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Performance and in-cylinder flow characteristics of LPG powered converted diesel bus engine: influence of air fuel blending system

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Abstract: Alternative fuels used in diesel buses are becoming the subjects of interest today. LPG, one of these fuels, has some economical and ecological benefits compared to diesel. Converting diesel engine into gaseous fueling is accompanied with performances degradation. Therefore, combustion must be improved to remedy this problem. Combustion Improvement is directly related to the enhancing of intake aerodynamic mixing movements which is influenced by the air-gas mixing system. This paper investigates the air-LPG engine mixer geometry effect on performance of a six-cylinder, heavy duty, IVECO engine, which is used to power urban buses in Tunisia. This engine was retrofitted from its diesel version into bi-fuel gasoline/LPG technology. The gaseous fueling technique used is the carburetion due to its benefits against the conversion cost for a wide range converted buses. Investigations were performed by numerical and experimental methods. A 3D modeling of the turbulent flow through engine intake system (with multiple-holes “MH” mixer and without mixer) was undertaken. The model is based on solving Navier-Stokes and energy equations in conjunction with the standard k-ε turbulence model. This model allowed the identity of the mixer effect. Results indicated that the addition of MH mixer can produce a better homogenous mixture. Experimental measurements are also carried out to validate the mixer influence by measuring the important engine performances. Comparative analysis of the experimental results showed the improvement of BP, BT, BTE and BSFC by 6.25 %, 3 %, 9 % and 6.6 % respectively using MHM mixer in LPG engine operation.

Key words: Alternative fuels; Converted engine; LPG mixing system; CFD; experiment; Engine performance.

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1. Introduction

Due to air pollution and restricted petroleum reserves, several fuels like CNG, LPG, hydrogen and biofuels, were regarded as promising alternatives for urban transport. To decrease traffic in many countries, including Tunisia, public transportation is encouraged. Urban buses are usually equipped with diesel engines whose emissions contribute in greenhouse gasses. Alternative fuels for internal combustion engines applications had emerged to substitute the conventional fuels such as gasoline and diesel. Among these alternative fuels, natural gas and liquefied petroleum gas (LPG) have significant impact on decreasing toxic emissions. Biofuels present additional possible alternatives engine fuels. They have a moderately high availability and enormous economic benefits, such as biogas and biodiesel [1-3]. In case of gaseous fueled engine (LPG and natural gas), promising results were achieved in terms of fuel economy and exhaust emissions reduction [4-5]. Numerous countries have used gaseous fuel for public transportation. In Japan, Canada, and Italy as much as 7% of the urban buses are powered with LPG [6]. LPG is produced from petroleum refining and from natural gas. Actually, LPG supply exceeds the demand in the mainly of petroleum countries, so its price is inferior to other fuels [7]. This gas is a mixture of propane (C_3H_8) and butane (C_4H_{10}). The proportions ratios fluctuate between countries but propane frequently comprises 80 to 95 % of the total. The use of LPG as an alternative fuel could be split into two main types depending on their fuel consumption. The first is gaseous engine type, which has been designed from the beginning to operate on gas. The second is converted engine type from diesel or gasoline version, which is also divided into two types: dual fuel system (both diesel and gas were introduced into cylinder) and bi-fuel system (engine is retrofitted to operate with gas). Numerous reliable researches on LPG fuelled engines have been done to enhance the benefits of this gas. [8-13].

LPG is generally used in private vehicles, but it's rarely in full-sized transit buses. Like all gas vehicles, LPG buses can profit from the gaseous fuels benefits, i.e. the very low proportion of sulfur and a low carbon hydrogen ratio, which tends to product relatively low CO_2 emissions [14-16].

To convert diesel engines to SI gaseous bi-fuel engine, many systems are changed, including the intake and air-fuel blending system. Consequently, a reliable design of this system leads to improve the combustion process, therefore to optimize engine performances. An air-fuel mixer is a mechanical system, which ensures optimal air-gas mixing in transient conditions. That's why the suitable design of this device directly affects engine performances. Gas mixer is a tool used to establish the amount of LPG mixed with air before entering in cylinder and able to provide the engine with the optimal blend of gas and air according to various engine speeds. The perfect blend of gas and air mainly helps to improve gaseous engine performance and reduce emissions [17]. To ensure good combustion, a correct fuel-air mixture is according to the required stoichiometric air-fuel ratio. Various studies are performed to examine mixer design effect on the engine behavior, and especially the engines converted from its diesel originality [18-21]. Two parameters should be fulfilled in the mixer design process: turbulence that enhances the mixing quality and losses reduction in order to increase charged gaseous mixture volume. To reach this point, numbers of optimal designs are proposed: venturi-type mixer, fan type mixer and multi-holes type. Our choice is focused on multiple-hole mixer. Literature has shown the benefits of this mixer type [17]. A clear understanding of the fluid motion and dynamic processes of the engine mixture preparation are necessary to enhance the concept of each part involved. Computational fluid dynamics "CFD" can be used to give information about the properties of engine fluid flow. CFD simulation helps in adapting engine components design, saving both time and money. Therefore, numerical simulations have been mostly employed in the majority of internal flow engine studies [22-26]. However, in most of former work, flow simulations through the air-gas homogenizing system have been considered without stressing on gas mixer effect on in-flow velocity and turbulence structures for diesel heavy duty engines converted into gaseous fueling.

In the present work, the fuel system of a LPG powered bus engine, focusing on the mixture formation unit was reviewed in the first case. A numerical simulation of the aerodynamic and turbulence fields is carried out, using the CFD code SolidWorks Flow Simulation, through an air-gas blending system of an IVECO urban bus engine. This engine has been converted from its diesel version into a spark ignition (gasoline-gas) bi-fuel version. This study is interested in the inlet system optimization and especially the gas mixer to ensure the correct operation of the retrofitted engine. Two possible designs are presented of intake system (Fig.1). The first design is added by a Multiple outlet gas Holes Mixer "MHM" (Fig.2). The second is consisted of intake system without mixer. In this case, LPG is injected directly in carburetor. This technique is used in order to ensure a mixture rich in gas, since heavy-duty engines require plenty of power which is lost using alternative gases [27]. The other benefit of this technique is the gain of mixers manufacturing cost since the number of diesel buses, to be converted in Tunisia, is very important. The comparison of the in-cylinder flow structures using the two geometries notifies the addition mixer effectiveness. The simulation objective is to examine the LPG mixer effect and, thus, making it possible to provide a fine knowledge of the mixture aerodynamics nature (velocity, turbulent characteristics and fuel mass fraction). Hence, the three-dimensional resolution of Navies-Stokes equations in conjunction with the standard k-ε turbulence model is achieved. The numerical modeling is realized considering the aspiration of one cylinder. The second objective of this paper is to identify the mixer influence on engine performances using experimental method. The MH mixer justified from numerical simulation was installed on a four-stroke engine type IVECO 8210.02. The used fuel in the experiments is the liquefied petroleum gas LPG. Engine power, torque, specific fuel consummation and thermal efficiency are determined.

