Investigation of Effects of Natural Gas Composition on One-Dimensional Comprehensive Engine Model Calibration

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1 Introduction

Natural gas as a potential clean alternative fuel is playing a growing role in stationary and transportation industries. Large reservoirs have made this fuel to be even more promising. On the other hand, employment of natural gas comes with an inevitable shortcoming of considerable deviation in composition of gases extracted from different resources, which causes change in these gases' combustion features.

Concerns over the energy crisis and rising environmental issues have made combustion system designers to face challenges in the design of modern systems. The development of numerous prototypes is not feasible or even possible anymore. Therefore, computer simulation can be an efficient tool. In spite of this, three-dimensional high-resolution modelling strategies are very time-consuming and won't be suitable tools for providing the rapid responses obliged by market demands or regulatory agenda. Reduced-order modelling strategies would be a

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favourable option in analysis and optimization of combustion systems, i.e. internal combustion engines. An example of these simulation strategies is comprehensive one-dimensional engine cycle modelling, which since its development in the late 1970s has undergone many developments and plays a pivotal role in the design and optimization processes of internal combustion engines. The main assumption in this modelling strategy is that the engine pipes and runners' length to diameter ratios are large enough, so that the flow could be considered to be one dimensional. It is also assumed that the flow rate is high enough to neglect the influences of viscosity (Benson 1982; Ramos 1989). It is therefore convenient to employ inviscid one-dimensional Euler system of equations known as gas dynamic system of equations (White 2003) to simulate the flow field inside the engine pipes. Other elements such as cylinder, air cleaner, plenums, junctions, injectors, catalytic converters, throttle, etc. are simulated using the thermodynamic zero-dimensional models. Meanwhile, another methodology for in-cylinder flow field simulation with more accuracy follows the general lines of multi-zone modelling technique, which considers the cylinder to be divided to different thermodynamic zero-dimensional zones, i.e. burned and unburned. The results of flow simulation in different elements are implemented as boundary conditions to gas dynamic system of equations. On the other hand, this approach suffers a great drawback. It has a very low accuracy where the three-dimensional effects take part in flow features, homentropic assumption does not apply or the flow bending radius is very small. Similar to all other reduced-order simulation procedures, this technique also involves employment of empirical correction factors which should be calibrated against experimental observations. Examples of these factors can be flow or discharge coefficients at the connection keypoints of elements and runners, in which the gas dynamic system of equations is solved. Besides these coefficients, several factors are required and should be calibrated for adjustment of results obtained from simulation of flow inside other elements. Examples of these coefficients would be factors considered in combustion thermodynamic function, i.e. Wiebe function (Ghojel 2010), or in-cylinder wall heat transfer coefficients and temperatures.

In this framework, the position of calibration process is of crucial necessity when employing reduced-order modelling techniques. A one-dimensional model may end in several results, one of which correctly predicts the engine intake and exhaust masses and reconstructs the in-cylinder pressure and heat release profile with high accuracy (Winterbone and Pearson 2000; Caton 2016). When the engine cycle 1D comprehensive model is fully calibrated, it could be employed for predicting of engine performance in several conditions (Benson 1982; Ramos 1989; Winterbone and Pearson 2000; Caton 2016), whereas in case of charge in many other performance conditions, such as valve lift timing and profile, it is mandatory to recalibrate the model with experimental observations and/or high-fidelity multidimensional simulation results.

It should be noted that by change in the fuel of an engine, the calibration of the thermodynamic cycle won't be reliable anymore. This fact follows the high dependency of burn rate function, Wiebe function as an example, on fuel bond types and energies, transport factors, etc. (Ghojel 2010). In addition, the change in the density of fuel and air mixture may influence the flow coefficients at the boundaries and may therefore harm the model correctness (Medina et al. 2014).

The natural gas consists of several small normal or branched aliphatic C_1 to C_5 hydrocarbons along with inert species, mainly nitrogen (N_2) and carbon dioxide (CO_2) molecules. On the other hand, its composition differs among various wells. Additionally, it may also change based on the petrochemical process. In most of the simulation tools, pure methane is considered as the main surrogate candidate for natural gas, whose composition may differ and alter significantly from one area to another, and this matter has caused concerns about the correctness of reduced models of CNG-fuelled combustion systems.

Considering the above-mentioned facts, and bearing in mind that natural gas shares about 70% of the Iran total energy consumption basket, along with the fact that approximately 3 million vehicles consume compressed natural gas (CNG) in Iran, the most in the world, this research aims to investigate the effects of deviations in composition of natural gas within Iran's range on the calibration of engine one-dimensional cycle model. To accomplish this, a gas-fuelled spark ignition engine is simulated using AVL BOOST v2013 software. The model results are calibrated for three different engine loads at three different compression ratios versus experimental observations for engine working with pure methane as fuel. The goal functions during the calibration process are air and fuel mass flow rates and in-cylinder open cycle pressure profile. After finalizing the model calibration, the models are used for predicting the performance of the same engine without change in the model-calibrated parameters when working with natural gases distributed in Mashhad and Tehran. The simulation results are compared with experimental observations on the same engine working with Mashhad and Tehran natural gases, which the latter is proved to have the maximum deviation from pure methane in composition. The findings indicate that the results of the model remain valid for air flow rate, fuel consumption and in-cylinder open cycle pressure profile, in spite of change in the composition of the fuel. This proves that the range of deviation in composition of natural gases distributed in Iran does not affect the calibration correctness of the engine comprehensive thermodynamic cycle model.

2 Studied Engine

The test engine employed in this study is a spark ignition single-cylinder natural gas-fuelled engine. The engine is developed by AVL GmbH and is currently installed in Vehicle, Fuel and Environment Research Institute, University of Tehran. The engine compression ratio can be varied between 6 and 16, by altering the shims between the cylinder head and engine block. General specifications of the engine are shown in Table 1. In addition, the test bed is also illustrated in Fig. 1. More information on the engine and test bed specifications is brought by Javaheri et al. (2014).

