Available online at www.elixirpublishers.com (Elixir International Journal)

Thermal Engineering

Elixir Thermal Engg. 85 (2015) 34533-34537

ISSN: 2229-712X

Computational Parametric Investigation on Single Cylinder SI Engine using LPG and Gasoline in Dual Fuel Mode under Constant Speed & Varying Load

Ufaith Qadri and M.Marouf Wani

Department of Mechanical Engineering, National Institute of Technology, Srinagar, India.

ARTICLE INFO

Article history:
Received: 16 June 2015;
Received in revised form:
10 August 2015;
Accepted: 20 August 2015;

Keywords
AVL Boost,
Dual fuel, Gasoline,
LPG (Liquefied Petroleum gas),
SI (Spark ignition).

ABSTRACT

This Paper presents the parametric study for a four stroke single cylinder SI Engine using a dual fuel Gasoline and LPG as an alternative fuel for investigating the performance and emissions of SI Engine. The performance parameters brake power, torque, Bsfc, Bmep were examined with using AVL BOOST Software. In addition the exhaust emissions like NOX, CO & HC were also measured.SI engine fuelled by LPG has slightly decreased on power output as compared to Gasoline. However, engine fuelled by LPG reduce on specific fuel consumption (SFC). This study investigates the effect of dual fuel LPG with gasoline in a four-stroke spark ignited single cylinder SI Engine. The study was carried out at varying load for constant engine speed. The main objective of the parametric study is to investigate the effects of replacing individual petrol and LPG with their optimum mixture of dual fuel in a spark ignition engine, and to prove reduction in emissions.

Introduction

Liquefied petroleum gas (LPG) has been widely adopted as an automotive fuel because of its potential for reducing emissions, relatively low fuel price, and CO2 advantages. Liquefied petroleum gas (LPG) has been demonstrated to be one of the cleanest alternative fuels for motor vehicles. LPG presents a useful combination of combustion and storage properties that makes it an attractive vehicular fuel. The lean combustion limit is also considerably leaner than that of gasoline, allowing the use of lean-burn calibration kits that increases efficiency and reduces emissions. On an energy basis, LPG has lower carbon content than gasoline or diesel fuel and produces less carbon dioxide during combustion. LPG, when used in spark ignition engines, should produce virtually zero emissions of particulate matter, very little carbon monoxide and moderate hydrocarbon emissions. NOx emissions are a function of the air–fuel ratio [1]. LPG probably makes the most sense as an alternative fuel for urban buses and delivery trucks operating in areas that are specially pollution sensitive. LPG refers to the propane or butane or the mixtures of propane (C_3H_8) and butane (C_4H_{10}) in same container with specific ratio [2]. Propylene and butylenes are usually also present in small concentration. A powerful odorant, Ethyl Mercaptan is added, in order to detect leak on the container (tank) or at any connection, it made leaking can be easily detected. The important characteristics of LPG on vehicle are, LPG has higher octane number of about 112 for pure propane, means it enables higher compression ratios to be employed and gives more thermal efficiencies [3-4]. Due to gaseous nature of LPG, engine operates smoother. Fuel consumption is reducing as compared to Gasoline and this is due to the high energy content in the LPG fuel. Power output is slightly reduced in LPG operation as compared to Gasoline due to the poor volumetric efficiencies effect. There is a several alternative method to improve the performance for the engine fuelled by LPG due to the losses of volumetric efficiencies. One of these methods can be implementing; increase compression ratio (CR) under natural aspirated operation and/or apply turbo charger and/or supercharger [5, 6, 7, 8,]. Liquefied petroleum gas (LPG) and compressed natural gas (CNG) have higher octane number than gasoline. It allows engines running on LPG/CNG to have higher compression ratios, and thus higher energy efficiencies, than gasoline engines. But gaseous Light-Duty Vehicles (LDVs), which are mostly retrofitted gasoline cars, do not exploit this advantage and do not have optimal engine efficiency.

This research generally focused on performance of small scale petrol engine fuelled by Dual fuel. In addition, this paper presents the Simulation results of one particular engine model Industrial plus 6.5 horsepower cm^3 air cooled single cylinder four stroke petrol engine with external carburetion from Briggs & Stratton.

