitted from IC engines, and increasing fuel prices due to the reduction of fossil fuel reserves lead researchers to work the incentive studies on the combustion technology of IC engines. In this regard, CAI combustion is one of the most promising alternative combustion methods in comparison with conventional spark ignition (SI) and compression ignition (CI) combustion engines. The CAI concept is based on the auto-ignition of a fuel mixture highly diluted gasses in order to achieve high indicated efficiency and low NOx and soot emissions through low-temperature combustion (Knop et al., 2009). In CAI combustion, the chemical kinetics of the mixture in a cylinder play a key role in combustion characteristics, since CAI combustion is achieved through auto-ignition of the mixture (Ebrahimi and Desmet, 2010, Yao et al., 2009). Therefore, CAI combustion is influenced by intake air temperature, air-fuel ratio, compression ratio, exhaust gas rate, auto-ignitable of a fuel. Several methods such as heating intake air, higher compression ratios, exhaust gas recirculation (EGR) and more ignitable fuels, has been investigated to achieve CAI combustion (Lee and Lee, 2007; Bai et al., 2010; Guo and Neill, 2013; Lee et al., 2013; Zhang and Wu, 2012). However, CAI combustion restricts the operating range of an engine due to challenges in the ignition timing and combustion phase control.

One of the most effective and practical means of achieving CAI combustion in an engine is EGR or exhaust trapping by negative valve overlap (NVO) (Hunicz and Kordos, 2011). EGR are used not only to initiate but also to control CAI combustion (Chen et al., 2003). The valve events are the main strategy to trap a certain rate of exhaust gas for CAI combustion. A number of studies have been performed by using various valve timing over CAI combustion. Kalian et al. (2008), studied on a modified engine for CAI combustion mode using a cam profile switching mechanism that can switch between high and low lift cam profiles for SI-CAI mode transition. Yeom et al. (2007), investigated the emissions and combustion characteristics of HCCI engine fuelled with LPG and gasoline by using variable valve timing to control the amount of residual gas in the cylinder. The characteristics of the LPG HCCI combustion mode was compared with those of the gasoline HCCI engine in their study. Cinar et al. (2015), studied on a single cylinder HCCI engine using four different cams to change valve lift. They reported that HCCI operating range can be extended by low lift cams.

Chen et al. (2014), carried out numerical and experimental studies on a single cylinder equipped with variable valve lifts and timings devices to expand the engine's operation range from low load limit to idle operation under HCCI combustion. During the engine operation, the valve lifts were adjusted from 0.3 mm to 9.5 mm with negative valve overlap strategy. In their study, it was stated that the intake backflow becomes greater and the highest temperature region occurs on the exhaust valve side when the intake valve opens earlier with a higher lift. The interaction between intake flow and residual gasses was considered as a possible means for the extended operation range of diluted HCCI combustion.

Mahrous et al. (2009), investigated 4-valve direct injection HCCI by using a 1D fluid-dynamic engine cycle simulation tool to determine the effects of variable valve timing strategy on the gas exchange process and the engine's performance. They used various valve timings and compared the results with using typical valve timing used for SI and HCCI engine. One of the results of this study is that the optimum engine performance could be achieved by actuating the intake valve at the symmetric crank angle position relative to the timing of exhaust valve closing.

CAI/HCCI combustion is achieved by controlling the temperature, pressure, and composition of the fuel and air mixture, so that it spontaneously ignites the air/fuel mixture in the engine. This special characteristic of CAI/HCCI allows the combustion to occur within very lean or diluted mixtures, it causes to achieve low exhaust temperatures that dramatically reduce engine NOx emissions. Similar to an SI engine the charge is well mixed which minimizes the particulate emissions, also inheriting the advantages of a CI engine being no throttling losses, therefore an overall higher efficiency can be achieved (Cao et al., 2005).

The use of natural gas as fuel in SI engines presents several positive aspects. The natural gas as a fuel provides high indicated thermal efficiency, low knock probability, and reduced NOx emissions. However, some problems such as low flame velocity and the rise of cyclic variability. One way to avoid these problems is Hydrogen addition to natural gas (Karim et al., 1996). Moderated concentrations of Hydrogen don't affect excellent on knock resistance properties of natural gas. Nevertheless, when the amount of Hydrogen fraction increases, the knock is observed earlier (Moreno et al., 2010).

Previous studies have been performed focusing on SI and CAI engine mode. Unlike these studies, SI-CAI combustion mode was investigated for various valve lifts (VVL), excess air ratio (λ) values and Methane-Hydrogen blends at full load. 1D cycle simulation tool, GT-power, as used for the numerical analyses. The 1D mode was generated for the numerical analyses, and numerical results were validated used experimental data. The experimental study was carried out by using a Lombardini LGW 523 two-cylinder engine, 505 cc with the compression ratio of 10:7. After the accuracy of the model was proven GT-power was used to explore the abilities of SI-CAI engines. This study reveals that the ability of the model for SI-CAI combustion model shows a good agreement with experimental studies.