Table 1 Main engine information (Javaheri et al. 2014)

Engine type	4 Stroke
Cylinder	1
Valve	4
Bore	86 mm
Course	86 mm
Connecting rod	143 mm
Compression ratio	6–16
Injection	Intake port injection

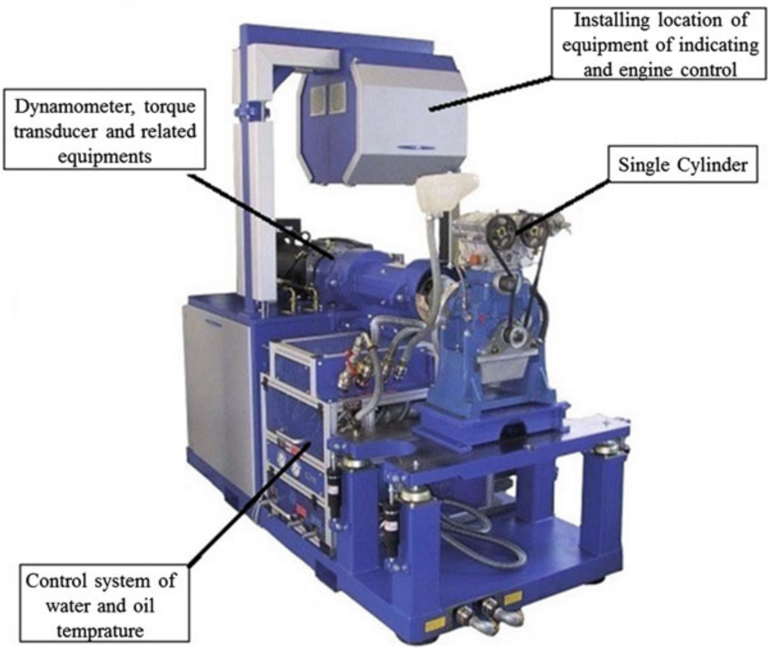

Fig. 1 Illustration of the test bed (Javaheri et al. 2014)

The experiments are carried out at three different compression ratios of 12, 14 and 16. Three different engine loads are also considered. For part-load conditions, the break mean effective pressure is set to be equal to 2 and 4 bar. The full load conditions are also investigated.

3 Simulation and Calibration Approach

In this research, it is aimed to create a baseline model with high accuracy based on experimental observations for a single-cylinder spark ignition natural gas-fuelled test engine at three different compression ratios, i.e. 12, 14 and 16. In all of the tests,

Table 2 Studied fuel compositions (mass fraction based)

	Fuel 1	Fuel 2	Fuel 3
Methane	100	94.778	77.696
Ethane	0	1.221	6.621
Propane	0	0.694	6.498
Carbone dioxide	0	2.052	2.689
Nitrogen	0	1.255	6.496

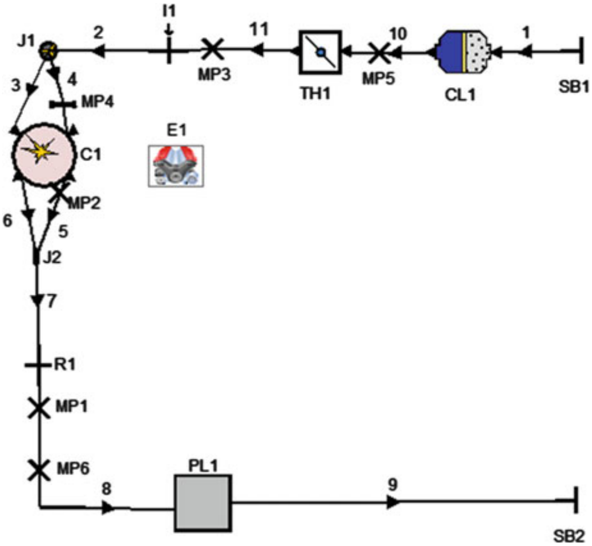

Fig. 2 One-dimensional comprehensive model developed in AVL BOOST v2013

the engine speed is held constant at 2000 rpm. All of the tests are repeated for three different throttle angles:

1. Wide open throttle (WOT) state, i.e. full load conditions
2. Partially open throttle state, so that the brake mean effective pressure equals to 4 bar
3. Partially open throttle state, so that the brake mean effective pressure equals to 2 bar

As described earlier, the tests are repeated for three different natural gas compositions. These compositions are shown in Table 2.

Fuel 1 is pure methane, fuel 2 is the natural gas distributed in Mashhad and fuel 3 represents the natural gas typically used in Tehran.

The engine one-dimensional comprehensive model is developed using AVL BOOST v2013 software, schematic of which is shown in Fig. 2. The general strategy of the modelling and calibration methodology will be described here briefly.

An internal combustion engine could be considered as a system of pipes connecting different elements. The important point which should be considered here is that each pipe's length to diameter ratio is large enough, so that the flow could be considered one dimensional (Benson 1982; Winterbone and Pearson 2000). On the other hand, with the relatively high flow speed, it would be convenient to employ the inviscid assumption. Hence, the one-dimensional Euler system of inviscid equations could be used for simulating the fluid flow characteristics inside the pipes. After creating the overall model topology, geometrical parameters, such as pipe lengths, elements, volumes, bore, stroke, connecting rod length, compression ratio, etc., are set. The simulation process could be started after setting up the operating parameters and required factors for numerical simulation process. The calibration will be started after the model setup is complete.

The air flow rate at each node of the pipe is calculated under the light of solving system of one-dimensional gas dynamics equations using fourth-order MacCormack (Tannehill et al. 1997) scheme. All the boundary conditions related to different elements are implemented using the non-homentropic approach (Benson 1982).

The fuel flow rate is found by setting the equivalence ratio in the injector element based on the results of gas dynamic system of equations. It is worth mentioning that the same applies for experimental results. During the test procedure, only the air flow rate is observed directly as raw data from the test cell and the fuel flow rate is calculated using the injector control instruments based on the input air-fuel ratio and is then reported to the user.

Combustion thermodynamic simulation is carried out using Wiebe two-zone algorithm, following the general lines of multi-zone modelling strategy (Heywood 1988; Medina et al. 2014). To simulate the heat transfer phenomenon, the AVL 2000 model is used (Schwarz 2012).