Nomenclature

NOx nitrogen Oxide	TDC top dead centre
CO carbon monoxide	BSFC brake specific fuel consumption (kg/kw-hr)
HC hydro Carbons	BMEP brake mean effective pressure (kg/kw-hr)
SFC specific Fuel Consumption	LPG liquefied petroleum gas
	CNG compressed natural gas
BTE brake thermal efficiency	RON research octane number
$\dot{m}$ mass flow rate (kg/s)	
m mass (kg)	

Simulation setup

The 1-Dimentional engine simulation model is developed by using the software AVL BOOST and has been employed to study the engine performance working on LPG-gasoline dual fuel blends.

Tele:
E-mail addresses: ufaith.qadri@gmail.com

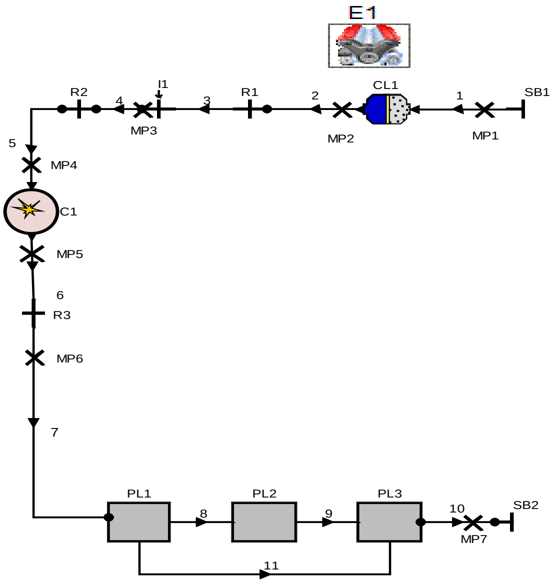

Fig A . Layout of Gasoline engine model [11]

The engine model used in this simulation was performed on a four stroke, Single cylinder spark ignition engine without catalytic convertor and port fuel injection. The gasoline engine model was calibrated by AVL and its layout is shown in Fig. A with engine specification shown in Table 2.

The pre-processing step of AVL Boost enable the user to model a 1-Dimensinal engine test bench setup using the predefined elements provided in the software toolbox. The various elements are joined by the desired connectors to establish the complete engine model using pipelines.

In Fig. A, E1 represents the engine while C1, represent the single cylinder of the engine. MP1 to MP7 represent the measuring points. PL1 PL2 and PL3 represent the plenum. SB1 and SB2 are for the system boundary. The flow pipes are numbered 1 to 10. CL1 represents the cleaner. R1 to R3 represent flow restrictions.

Model formulation

The basic equation for the engine model that is derived from first law of thermodynamics is:

$$\partial E = -\partial Q - \partial W + \sum_i h_i dm_i$$

where E is the internal energy of the cylinder gas mixture, Q is the heat exchange of the cylinder contents with the cylinder walls, W is the work, hi is the specific enthalpy of gas which enters or leaves the cylinder, and dmi is the mass flow into (+) or out of (−) the cylinder, ∂W can be expressed as P.dV, where P is the pressure and V is the cylinder volume [10].

Mathematical SI engine cycle model

The present model is originally based on the model previously developed for gasoline fuelled SI engines. Details of this model were given by Bayraktar [12], Bayraktar and Durgun [13] and Bayraktar and Durgun [14]. Here, the mathematical model is briefly described, and some basic details are given.

Governing equations

The governing equations of the mathematical model have been obtained by applying the first Law of thermodynamics for the cylinder charge, which is assumed to be an ideal gas mixture. The thermodynamic state of the cylinder contents at any instant during the cycle is determined by solving the following time dependent; first order ordinary differential equations for pressure and temperature:

$$\dot{T}_i = \left(\frac{B}{A}\right)_i \left[\left(\frac{\dot{m}}{m}\right)_i \left(1 - \frac{hi}{B}_i\right) - \left(\frac{\dot{V}}{V}\right)_i + \frac{1}{(Bm)_i}\left(-\dot{Q}_{wi} + \dot{m}_{ih_u}\right)\right] \qquad (1)$$

$$\dot{P} = 10^{-5\left(\frac{\rho}{\partial\rho/\partial p}\right)_i}\left[-\left(\frac{\dot{V}}{V}\right)_i - \left(\frac{\frac{\partial\rho}{\partial T}}{\rho}\right)_i \dot{T}i + \left(\frac{\dot{m}}{m}\right)_i\right] \qquad (2)$$