NUMERICAL METHOD

Numerical studies instead of experimental studies have been shown to be useful both in terms of cost and time. Numerical solution can work faster in terms of adjusting the valve lift in an experimental system. GT-Power is software that is used to perform the engine simulation by Gamma Technologies. GT-Power calculates the flow motion in time using many different models for all parts of the engine (Fjallman, 2014). Figure 1 shows the engine simulation model as built in GT-power. The characteristic features of the used engine model used are given in Table 1. Exhaust gasses were taken from ex-port1 and ex-port2 line and send EGR valve. After that, EGR mixture was given in both manifolds.

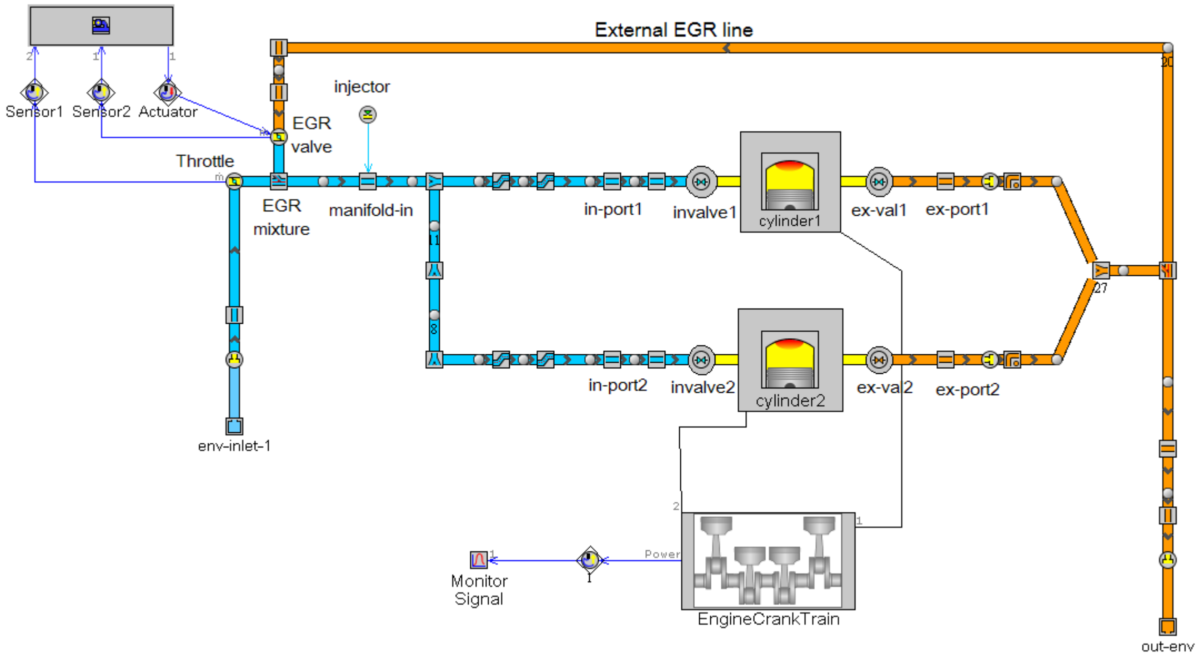

Figure 1. GT-POWER simulation model.

Table 1. The engine specifications

Description	Unit	Value
Cylinder bore	[mm]	71.5
Stroke	[mm]	62
Displacement volume	[cc]	505
Compression ratio	[-]	10.7
Intake valve diameter	[mm]	32
Exhaust valve diameter	[mm]	27

The cam profiles are defined the sub-module VT-Design of GT-Power which is a software to design camshaft and rocker mechanism for engines. The valve profiles are defined depending on valve lifts and exhaust valve opening duration. Polynomial model with the full-cam-16 method, in which the polynomial equation is divided into 16 regions, which was used in the design of cams. Valve train model is shown in Figure 2. The valve lifts based on the parameters defined in Table 2 are set to 3, 3.5, 4, 4.5 and 5 mm from this program. Figure 3 shows the valve opening width versus crank angle degree. Especially exhaust valve lifts were changed in this study.

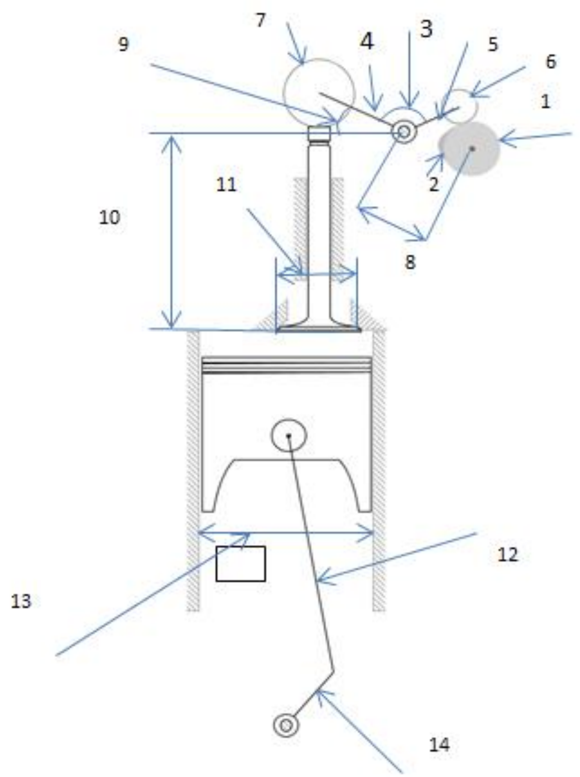

Figure 2. Valve train model.