The calibration process is started for motoring conditions and the main focus is on the in-cylinder pressure profile. To finalize the calibration process at this step, various model parameters and coefficients are set based on trial and error strategy. These parameters include 24 flow coefficients, 7 heat transfer coefficients and temperatures for cylinder components, i.e. piston, cylinder head, liner at both conditions of piston at bottom dead centre and top dead centre. In addition, due to non-homentropic treatment of boundary conditions, and the necessity for correct positioning of interaction point of the characteristic curve with energy ellipse, the temperatures and heat transfer coefficients at the intake and exhaust ports are also set. After finalizing the calibration process for motoring condition, fuel injection and combustion-related parameters are added to begin the second step of calibration process. Therefore, four factors are added to calibrate the two-zone Wiebe function. It should be considered that, after adding fuel injection and combustion to the model developed initially for motoring conditions, all the parameters including flow coefficients, heat transfer coefficients and temperatures should be readjusted.

As it was mentioned earlier, one of the main goal functions during the calibration process is the minimum error between the experimentally achieved in-cylinder pressure profile and the numerical simulation results. To this end, the trial and error procedure was designed so that minimum error exists during combustion, expansion, exhaust and intake periods of the cycle. In addition, during the

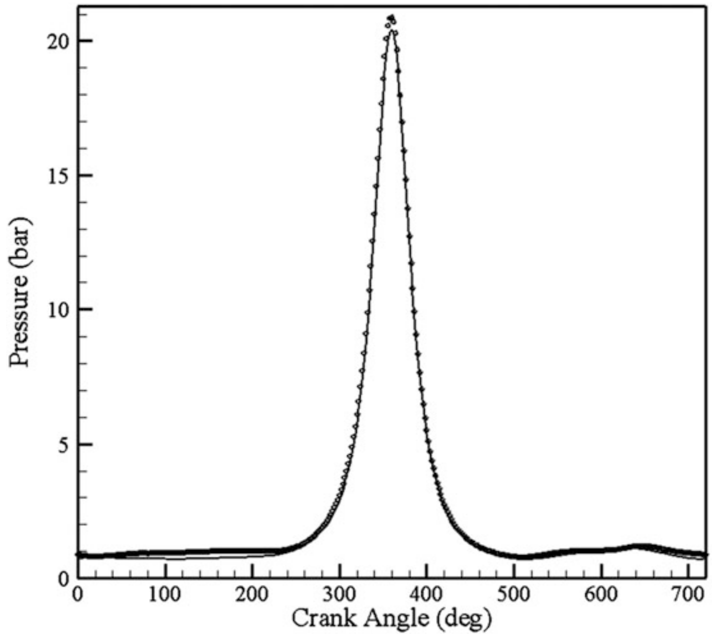

Fig. 3 In-cylinder pressure profile for motoring condition at CR = 12

compression stroke, the numerically and experimentally obtained pressure profile should be completely identical, so that one can assure the agreement between the in-cylinder-trapped mass obtained from experiment and numerical simulation. The calibrated results for motoring condition are shown in Figs. 3, 4 and 5 for the compression ratios 12, 14 and 16, respectively.

From Fig. 3, it is evident that the root mean square of errors between experimental and numerical in-cylinder profile during the compression stroke does not exceed 0.8%. On the other hand, the overall root mean square of experiment and simulation results on in-cylinder pressure profile remains less than 1.5%. For compression ratio of 14, as depicted in Fig. 4, the situation is the same. The root mean square of errors during the compression stroke remains less than 0.65%, while the overall error does not exceed 1.2%. For compression ratio of 16, as illustrated in Fig. 5, the root mean square of errors during the compression stroke remains less than 0.9%, while the overall error does not exceed 1.5%.

The air and fuel mass flow rates obtained from experimental data and numerical simulation results for engine working with stoichiometric mixture of air and pure methane (fuel 1) at constant engine speed of 2000 rpm in three different compression ratios of 12, 14 and 16 are shown in Tables 3, 4 and 5 for engine working at part-load condition of brake mean effective pressure (BMEP) of 2 and 4 bar and full load conditions, respectively. From the tables, it is obvious that the errors always stay below 1%, which proves acceptable consistency of the numerical simulation results and experimental findings.