During the compression and expansion periods, the terms in the above equations have been expressed for only the unburned gases (i = u) or only the burned gases (i = b), respectively, whereas during the combustion period, these terms have been determined separately for both the unburned and burned mixtures. Coefficients A and B have been arranged in the following form

$$A_u = c_{pu} + \frac{(\partial\rho/\partial T)_u}{(\partial\rho/\partial P)_u}\frac{1}{\rho_u}; \quad B_u = \frac{1}{(\partial\rho/\partial P)_u} \qquad (3)$$

$$A_b = c_{pb} + \frac{(\partial\rho/\partial T)_b}{(\partial\rho/\partial P)_b}\left[\frac{1}{P_b} - 10^{-5}\left(\frac{\partial h}{\partial P}\right)_b\right];$$

$$B_b = \frac{1}{(\partial\rho/\partial P)_b}\left[1 - 10^{-5\left(\rho\frac{\partial h}{\partial P}\right)} b\right] \qquad (4)$$

In the above equations, the dots denote differentiation with respect to time or crank angle, and ∑ mi. hi is the net rate of influx of enthalpy. From the conservation of mass, it is obvious that throughout combustion, ṁb = - mu.

Engine Performance and Emissions

SI engine performance and emission characteristics are directly affected by the type of fuel. These characteristics include: power, torque, brake mean effective pressure, brake specific fuel consumption, brake specific NOx, HC-Emissions and produced CO.

Results and Discussion

Brake power

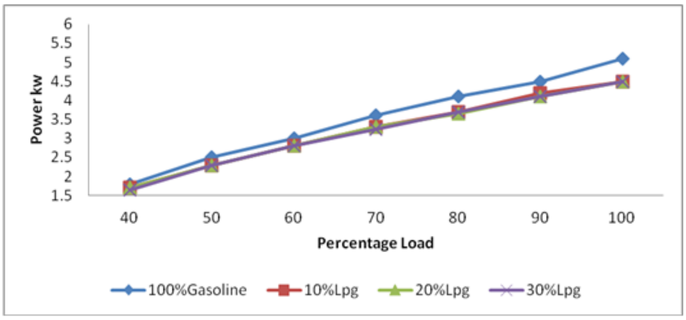

Fig 1. Shows the graph betweens Power & Load

The power produced by methane, methanol, hydrogen, propane and ethanol is less of gasoline by 20%, 13%, 19%, 10% and 10%, respectively. As the engine is designed for operating on gasoline, more power is obtained when gasoline is applied. All the other fuels have a higher octane number than gasoline, so engine compression ratio could be higher if the engine was dedicated to those fuels, and therefore engine performance could be improved. In Fig1.it is clearly defined that the maximum power is obtained for 100% Gasoline. After we start increasing percentage of LPG content in Gasoline the Power shows decrease with increase in load. The dual fuel mode of LPG in Gasoline increases first and then shows slight variation in power increase and remained constant with increase in percentage Load. During the compression stroke, mixture of LPG air-fuel mixture is easily passing through the piston ring gap and this will reduce the quantity of air-fuel mixture to combust. However, this phenomenon is not critical compared when the mixture of Gasoline fuel-air is passing through the piston ring, where it will reduce the viscosity of the lubrication oil and definitely, engine is exposing to the serious friction. In order to control wear and tear of the engine, change the engine oil and engine oil filter regularly and it is definitely costly. Standard ignition timing and low flame speed of LPG also contribute to the power loss, so the ignition time needs to be advance.

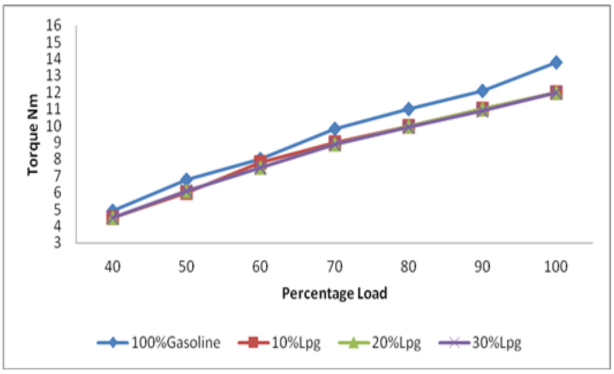

Fig 2. Shows the graph between torque & Load

It is known that the heat of vaporization and oxygen content of alternative fuels are more than those of gasoline fuel. That is why the brake thermal efficiency, BTE and volumetric efficiency increase when the alcohols are used in the engine. Fig. 2 shows the variation in engine torque at different engine Loads for the test fuels. As seen in Fig. 2 it is observed that the maximum torque is obtained at 100% Load using Gasoline as the fuel.

