COMPUTATIONAL ASSESSMENT OF FOUR CYLINDERS SI ENGINEWORKING ON PETROL OR DUAL FUEL MODE(LPG-PETROL)

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ABSTRACT

Liquefied petroleum gas LPG is a good alternative to gasoline fuel. It has emerged as a solution to the deteriorating urban air quality problem especially in an oil country like Iraq. Computational model is used for parametric study of spark ignition engine works on Iraqi fuel (gasoline or LPG). Transient one dimensional continuity, momentum and energy equations are solved by two – step Lax wender off (Ritchmyer) approach to evaluate brake specific fuel consumption BSFC, brake power, brake thermal efficiency, volumetric efficiency, air fuel ratio, in cylinder pressure and exhaust gas temperature. Results revealed that LPG fuel improves BSFC by 3.11% as a maximum compare to gasoline for 10 kW brake power and 9.9:1 compression ratio. The maximum cylinder pressures predicted for LPG are lower than that for gasoline fuel. The volumetric efficiency was 76.8 % for engine works with, LPG at compression ratio 9.9:1. While that for gasoline was 85.9%. The equivalence ratio is higher for gasoline than that for LPG, since the first required higher air-fuel ratio for combustion. The reported maximum exhaust gas temperature for LPG is 706°C, while that for gasoline is 741.4°C.

Keywords: SI engine, Gasoline, Liquefied Petroleum Gas (LPG), Performance, Exhaust gas temperature

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تقييم حسابي لمحرك احتراق داخلي ذو أربع أسطوانات يعمل بوقود (بنزين)او ثنائي الوقود
(غاز البترول المسال- البنزين)

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الخلاصة

يعتبر غاز البترول المسال LPG بديلا جيدا للكازولين. اظهر هذا الوقود حلا لمشكلة تردي نوعية هواء المناطق الحضرية في الدول النفطية مثل العراق. استخدم نموذج حسابي نظري لدراسة واسعة لمحرك احتراق بالشرارة يعمل على وقود عراقي (بنزين او غاز البترول المسال). تم حل معادلات الحفظ أحادية البعد والغير مستقرة للاستمرارية والزخم والطاقة بمنهجية لاكس وندروف ثنائية الخطوة لتحديد صرف الوقود المكبحي النوعي BSFC ، القدرة المكبحية، الكفاءة المكبحية الحرارية، الكفاءة الحجمية، نسبة هواء/وقود ، ضغط الأسطوانة و درجة حرارة العادم. أظهرت النتائج ان القيمة العظمى لتحسين صرف الوقود المكبحي النوعي BSFC لغاز البترول المسال كان بمقدار 3.11% مقارنة بوقود البنزين لقدرة مكبحية مقدارها kW10 ونسبة انضغاط 9.9:1. وكان اعظم ضغط الاسطوانة الذي تم الوصول اليه لغاز البترول المسال اقل مما هو عليه لوقود البنزين. ان الكفاءة الحجمية للمحرك كانت 76.8% لغاز

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البترول المسال عند نسبة انضغاط 9.9:1 بينما كانت 85.9% لوقود البنزين. ان نسبة المعادلة كانت لوقود الكازولين اكبر مما هي عليه لغاز البنرول المسال نظرا لكون الأول يحتاج للاحتراق بنسبة هواء/وقود عالية. ان اعظم درجة حرارة تم تحديدها لغاز العادم للوقود الغاز المسال كانت 706 °م بينما تلك الخاصة بوقود البنزين فكانت 741.4 °م.

NOMENCLTURE

Latin Symbols	Description
BSFC	brake specific fuel consumption (kg/kWh)
BTDC	before top dead center
b	cylinder bore (mm)
bmep	brake mean effective pressure (bar)
CNG	compressed natural gas
D	diameter of the duct (mm)
dV	Volume (mm^3)
DI	direct injection
DOHC	double overhead camshaft
e	internal energy (kJ)
F	force (N)
f	pipe wall friction coefficient (mm.sec)
FA	fuel /air ratio
FA$_s$	stoichiometric fuel /air ratio
fmep	friction mean effective pressure
h	Enthalpy (kJ)
H/C	hydrogen to carbon ratio
imep	indicated mean effective pressure (bar)
LPG	liquefied petroleum gas
$\dot{m}_{ia}$	inlet air mass flow rate (kg/h)
$\dot{m}_f$	fuel mass flow rate (kg/h)
N	engine rotational speed (rpm)
n_r	revolution per cycle and is taken as 2 for four stroke engines
P	pressure (bar)
P_e	brake power (kW)
q	rate of heat transfer per unit mass (W/kg)
Q_{LHV}	lower heating calorific value of fuel (kJ/kg).
R	air constant (kJ/K.mol)
s	piston stroke (mm)
SOHC	single overhead camshaft
T	temperature (k or °C)
t	time (sec)
V_d	displacement volume (mm^3)
W_c	net indicated work per cycle (kJ).
y	distance (mm)
Z	the cylinder number

Greek letters

η_{th}	brake thermal efficiency
η_v	volumetric efficiency
ϕ	equivalence ratio
λ	ratio of actual to stoichiometric air/fuel ratio
ρ	density (kg/m^3)

Subscripts	Description
BDC	bottom dead center
ia	inlet air
o	**Stagnation**
TDC	top dead center

INTRODUCTION

Most internal combustion engines are used in automobile applications such as cars, aircrafts, boats and power generation etc. Environmental concerns and depletion in petroleum resources have forced researchers to concentrate on finding alternatives to conventional petroleum fuels. Liquefied Petroleum Gas LPG has efficient combustion characteristics and low gases emission, so it is considered as an alternative fuel for gasoline in spark-ignition (SI) engines. Higher octane number for LPG enables higher engine compression ratio, and results higher thermal efficiency **Snelgrove** (**1996**).LPG has high octane rating 110^+ that allows compression ratios to be high up to 15:1, (the range of 8:1 to 9.5:1 for gasoline engines). Also, since LPG enters an engine's combustion chambers as a vapor, it does not strip oil from cylinder walls or dilute the oil when the engine is cold. Cycle simulation studies had great interest **Heywood (1988), Bayraktar (1997)**. Recently, much researches are conducted on alternative fuels due to increasing demand to lower fuel consumption and exhaust emissions. These improve the awareness of the effective use of present reserves and slowly switches over to the alternative friendly environment fuels, **Nirendra(2008), Lata et al. (2011) and Krishna et al. (2011)**. One of the solutions to accomplish this target is the use of gaseous fuels in SI engine. The use of alternative gaseous fuels e.g. liquefied natural gas LNG, liquefied petroleum gas (LPG), etc can be adopted in SI engine for automobiles, power plants, ships etc. SI engines are a premixed air-fuel mixture followed by compression before a spark ignites the mixture and burned. Octane rating of a fuel indicates how slowly the fuel will burn and how fuel will resist pre-ignition before the spark plug fires. This helps to have a longer service life and reduced maintenance costs of engine. **Ozcan and Yamin (2008)**, simulated the effect of variable the stroke length (compression ratio) of LPG fueled engine. The results obtained were compared with the experimental results of both constant and variable stroke lengths. It was concluded that brake power registered an increase of about 7-54% at low speed and 7-57% at high speed for all investigated stroke length. BSFC reported a reduction of about 6% to an increase of about 3% at low speed and from a reduction of about 6% to an increase of about 8% at high speeds. The concentration of pollutants is increased from 0.65% to 2% at low speed while larger stroke lengths resulted a reduction of pollutant level of about 1.5% at higher speed while at lower stroke length an increase of about 2% is indicated. **Mustafa and Briggs (2009),** tested LPG-gasoline engine with different LPG volume content of (5%, 10% and 20%) at constant engine speed (4000 rpm) but at varying values of air-fuel ratio. Power output is decrease by 20% and 41%, for 5% and 20% LPG-gasoline blended as compared to 100% gasoline. Brake specific fuel consumption (BSFC) for LPG-gasoline blends is slightly lower than that of 100% gasoline, the difference in the minimum value of BSFC was found to be 47% and 58% for 5% and 20% LPG-gasoline blended respectively. It was reported that LPG addition does not lead to a significant improvement in brake thermal efficiency. The concentration levels of CO, CO_2, unburnt HC and NO_x for LPG fueled engine were found lower than the gasoline fueled engine. **Pourkhesalian et al. (2009)** investigated the performance of a four stroke, four cylinders SI engine fueled with different alternative fuels. when the engine is fed with gaseous fuels results a decrease in volumetric efficiency