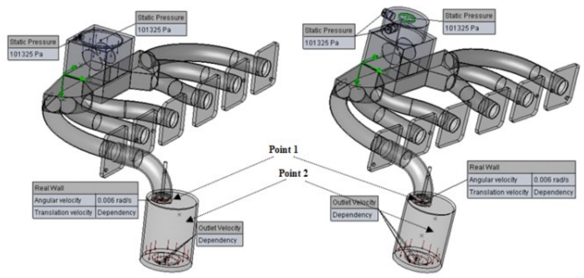


Fig.1. Air LPG intake system, a- Without mixer, b- with MH mixer

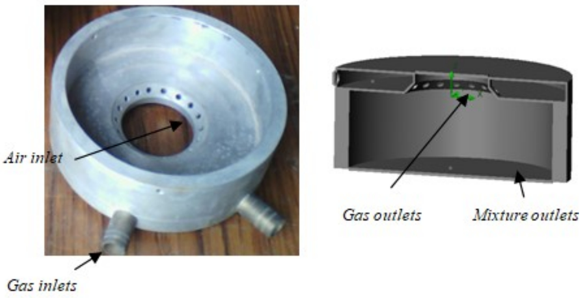


Fig.2. MH mixer design

2. Engine configuration

An IVECO engine, six-cylinder, 13.8 liter displacement, water-cooled, heavy duty, direct injection (DI), installed at the authors' laboratory was modified to bi-fuel spark ignition (SI) engine gasoline and gas fuelling (LPG and CNG). It is used to power the urban diesel buses in Sfax, Tunisia. The engine was operated at wide open throttle condition. The main engine specifications are given in Table. 1.

Table.1. Characteristics of the engine model

Engine parameters	Value
Engine (four cycle)	IVECO
Reference	8210.02. AP 160
Type	6 Cylinders – Inline
Bore × Stroke (mm)	137 x 156
Displacement (dm^3)	13.8
rod length (mm)	260
Crank radius (mm)	78
Compression ratio	16:1 (diesel), 12 : 1
Engine speed range (rpm)	(bi-fuel)
Cooling system	700 - 2000
Firing order	Water cooling
Maximum power at 2000 rpm (kW)	1-5-3-6-2-4
(diesel)	191,36
Maximum torque at 1200 rpm (daN.m) (diesel)	101

3. Fuel system of LPG engine

The major components of LPG fuel system are gas storage tank, vaporizer and the air-fuel mixer. On the whole, LPG is brought in at high pressure gaseous form in the storage tank. Pressure is dropped by passing through regulator and vaporizer. Finally it reaches the intake manifold, passing through the mixer or injected directly into the carburetor, before entering inside the cylinder. Pressure difference in the mixer contraction passage produces a velocity variation. A change in fuel flow is created to join and blend with the main airflow in the necessary proportion [28]. The mixer size was based on the air flow drawn into the engine.

In this study, a multiple holes type mixer, with 17 holes, based on venturi system is chosen. Despite its manufacturing problems, it brings gains in engine filling [24]. Holes were made along the circumference of the throat to dispense the propane fuel to the air stream evenly, with a radial arrangement.

4. CFD modeling of intake system

CFD is largely used to simulate the transition engine flows. In this study, the commonly available CFD means SW Flow Simulation is used. It has the benefit of importing geometry directly from CAD software such as SolidWorks. Firstly, 3D geometry of the two intake system configurations (with and without mixer) is built using the SolidWorks (SW) software. The files created by SW are imported in CFD code to build the grid for the final calculation of simulation. The 3D model can simulate the in-cylinder flow under unsteady engine conditions [29]. The simulation methodology is indicated in Fig.3.

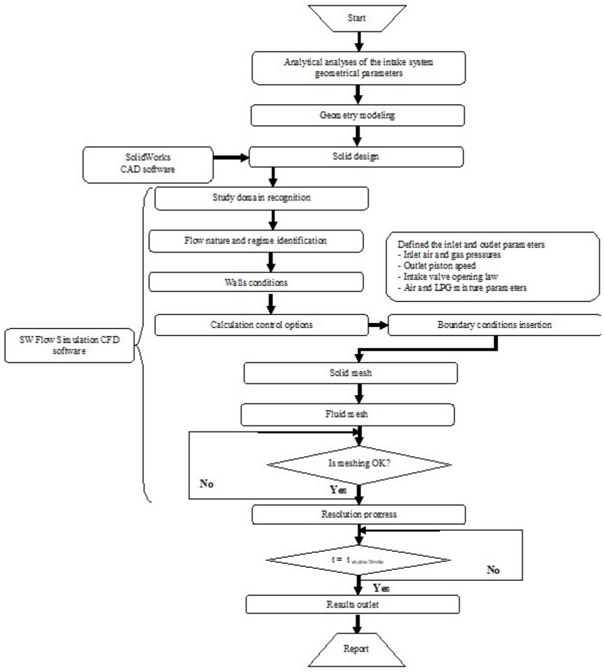


Fig.3. Intake system numerical simulation steps

4.1. Mathematical model

In this paragraph, the conservation equations of CFD simulation required for modeling inlet premixed air gas flow is presented. The equations governing the dynamics of gases are expressions of the conservation and thermodynamics laws. Simulation is based on the steady and unsteady flow. Gases are defined as compressible fluid. The standard model k-ε is used to solve the problems of intake flows reigning inside two inlet system configurations. Equations governing the flow model, including the conservation equation of mass, conservation of momentum, and the conservation equation of energy, are summarized in the Navier-Stokes equations:

$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x_i}(\rho u_i)=0\,, \qquad (1)$$

$$\frac{\partial}{\partial t}(\rho u_i)+\frac{\partial}{\partial x_i}\left(\rho u_i u_j\right)+\frac{\partial p}{\partial x_i}=\frac{\partial}{\partial x_i}\left(\tau_{ij}+\tau_{ij}^R\right)+S_i\;;\quad i=1,2,3$$

(2)

$$\frac{\partial \rho H}{\partial t}+\frac{\partial \rho u_i H}{\partial x_i}=\frac{\partial}{\partial x_i}\left(u_j\left(\tau_{ij}+\tau_{ij}^R\right)+q_i\right)+\frac{\partial p}{\partial t}-\tau_{ij}^R\frac{\partial u_i}{\partial x_j}+\rho\varepsilon+S_i u_i+Q_H$$

(3)

$$H=h+\frac{u^2}{2},$$

The following energy equation is:

$$\frac{\partial \rho E}{\partial t}+\frac{\partial \rho u_i\left(E+\frac{p}{\rho}\right)}{\partial x_i}=\frac{\partial}{\partial x_i}\left(u_j\left(\tau_{ij}+\tau_{ij}^R\right)+q_i\right)-\tau_{ij}^R\frac{\partial u_i}{\partial x_j}+\rho\varepsilon+S_i u_i+Q_H$$

(4)

$$E=e+\frac{u^2}{2} \qquad (5)$$

4.2. Turbulence Model

Turbulence modeling is imperative in engine intake flows. Since the turbulence directly influences spray, blending, in-cylinder movements and combustion, adequate prediction of turbulence behavior is required. The study of this fact is essential in order to improve engine performance and emissions. In most of the computational codes developed so far, the characteristics of velocity are directly related to the corresponding turbulence [30]. Therefore, an appropriate modeling of intake mixing processes is dependent on the accuracy of the turbulence prediction. In this section, the turbulence models employed for current work will be developed according to the standard k-ε turbulence model [31]. The typical turbulence parameter consists of turbulent kinetic energy and its dissipation rate. The equations of turbulent kinetic energy k and its dissipation rate ε are written as:

$$\frac{\partial \rho k}{\partial t}+\frac{\partial}{\partial x_i}(\rho u_i k)=\frac{\partial}{\partial x_i}\left(\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right)+\tau_{ij}^R\frac{\partial u_i}{\partial x_j}-\rho\varepsilon+\mu_t P_B.$$