On average, the maximum engine torque for pure Gasoline increased. When compared with the values of LPG. The value of the best engine torque obtained with Gasoline is 14 Nm. It is shown that the decrease in torque of LPG usage is higher than that of the Gasoline. As shown in Fig.4, it was attained that the value of the minimum BSFC with gasoline is 0.3 kg/kW h. It was observed that the minimum BSFC for LPG is 0.28 kg/kW h. when compared with the least values of gasoline, respectively.

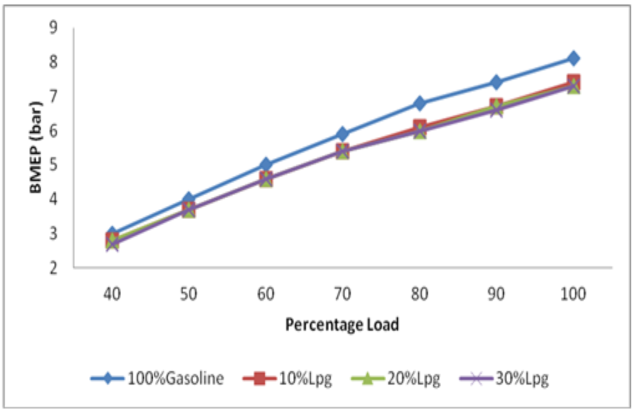

Fig 3. Shows the graph between BMEP & Load

Fig. 3 presents a comparison between brake mean effective pressures (BMEP) of different fuels. For naturally aspirated spark ignition engines, maximum values for BMEP are in the range 7.5-8.5 bar at the Load where maximum torque is obtained. At the percentage load where maximum power occurred, BMEP values are 10–15% lower. The variation of BMEP and brake power is primarily due to the variation in volumetric efficiency [10]. In Fig. 3, it can be seen that the shape and trend of BMEP curve follows the volumetric efficiency curve.

The reduction in BMEP with Propane operation is seen after the Load range is increased beyond 50%. Part of this BMEP loss is due to longer ignition delay and lower flame speed of propane. In more spark advance, combustion starts earlier with respect to TDC and there is a greater amount of negative work done on the piston before TDC compared to gasoline. The remainder of the BMEP loss is due to the lower volumetric efficiency.

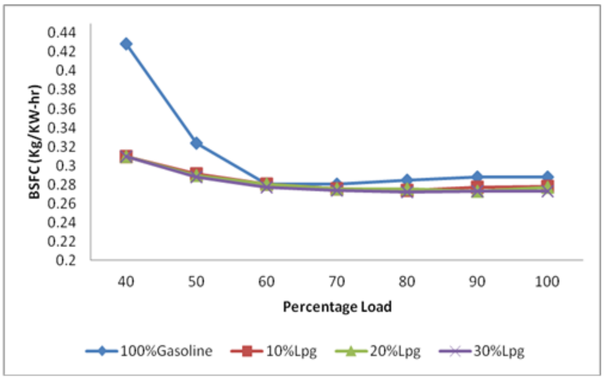

Fig 4. Shows the graph between BSFC & Load

Variation of BSFC according to Load is shown in Fig. 4. In general, BSFC decreased with the increase in Load on engine. BSFC is decreasing for all percentage of LPG in Gasoline after increasing Loads. The higher mass heating value of LPG and its positive effect on combustion efficiency are the most evident factors for this result. BSFC was typically increased for 100% Gasoline at lower loads.

Maximum fuel consumption was obtained with 100% gasoline at low engine loads, as for high engine loads, it remained constant with the 100% Gasoline fuel. With the use of mixture containing 10%, 20% and 30% LPG, BSFC decreased with increase in Load. With the use of 100% Gasoline, BSFC increased with decrease in Load. From the graph of BSFC, generally engine fuelled by LPG is more efficient compared to engine fuelled by Gasoline. This phenomenon occur because of the dissimilar properties of the fuel itself, where LPG has a higher heating value and higher stoichiometric air-fuel ratio compared to Gasoline, so high specified amount of heat can be released with less amount of fuel. [15]

Emissions:

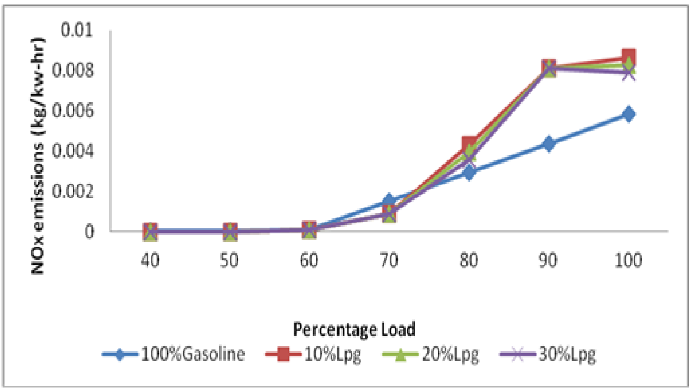

Fig 5. Shows the graph between NOx and Load

Fig. 5 shows the NO_x emission characteristics in relation to the Percentage Load. The NO_x emission slightly increased with increase in Load for 100% gasoline as shown in the graph enlarging the lean combustion region. The thermal NO_x increased under a high pressure and temperature by changing the Compression ratio CR.

Because the NO_x emission is largely affected by the excess air ratio and ignition timing. The NOx is mostly increased for LPG duel fuel with Gasoline with increase in load at all percentage of LPG blends i.e. (10%, 20%, &30% LPG) dual fuel in Gasoline. Then it again increase after the Load is increased further the increase in the NO_x emission from increasing the Load should be solved by using lean combustion or by retarding the ignition timing.

Table 1. Fuel properties [10]

Fuel Type	RON	Formula	Molecular Weight	Density (kg/m3)	Heat of Vaporization 298K (Kj/Kg)	Lower Heating Value (MJ/Kg)	Stoichiometric air/fuel Ratio
Gasoline	95.8	C8H18	106.22	750	305	44	14.6
Methane	120	CH4	16.04	720	305	50	17.23
Methanol	106	CH4O	32.04	792	1103	20	6.47
Ethanol	107	C2H6O	46.07	785	840	26.9	9.00
Propane	112	C3H8	44.10	545	426	46.4	15.67

Table 2. Engine Specifications

Criteria	Description
Make	Briggs & Stratton
Type	Air Cooled 4stroke OHV Gasoline
Displacement	206cc
Compression ratio	9
Bore & Stroke	65.09 × 61.91
Maximum HP	6.5
Cooling System	Air Cooled
Ignition System	Magnetron Electronic

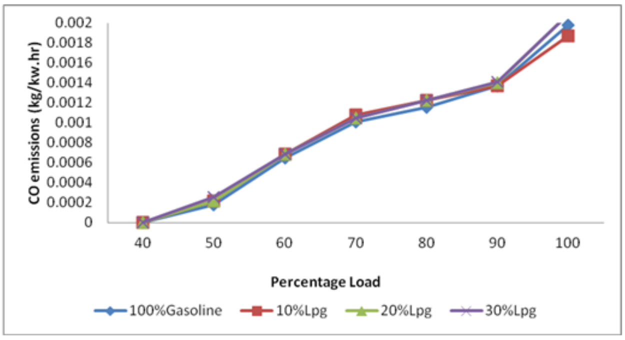

Fig 6. Shows the graph between CO and Load

Fig. 6 shows the CO emission in relation to the Percentage load. The CO emission with Gasoline is higher than that with LPG, The CO emission with a Compression ratio CR of 9 increased after increasing load. This was believed to be caused by insufficient time for the oxidation of CO to CO_2 because of the shortened combustion duration under a higher pressure and temperature. Nevertheless, the CO emission could be reduced by around 95% if an oxidation catalyst was installed downstream of the turbocharger.

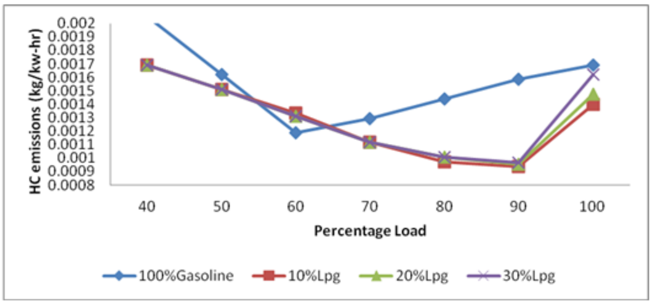

Fig 7. Shows graph between HC and Load

Exhaust hydrocarbon measurements (HC) are presented in Fig.7. As LPG was used as dual fuel with Gasoline, the hydro carbon (HC) increased at lower loads. However, the hydro carbon HC emissions for dual fuel stayed very low, less than 0.001 kg/kw.hr, at higher loads. But same is not the case with 100% Gasoline. Since we can see from the graph the HC emissions are comparatively higher than dual fuel using LPG and gasoline at various percentage of LPG in gasoline. After increasing percentage Load through injector the hydro carbon emission decreased, and then again increases. This is due to the incomplete combustion in a fraction of the engines operating cycles (either partial burning or complete misfire), occurring when combustion quality is poor.