and power output, since gaseous fuels have low density as compared to gasoline. Brake specific fuel consumption of engine, operating on, propane is lower than that of gasoline. **Arslan et al. (2010)** tested LPG fueled four cylinders SI engine to estimate its performance and exhaust emission under cold climatic conditions in Turkey. SFC increases with the increasing of engine speed for both fuel types with higher values for LPG by 22–32%. Despite this fact, the cost of LPG consumption per km is less. Concentration of CO was found to be reduced with increasing of engine speed, but for LPG this reduction was higher and it was accounted to be 91.9%. A decrease in HC emission can be achieved as a percentage ranging between 10.1 and 29.7, when there is a mixer with no load on the engine. An increase in HC emissions at 1,000 rpm of 1.5% and 2.9% is reported for the cases of off-load and on-load, respectively. Concentration of CO_2 was decreased at off-load in a percentage range between 13.4 and 16.7. This value ranges 13.4-22.7, for full load case. **Shankar and Mohanan (2011)**, studied the effect of variation in ignition timing on the performance of four cylinders multipoint port fuel injection gasoline engine which is retrofitted to run with LPG injection. Since the ignition temperature is high for LPG therefore combustion duration is longer and this decreases average burning rate. So, engine consumes more fuel and thus decreases in brake thermal efficiency. But at higher engine speeds, the flame propagation speed of LPG is increased which lowers the time duration for each cycle and thus demands more rate of combustion and hence efficiency increases. CO and HC emission are reduced drastically for LPG at all the throttle positions while NO_x emission were found to be increased for LPG and was almost doubled at 4500 rpm. At 6° BTDC, CO and HC emissions were found to be lower than that at 3° and 4° BTDC due to complete combustion resulted from high propagation speed of LPG. Concentration of NO_x was 1400 PPM at 6° BTDC and 500 PPM at 3° BTDC due to low combustion temperature because of incomplete combustion at this angle. **Gumus (2011)** investigated the effect of variation in volumetric efficiency and SI engine performance (for Renault Clio 1.4i RTA vehicle and 4 cylinder) at different LPG usage levels with gasoline (25%, 50%, 75% and 100%) on an engine equipped with electronically controlled multipoint sequential gas injection system. The experiments were carried out at constant engine speed of (3800 rpm) and at different loads conditions (5%, 30%, 60% and 90%). The volumetric efficiency is decreased considerably with the use of 25% LPG. Best results of exhaust emissions were obtained for 100% LPG.**Pundkar et al. (2012)**, presented the effect of engine torque, power and specific fuel economy on engine efficiency. As the gaseous fuel requires 4 to 15 percent of more intake passage volume than liquid fuels which reduces the volumetric efficiency and hence maximum power output will also be reduced (by 4%). **Erkus et.al. (2013),** investigated the effect of electronically controlled gas phase manifold LPG injection system on the SI engine performance fueled with LPG at 2000-4000 rpm and fixed throttle position of 25% as well as 45% of full opening. During tests, the excess air coefficient was maintained in the range of 0.95 to 1.05. At 25% throttle position with the LPG injection system, the maximum improvement in the brake power achieved were 99.52% and 84.84% compared to gasoline and LPG carburetion, at 3500 rpm. While at 50% throttle position, brake power of the engine was always higher with LPG injection system than with the carburetion systems at 3000 rpm. Volumetric efficiency is improved with LPG injection system as compared to carburetion systems at both 25% and 50% throttle positions. Brake-specific fuel consumption (BSFC) was lower for LPG injection system at both the throttle positions except at 2500 rpm for 25% throttle position. Although, experimental investigation provides actual results for internal combustion engine operation, but it is more expensive and requires more time than computational methods. The aim of this work is to establish specifications of four cylinders four stroke engine fueled with LPG to reduce emission gases compared with gasoline fuel. Contribution of this work is that simulation

program is used for a four-cylinder SI engine to perform variation of engine parameters. Study engine efficiency with the aid of superior physical or chemical properties software (lower calorific vale, density and H/C ratio … etc) of alternative fuels compared to those of conventional fuels under different compression ratios (8:1, 9.9:1, 10.25 and 15). This work presents a first try to investigate the performance of 4-cylinder 4-stroke SI engine works on Iraqi LPG fuel as an alternative engine fuel.

MATHMETICAL AND NUMERICAL MODEL

Governing equations of gas flow in pipes, cylinders, heat transfer and combustion modeling, ports, valves, charge cooling and engine dynamics are individually described. To build the computational model, of the multi cylinder engine, and calculate the cycle performance data, the brake power (P_e), the brake thermal efficiency (η_{bth}) and the brake specific consumption of fuel (BSFC). The following steps were followed:-

1. Defining steady-state test condition data.
2. Engine and fuel system building.

Steady State Test Condition Data

The full load performance steady-state operating conditions simulation of an engine model consists of the intake manifold system, valve train, displacement volume and the exhaust manifold system. The development of an in-line four cylinder and four stroke indirect injection, liquid cooling SI engine model in the simulation software required the following engine and manifold specifications. The input data required for engine simulation can be presented as (see table (3.1.a, b, c and d):

a. The initial conditions (Patm=1 bar, Tatm=25 $^{\circ}$C).
b. The engine bore.
c. Stroke dimension.
d. Number of cylinders.
e. Compression ratio.
f. Valve timing data.
g. Intake and exhaust manifold dimensions.
h. Engine model the requirements can be defined as the engine speed (rpm) which was from 1000 rpm to 3500 rpm, with an increment of 500 rpm
i. The Air / fuel ratio.

THE ENGINE MODEL
Building Of The Engine

Building simulation model (cylinder, inlet and exhaust manifold ports and valves) is done only for one cylinder and the other cylinders shared the same specifications, figure (1) shows the complete one-cylinder model. After that it was copied with all the configuration four times to produce a four cylinders engine with a consideration of setting the fire order to (1-3-4-2). Figures (2 and 3) show the real model of the petrol engine. The next part is adding the inlet and exhaust manifold ports and valves. Which inched valve timing. The specifications of the engine are given in table (1).

Steps of Les Engine Model

After the schematically steps of engine model built in LES as shown in figure (4). Every step, the program checks the validity of the built engine model or inserted data and specifications. Finally, when the operating conditions are determined to be true, then the simulation is run at the specified engine speed to achieve the demanded engine loading with the inserted fuel injection rate.

GOVERINING EQUATIONS

One-dimensional transient model of pipe gas dynamics are applied for the gas flow in pipes. At each time step (crank angle), conservation differential equations for mass, momentum and energy are solved for calculating the conditions within pipe elements. The

flow of a compressible fluid through an infinitesimal section of pipe is shown in Figure (5). When the area variation is small, the fluid properties can be taken uniform across any cross-section and can be considered as a function of x and time only. Therefore, the flow can be assumed as quasi-one-dimensional (**Winterbone and Pearson 1999 and Winterbone and Pearson2000**). The related properties are pressure, density, velocity of flow and cross-sectional area.

The governing equations (continuity, momentum and energy equations in order) for the transient one-dimensional flow of a compressible fluid in a pipe with area variation are (**Winterbone and Pearson 2000**):

$$\frac{\partial(\rho F)}{\partial t} + \frac{\partial(\rho u F)}{\partial x} = 0 \tag{1}$$

$$\frac{\partial(\rho u F)}{\partial t} + \frac{\partial(\rho u^2 + p)F}{\partial x} - p\frac{dF}{dx} + \frac{1}{2}\rho u^2 f\pi D = 0 \tag{2}$$

$$\frac{\partial(\rho e_0 F)}{\partial t} + \frac{\partial(\rho u h_0 F)}{\partial x} - q\,\rho F = 0 \tag{3}$$

The equations can also be presented in vector form as:

$$\frac{\partial W}{\partial t} + \frac{\partial F(W)}{\partial x} + C = 0 \tag{4}$$

where
$$W = \begin{pmatrix} \rho F \\ \rho u F \\ \rho e_0 F \end{pmatrix} \tag{5}$$

$$F(W) = \begin{pmatrix} \rho u F \\ (\rho u^2 + p)F \\ \rho u h_0 F \end{pmatrix} \tag{6}$$

$$C = \begin{pmatrix} 0 \\ -p\dfrac{dF}{dx} \\ 0 \end{pmatrix} + \begin{pmatrix} 0 \\ \rho Gf \\ -q\,\rho F \end{pmatrix} \tag{7}$$

The two-step Lax-Wendroff (Richtmyer) is used to solve the governing equations above. This numerical method is a space- centered scheme based on the computational stencil shown in Figure (6) (**Winterbone and Pearson 2000**). This scheme can be modified to include the source terms and can be expressed by explicit forward finite difference scheme at node (i+1) such that

$$W_{i+1/2}^{n+1/2} = \frac{1}{2}(W_{i+1}^n + W_i^n) - \frac{\Delta t}{2\Delta x}(F_{i+1}^n - F_i^n) - \frac{\Delta t}{4}(C_{i+1}^n + C_i^n) \tag{8}$$

Where the super script $n+1$ refers to the new value at time t $= t + \frac{1}{2}\Delta t$. While n refers to the current value.

Explicit forward finite difference scheme is adopted at (i-1) such that:

$$W_{i-1/2}^{n+1/2} = \frac{1}{2}(W_i^n + W_{i-1}^n) - \frac{\Delta t}{2\Delta x}(F_i^n - F_{i-1}^n) - \frac{\Delta t}{4}(C_i^n + C_{i-1}^n) \tag{9}$$

Implicit forward finite difference scheme is adopted at (i) such that:

$$W_i^{n+1} = W_i^n - \frac{\Delta t}{\Delta x}\left(F_{i+1/2}^{n+1/2} - F_{i-1/2}^{n+1/2}\right) - \frac{\Delta t}{2}\left(C_{i+1/2}^{n+1/2} - C_{i-1/2}^{n+1/2}\right) \tag{10}$$

Calculations of the engine performance parameters

The calculated performance parameters includes **(Heywood 1988) and (Pearson et al 2002)**:
The brake mean effective pressure (bmep) of the SI engine is evaluated as:

bmep= imep**cycle** – fmep   (11)

where:

fmep is friction mean effective pressure (bar).

imep**cycle**is indicated mean effective pressure (bar). It is calculated as:

$$\text{imep}_{\text{cycle}} = \frac{W_c}{V_d} \tag{12}$$

W_c is defined as:

$$W_c = \int p \, dV \tag{13}$$

The displacement volume (V_d) is equal to the volume of part of the cylinder envelope between top dead center (TDC) and bottom dead center (BDC) shown in figure (7). This volume can be calculated by:

$$V_d = \frac{\pi}{4} b^2 s \tag{14}$$

The brake power, P_e (kW), and torque τ_e N.m, are evaluated as:

$$P_e = \left(\frac{bmep \, V_d \, N \, Z}{n_r \, 60}\right) \tag{15}$$

$$\tau_e = \left(\frac{10^3 \, P_e}{\omega}\right) \tag{16}$$

Angular speed of the engine, ω (rad/s), in equation (17) is defined as:

$$\omega = 2 \, \pi N/60 \tag{17}$$

The brake specific fuel consumption, BSFC (g/kWh), is calculated with the following formula below:

$$\text{BSFC} = \frac{\dot{m}_f}{P_e} \tag{18}$$

The brake thermal efficiency of the system,η_{th}, can be calculated as:

$$\eta_{th} = \frac{3600 \, P_e}{\dot{m}_f * Q_{LHV}} \tag{19}$$

Finally, volumetric efficiency is defined by:

$$\eta_v = \frac{2 m_{ia}^. \, 10^3}{60 \, \rho_{ia} V_d \, N} \tag{20}$$

Inlet air density is calculated as:

$$\rho_{ia} = \frac{P_{ia} \, 10^3}{R \, T_{\rho_{ia}}} \tag{21}$$

where:

R is air constant (287 kJ/K.mol).