Table 2. Valve train characteristics.

1	Base Radius	12 mm
2	Cam width	5 mm
3	Included Angle	130º
4	Valve Side Arm Length	40 mm
5	Cam Side Arm Length	26 mm
6	Roller or Contract Radius	7.5 mm
7	Valve Side Contract Radius	15 mm
8	Pivot-Cam Distance	30 mm
9	Valve Side Arm Angle	25º degree
10	Valve Height	90 mm
11	Valve Diameter	36 mm
12	Connection Rod Length	107 mm
13	Piston Bore	71.8 mm
14	Crank Throw Radius	31 mm

EXPERIMENTAL APPARATUS AND PROCEDURE

Experiments were performed to validate the numerical model developed for SI-CAI combustion mode. The experimental studies were carried out for 5 mm valve lift by reducing the cam width illustrated as corresponding to number 2 in Figure 2, from the original dimension, 7 mm. For SI-CAI operation, the engine was equipped with an air pre-heating system of 1.2 kW that can control the intake air temperature in the range 90–110 °C.

A water-cooled piezo-electric pressure transducer (Kistler 6041A) was used with connected to a charge amplifier for the in-cylinder pressure measurements; absolute pressure was calculated considering the mean intake manifold pressure. The power output of the test engine was measured by Baturalp Tayland model hydrokinetic dynamometer. The experimental setup is depicted in Figure 4. The engine was previously operated with gasoline for SI mode and after experiments were repeated for SI-CAI engine mode. All experiments were performed at full-load and steady-state conditions. For SI-CAI engine mode, intake air temperature, and EGR were used with the adjusted valve lift, 5 mm. The experiments were performed at an engine speed of 2000 rpm with Methane fuel for a stoichiometric mixture. The study was run at 2000 rpm engine speed that is one of the most common driving ranges. Methane-Hydrogen blends were used at different excess air ratio values for the other experiments.

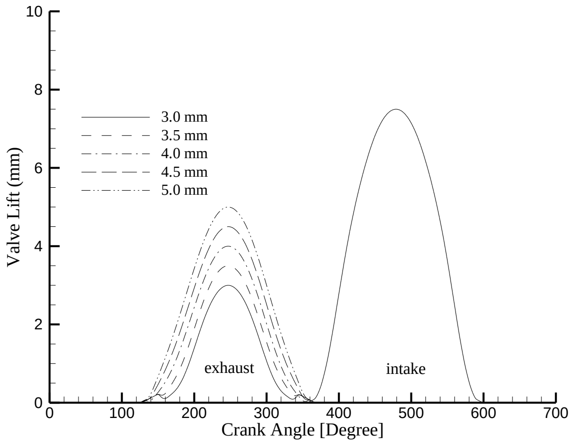

Figure 3. The amount of opening with the opening duration of the intake and exhaust valves at different valve lifts.