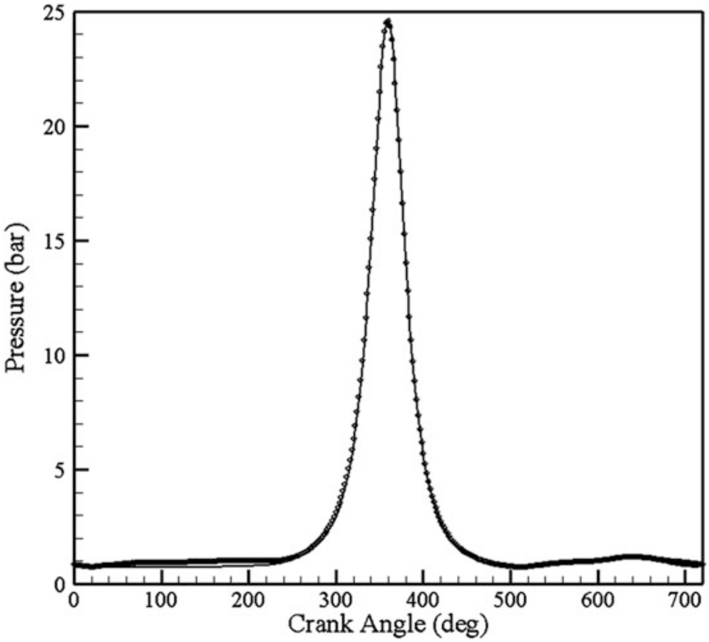

Fig. 4 In-cylinder pressure profile for motoring condition at CR = 14

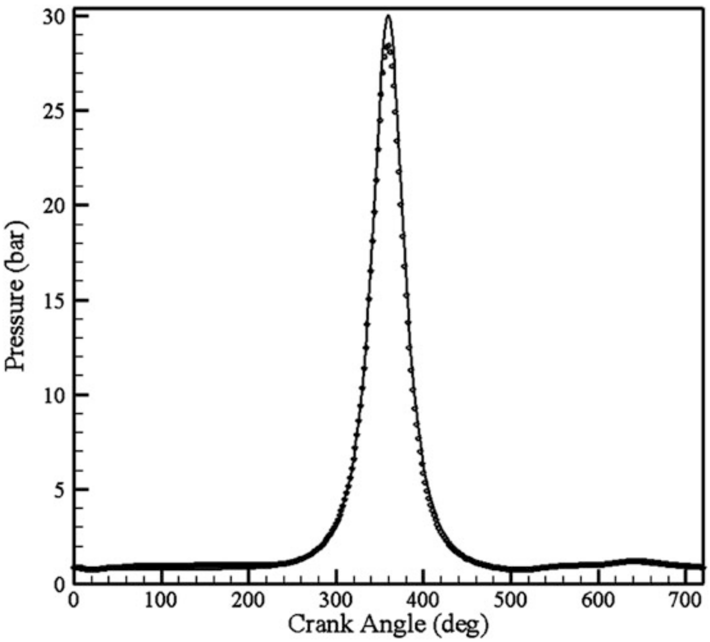

Fig. 5 In-cylinder pressure profile for motoring condition at CR = 16

Table 3 Experimental and simulation results of air and fuel mass flow rates for fuel 1 in BMEP = 2 bar

CR		Exp. (gr/s)	Sim. (gr/s)	Error (%)
12	Air	2.860	2.866	0.209
	Fuel	0.166	0.167	0.602
14	Air	2.764	2.779	0.542
	Fuel	0.161	0.162	0.621
16	Air	2.907	2.889	0.619
	Fuel	0.169	0.168	0.591

Table 4 Experimental and simulation results of air and fuel mass flow rates for fuel 1 in BMEP = 4 bar

CR		Exp. (gr/s)	Sim. (gr/s)	Error (%)
12	Air	4.004	4.034	0.749
	Fuel	0.233	0.235	0.858
14	Air	3.956	3.971	0.379
	Fuel	0.230	0.231	0.434
16	Air	4.051	4.018	0.814
	Fuel	0.236	0.234	0.847

Table 5 Experimental and simulation results of air and fuel mass flow rates for fuel 1 in full load

CR		Exp. (gr/s)	Sim. (gr/s)	Error (%)
12	Air	5.434	5.407	0.501
	Fuel	0.316	0.315	0.316
14	Air	5.196	5.252	1.082
	Fuel	0.302	0.305	0.993
16	Air	5.106	5.093	0.244
	Fuel	0.297	0.296	0.336

The in-cylinder pressure profile for three different engine loads and compression ratios of 12, 14 and 16 are depicted in Figs. 6, 7 and 8, respectively. In addition, the root mean squares of the errors between numerical and experimental results are shown in Table 6. These results indicate that in spite of negligible discrepancy of simulation results from experimental findings, the model calibration for engine working on pure methane (fuel 1) is promising.

During the calibration, the following results are also obtained for tuning the temperatures of different combustion chamber parts at different engine loads and compression ratios.

As the compression ratio rises, the gas temperature at the exhaust will decrease, due to higher work obtained during combustion period (Ferguson and Kirkpatrick 2000; Heywood 1988). Experimental data on exhaust gas temperatures at 5 cm after cylinder for pure methane is shown in Table 7. Therefore, the following results are obtained during the calibration of cylinder part temperatures:

1. All the temperatures are lowered about 2.5 °C when the compression ratio is reduced from 16 to 14 and from 14 to 12.

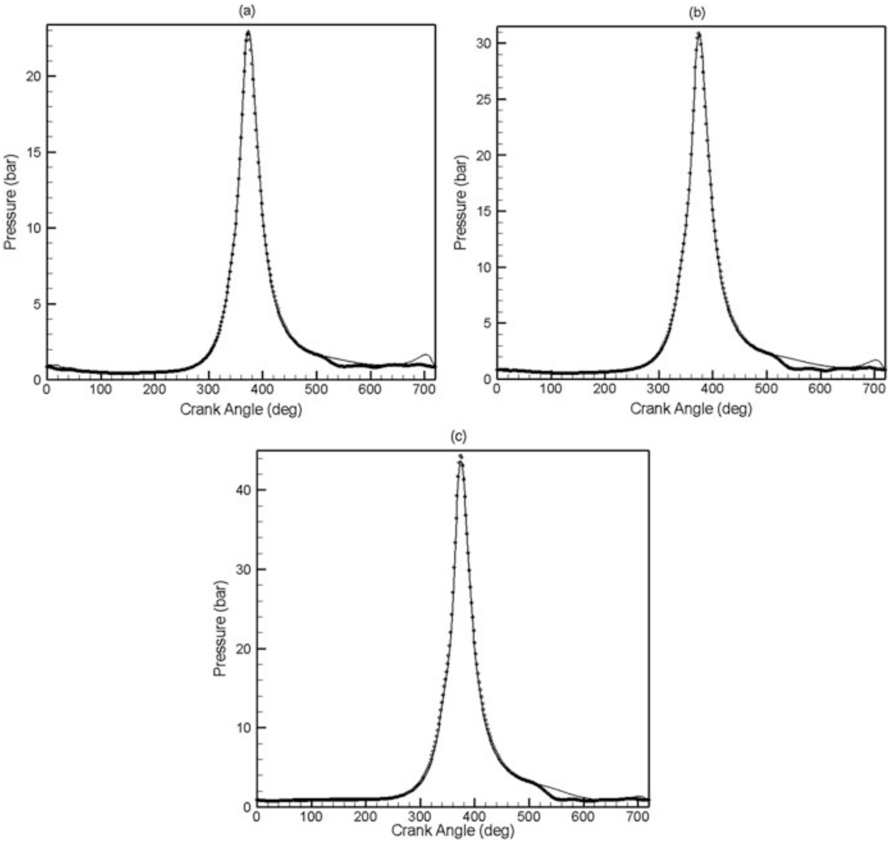

Fig. 6 In-cylinder pressure profile for fuel 1 at compression ratio of 12 for (**a**) BMEP = 2 bar, (**b**) BMEP = 4 bar and (**c**) full load conditions. Solid lines denote simulation and symbols denote experimental results

2. The engine part temperatures don't change when the engine load is lowered from full load to brake mean effective pressure of 4 bar.
3. With decrease in load at part-load condition from 4 to 2 bar, the engine part temperatures are reduced about 4 °C at all the compression ratios.