RESULTS AND DISSECTION

Engine Performance

Brake Specific Fuel Consumption

Figure (8) shows the variation of BSFC with engine speed. As the engine speed increases, BSFC is decreased gradually to its minimum value at engine speed of 3000 rpm, then increases as the speed was increased to 3500 rpm. The maximum percentage deviation of BSFC for gasoline fuel compared to LPG fuel is 3,3.11%, 3.51% and5.53% for compression ratio equals 8:1,9.9:1,10.25:1 and 15:1respectively for gasoline compared with LPG fuels since the later needs higher combustion chamber size i.e. higher temperature and pressure. Since the BSFC is a measure of the "fuel economy", lower BSFC should be the ultimate target in all engine designs. Result show that adding LPG system to the SI engine was

beneficial towards improving BSFC. It is also evident that BSFC deteriorates as the engine speed approaches 3500 rpm, and this was attributed to the increase power required to overcome the frictional power. Since frictional power loss is proportional with engine speed, extra fuel is required to ensure sustainable engine operation which explains the increase in BSFC during the window of operation from 3000 to 3500 rpm.

Brake power

Peaks of brake power was indicated at engine speed of 3500 rpm for as illustrated in Figure (9). Brake power increases rapidly with increasing of engine speed for each compression ratios 8:1, 9.9:1, 10.25:1 and 15:1. Maximum percentage increase of brake power for gasoline fuel compared to LPG fuel is 17.6%, 11.51%, 9.35%, 6.4% and5.35%at engine speeds 1500, 2000, 2500, 3000 and 3500 rpm respectively for compression ratio equal 8:1.For compression ratio 9.9:1 it is 16.7%, 11.1%, 9.1%, 6.25% and 5.26% at engine speeds 1500, 2000, 2500, 3000 and 3500 rpm respectively for compression ratio equal 9.9:1. For compression ratio 10.25:1 it is 15.4%, 8.5%, 7.8%, 5.9% and 5.3%. While for compression ratio 15:1 it is 13.55%, 7.9%, 7.1%, 5.6% and 5.13%. Further, for a given engine speed, the engine brake power increases as compression ratio increases, since it is function of torque and speed. The reduction in the brake power varies between 5% to 16% for LPG compared to gasoline fuel. Due to the displacement of the fresh charge at the chamber during the intake stroke, which resulted in a decrease of engine volumetric efficiency.

Brake thermal efficiency

The peak value of brake thermal efficiency is indicated at 3000 rpm as presented in figure (10). The range of brake thermal efficiency varies from 24.1% to 27.3% for gasoline and LPG respectively at compression ratio equals to 8:1. It varies from 25.1% to 30.3% for gasoline and LPG at compression ratio equals to 9.9:1. It varies from 25.9% to 31.4% for gasoline and LPG at compression ratio equals to 10.25:1. While, it varies from 26.4% to 32.2% for gasoline and LPG at compression ratio equals to 15:1. The maximum percentage deviation of peak brake thermal efficiency for gasoline fuel compared with LPG fuel at 3000 rpm for compression ratio 8:1, 9.9:1, 10.25:1 and 15:1 are 3.84, 4.2%, 4.7% and 5.14% respectively. LPG addition has inconsiderable improvement in terms of brake thermal efficiency due to the properties of gasoline and LPG. The heat generated in engine will be increased for LPG compared to gasoline due to the higher heating value of LPG (45.25 MJ/kg) compared to gasoline (43.96 MJ/kg). Thus, at a given engine operational speed, the amount of heat released will be greater for LPG fuel leads to higher engine brake thermal efficiency. Higher H/C ratio for LPG compared to gasoline also ensures adequate oxygen molecule for complete carbon combustion for useful thermal conversion which improves the thermal efficiency of the engine. Higher rate of fuel utilization and higher air to fuel ratio greatly increases the combustion rate of the engine since more air and fuel is introduced to the system. At engine speeds higher than 3000 rpm, the thermal efficiency starts to decrease slightly due to the effects of engine knocks that causes loses of output power and damages the engine.

Volumetric efficiency

The volumetric efficiency increases with an increase of compression ratio and engine speed as illustrated in figure (11). The reported volumetric efficiency for LPG was 76% and 79.9% for gasoline at 3500 rpm for compression ratio of 8:1. It was 80%for LPG and 82.9% for gasoline at compression ratio of 9.9:1. it was 81.3% for LPG and 84.9% for gasoline at compression ratio of 10.25:1. While, it was 83.3% for LPG and 87.2% for gasoline at compression ratio of 15:1. The maximum percentage difference of volumetric efficiency at 3500 rpm between gasoline and LPG was 4.88, 4.65%, 4.2% and 5.3% for compression ratio 8:1,9.9:1,10.25 and 15:1 respectively. Increasing compression ratio improved engine volumetric efficiency for LPG fuel and bring it close to gasoline. Volumetric efficiency of

gasoline is higher than LPG because gasoline is injected as a liquid phase and as it vaporizes, it cools air, thus produces improved volumetric efficiency. The vaporized LPG fuel leads to discharge intake air because of its gaseous nature that occupied the space and reduced the amount of air entering in the engine cylinder. Most of the LPG conversion injects the fuel at intake system or intake manifold and the vaporization of fuel depends on few factors such as pipes thermal insulation, ambient air and liquid fuel temperatures, shape and dimensions of the intake system, heat of vaporization of the fuel, load and rotational speed of the engine. Pumping loss associated with conversion system hardware also is the reason to loss part of volumetric efficiency.

In-cylinder pressure

Same trend for cylinder pressure behavior via crank angle is indicated for both LPG and gasoline, except the differences in the value and timing of peak pressure which depends on the difference in brake power, engine speed, and compression ratio as shown in figure (12). It can be seen that, as engine speed increases, maximum pressure increases reaching its maximum value at 3000 rpm. The increase of speed from 1000 to 3000 rpm, causes an increase in the maximum pressure from (56.2 to 60) bar for gasoline and from (53.8 to 58) bar for LPG at compression ratio 8:1. For compression ratio 9.9:1, the maximum pressure increases from (62.5 to 65.6) bar for gasoline and from (60 to 64.12) bar for LPG. For compression ratio 10.25:1, the maximum pressure increases from (66 to 70.8) bar for gasoline and from (63.25 to 68.8) bar for LPG. While that for compression ratio 15:1, the maximum pressure increases from (69.7 to 74.7) bar for gasoline and from (73.5 to 68.8) bar for LPG at brake power equals to 10 kW. This is due to high flame speed for gasoline fuel and higher content of carbon atoms than LPG, therefore the produced energy is higher. That causes clear reduction in the combustion duration.

Exhaust gas temperature

Gasoline exhaust gas temperature increases with increased engine speed and compression ratio. Similar trends but with lower values is observed when LPG fuel is used for the same brake power and engine speeds as shown in figure (13). The maximum exhaust gas temperature reported is 680.3 $^{\circ}$C for LPG fuel while that for gasoline is 715.6°C at compression ratio equals to 8:1. For compression ratio of 9.9:1 was706 $^{\circ}$C for LPG fuel and 741.4°C for gasoline. For compression ratio of 10.25:1 was 742 $^{\circ}$C for LPG fuel and 773 $^{\circ}$C for gasoline, While that for compression ratio of 15:1 was 762.5°C for LPG fuel and 794.1°C for gasoline. The increase in exhaust gas temperature is due to the increase in peak temperature and peak pressure, causes faster running of fuel combustion. Gasoline produced higher exhaust gas temperatures taking advantages of its higher volumetric efficiency and its higher heating value. The depreciation of LPG volumetric efficiency causes a reduction in exhaust temperature. Many parameters operated against LPG, lower pressure and lower temperature inside combustion chamber results a temperature rise with increasing of the speed to inject largest quantity of fuel. Gases temperature is gradually decreased as they pass through the converter catalytic and muffler, due to heat lost to atmosphere by convection, conduction and radiation.

Comparison between computational and experimental results

Figure (14) shows a comparison of experimental [Gatea 2019] and computational results of brake specific fuel consumption (gasoline or LPG) against engine speed (rpm) for two compression ratios (9.9:1 and 10.25:1) and for brake power of 10 kW. The results show that as the engine speed increases, the BSFC is decreased gradually to its minimum value at engine speed of 3000 rpm, then increases as the speed was increased to 3500 rpm. Figure (15) presents the relation between brake power (kW) and engine speed (rpm) for LPG and gasoline fuels for the two-compression ratio (9.9:1 and 10.25:1). Peaks of brake power was indicated at engine speed of 3500 rpm for all experimental and computational results. Brake power

increases rapidly with increasing of engine speed for each compression ratios equal to 9.9:1 and 10.25:1. While, figure (16) shows the brake thermal efficiency for different values of engine speed (rpm), different operating fuel (LPG and gasoline), brake power (10 kW) and two compression ratio of (9.9:1 and 10.25:1). The peak brake thermal efficiency in all cases is reached at 3000 rpm.

CONCLUSIONS

An extensive investigation of the combustion, cycle, performance parameters and exhaust emissions of SI engine running on gasoline and LPG has been performed in this work. From the obtained results, the following conclusions can be drawn:

1. LPG fuel improves the engine brake specific fuel consumption and brake thermal efficiency but reduces the brake power compared to gasoline fuel. The maximum of percentage deviation BSFC in gasoline fuel compare with LPG fuel reported is 3.11% for engine brake power equals to 10 kW when compression ratio equal 9.9:1.

2. The maximum cylinder pressures predicted for LPG are lower than that produced by gasoline fuel. The maximum pressure from (62 to 71.5) bar for gasoline and from (60 to 67.12) bar for LPG when 9.9:1 compression ratio and for equals to 10 kW. LPG reduces the engine volumetric efficiency compared to that produced by gasoline fuel, thus, engine effective power is decreased. The maximum volumetric efficiency in all cases is reached at 3500 rpm and for equals to 10 kW. The volumetric efficiency was 76.8% for LPG and 85.9% for gasoline when compression ratio equals to 9.9:1.

3. Compression ratio and equivalence ratio have a significant effect on both performance and emission characteristics of the engine and have to be carefully designed to achieve the best engine performance characteristics. The equivalence ratio of gasoline is higher than LPG, the maximum difference of equivalence ratio brake power equals to 10 kW between gasoline and LPG was 3.65% and 6.6% for compression ratio 9.9:1 and 10.25 respectively.