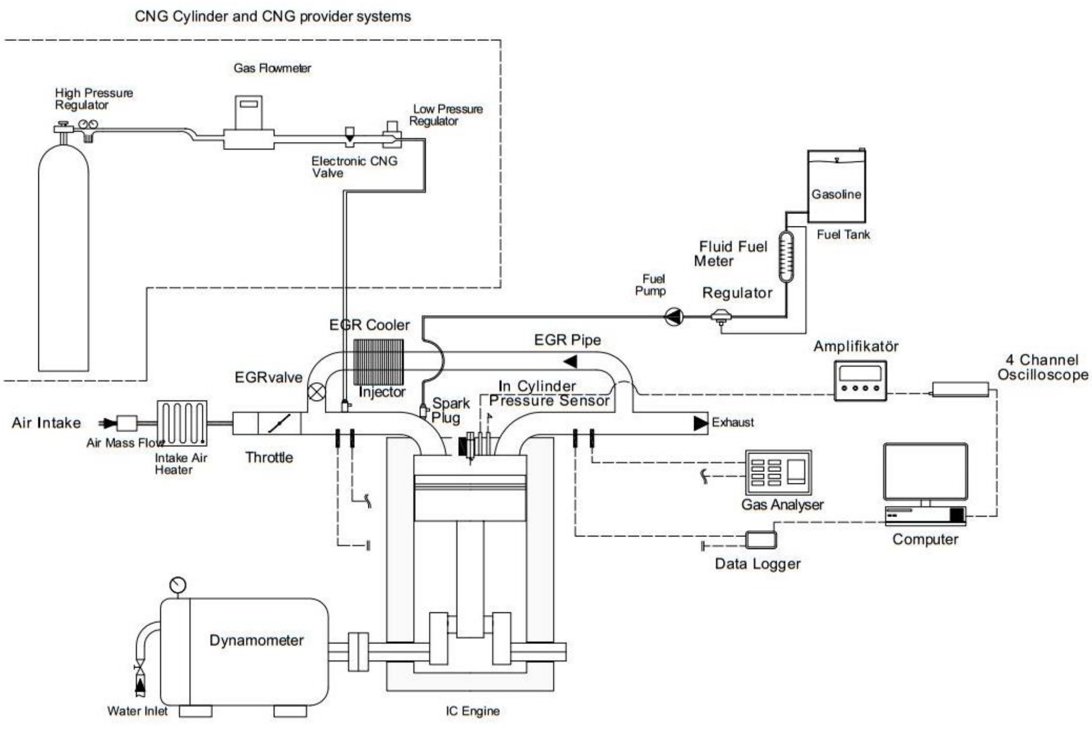

Figure 4. Experimental setup

RESULTS AND DISCUSSION

Model Validation

In order to prove the accuracy of the numerical model, pressure results of the experimental and numerical study were compared with each other. Figure 5 shows the comparison of the numerical and experimental results for both SI and SI-CAI combustion mode at 2000 rpm with a stoichiometric mixture of Methane fuel. Valve lifts were taken as 5 mm for both cases.

As seen in Figure 5, the numerical results show a good agreement with experimental data (within ± 5%). Maximum pressure values are obtained as 41 bar and 39.2 bar for numerical and experimental studies, respectively, for SI mode. For the case of SI-CAI mode, the maximum pressure values are obtained as about 30.8 and 30.2 bar for numerical and experimental studies respectively. Therefore, the based on the model was used to predict the effects of different parameters (valve lifts, λ, methane -hydrogen blends) on in-cylinder pressure and temperature development, engine performance and emissions.

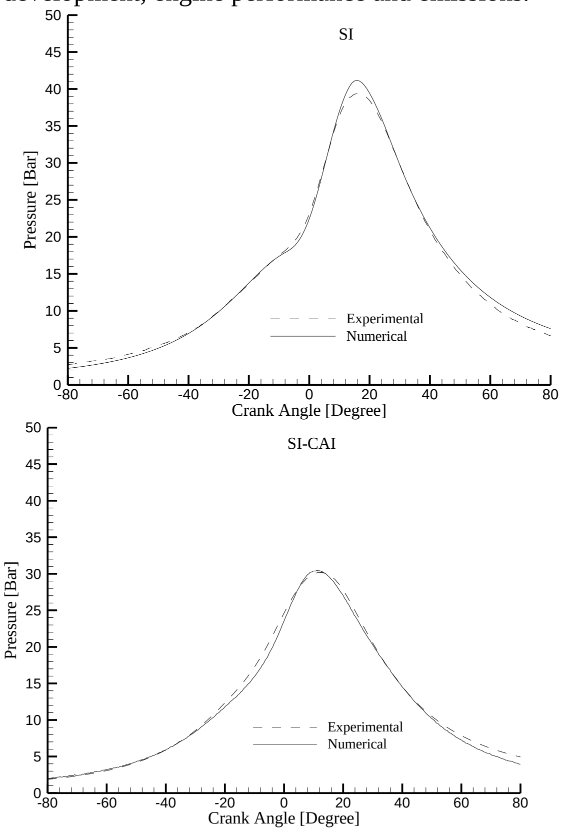

Figure 5. Comparison of experimental and numerical results of SI and SI-CAI engine mode.

Pressure development

Figure 6 illustrates the in-cylinder pressure and temperature values against crank angle for Methane fuel at 2000 rpm and stoichiometric mixture (λ=1.0). Maximum pressure values are observed about 29.0 and 27.5 bar for 5.0 and 3.0 mm valve lifts, respectively. As seen in this figure, the cylinder pressure and temperature value tend to increase, the valve lift value increase.

The temperature values have a similar trend to the pressure value which tends to rise with increasing of valve lift. SI-CAI mode has a low temperature due to the fact that EGR includes the compounds such as CO_2 and H_2O that have a higher value of heat capacity. It can also be seen that the effects of valve lifts on temperature development are in the form of an increase.

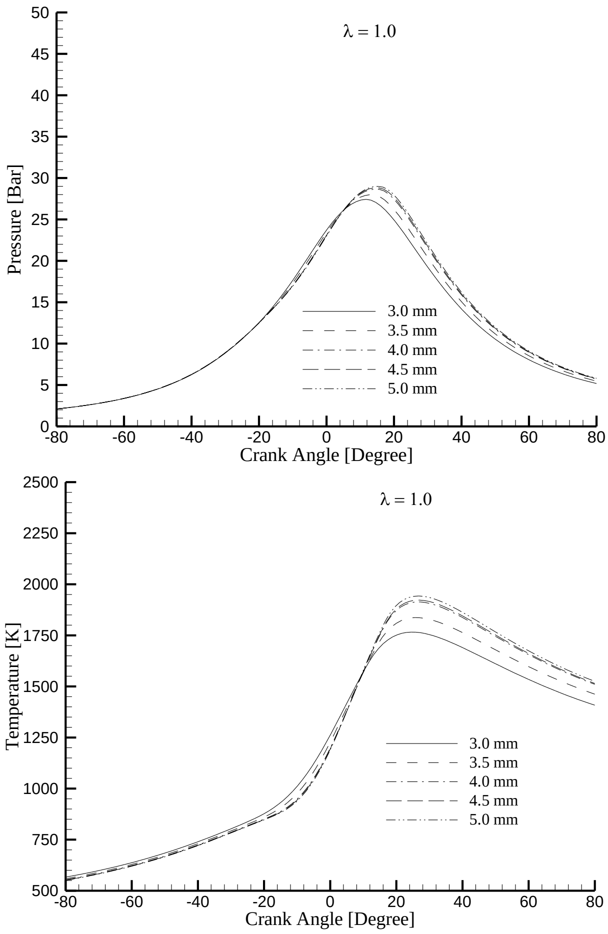

Figure 6. Pressure and temperature values versus crank angle for different valve lifts