4 Results and Discussion

In the next step, the model parameters are held fixed, and the fuel composition is changed. Figure 9 illustrates the in-cylinder pressure profile obtained from simulation and experiment at three different engine compression ratios of 12, 14 and 16, respectively. The natural gas is gathered from Mashhad (fuel 2) and the same composition is considered for the simulation. As it is evident from the figures, the

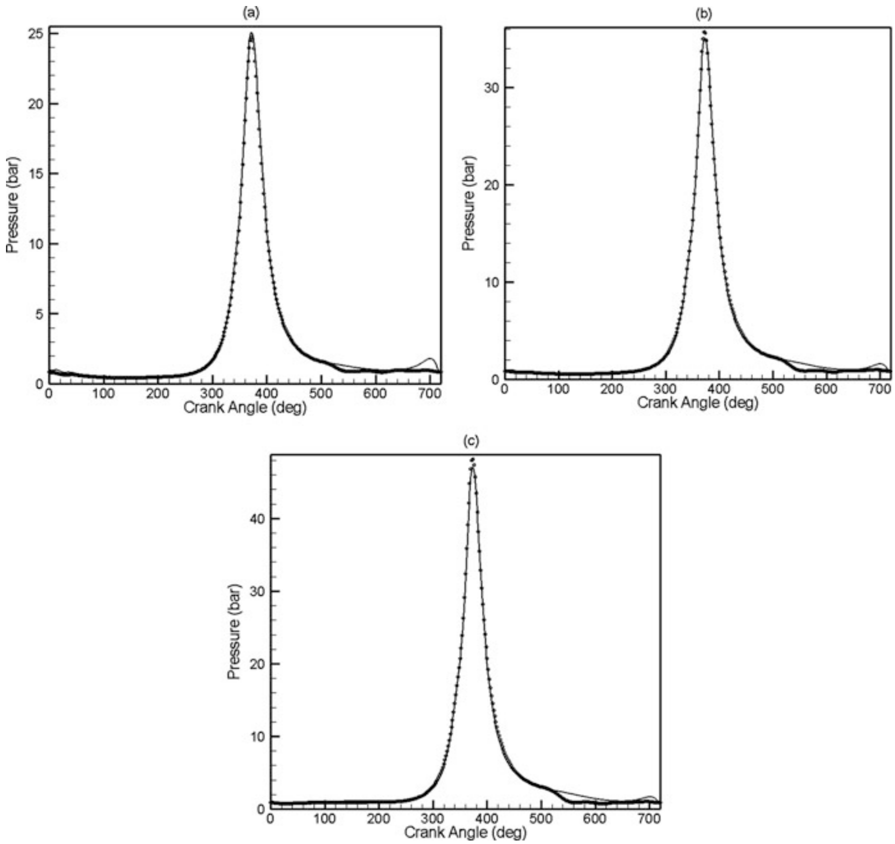

Fig. 7 In-cylinder pressure profile for fuel 1 at compression ratio of 14 for (**a**) BMEP = 2 bar, (**b**) BMEP = 4 bar and (**c**) full load conditions. Solid lines denote simulation and symbols denote experimental results

root mean squares of the errors don't exceed 1.1%, proving proper agreement of the simulation results and experimental data.

Once again, it is important to note that all of the experiments and simulations are carried out at a constant engine speed of 2000 rpm and stoichiometric mixture conditions at the injector. Meanwhile, the mass flow rates of air and fuel are shown in Table 8. The numerical and experimental results in the table are at the full load condition (throttle at wide open state).

It is evident from the table that the errors between numerical and experimental results for natural gas distributed in Mashhad (fuel 2) always stay below 4.5%. This small and negligible discrepancy indicates good agreement of the gas dynamic system of equation results with experimental data. These numerical results are calibrated using flow coefficients at the boundaries of each pipe, i.e. engine elements. Therefore, one can conclude that in addition to the thermodynamic combustion model (Fig. 9), the Euler equation calibration remains also valid.

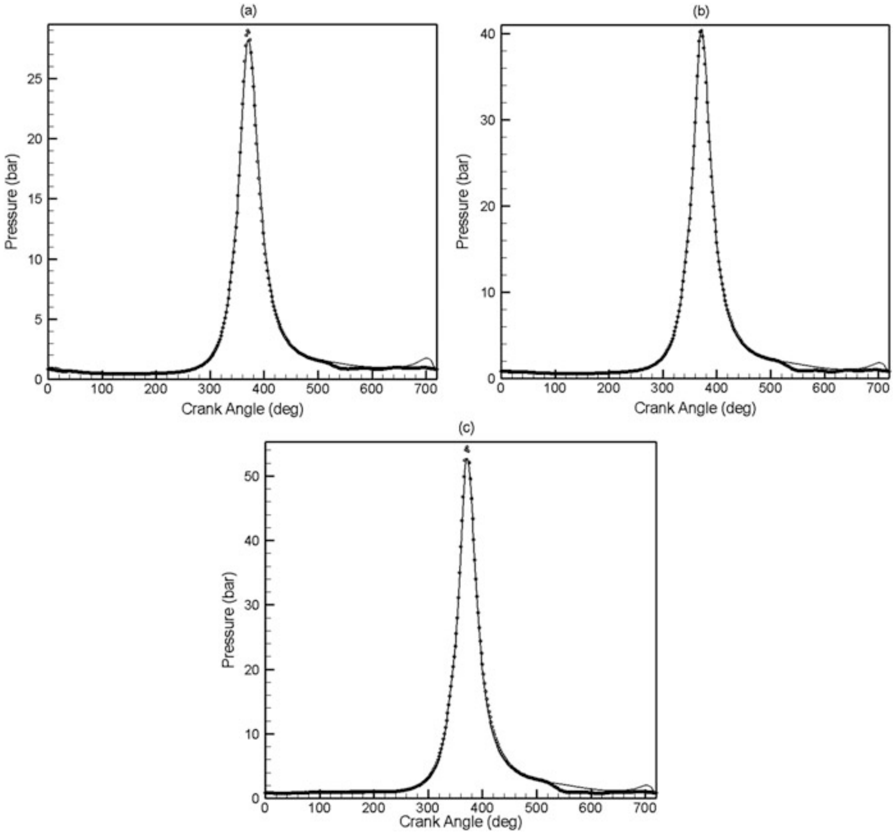

Fig. 8 In-cylinder pressure profile for fuel 1 at compression ratio of 16 for (**a**) BMEP = 2 bar, (**b**) BMEP = 4 bar and (**c**) full load conditions. Solid lines denote simulation and symbols denote experimental results