4. The exhaust gas temperature increases with the increase of engine speed, compression ratio and brake power. Similar trends are also observed when LPG fuel is used. The exhaust gas temperatures for gasoline fuel is higher than that for LPG fuel at the same brake power and engine speed. The maximum exhaust gas temperature reported is 706°c for LPG fuel while that for gasoline is 741.4^{o}c at point 1 for brake power on engine equals to 10 kW and compression ratio equals to 9.9:1.

Table 1: Specifications of the test engine.

Details		Specifications
Engine type		MR18DE
Cylinder arrangement		In-line 4
Displacement (cm^3)		1,797
Bore and stroke(1) and (2) (mm)		84.0*81.1*84.25
Valve arrangement		DOHC
Firing order		1-3-4-2
Number of piston rings	Compression	2
	Oil	1
Compression ratio		8, 9.9 and 10.25, 15
Compression pressure (bar)	Standard	15.0
	Minimum	12.0
	Differential limit between cylinders	1
Valve Timing		

a	b	c	D	e	f
212	224	-8	52	7	25

Table 2: Properties of utilized fuels

Properties	LPG	Gasoline
Density at 15 $^{\circ}$C (kg/cm^3)	0.554	0.737
Lower Heating Value (MJ/kg)	45.25	43.96
Octane number	104	82
Boiling temperature ($^{\circ}$C)	2.2	210
Vapor pressure at 37.8 $^{\circ}$C (kg/cm^2)	8-10	0.45-0.62
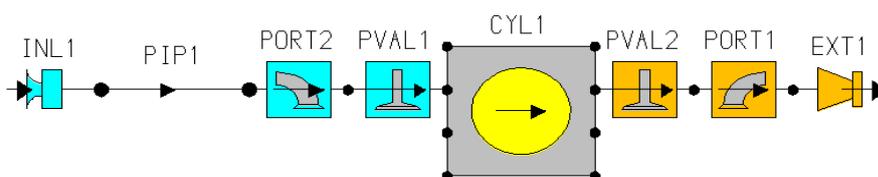
Figure (1): Model of one cylinder
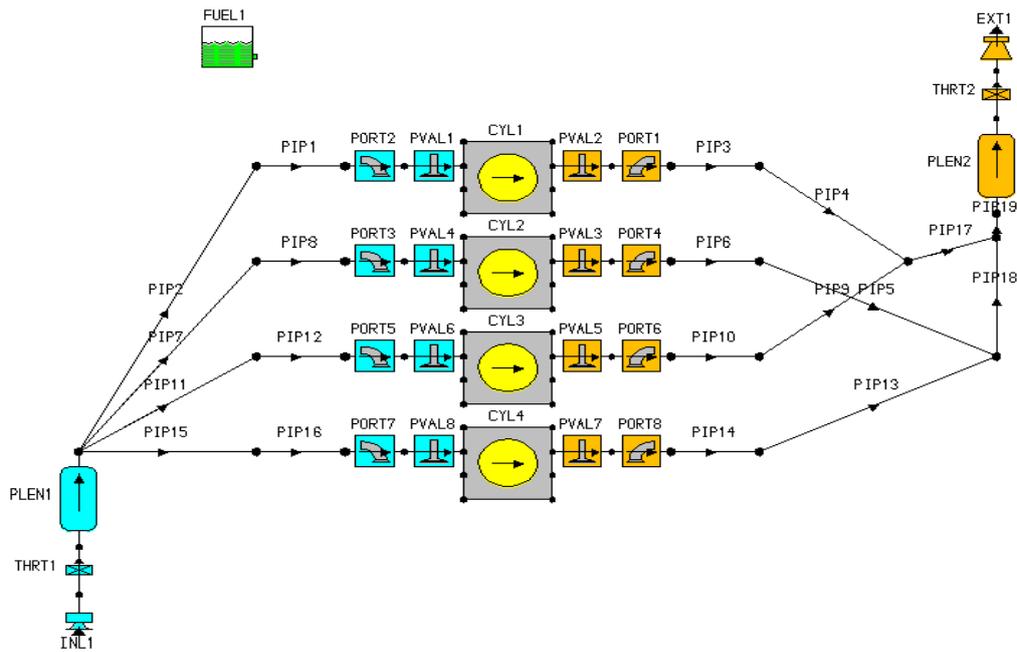
Figure (2): LOTUS model of four cylinder IC engine