Figure 7 shows the pressure values of Methane fuel for considered λ values and valve lifts at 2000 rpm. When the valve lift values increase from 3 to 5 mm, it is seen that the pressure values increase. The peak pressure values are 27.0, 26.0, 24.5 and 23.0 bar for 5 mm valve lift at λ values of 1.1, 1.2, 1.3 and 1.4, respectively. With the increasing of excess air ratio values, the peak pressure values decrease because of less amount of fuel mixture, in turn, in-cylinder pressure developed in a lower value.

Similarly, temperature values inside the cylinder tend to decrease when the λ increases (Fig. 8). For λ value of 1.1, 1.2, 1.3, 1.4 at 5.0 mm valve lift, the temperature values are obtained as 1875 K, 1800 K, 1720 K, 1648 K, respectively. As seen in the figures, the temperature

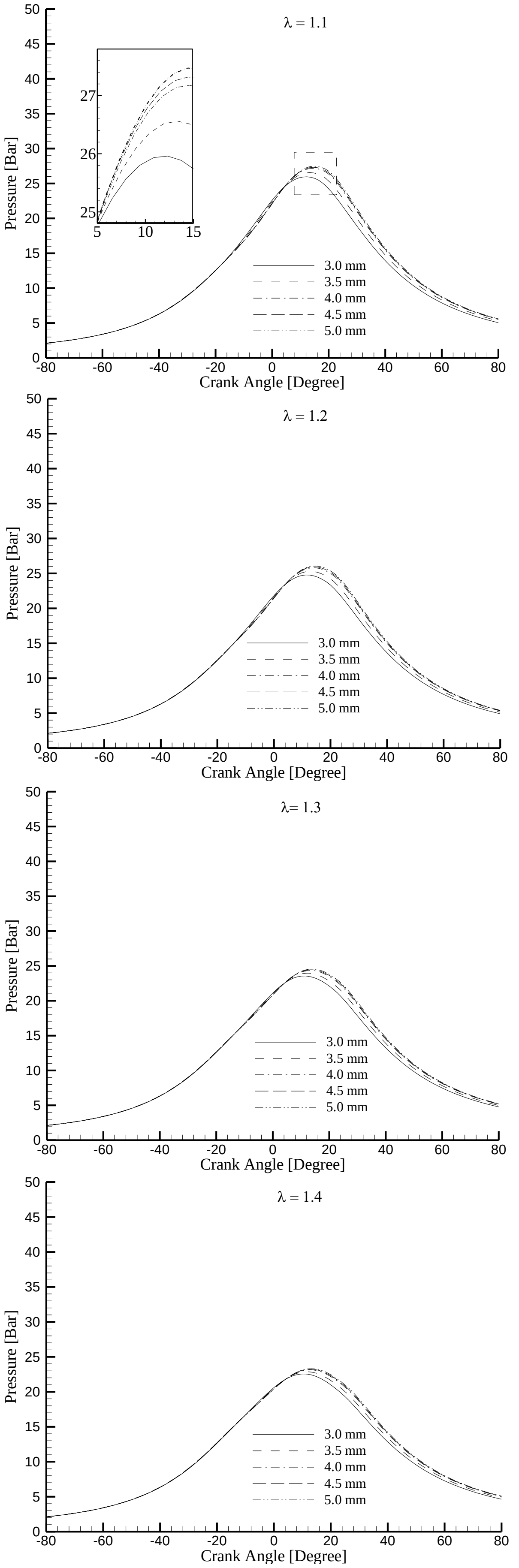

Figure 7. Pressure values of Methane fuel versus crank angle for the different exhaust valve lifts at 2000 rpm