(6)

"P_B" represents the turbulent generation due to buoyancy forces [32] and can be written as:

$$P_B=-\frac{g_i}{\sigma_B}\frac{1}{\rho}\frac{\partial \rho}{\partial x_i},$$

(7)

where "g_i" is the component of gravitational acceleration in direction x_i, the constant $\sigma_B = 0.9$

$$\frac{\partial \rho\varepsilon}{\partial t}+\frac{\partial}{\partial x_i}(\rho u_i\varepsilon)=\frac{\partial}{\partial x_i}\left(\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_i}\right)+C_{\varepsilon 1}\frac{\varepsilon}{k}\left(f_1\tau_{ij}^R\frac{\partial u_i}{\partial x_j}+\mu_t C_B P_B\right)-C_{\varepsilon 2}f_2\frac{\rho\varepsilon^2}{k}.$$

(8)

"C_B" is defined as: $C_B = 1$ when $P_B > 0$, and 0 otherwise;

$$f_1=1+\left(\frac{0.05}{f_\mu}\right)^3\;,\; f_2=1-e^{\left(-\left(\frac{\rho k^2}{\mu\varepsilon}\right)^2\right)}.$$

(9)

The constants C_μ, $C_{\varepsilon 1}$, $C_{\varepsilon 2}$, σ_k, σ_ε are defined empirically [33]. In Flow Simulation the following typical values are summarized in Table 2.

Table.2. (k- ε) model empirical constants

C_μ	$C_{\varepsilon 1}$	$C_{\varepsilon 2}$	σ_k	σ_ε
0.09	1.44	1.92	1	1.3

4.3. Boundaries Conditions and Computational Mesh

An air ideal mass fraction equal to 1 and a pressure equal to 1.013 bars are considered as the air inlet initial conditions. The inlet fuel initial boundary conditions are the gas mass fraction as 1 and 1.013 bars for propane pressure. The outlet boundaries conditions are determined from the engine filing parameters, intake valve opening and alternative piston motion as is shown in Fig. 4. The equation describing the transient evolution of piston motion "U_{piston}" is presented as follows:

$$U_{piston}=\omega\, l\left(\lambda\sin(\theta)+\frac{1}{2}\frac{\lambda^2\sin(2\theta)}{\sqrt{\left(1-\lambda^2\sin^2(\theta)\right)}}\right),\;\lambda=\frac{r}{l},\;\;\theta=\omega t$$

(10)

"r" is the radius of the crankshaft, "l" is the length of the connecting rod.

The intake valve lift speed is determined from derivative form of the valve lift equation:

$$L(\theta)=\frac{1}{2}L_{\max}.\left(1+\tanh\left(\theta+IOA\right).\tanh\left(180-ICD\right)\right)$$

(11)

L_{max}: maximum intake valve lift, IOA: intake opening advance, ICD: intake closure delay

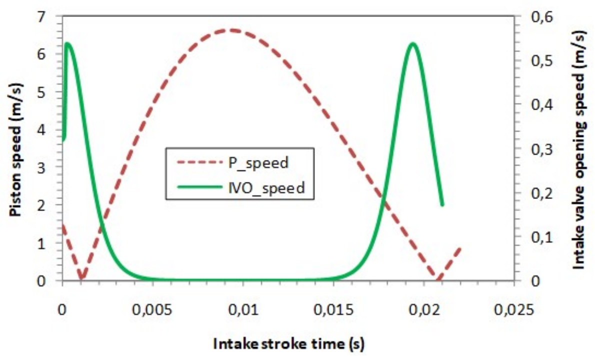
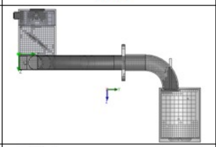
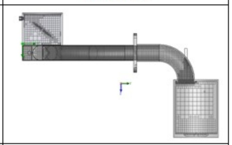
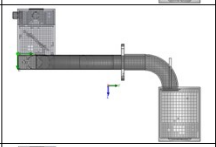
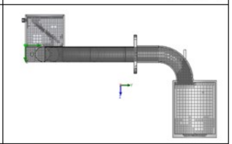
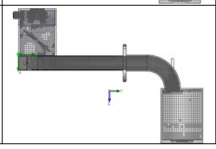
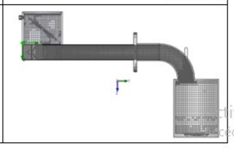


Fig.4. Piston and intake valve opening speeds temporal evolution

The inlet flow is in practice an unsteady process because of the valves motion, the piston motion, and the effects of the pressure [34]. To simplify simulation, the inlet system walls are assumed adiabatic: no transfer of heat with outside. The piston speed is taken for the engine speed corresponding to the maximum torque (n=1500 rpm). The LPG fuel simulated in the present study is C_3H_8, (almost 100% propane).

A rectangular mesh was used to model the engine intake system investigated in this study. The intake manifold of this engine is divided into two separate parts. Each three cylinders are fueled from a separate way. A separation procedure was adopted in this study. It splits up the cells of mesh in high-gradient flow sections, which cannot be determined before simulation, and merges cells in low-gradient flow sections (see Table. 3).

Table.3. Different dimensions of mesh cells

Mesh level	Total cells number		Meshing plan view	
	MHM	Without mixer	MHM	Without mixer
Mesh_1 (M_1)	126635	176764		
Mesh_2	203875	391096		
Mesh_3	619498	876449		

To achieve grid-independence computational results, three mesh levels were generated and analyzed. Suitable mesh size is selected based on the convergence and stability of results. The variation in Flow velocity at the intake system input was chosen as an acceptable criterion to validate the chosen resolution. As indicated in Fig. 5, it is clear that the grid sizes with Mesh_2 level and Mesh_3level give way very similar results for both configurations. Consequently, results are grid independent using a mesh_2 or more cells. Taking into account the time resolution constraints and availability of computer equipment, the decision was taken to work with Mesh_2 level in further analyses. In this work, only one cylinder is taken in intake stroke. Therefore the mesh of the manifold half is sufficient. Mesh features are summarized in Table 4.

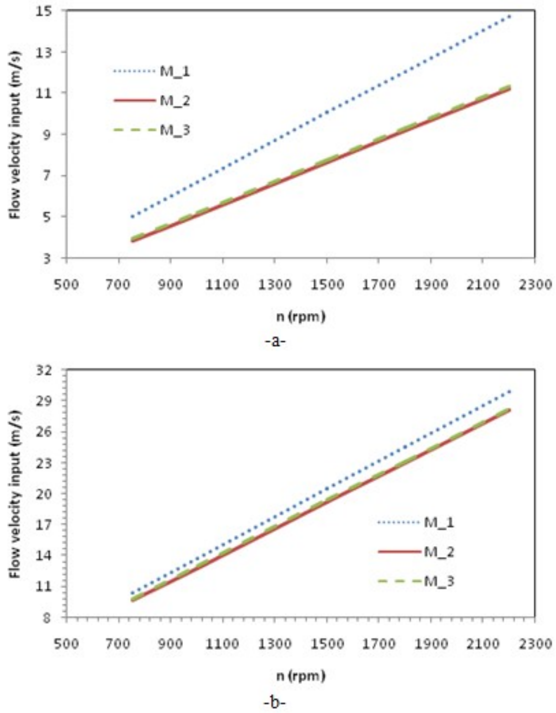
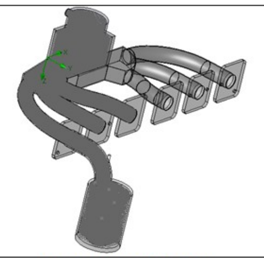
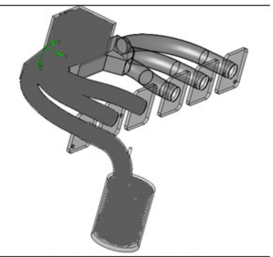