| 1. Throttle position sensor | 2. Air flow sensor | 3. Fuel pressure sensor | 4. Crankshaft speed sensor | 5. Knock sensor |
| 6. Temperature sensor | 7. Exhaust Temp. sensor | 8. Oxygen sensor | 9. Fuel injector | 10. Ignition control |
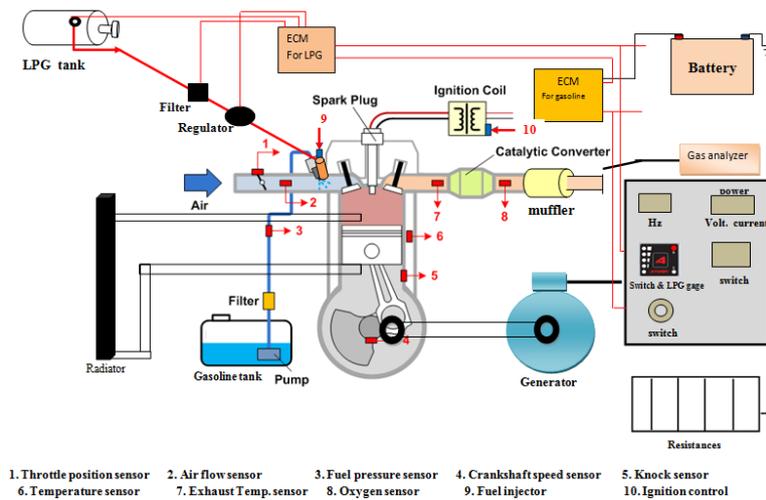
Figure (3): Schematic diagram of experimental setup
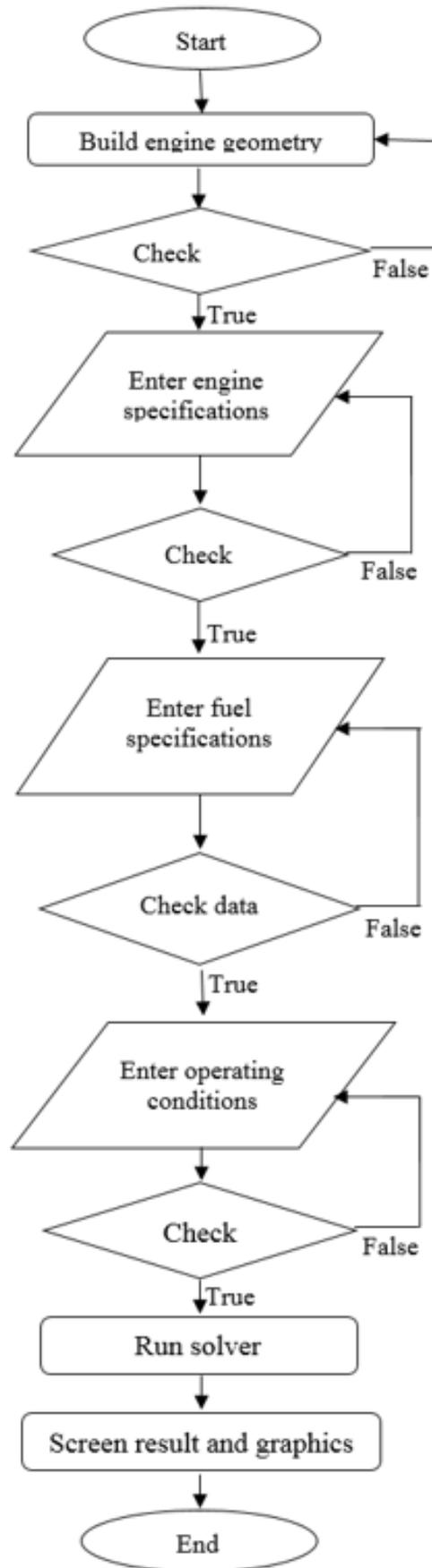
Figure (4): The schematically steps of LOTUS Engine Simulation model.
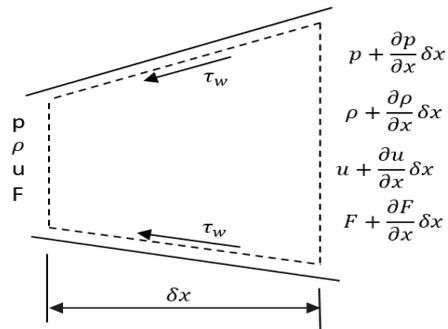
Figure (5): Fluid control volume in duct
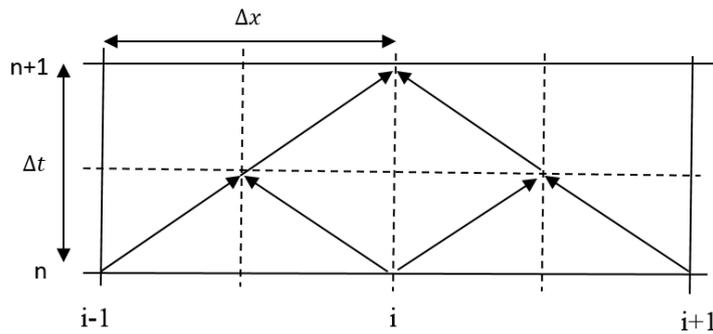
Figure (6): Schematical representation of the two-step Lax-Wendroff scheme.