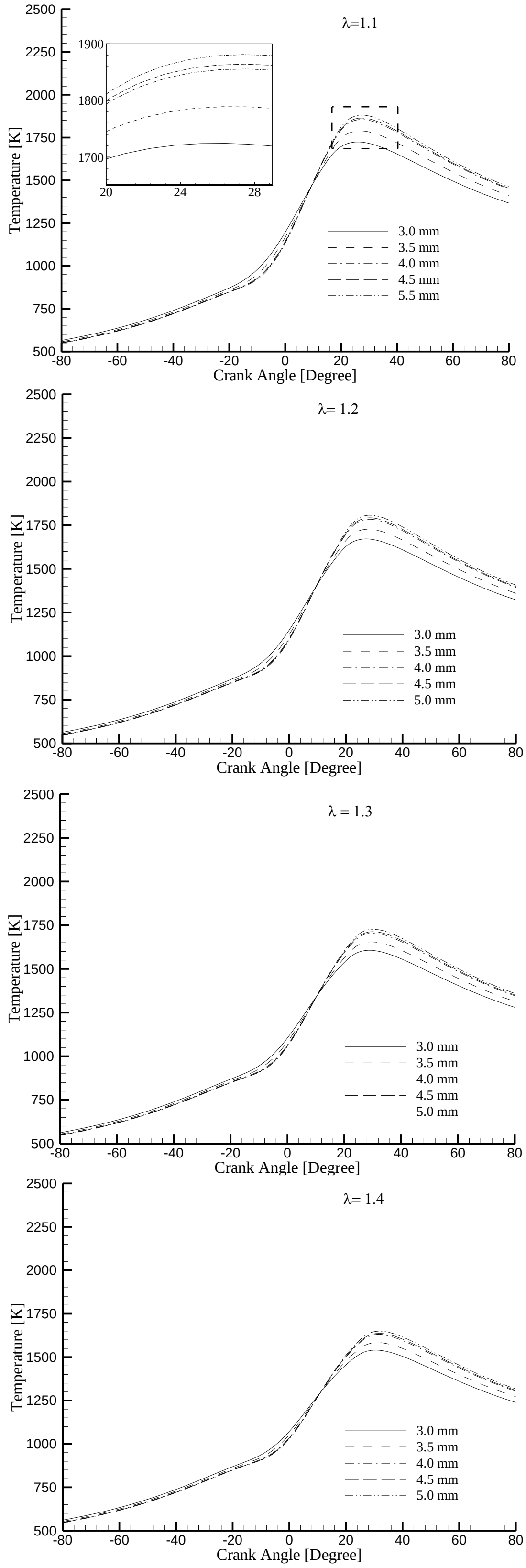

Figure 8. Temperature values versus crank angle for the different exhaust valve lifts at SI-CAI mode

The pressure and temperature values of different Methane-Hydrogen blends at 2000 rpm for λ=1.0 and 5 mm valve lift are presented in Figure 9 and 10. With the addition of Hydrogen to Methane, in-cylinder pressure values increase. The maximum pressure values are obtained as 48 bar at 70M30H mixture. The reason of this is that with Hydrogen addition to Methane, heat release, which leads to pressure development, occurs earlier, as seen in this Figure.

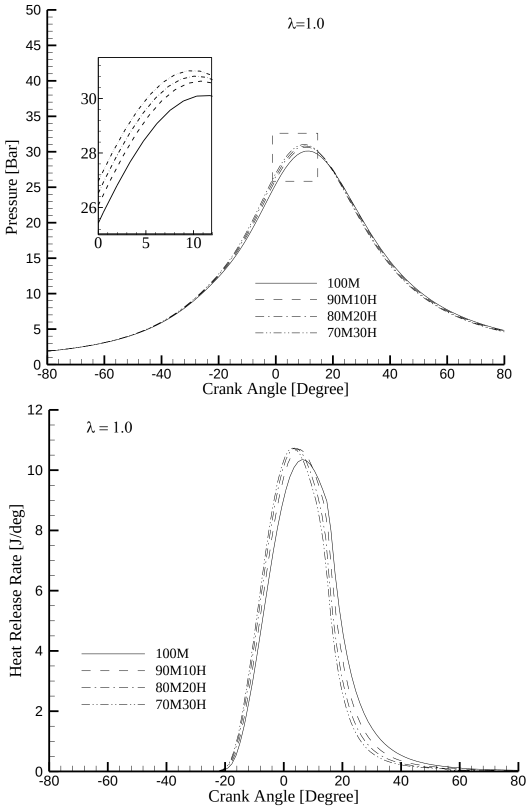

Figure 9. Pressure values and heat release rate versus crank angle for the different fuel blends at

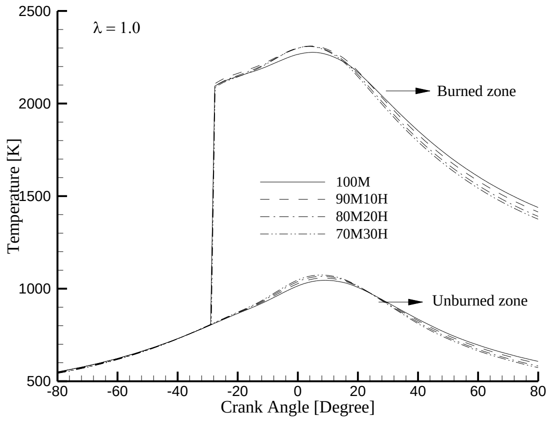

Figure 10. Temperature traces versus crank angle for the different fuel blends

Heat release rate (HRR) is a measure of how fast the chemical energy of the fuel is converted to thermal energy. Therefore, it directly affects the pressure rise rate (dP/dt). The heat release rate was calculated using the first law of thermodynamics as stated by Stone (1999) and Yildiz et al. (2015). In order to calculate heat transfer from gas to the wall, Woschni heat transfer model was used. Thus, the temperature in the cylinder was estimated from the energy balance (Heywood, 1988).

Figure 11 shows also the pressure traces depending on excess air ratio for the different fuel blends. As seen in this figure, the peak pressure values decrease with the increase of λ. It can also be noted that the effect of Hydrogen addition on pressure development decreases with the increase of λ.