Conclusions

In this study, Power losses arising from the use of LPG has been tried to control by using dual fuel (gasoline and LPG). For this aim, effects of variation in brake power, Torque, BSFC, on the engine performance and emissions with different LPG usage levels on an engine operated with new generation single cylinder SI engine fitted to a petrol generator and sequential gas injection system were investigated. Simulations were carried out under constant engine speed and different Load conditions. The variations in Brake power, Torque, BMEP, BSFC, and exhaust gasses were examined. Results obtained in this study were outlined below.

• The Power was shown a decrease depending on the LPG usage. The brake power decreased considerably with the use of 30% LPG. As for the 10%, 20% LPG usage, power decreased in proportion to LPG usage level. BSFC decreases with the increase in LPG usage level and the minimum BSFC value was obtained at 30% LPG usage.

• Positive results in terms of engine performance were only achieved when using 10% LPG mixture ratio. With the use of mixture containing 10% LPG, BSFC decreased, while the BMEP was maintained.

• During Simulation carried out on single cylinder SI engine fitted to a Petrol Generator using various engine loads on AVL Boost Software, positive results were obtained in terms of exhaust emissions only for hydro carbons HC for LPG usage levels. Best results in terms of exhaust emissions were achieved at using 100% Gasoline for NOx emissions. But for HC and CO emissions were increased at 100% Gasoline usage.

References

[1] Bass E, Bailey B, Jaeger S. LPG conversion and HC emissions Speciation of a light-duty vehicle, SAE Paper 932745, 1993.

[2]World LP Gas Association and United Nation Environment Programme. *Guidelines for Good Safety Practices in the LP Gas Industry*. France, 2008.

[3] K. Owen and T. Coley. *Automotive Fuels Reference Book: Second Edition*. Society of Automotive Engineering, Inc. Warrendale PA, 1995.

[4]. Ali M. Pourkhesalian, Amir H. Shamekhi and Farhad Salimi. Alternative fuel and gasoline in an SI engine: A

comparative study of performance and emissions characteristics. Elsevier, *Fuel Vol. 89*, Issue 5, 2009, pp 1056-1063. K. Owen and T.
[5].Coley. *Automotive Fuels Reference Book: Second Edition*. Society of Automotive Engineering, Inc. Warrendale PA, 1995.
[6].Mustafa K. F., Gitano-Briggs H. W. Liquid Petroleum Gas (LPG) as an alternative fuel in spark ignition engine: Performance and emission
Characteristics. *IEEE Energy and Environment*, 2009. ISBN 978-1-4244-5144-9, 2010, pp 189-194.
[7].Ali M. Pourkhesalian, Amir H. Shamekhi and Farhad Salimi. Alternative fuel and gasoline in an SI engine: A comparative study of performance and emissions characteristics. Elsevier, *Fuel Vol. 89*, Issue 5, 2009, pp 1056-1063.
[8]. O. A. Kutlar, H. Arslan, and A. T. Calik. Method to improve efficiency of four stroke, spark ignition engine at part load. Elsevier, *Energy Conversion & Management vol 46*, 2005, pp 3202-3220.
[9] Bayraktar Hakan, Durgun Orhan. Investigating the effects of LPG on spark ignition engine combustion and performance. Energy Conver Manage 2005;46:2317–33.
[10].Heywood JB. Internal combustion engines fundamentals. New York: McGraw-Hill; 1988.
[11]. AVL List Gmbh, AVL Boost – Theory, 2013
[12]. Bayraktar H. Theoretical investigation of using ethanol–gasoline blends on SI engine combustion and performance, Ph.D. Thesis, Karadeniz Technical University, Trabzon, Turkey, 1997 [in Turkish].
[13]. Bayraktar H, Durgun O. Mathematical modelling of spark-ignition engine cycles. Energy Sour 2003;25(7):651–66.
[14]. Bayraktar H, Durgun O. Development of an empirical correlation for combustion duration in spark ignition engines. Energ Conv Manage 2004;45:1419–31.
[15]. M. A. Ceviz and F. Yuksel. Cyclic variation on LPG and gasoline fuelled lean burn SI engine. Elsevier, *Renewable Energy vol 3*, 2005, pp 1950-1960.