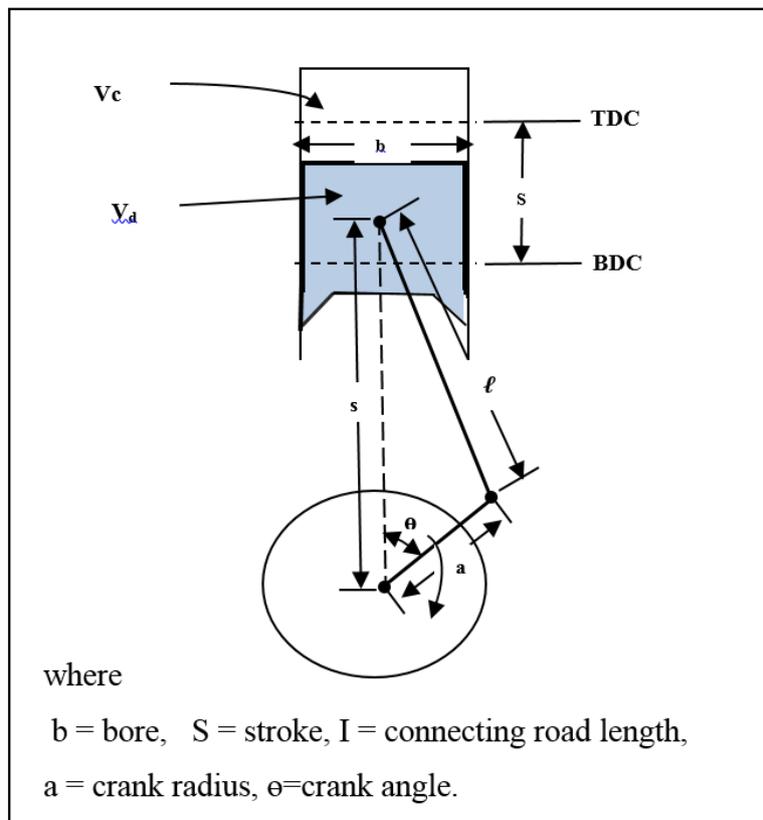
Figure (7): Geometry of cylinder
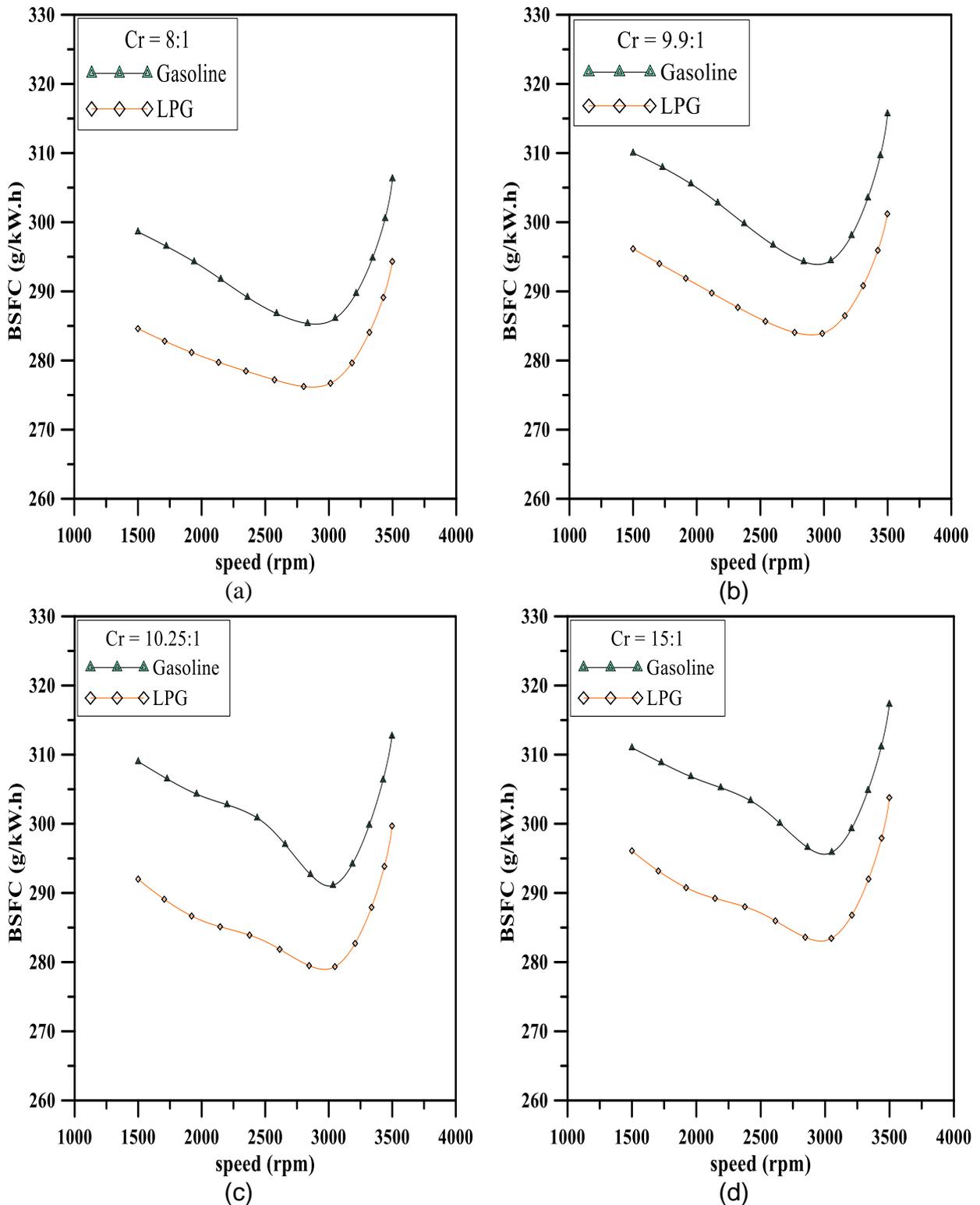
Figure (8): Variation of brake specific fuel consumption with engine speed
for brake power equals to (10 kW).
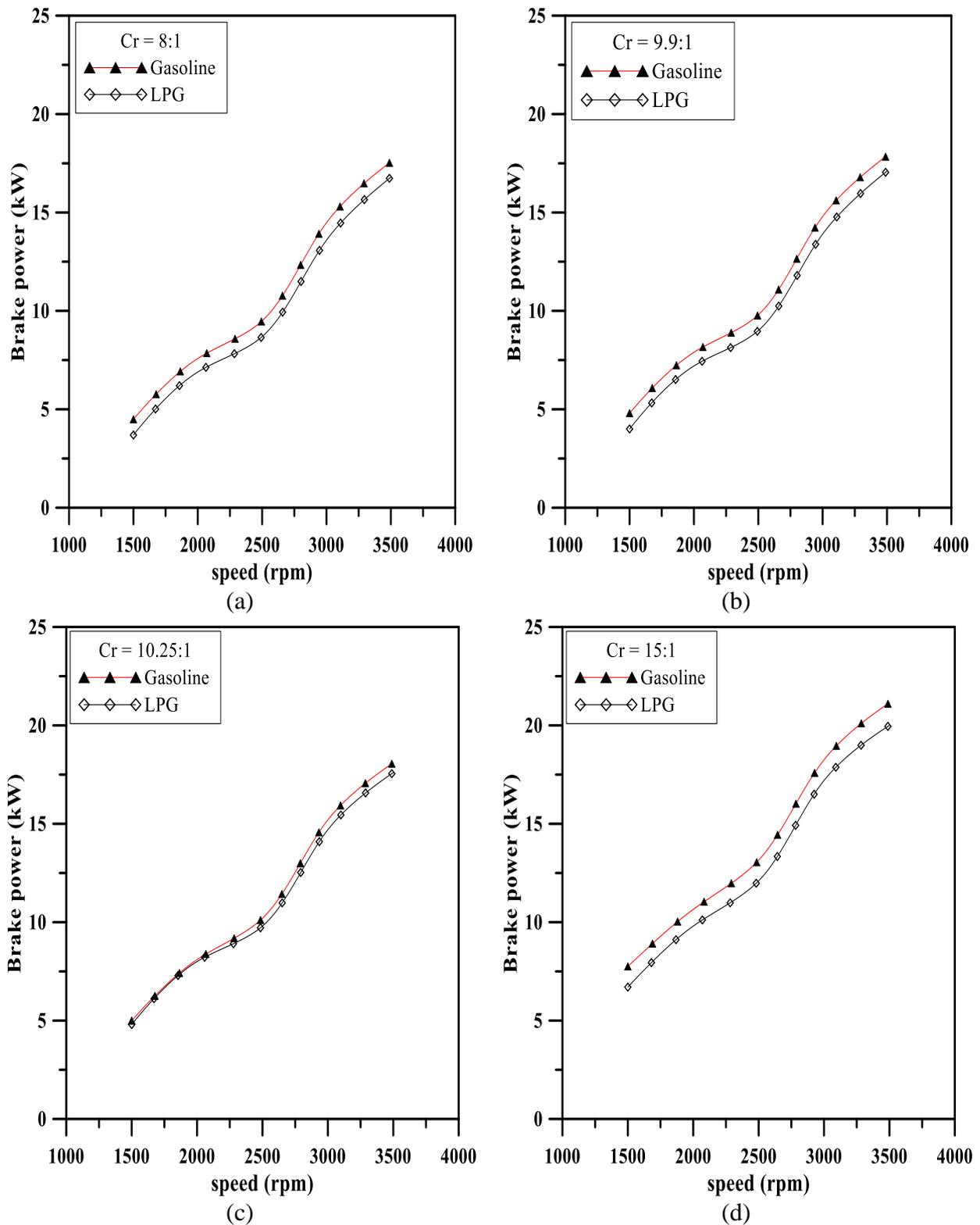
Figure (9): Variation of brake power versus engine speed.
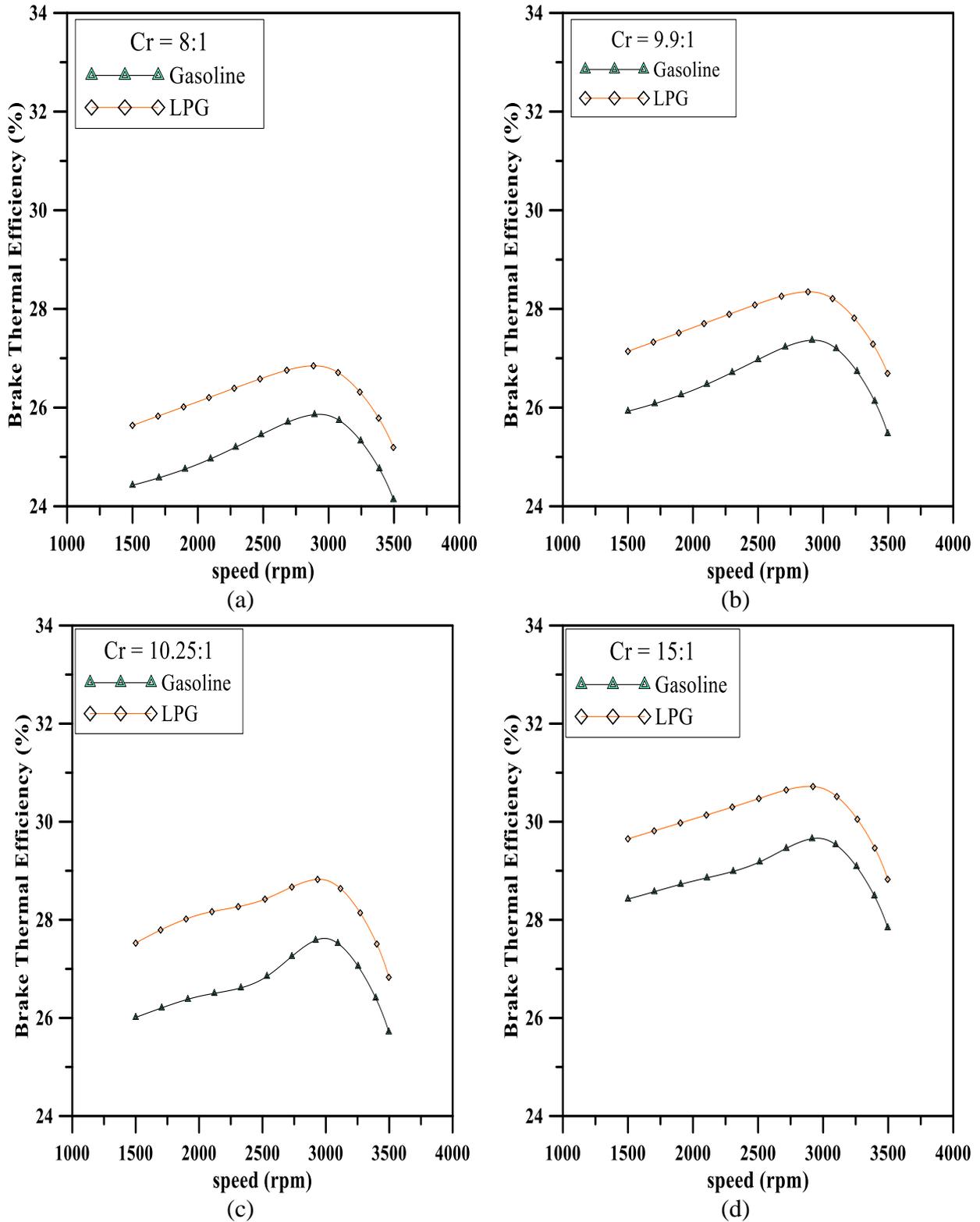
Figure (10): Variation of brake thermal efficiency versus engine speed
for brake power equals to (10 kW).
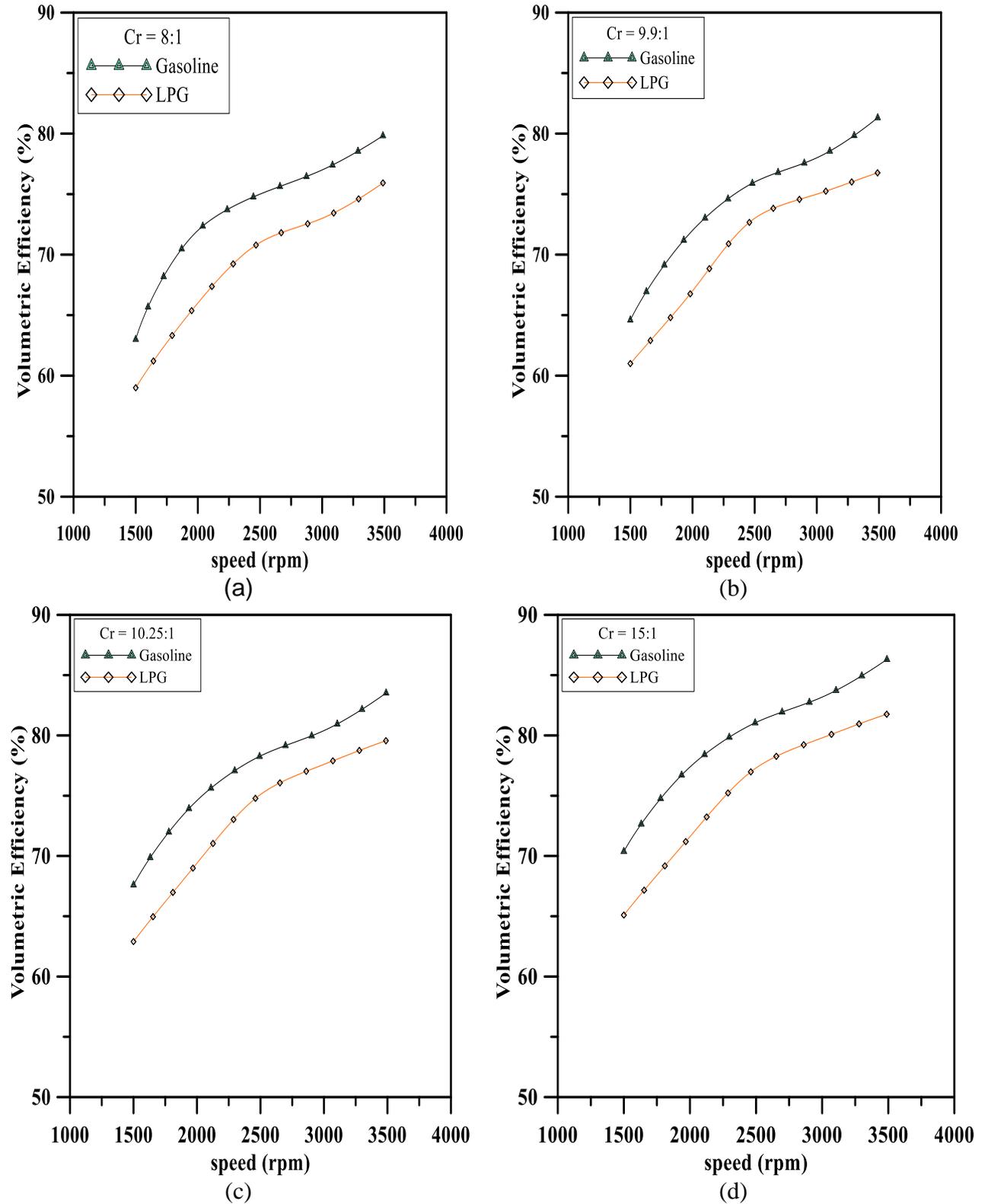
Figure (11): Variation of volumetric efficiency versus engine speed for brake power equals to (10 kW).