Fig.5. Effect of CFD mesh resolution on Flow velocity at the intake system input, a/ without mixer, b/ with MHM

Table.4. Intake system models grid

	
Intake system with MHM mixer Total cells: 203875, Fluid cells: 76975, Solid cells: 68556, Partial cells: 58344	**Intake system Without mixer** Total cells: 391096, Fluid cells: 166230, Solid cells: 115113, Partial cells: 109753

4.4. CFD Results and discussion

Mixing efficiency is an important parameter that determines the quality of combustion. A high combustion quality will produce low exhaust gas emission components. The purpose of numerical simulation tests is the identification of the mixer effect on mixing quality of fuel and air. Flow structures through the two configuration of intake system (with and without mixer) are studied. The cyclic variations of the mixture velocity, fuel air ratio, propane mass fraction and turbulent kinetic energy are determined. Results are taken in the intake valve measurement point (P1) and inside cylinder point (P2), which taken for a crank angle equal to 38° after top death centre, as can be seen in Fig. 1.

4.4.1. In-cylinder flow velocity variation

The control of the velocity distribution in the intake system improves quality of cylinder filing. In Fig. 6, the cyclic variation of mixture velocity is shown during in the intake stroke using MHM mixer and without mixer. It has noted the presence of four regions in the evolution of the two curves. The first, with linear evolution, is located in the first moments of intake stroke. This linearity is due to the intake valve opening advance. The second region presents a sudden peak of in-flow velocity. This is due to the admission of the pressurized air/fuel mixture stored in the intake manifold just after the valve opening. The third and fourth regions have parabolic evolution growing then decreasing in accordance with the piston movement. The dominance of the velocity, using the mixer, appears only in the fourth region when the piston approaches to the bottom dead center BDC. The velocity priority of the in-cylinder flow without mixer, at the intake stroke beginning, is explained by the large flow rate entering directly into the manifold without passing through the mixer rooms. When the piston continues its upward movement inflow velocity, with mixer, increases which indicates the effect of the mixer addition on volumetric efficiency. According to Fig. 6 a, the maximum velocity values approximates 160 m/s. It is explained by the restriction of passage section through the valve. Inside cylinder, velocity values did not exceed 13 m/s due to the sudden increase of volume in the combustion chamber.

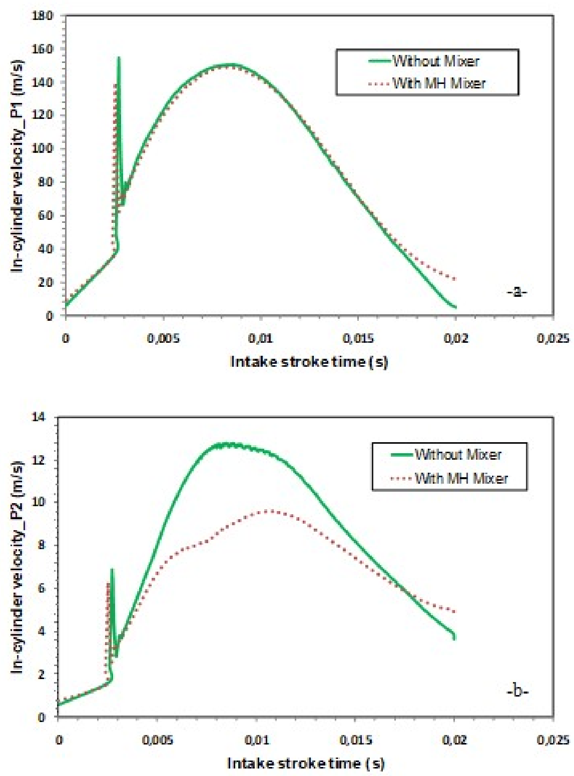


Fig. 6. Variation of in-cylinder velocity with and without mixer, a/ Intake valve level, b/ Inside cylinder level

4.4.2. Turbulent kinetic energy "TKE" variation

The turbulence affects the flow dissipation rate, heat transfer and the flame propagation rate. It is quantified by TKE inside cylinder. Turbulence can be beneficial to reach an optimum air-fuel blend and to qualify combustion. Its intensity during the intake stroke is presented in Fig. 7. It is observed that TKE are reaching their highest values at 0.01s intake stroke time "IST" for both curves with and without mixer. This expected by the higher in-cylinder velocity according to this IST. Variation of TKE is almost in line with the variation of the average air/fuel blend velocities. It means that almost it follows the variation of the inlet average blend velocity at a given engine speed. During the first moments of the intake stroke, the interactions between the blend jet (air and fuel) through the intake valve are the most important mechanism for the production of the TKE intensity, which explains the sudden peak in the curves at 0.0027 s (Fig. 7. a). The abrupt fluctuation turbulence intensity is due to the intake valve sudden opening and exhaust valve closing. In the middle of the piston path, a priority of the energy without mixer curve almost equal to 19 % is noticed. But, at the end of the intake phase, this priority is lost by using the MH mixer. Adding that element generates causes the increase in the turbulence gradually. As a result, the agitation of the particles rises causing a better homogenization of the air-gas mixture. From Fig. 7.b, it can be observed that TKE inside cylinder present similar temporal variation comparing to that of the variation through the valve but with reduced values. According to Figs. 7, the MH mixer improved the average TKE in the intake phase end.

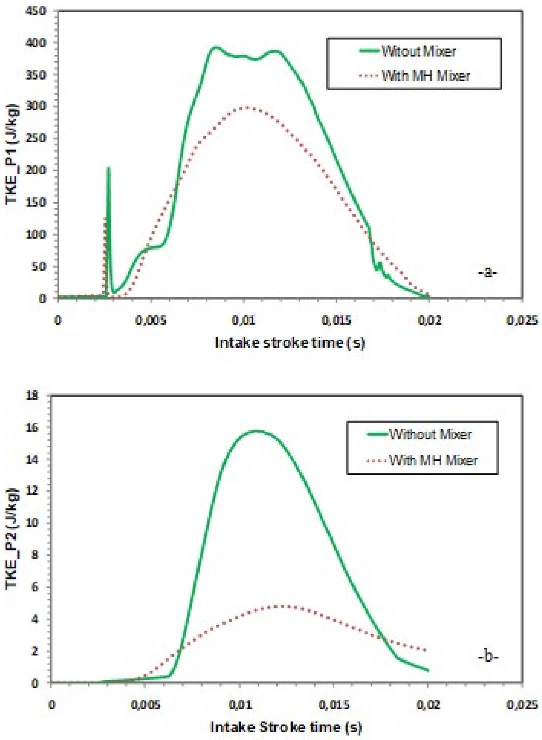


Fig. 7. Variation of kinetic turbulent energy, a/ Intake valve level, b/ Inside cylinder level

4.4.3. LPG-air ratio variation

In optimal combustion process, the fuel is burned totally with oxygen and it is called stoichiometric combustion. In real combustion processes, more of air is used to raise the chances of optimal combustion.