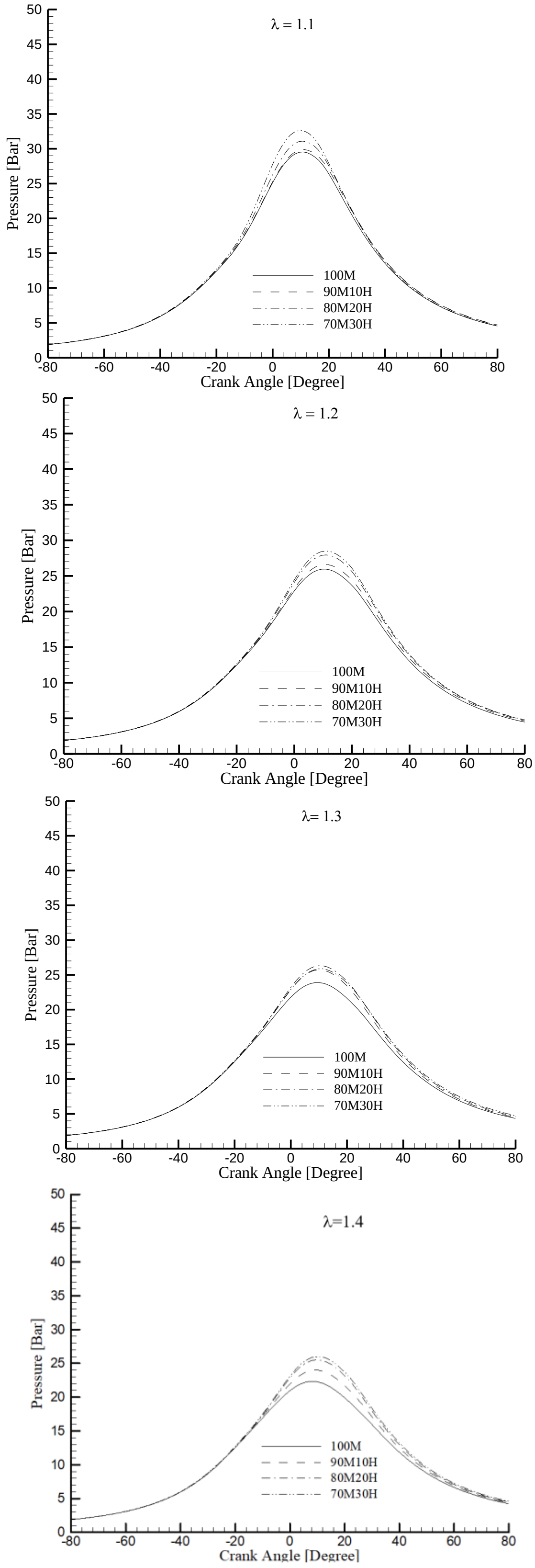

Figure 11. Pressure values versus crank angle for the different Methane-Hydrogen blends at various λ values.

The results of indicated thermal efficiency, brake specific fuel consumption (BSFC) and indicated mean effect pressure (IMEP) for different Methane-Hydrogen blends and λ values at 2000 rpm are given in Figure 12. On considered the results with regard to Hydrogen addition to Methane, the increase of Hydrogen addition lowered indicated thermal efficiency and IMEP considerably at λ value of 1.0.

This can be attributed to early pressure development which leads to an increase in the compression work during the cycle, although peak pressure values increase (Figure 11). However, the results except for BSFC of Methane-Hydrogen blends are in an upward trend up to λ value of 1.1. After λ value of 1.2, the change in indicated thermal efficiency and BSFC are not remarkable for Methane-Hydrogen blends.

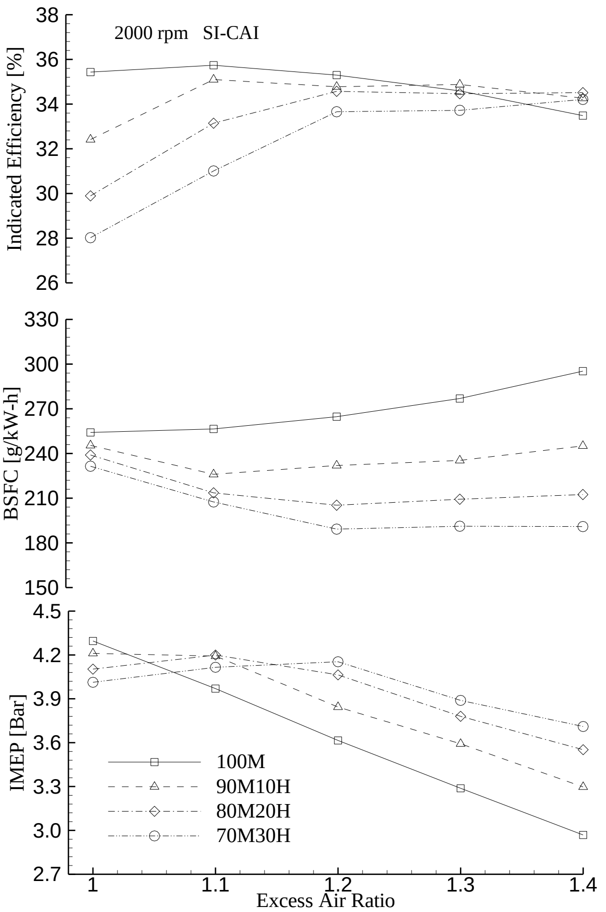

Figure 12. Changes in indicated efficiency, BSFC and IMEP with λ value and fuel blends