(a)

(b)

(c)

(d)
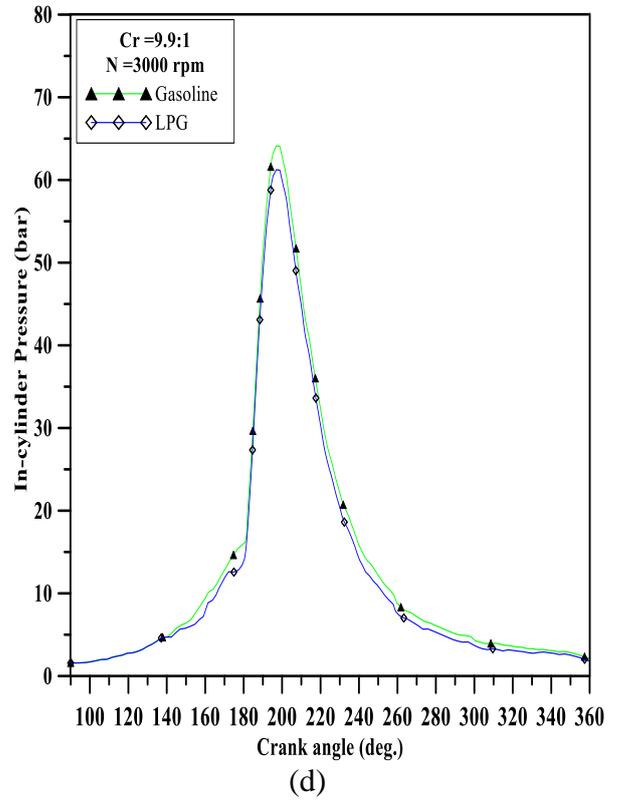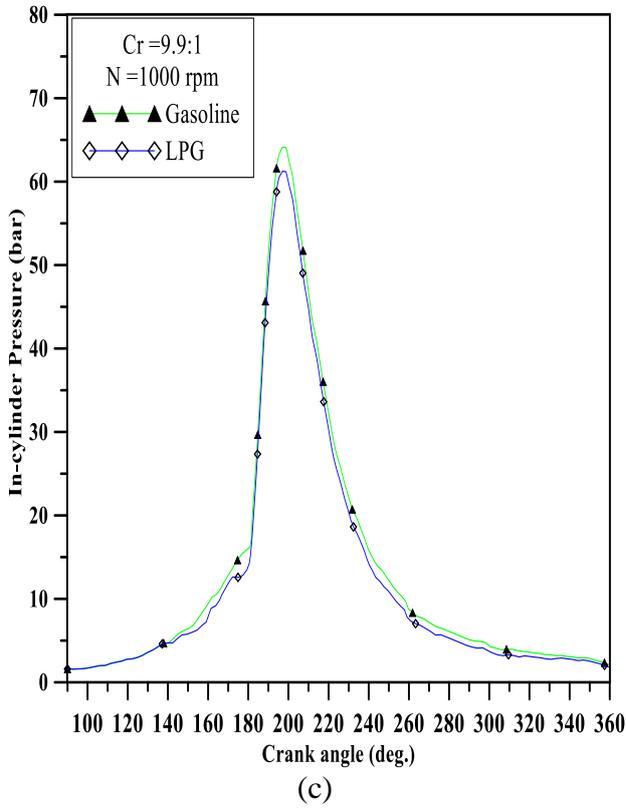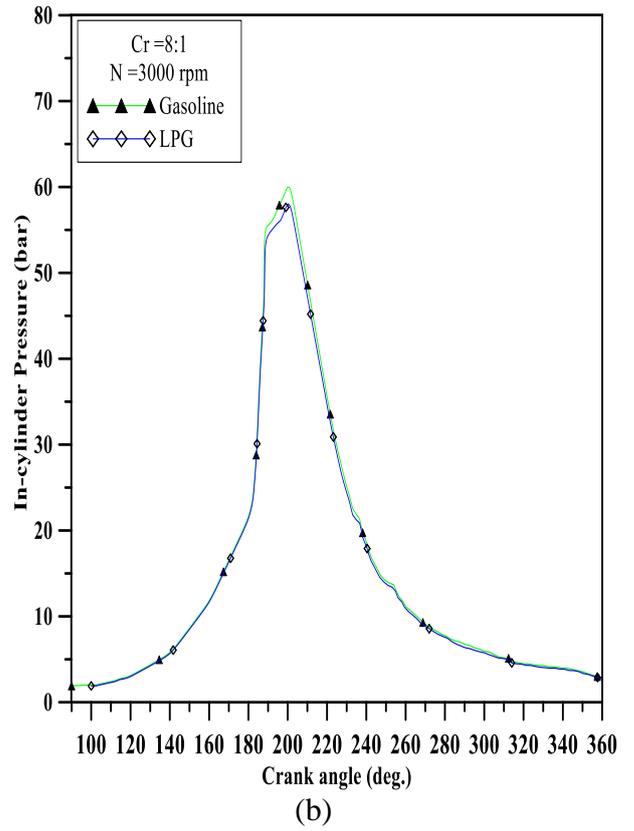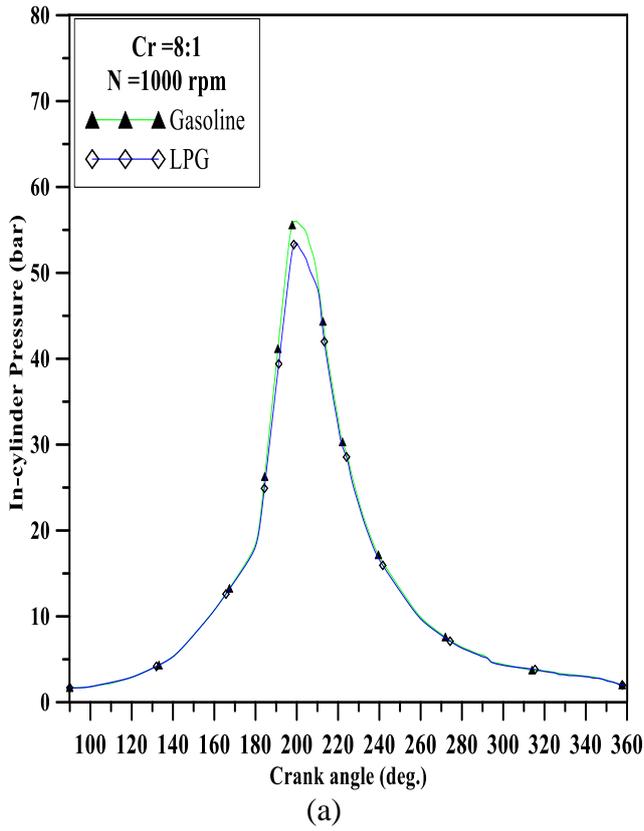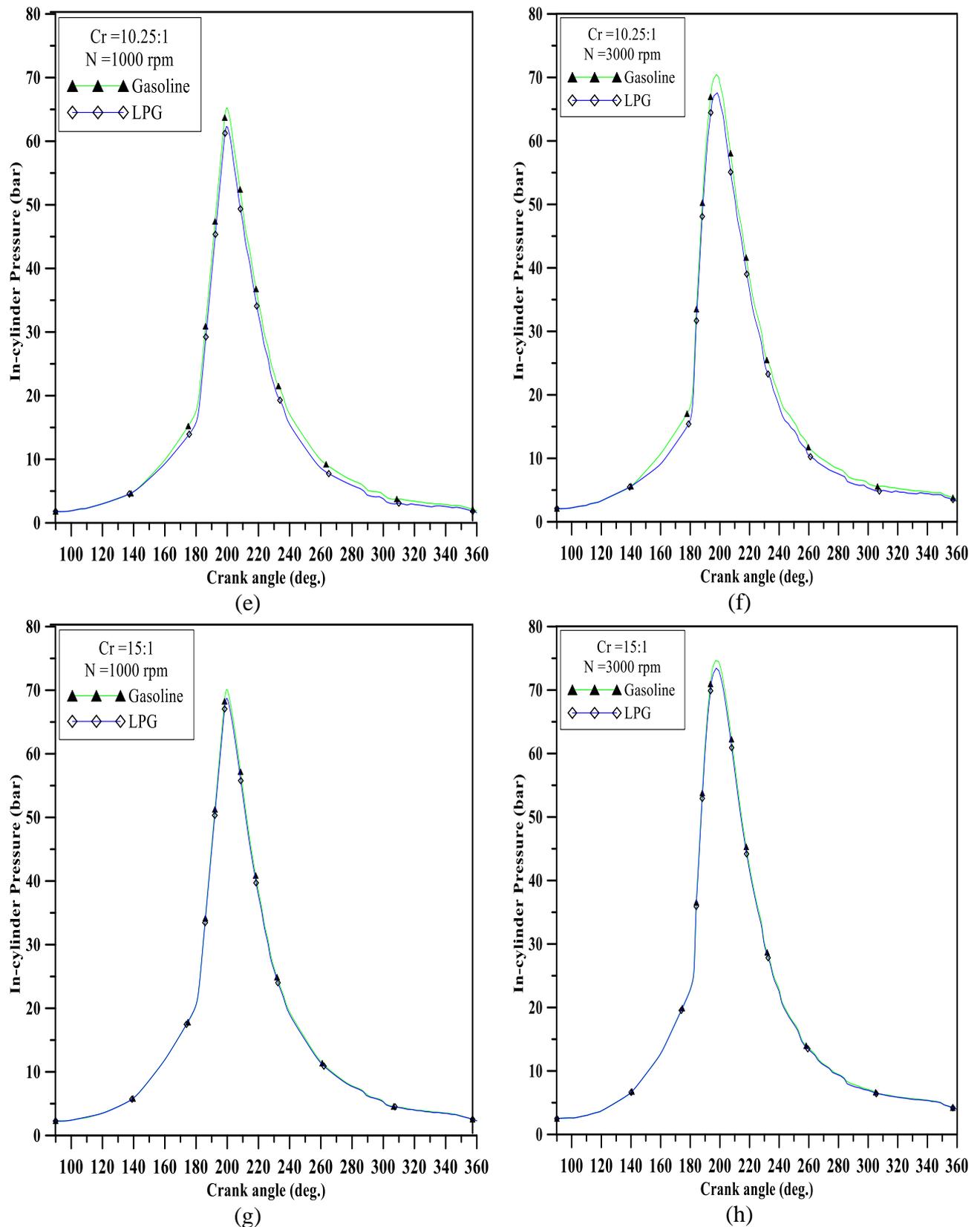
Figure (12): Variation of In-cylinder pressure versus crank angle for
brake power equals to (10 kW).
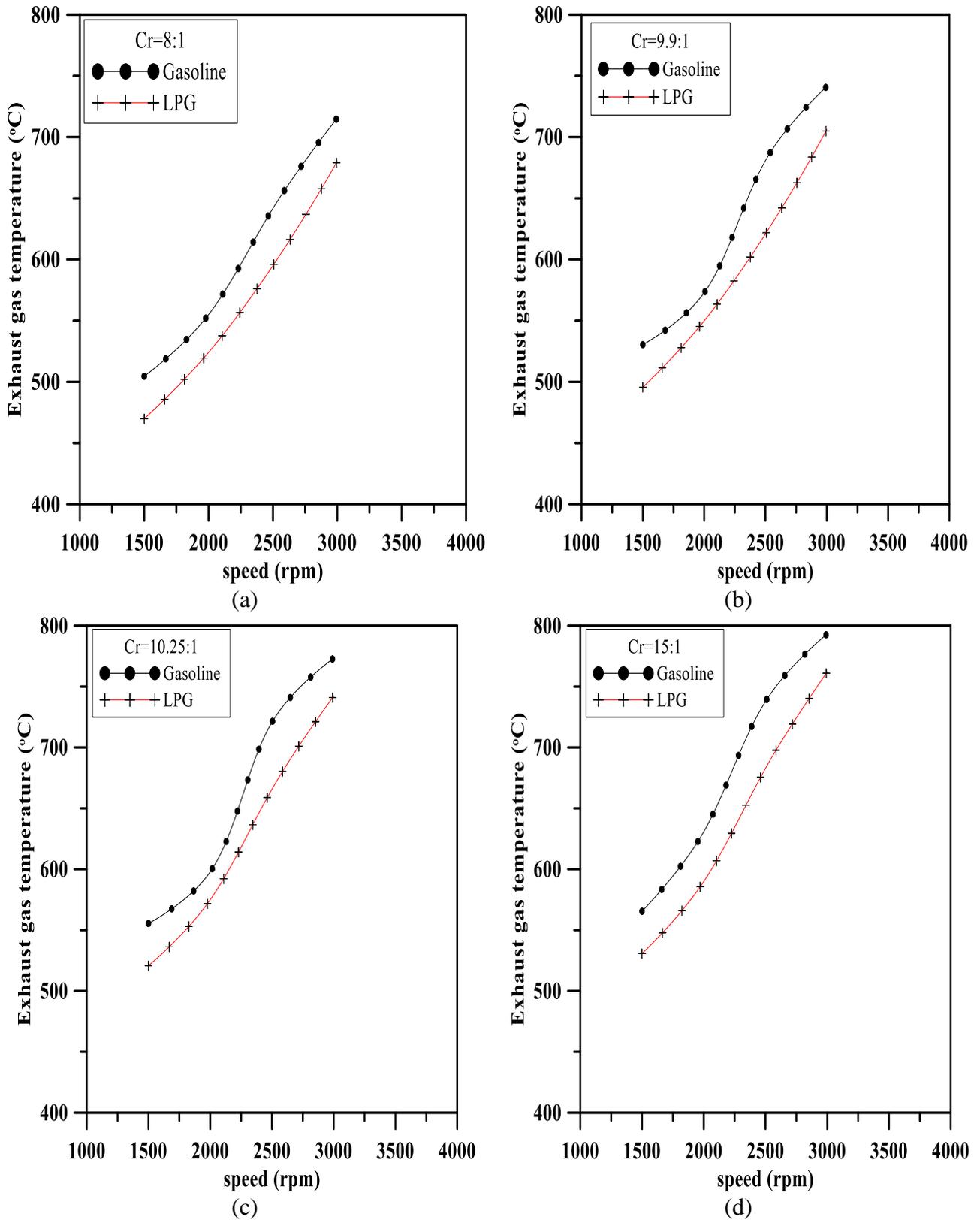
Figure (13): Variation of exhaust gas temperature versus engine speeds for brake power equals to (10 kW).