Fuel-air ratio FAR is the mass ratio of fuel to air present in internal combustion engine. The FAR is expressed on a mass basis and defined as:

$$FAR = \frac{m_{fuel}}{m_{air}} = \frac{N_{fuel}\left(M_{fuel}\right)}{N_{air}\left(M_{air}\right)}. \quad (12)$$

Where m represents the mass, N represents number of moles and M molar mass.

The air quantity used in combustion processes is also expressed in terms of the fuel-air equivalence ratio "Φ" called also "fuel richness":

$$\Phi = \frac{FA_{actual}}{FA_{stoich}} \begin{cases} <1 & \text{lean mixture (excess air)} \\ >1 & \text{rich mixture (default air)} \end{cases} \quad (13)$$

In the case of LPG, 15.5 *kg* of air is used to burn each 1 *kg* of gas (which is taken in this study 100% propane) [35], i.e. the stoichiometric FAR is equal to 0.065.

Fig. 8 depicts the LPG-air ratio, called also fuel-air ratio, variation of in-cylinder blend. At the intake valve level, the FAR is equal to 0.075 in the beginning of intake stroke for the two configurations. It means that a rich blend is introduced at opening valve. After a few moments of fluctuation, the FAR has stabilized using MH mixer. This stabilization is due to geometry effect of the mixer. Although the presented values are greater than stoichiometric FAR (which is equal to 0.065), using mixer approximates the blend nature to the stoichiometry. The inlet flow without mixer results a rich blend. Such a mixture causes numerous pollution problems due to unburned particles despite the advantages in engine power. Fig. 8 b depicts the predicted FAR evolutions for both intake configurations inside cylinder (point P2). It has noted that the ratio is almost zero in the first moments of the cycle. This is due to the elapsed time until arrival of first blend particles. In the rest of intake stroke, the FAR is almost equal to 0.07 using multi-holes mixer, such a very close value to stoichiometric FAR. The average difference between the two curves is equal to 8 %. Consequently, according to the two preceding figures, it is noticed that the use of the mixer results an acceptable blend for proper combustion.

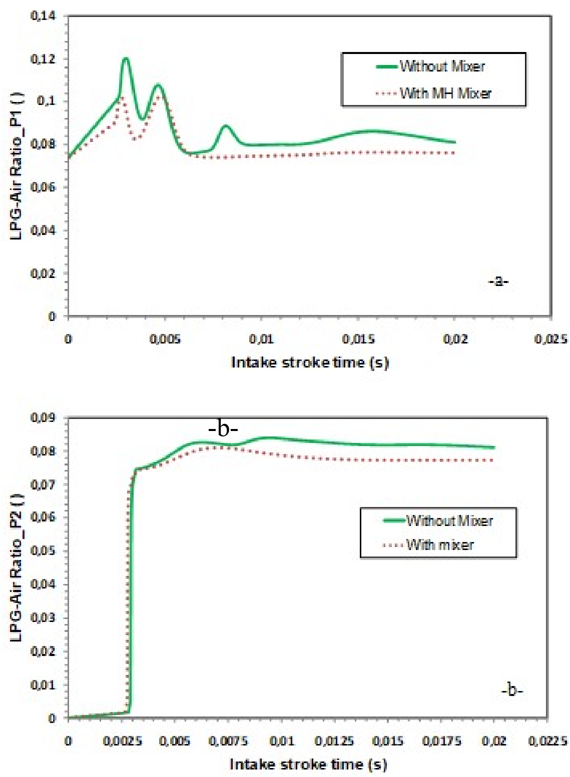


Fig. 8. Variation of LPG-air ratio, a/ Intake valve level, b/ Inside cylinder level

4.4.4. LPG mass fraction streamlines

Studying the flow along the entire intake system for the piston position corresponding to a crank angle equal to 170°, it is noteworthy in Fig. 9 that the progression of streamlines is more optimized using MH mixer. The LPG mass fraction (fuel mass quantity versus the entire mixture) inside cylinder is equal, in an average value, to 0.07 which is a value close to the FAR. The in-cylinder streamlines evolution without mixer (Fig. 9 b) presents dead zones in the combustion room due to the sudden jet entering without volume mixture preparation. The LPG mass fraction in this case is equal, in an average value, to 0.077, such value generates a rich blend which directly affects the quality of combustion, thus affects engine performance.

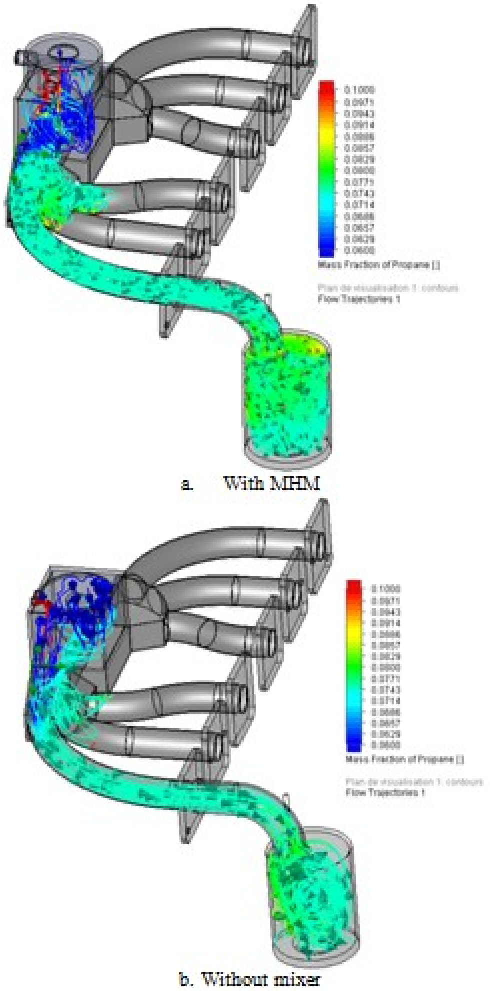


Fig. 9. LPG mass fraction flow trajectories

As a general conclusion, it can be deduced that the distributions of the velocity, turbulence energy and LPG-air ratio variation, depend strongly on air-gas mixer addition in inlet system. Also, from this numerical prediction of air-LPG mixture structure, it can be concluded that the optimal mixture homogeneity is tend to be occurred using MH mixer, especially with its advantages of streamlines and diffusion of propane in air. Therefore, the MH mixer was made and experimentally tested on IVECO bi-fuel gaseous converted engine.

5. Experimental study

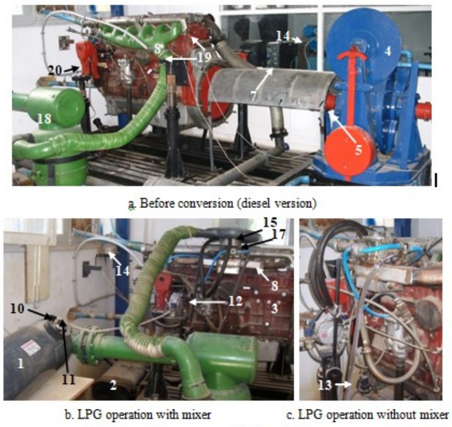
An experimental study is carried out to identify the mixer influence on engine performance. An engine test bench installed at the authors' laboratory is used. The brake power (BP), brake torque (BT), brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE) of the engine are measured before and after conversion with and without air/LPG MHM mixer. The object is to manage each series of experimental statements in order to plot diagrams which characterize the behavior of the test engine for bi-fuel operation. The test fuel used in experiments is the LPG. Its properties are illustrated in Table 5.