Table 6 Exhaust gas temperatures at different engine loads and compression ratios

	CR = 16	CR = 14	CR = 12
Full load	458	467	481
BMEP = 4 bar	457	467	479
BMEP = 2 bar	442	453	460

Table 7 Root mean squares of errors for numerical and experimental pressure profiles for fuel 1

	CR = 16 (%)	CR = 14 (%)	CR = 12 (%)
Full load	0.93	0.69	0.73
BMEP = 4 bar	0.64	0.77	0.82
BMEP = 2 bar	0.88	0.79	0.77

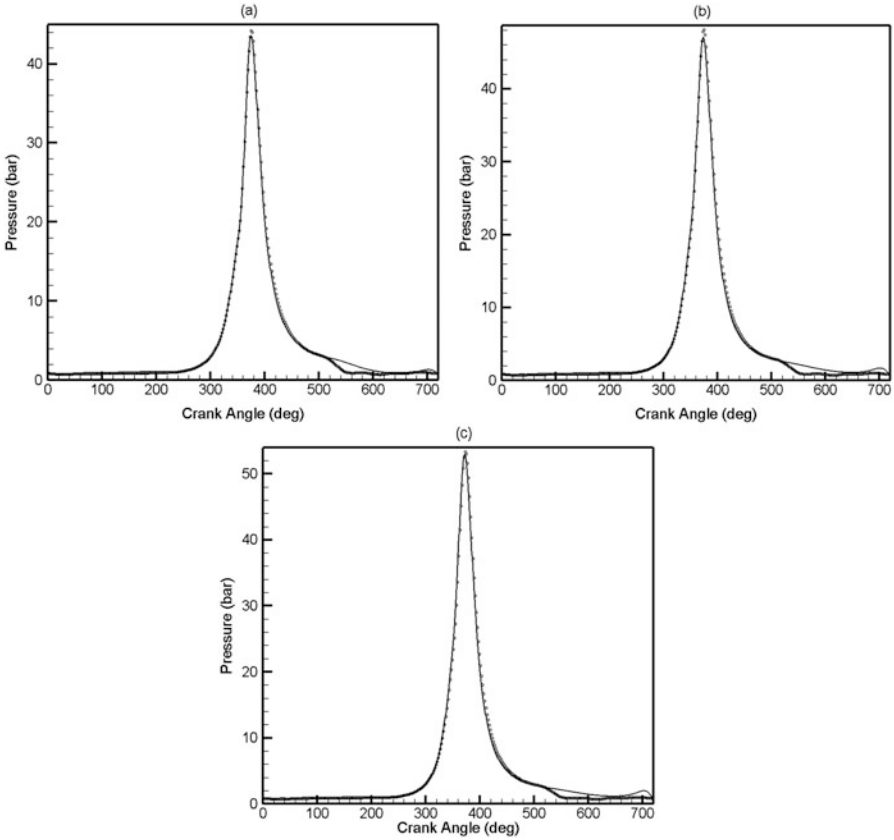

Fig. 9 In-cylinder pressure profile for fuel 2 at (**a**) CR = 12 bar, (**b**) CR = 14 bar and (**c**) CR = 16. Solid lines denote simulation and symbols denote experimental results

Table 8 Experimental and simulation results of air and fuel mass flow rates for fuel 2 in full load

CR		Exp. (gr/s)	Sim. (gr/s)	Error (%)
12	Air	5.527	5.534	0.132
	Fuel	0.331	0.332	0.591
14	Air	5.573	5.374	3.563
	Fuel	0.333	0.324	2.664
16	Air	5.480	5.234	4.485
	Fuel	0.328	0.317	3.228

The simulation and experiment procedures are then repeated for the natural gas distributed in Tehran. It was mentioned earlier that Tehran natural gas has the maximum deviation from pure methane in composition. Hence, if the model remains calibrated for this composition of natural gas, it will be valid for all types of natural gas found in Iran.

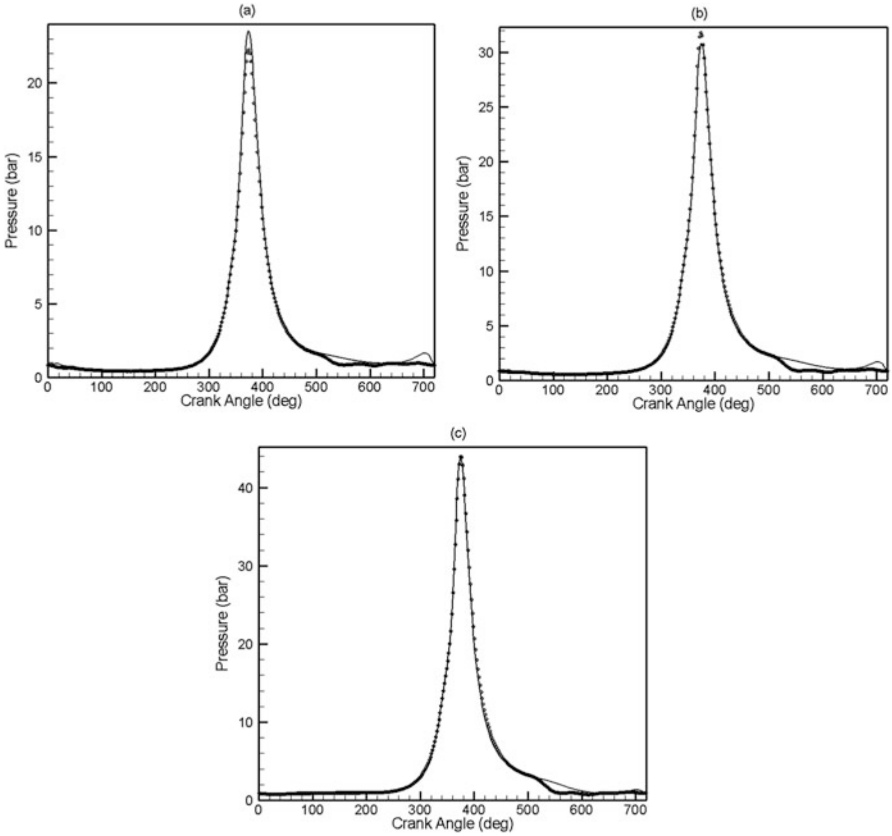

Fig. 10 In-cylinder pressure profile for fuel 3 at compression ratio of 12 for (**a**) BMEP = 2 bar, (**b**) BMEP = 4 bar and (**c**) full load conditions. Solid lines denote simulation and symbols denote experimental results

As for fuels 1 and 2, the verification of the model's calibrated results is investigated at two different steps.