(a)   (b)
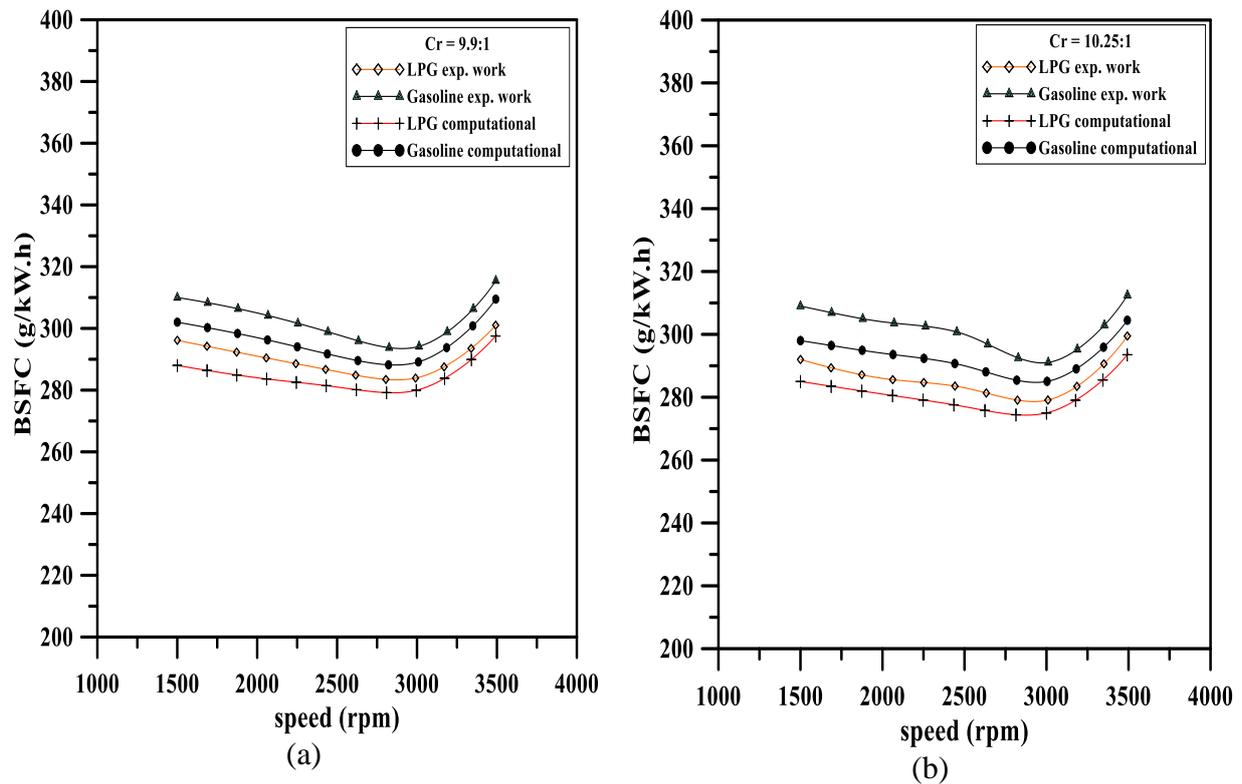
Figure (14): Variation of brake specific fuel consumption versus engine speed for brake power equals to (10 kW).

(a)   (b)
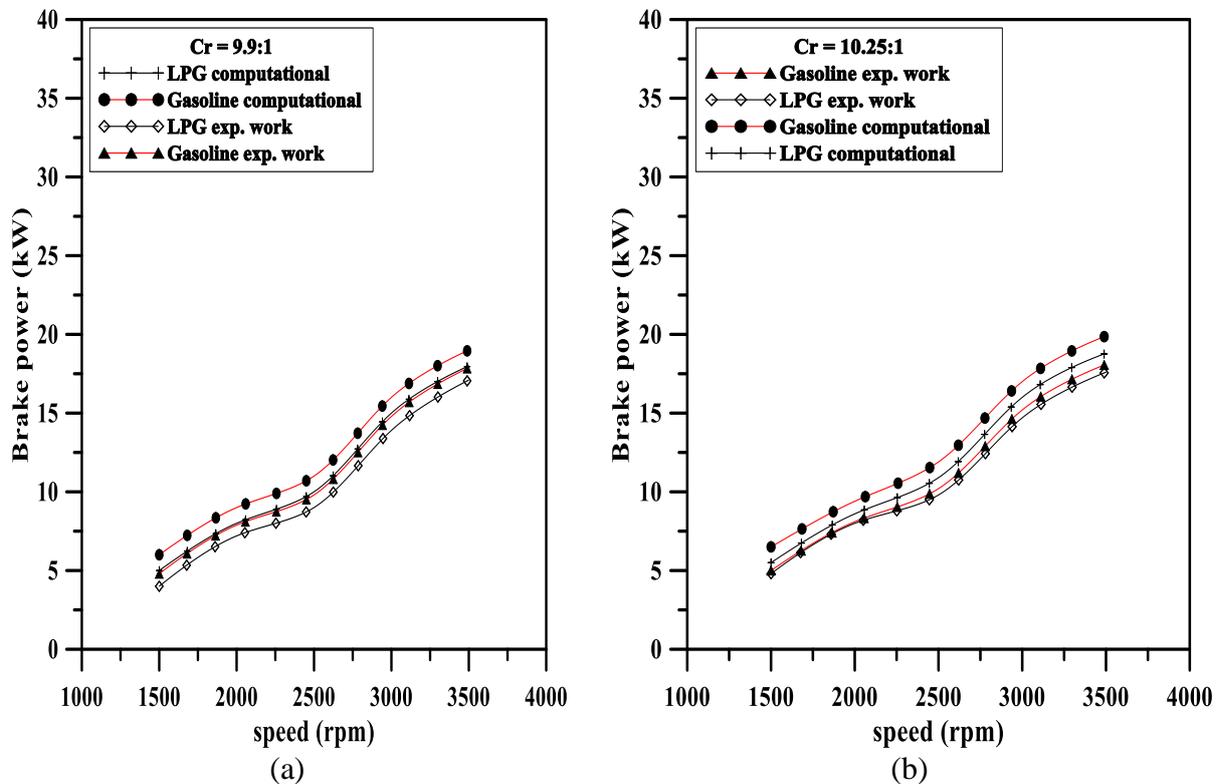
Figure (15): Variation of brake power versus engine speed.
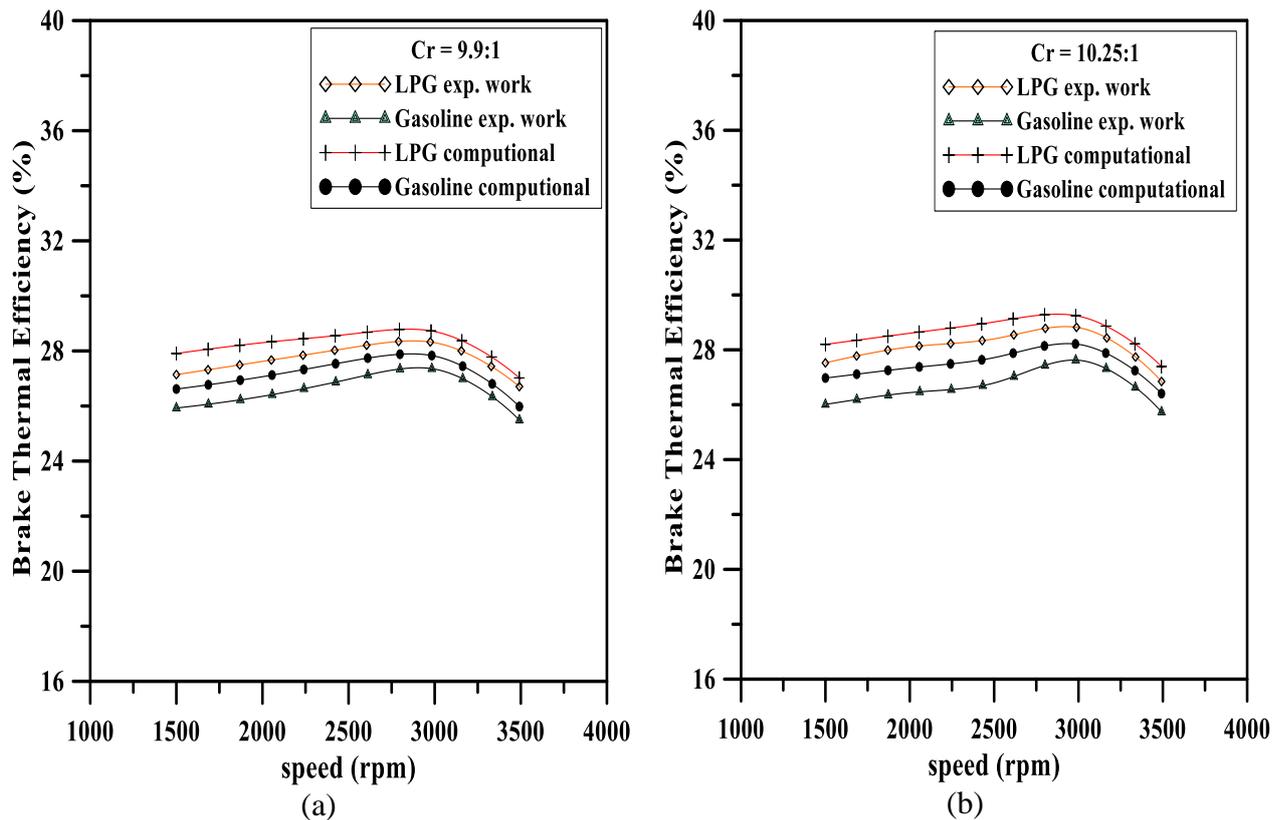
Figure (16): Variation of brake thermal efficiency versus engine speed for
brake power equals to (10 kW).
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