Table.5. LPG properties [35]

Property	**Value**
Molecular formula	C_3H_8 (propane)+ C_4H_{10} (butane)
Octane number	100-110
Stoichiometric air-fuel ratio	15.5
Lower heating value LHV (kJ/kg)	46000
Density ρ at 1 atm and 15°C (g/l)	510-580
Ignition temperature (°C)	480

5.1. Engine specifications

The engine is derived from a heavy duty, urban bus diesel engine type IVECO. This engine was modified to bi-fuel spark ignition (SI) engine gasoline and gas fuelling. The main engine specifications are summarized in Table 1. The engine combustion chamber was adapted to the spark ignition operation through a modification of pistons geometries and a reduction of the volumetric compression ratio from original version (16:1) into bi-fuel version (12:1). Through a thermodynamic analysis, the compression ratio is considered for the proper operation on spark ignition, according to the two following conditions: avoiding the risk of fuel mixture self ignition at the finish of compression stroke, and ensuring maximum mechanical power, combustion chamber pressure and thermal performance. Reducing this ratio is performed through the reduction of the height of the pistons according to the value of the selected compression ratio. The cylinder heads are modified, in order to adapt the fitting of spark plugs at the location of the diesel injectors. The fuel injection system was replaced with an electromechanical ignition system. LPG components were installed. The LPG was stored in a cylindrical tank at a maximum pressure of 30 bars. The gas regulator system used was a two-stage diaphragm type regulator, filter, gas solenoid valve and safety valve which was duly calibrated for a supply pressure of 2 bars above the pressure of the intake manifold. Electronic switch regulator was used to monitor the current fuel (LPG or gasoline). The experimental setup is shown schematically in Fig. 10.



1- LPG tank, 2- Gasoline tank, 3- IVECO Engine, 4- Hydraulic dynamometer, 5- Rigid coupling, 6- Engine control system, 7- PC controller, 8- Intake manifold, 9- Muffler, 10- Pressure-gauge, 11- Regulator, 12- LPG vaporizer, 13- Gasoline pump, 14- Fuel flow meter, 15- *Air-gas mixer*, 16- Air flow meter, 17- Carburetor, 18- Air filter, 19- thermocouple, 20- Optical tachometer, 21-Vibration dampers

Fig.10. Engine test bench

5.2. Experimental setup and procedure

To determine brake power "BP" and brake torque "BT", the engine was coupled with a hydraulic dynamometer brake type "H3 BIS" of maximum power 400 kW. It was aligned and leveled to avoid vibrations in the transmission system and measurement errors. The quantity of consumed fuel is measured using a mass flow meter for purpose of Carioles effect type "Kroohne Optimass MFC 050 F". Engine speed is measured by an optical tachometer type "Chauvinistic Arnoux CA 27".

Engine performances are measured on the test bench while acting on two parameters. The first parameter is engine speed starting from the position of the regulation process (accelerating) which operates on the injection pump or the carburetor and makes it possible to vary the quantity of fuel introduced into the cylinders. The second is the load exerted by the hydraulic brake on the engine while acting on the quantity of water introduced into the brake. The engine was started using gasoline fuel and allowed to warm up. After 5 min, LPG liquid has evaporated by the vaporizer, and through the gasoline-gas switch, the engine started to operate with LPG. Soon as the engine reaches a stable operation (after approximately 30 min of operation), it is charged gradually while ensuring a constant flow of feed water of the brake for the same engine speed. The BP value is raised by a direct reading as well as the mass throughput of the fuel. Experiments were conducted with two modes:

- Tests without air-gas mixer. In this mode, LPG pipes are connected directly to the carburetor,
- Tests with MH mixer. Multiple holes mixer was installed and tightens at the right position (above the carburetor) and it is connected with gas pipes.

Operations are set up at compression ratio equal to 12:1. Engine was operated at different speeds (680 - 1800 rpm).

5.3. Experiments instruments and error analysis

For all the results, three runs of tests were performed under identical conditions to check the data repeatability, which appeared within an uncertainty of 4%. The error in the measurement of the engine brake power was ± 0.3 %. Hence, the accuracy of the hydraulic dynamometer brake was ± 0.5 %. The error in the measured value of the instantaneous speed of the engine could be estimated by considering the error with the tachometer, which is ± 0.015 %. Thus, the accuracy in the values of rpm is estimated to be ± 0.05 %. In mass flow measurements, the relative error is ± 0.1 % and with accuracy equal to ± 0.5 % of the mass flow meter. Errors in experiments can arise from instrument conditions, calibration, environment, observation, reading and test planning. To reduce the measurement errors, these experiments are repeated three times for all the results. Finally, engine characteristics are determined by taking the average values. All instrumentation was calibrated prior to engine testing. The characteristics of engine performance with various rates were calculated as follows [28]:

$$BP = 2\, n\, F\, .10^{-3},\ BT = \frac{30\, BP}{\pi\, n},\ BSFC = \frac{C}{BP} = \frac{3{,}6.10^{6}\, \dot{m}_f}{BP},\ BTE = \frac{BP}{\dot{m}_f\, Q_{fuel}},\qquad (14)$$

F is a characteristic of the hydraulic dynamometer read directly on the brake graduated disc.

5.4. Results and discussion

Figs. 11-14 show the effect air-gas mixing device impact on the engine performance characteristics.

5.4.1. *Brake power and torque*

It is clearly plotted on Fig.11 that engine fueled by LPG with multiple holes mixer has the highest power rise compared to LPG without mixer (almost 6.25% in the average). This outcome is explained by two processes. 1- The turbulent flow, generated by using the mixer gas pre-chamber increased the homogenous mixture that affected the better combustion performance. 2- The control of fuel flow promotes mixing fuel and closer to its stoichiometric reports. Consequently, rich mixture problems are removed.

The brake torque variation is shown in Fig.12. It is observed that the measured value from the dynamometer shows a higher torque for the engine operation with MH mixer. It gives rise to 3 % increase, in average, compared to without mixer operation. The dominance is more readable in high speeds. It has noted that the mixer addition for the gaseous fueling shows a slight increase in torque in the interval 1400 - 1600 rpm equal to 4% in the mean.

BP and BT are essentially dependent on in-cylinder mixture quantity and quality. Consequently, volumetric efficiency has a significant role to improve engine operation. According to the BP and BT curves, it is clear that the use of this mixer has advantages on the engine performances.