First, it is investigated that the cylinder-related part temperatures and heat transfer coefficients are still correct or not. In addition, the calibration of the combustion zero-dimensional thermodynamic model, i.e. Wiebe function, is also controlled. To accomplish this step, the in-cylinder pressure profiles obtained from simulation and experiment should be compared. This is done in Figs. 10, 11 and 12. As can be seen in the figures, for all the case studies, the pressure curves are still in good agreement. The root mean squares of experimental and simulation data discrepancies are shown in Table 9. The root mean square of errors does not exceed 1.9%, which could be regarded as a very good agreement.

For the second step, the performance of the gas dynamic system of equations with calibrated flow coefficients is qualified. To this end, the experimental and simulation results for the mass flow rates of air and fuel are shown in

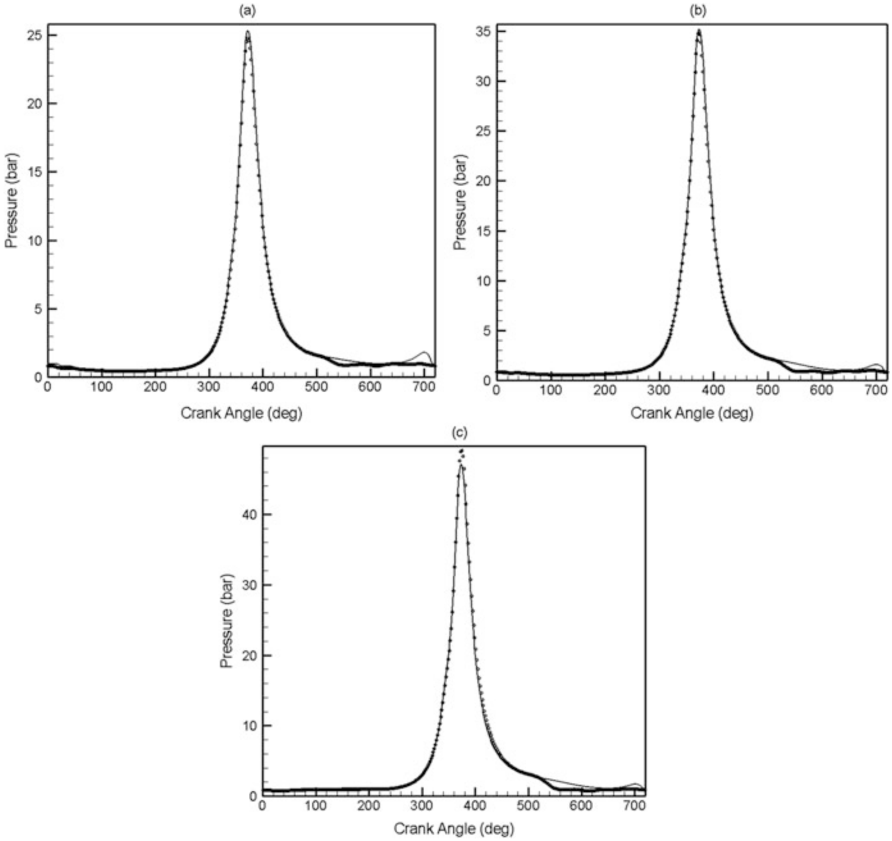

Fig. 11 In-cylinder pressure profile for fuel 3 at compression ratio of 14 for (**a**) BMEP = 2 bar, (**b**) BMEP = 4 bar and (**c**) full load conditions. Solid lines denote simulation and symbols denote experimental results

Tables 10, 11 and 12. The data in the table indicate that errors are larger than those for Mashhad gas, which was also predictable due to larger deviation of Tehran gas composition. Nevertheless, these errors do not exceed 4.5%, proving that the results are still in good agreement and the model results are still reliable.

5 Concluding Remarks

Natural gas burns through a clean combustion process and has large resources throughout the world. These key features let natural gas be considered as a promising alternative fuel. In spite of all these, it faces severe drawback of deviation in composition between different reservoirs, which affects its combustion characteristics.