The results of indicated thermal efficiency, brake specific fuel consumption (BSFC) and indicated mean effect pressure (IMEP) for different Methane-Hydrogen blends and λ values at 2000 rpm are given in Figure 12. On considered the results with regard to Hydrogen addition to Methane, the increase of Hydrogen addition lowered indicated thermal efficiency and IMEP considerably at λ value of 1.0. This can be attributed to early pressure development which leads to an increase in the compression work during the cycle, although peak pressure values increase (Figure 11). It can also be seen that there is a decrease in BSFC with the increment of excess air ratio from 1.0 to around 1.2 for all blends. But the further increase of excess air ratio leads to the increasing of BSFC due to a reduction in combustion efficiency (Syed et al., 2012). As volume fraction of Hydrogen in the mixture increases, BSFC values decreases. Besides, BSFC values reach to the minimum value when excess air ratio is about 1.1 and 70M30H.

Figure 13 shows brake specific emissions of CO, CO_2, and HC at the exhaust line without an after-treatment system against λ value for Methane-Hydrogen fuel blends at 5mm exhaust valve lift. As seen in the figure, BSCO emissions increase with increasing of Hydrogen addition. On the other hand, when the λ value reaches 1.3 value, there is not any effect of Hydrogen addition on BSCO because its values are already so low. $BSCO_2$ emissions reduce with increasing of Hydrogen addition rate due to reducing the content of carbon in the fuel blends, as expected. BSHC emission reaches its minimum value when λ is slightly greater than nearly 1.0. BSHC emission increases with the increased λ values. However, Hydrogen addition provides lower BSHC emissions. Because of the increase of excess air ratio, lean mixture and the decrease of peak temperature in the cylinder, incomplete combustion occurs easier to happen and burning velocity becomes slow, so the amount of unburned hydrocarbons increase (Zhang et al., 2011). HC emission for combustion process composes mainly due to incomplete combustion and wall quenching effect (Naeve et al., 2011).

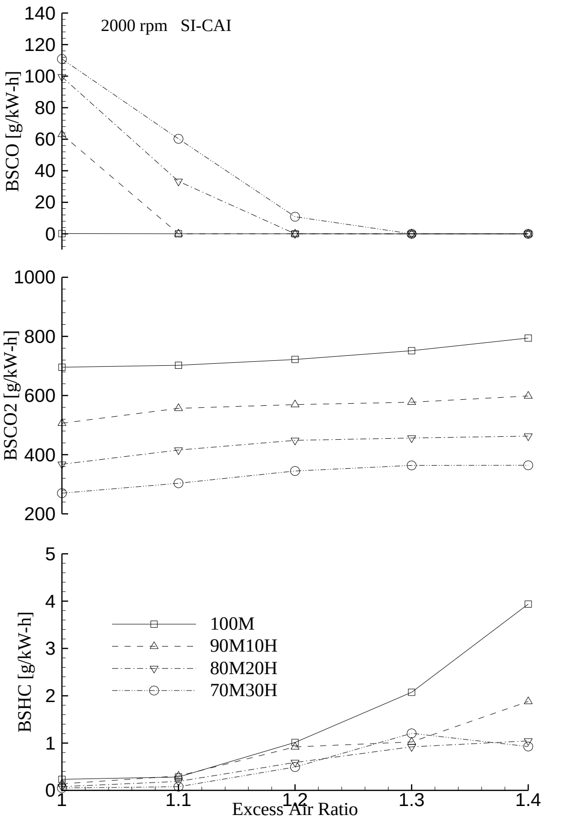

Figure 13. Changes in the specific emissions of.CO, CO_2, HC with λ value and fuel blends

CONCLUSIONS

In this study, numerical and experimental studies were performed for an engine model of Lombardini LGW-523 engine using GT-Power analysis tool. SI-CAI engine mode was investigated at 2000 rpm engine speed for different excess air ratios, valve lifts and, Methane-Hydrogen blends. The main conclusions are ordered at below:

- The numerical engine model was accomplished in good agreement with experimental results.
- The effect of the valve lifts (3 to 5 mm) on pressure development were investigated under SI-CAI combustion mode. When the considered valve lift values increased, the peak pressure and temperature values increased. Furthermore, the higher peak pressure values were obtained at 5 mm valve lift.
- The effect of valve lifts (3 to 5 mm) on pressure development with different λ were investigated under SI-CAI mode. It is observed that greater than a valve lift value of 3.5, the change in the pressure developments are very close to each other for each λ value.
- It was also investigated the Hydrogen addition to Methane for different λ values at 5 mm valve lift. The addition of Hydrogen influenced on the peak pressure in an increasing trend. Moreover; in terms of engine performance, the lowest BSFC and BSHC emission values were obtained at 70M30H blends.

Since the accuracy of the numerical procedure is proved, different parameters on an SI-CAI engine can be investigated in next studies.

ACKNOWLEDGEMENTS

The authors of this paper would like to acknowledge the financial support provided by TUBITAK Turkey under contract number 113M101 and the Unit of the Scientific Research Projects of Erciyes University under contract number FYL-2014-5503.

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