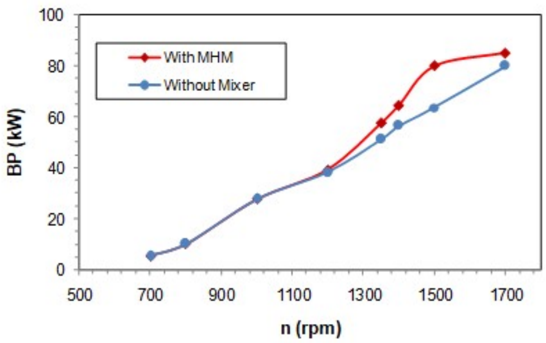


Fig.11. Variation of brake power with engine speed

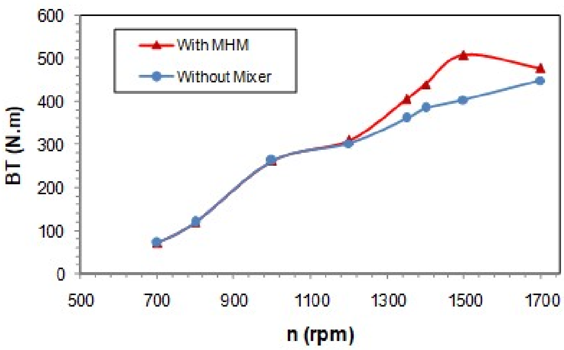


Fig.12. Variation of brake torque with engine speed

5.4.2. Brake Specific fuel consumption

Specific fuel consumption is the measure of fuel flow rate per unit power outlet and relates to the fuel efficiency of an engine. It is inversely proportional to efficiency of the engine as lower values of specific fuel consumption are favorable for higher performance. Fig. 13 presents the brake specific fuel consumption (BSFC) characteristics with the engine speeds. The BSFC variation decreases gradually when the engine speed is increased. At lower engine speeds, the high frictional forces acting on the piston and cylinder walls results in a greater fuel consumption rate since the heat of combustion is lost to friction. It is clear from this figure that fuel consumption per engine power outlet was the lowest for the engine speed range of 680–1100 rpm by using the multiple holes mixer in LPG engine operation. The presence of this device forms a homogeneous nature of the combustible mixture and gas optimal consumption. The resulted combustion gives rise to a significant torque on the crankshaft with a minimum of consumed propane. Therefore the higher combustion efficiency compensates the lower gas filling efficiency caused by the increased pressure drop of the mixer. Furthermore, the turbulent aerodynamic in-cylinder flow structures are increasingly favorable, which improves necessarily the combustion process. In a mean value, BSFC is increased by 6.6 %.

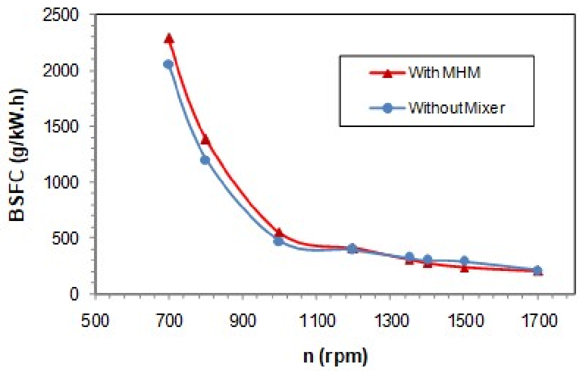


Fig.13. Variation of brake specific fuel consumption with engine speed

5.4.3. Brake thermal efficiency

Engine thermal efficiency is crucial on evaluating engine economic and overall performance. The brake thermal efficiency "BTE" in the three engine modes of operation is reported in Fig. 14. It can be seen from this figure that brake thermal efficiency obviously increases with the increase of engine speed. It can be seen that the mixer addition causes a BTE improvement. The improvement is almost equal to 9 % in the average. For the without mixer engine operation, it ranges from 1.5 % to 23.6 % whereas in the case of with mixer operation it varies from 1.64 % to 25.1 %. Because of the stoichiometric ratio uncertain values (incompatibility between number of gas molecules and that of oxygen), the chemical process of combustion is incomplete. The absence of mixer causes a discrepancy between the gas and air flow, which increases the availability of these errors.

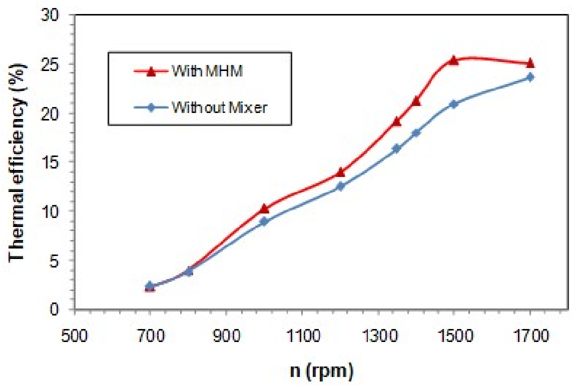


Fig.14. Variation of brake thermal efficiency with engine speed

6. Conclusion

The investigation and the analysis of the air-gas intake flow characteristics, during intake stroke, are numerically carried out using a CFD code. The effectiveness of a liquefied petroleum gas mixer is determined in order to provide a good air fuel mixing quality. Through the numerical approach, two intake system configurations are compared. Simulation results showed that the air-fuel mixing system plays an important role on the gaseous fueling engine performance. In addition, it is noted that the use of multiple holes mixer leads to an optimal homogenization compared intake system without mixer.

Experiments were performed on an urban bus diesel engine converted into bi-fuel LPG fuelled engine to study the influence of fueling type and mixer on the engine performance. From experimental tests it is noted that the gaseous fueling engine shows an average of 6.25 %, 3 %, 9 % higher BP, BT and BTE respectively using MH mixer within intake system. The consumption is reduced by 6.6 %.

The numerical and experimental results affirm each other well on the optimal design efficiency of the selected gas mixer that yields the better cylinder filling.

Nomenclature

C ($kg.min^{-1}$)	Fuel consummation
C_p ($J\ kg^{-1}\ K^{-1}$)	Specific heat at constant pressure
C_v ($J\ kg^{-1}\ K^{-1}$)	Specific heat at constant volume
e (J)	Specific internal energy
f_μ	Turbulent viscosity factor
k ($J.kg^{-1}$)	Turbulence kinetic energy
m($kg.s^{-1}$)	Mass flow rate of fuel
n (rpm)	Engine speed
P_B (s^{-2})	Turbulent generation due to buoyancy forces
p (Pa)	Pressure
Q (kJ/kg)	Lower heating value
Q_H (J)	Heat source or sink per unit volume
q_i (W/m^{-2})	Diffusive heat flux
S_i	Mass-distributed external force per unit mass
R ($J\ K^{-1}\ kg^{-1}$)	Ideal gas constant
T (°K)	Temperature
u ($m.s^{-1}$)	Fluid velocity
ρ (kg/m^{-3})	Density
Greek symbols	
ε ($W.kg^{-1}$)	Turbulence dissipation rate
μ_t (Pa.s)	Turbulent viscosity
τ_{ij} ($kg\ m^{-1}\ s^{-2}$)	Viscous shear stress tensor
Abbreviations	
BP (kW)	Break power
BT (N.m)	Break torque
BSFC (g/kW.h)	Break specific fuel consummation
BTE (%)	Brake thermal efficiency
CNG	Compressed natural gas
CFD	Computational fluid dynamics
CAD	Computer associated design
LPG	Liquefied petroleum gas

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References

1. Priyank Gupta, Caneon Kurien, Mayank Mittal. (2023) Biogas (a promising bioenergy source): A critical review on the potential of biogas as a sustainable energy source for gaseous fuelled spark ignition engines. International Journal of Hydrogen Energy 48: 7747-7769, https://doi.org/10.1016/j.ijhydene.2022.11.195
2. Hüsamettin Alperen Alabaş, Bilge Albayrak Çeper (2023) Effect of oxygen enrichment on the combustion characteristic and pollutant emissions of kerosene-biogas mixtures on a mini jet engine combustion chamber. Journal of the Energy Institute 111: 101420, https://doi.org/10.1016/j.joei.2023.101420

3. Qian Y, Sun S, Ju D, Shan X, Lu X (2017) Review of the state-of-the-art of biogas combustion mechanisms and applications in internal combustion engines. Renewable and Sustainable Energy Reviews 69 : 50-58. doi.org/10.1016/j.rser.2016.11.059
4. Erkus B, Karamangil MI, Sürmen A (2015) Enhancing the heavy load performance of a gasoline engine converted for LPG use by modifying the ignition timings. Applied Thermal Engineering 85: 188-194. doi.org/10.1016/j.applthermaleng.2015.03.076
5. Nitnaware PT, Suryawanshi JG (2016) Effects of equal spark timing on performance emission and combustion characteristics of SI engine using hydrogen and CNG blends. J Braz. Soc. Mech. Sci. Eng. (2016) 38:2245-2253. Doi: 10.1007/s40430-016-0508-8
6. Tzeng G H, Lin C W, Opricovic S (2005) Multi-criteria analysis of alternative-fuel buses for public transportation. Energy Policy 33: 1373-1383. doi.org/10.1016/j.enpol.2003.12.014
7. Gramm P et al. (1999) Mass Transit - Use of Alternative Fuels in Transit Buses. United States General Accounting Office
8. Ozcan H, Yamin J AA (2008) Performance and emission characteristics of LPG powered four stroke SI engine under variable stroke length and compression ratio. Energy Conversion and Management 49: 1193-1201. doi.org/10.1016/j.enconman.2007.09.004
9. Gumus M (2011) Effects of volumetric efficiency on the performance and emissions characteristics of a dual fueled (gasoline and LPG) spark ignition engine. Fuel Processing Technology 92: 1862–1867. doi.org/10.1016/j.fuproc.2011.05.001
10. Murillo S, Miguez JL, Porteiro J, Gonzalez LML, Granada E, Moran JC (2005) LPG: Pollutant emission and performance enhancement for spark-ignition four strokes outboard engines. Applied Thermal Engineering 25: 1882-1893. doi.org/10.1016/j.applthermaleng.2004.12.002
11. Bayraktar H, Durgun O (2005) Investigating the effects of LPG on spark ignition engine combustion and performance. Energy Conversion and Management 46: 2317-2333. doi.org/10.1016/j.enconman.2004.09.012
12. Saleh HE (2008) Effect of variation in LPG composition on emissions and performance in a dual fuel diesel engine. Fuel 87: 3031-3039. doi.org/10.1016/j.fuel.2008.04.007
13. Choi GH, Bae SC, Han SB and Chung YJ (2004) A study on the characteristics of combustion with butane and propane in a retrofitted diesel engine. Proc. IMechE, Part D, Journal of automobile engineering 218: 915-920. doi: 10.1243/0954407041581048
14. Beer T, Grant T, Williams D, Watson H (2002) Fuel-cycle greenhouse gas emissions from alternative fuels in Australian heavy vehicles. Atmospheric Environment 36: 753-763. doi.org/10.1016/S1352-2310(01)00514-3
15. Ou X, Zhang X, Chang S (2010) Alternative fuel buses currently in use in China: Life-cycle fossil energy use, GHG emissions and policy recommendations. Energy Policy 38: 406-418. doi.org/10.1016/j.enpol.2009.09.031
16. Psaraftis HN (2016) Green Transportation Logistics. The Quest for Win-Win Solutions. International Series in Operations Research & Management Science. Volume 226, ISSN 0884-8289, Springer.
17. Mardani AS, Rosli A B and Leong S K (2003) CNG Engine Performance Improvement Strategy Through Advanced Intake System. Society of Automotive Engineers. SAE paper 2003-01-1937. Doi: 10.4271/2003-01-1937
18. Ji R, Lin K, Li C and Liu X (2011) Research and Design of Small-Engine LPG Motorcycle Mixer. Advanced Materials Research154-155: 282-286. Doi: 10.4028/www.scientific.net/AMR.154-155.282
19. Furuyama M and Yan X (1998) Mixing Flow Phenomena of Natural Gas and Air in the Mixer of a CNG Vehicle. SAE paper 981391. Doi: 10.4271/981391
20. Xu BY and Furuyama M (1997) Visualization of natural gas-air mixing flow in the mixer of a CNG vehicle, JSAE Review 18: 57-60. Doi:10.1016/S0389-4304(96)00053-7
21. Xu BY, Sun DZ, Qi YL, Zheng YW, Tian HY, Cai SL (2011) Study on Mixture Formation of Liquid LPG for a Center Injection DISI Engine. Advanced Materials Research 201-203: 622-626. Doi: 10.4028/www.scientific.net/AMR.201-203.622

22. Kilpinen P (2010) Optimization of a simplified sub-model for NO emission prediction by CFD in large 4-stroke marine diesel engines. Fuel Processing Technology 91: 218-228. doi.org/10.1016/j.fuproc.2009.10.001
23. Shuai W, Guodong L, Huilin L, Juhui C, Yurong H, Jiaxing W (2011) Fluid dynamic simulation in a chemical looping combustion with two interconnected fluidized beds. Fuel Processing Technology 92 385-393. doi.org/10.1016/j.fuproc.2010.09.032
24. Tavakoli S, Ganji DD, Gorji M, Rasekh, A, Naeejee S (2016) Different camshaft profile analyses for natural gas engine performance and emission. J Braz. Soc. Mech. Sci. Eng. 38: 355-364. Doi: 10.1007/s40430-015-0317-5
25. Palumbo MF (2007) CFD-Analysis of intake-system performances of a small turbocharged spark-ignition engine. The Society of Automotive Engineers SAE, 24-0047. Doi: 10.4271/2007-24-0047
26. Torregrosa AJ, Olmeda P, Gil A and Cornejo O (2015) Computational fluid dynamics assessment of subcooled flow boiling in internal-combustion engine-like conditions at low flow velocities with a volume-of-fluid model and a two-fluid mode. Proc IMechE Part D: J Automobile Engineering 229 (13): 1830-1839. doi: 10.1177/0954407015571674
27. Syed K and Gaddale RPA (2010) Conversion of diesel engine into spark ignition engine to work with CNG and LPG fuels for meeting new emission norms. Thermal science. 14(4): 913-922. Doi: 10.2298/TSCI1004913K
28. Heywood JB (1988) Internal combustion engine fundamentals. New York: McGraw-Hill
29. Sobachkin A, Dumnov G (2014) Numerical Basis of CAD-Embedded CFD. SolidWorks, Dassault Systemes. February.
30. Amsden AA, O'Rourke PJ, Butler TD (1989) KIVA-II- A Computer program for chemically reactive flows with sprays. Los Alamos National Labs, United States of America, LA-11560-MS
31. Arcoumanis C, Kamimoto T (2009) Flow and Combustion in Reciprocating Engines, Springer-Verlag Berlin Heidelberg
32. SolidWorks Flow Simulation (2014). Technical Reference
33. Poroseva SV, Bezard H (2001) On ability of standard k-ε model to simulate aerodynamic turbulent flows. CFD Journal. Special Number 2001: 627 – 633
34. Han Q (2013) A Preliminary Study on Intake Flow to Improve In-Cylinder Air Motion. Thesis, University of Windsor, Canada, April
35. Guibet JC (1997) Fuels and engines: technology, energy, environment, Volume 2. Editions Technip, Paris