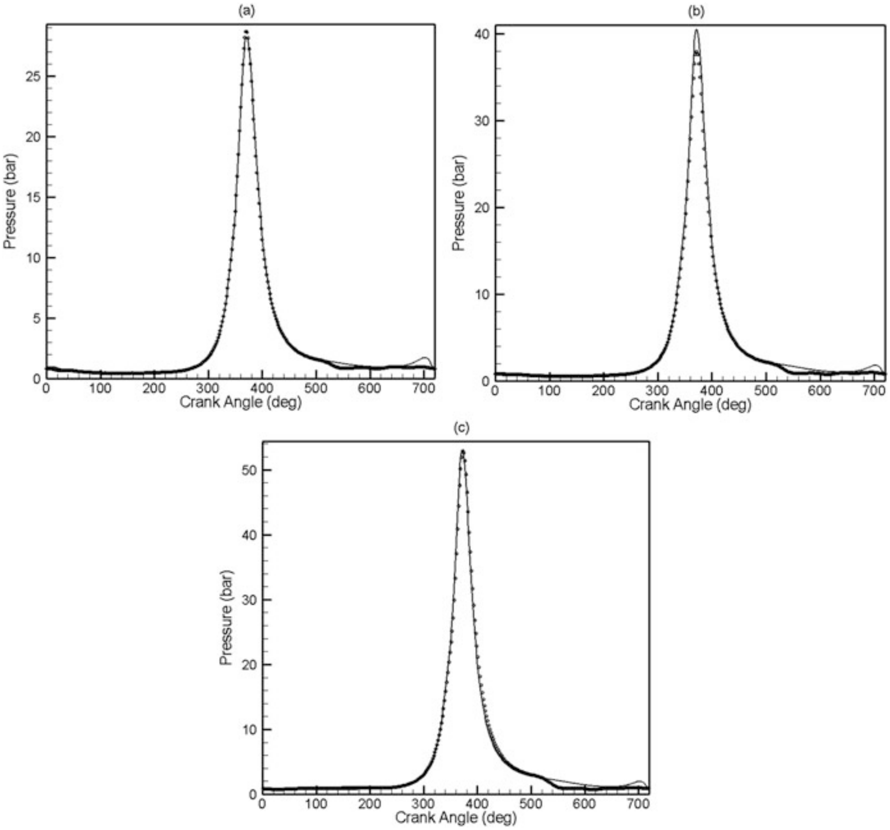

Fig. 12 In-cylinder pressure profile for fuel 3 at compression ratio of 16 for (**a**) BMEP = 2 bar, (**b**) BMEP = 4 bar and (**c**) full load conditions. Solid lines denote simulation and symbols denote experimental results

Table 9 Root mean squares of errors for numerical and experimental pressure profiles for fuel 3

	CR = 16 (%)	CR = 14 (%)	CR = 12 (%)
Full load	0.87	1.82	0.93
BMEP = 4 bar	1.84	0.89	1.80
BMEP = 2 bar	1.31	1.10	1.89

Table 10 Experimental and simulation results of air and fuel mass flow rates for fuel 3 in BMEP = 2 bar

CR		Exp. (gr/s)	Sim. (gr/s)	Error (%)
12	Air	2.728	2.827	3.641
	Fuel	0.197	0.203	2.861
14	Air	2.775	2.840	2.332
	Fuel	0.180	0.184	2.377
16	Air	2.842	2.930	3.071
	Fuel	0.184	0.190	3.008

Table 11 Experimental and simulation results of air and fuel mass flow rates for fuel 1 in BMEP = 4 bar

CR		Exp. (gr/s)	Sim. (gr/s)	Error (%)
12	Air	3.970	3.955	0.377
	Fuel	0.258	0.257	0.256
14	Air	3.800	3.967	4.392
	Fuel	0.247	0.257	4.269
16	Air	3.872	4.047	4.505
	Fuel	0.252	0.263	4.420

Table 12 Experimental and simulation results of air and fuel mass flow rates for fuel 3 in full load

CR		Exp. (gr/s)	Sim. (gr/s)	Error (%)
12	Air	5.379	5.518	2.583
	Fuel	0.350	0.359	2.662
14	Air	5.294	5.316	0.414
	Fuel	0.344	0.346	0.486
16	Air	5.150	5.187	0.727
	Fuel	0.335	0.337	0.687

One-dimensional comprehensive engine models have key importance during the engine design or optimization processes. This modelling technique consists of solving one-dimensional Euler system of inviscid flow equations inside the engine pipes and embedding them with zero-dimensional thermodynamic models for other engine elements such as cylinders. This methodology engages employment of several empirical factors which have to be calibrated and may differ as the working fluid composition changes. These parameters can be divided into three main categories:

1. Flow coefficients: The one-dimensional system of gas dynamic equations considers the flow to be homentropic (Benson 1982; Winterbone and Pearson 2000), i.e. having constant entropy over the flow domain. Therefore, the mass flow rates of working fluid into and out from the elements should be adjusted with flow coefficients, which are computed as the ratio of actual to isentropic flow rates.
2. Zero-dimensional combustion thermodynamic function is Wiebe function in this work. For Wiebe function, there are four calibration parameters.
3. Cylinder parts, i.e. liner, cylinder head, piston, valve and port temperatures and heat transfer coefficients which influence the in-cylinder thermal losses during the cycle.

Hereby, it is worth mentioning again that the present work's aim is not to investigate the effects of natural gas composition on the performance of the gas engine, but it aims to answer the question that does the change in composition of natural gas affect the calibration of the engine comprehensive gas dynamics model?

To answer this question, a one-dimensional comprehensive gas dynamics model is developed and calibrated for a single-cylinder spark ignition gas engine working with pure methane as fuel for three different engine compression ratios and three different engine loads. All the tests are carried out at constant engine speed of

2000 rpm at stoichiometric condition. The engine used in this research is designed and manufactured by AVL List Gmbh. Details on the test cell and engine are provided by Javaheri et al. (2014). In addition, the AVL BOOST v2013 software is employed for the simulations. The following results are achieved during the calibration process:

1. As the compression ratio rises, the engine part temperatures are reduced. This was predictable, because with increase in compression ratio, the temperature of cylinder exhaust gas will reduce (Heywood 1988; Ferguson and Kirkpatrick 2001; Pulkrabek 1997). In this work, as the compression ratio is increased from 12 to 14 and from 14 to 16, the temperature of piston, cylinder head, liner, ports and valves is considered to be 2.5 °C lower. This consideration yielded to the numerical predictions with most agreement with experimental findings.
2. In relatively high engine loads, the engine part temperature can be considered equal to those at engine full loads. In the present work, as the engine brake mean effective pressure is reduced from 6.2 at full load condition to 4, the engine part temperature remains constant in the model. Nevertheless, with further reduction of engine load to 2, the engine part temperatures should be reduced by 4 °C to achieve the best possible agreement.

To study the effects of changes in gas composition, the range of natural gas compositions in Iran is brought under consideration. Therefore, the natural gas distributed in Tehran which has the maximum deviation from pure methane is chosen. Another natural gas is also chosen from Mashhad, the second great city in Iran. Nine tests (three engine loads at three different compression ratios of 12, 14 and 16) are repeated for the engine working with each of the fuels. It is worth mentioning that in these tests, the engine speed is held constant at 2000 rpm, and the mixture is stoichiometric. The calibrated model is employed to simulate the engine cycle, without any change in parameters. The following results are obtained:

1. Despite the fact that the discrepancies between numerical predictions and experimental observations for in-cylinder pressure profile and mass flow rates of air and fuel increase as the deviation in the composition rises (the discrepancies of the simulation results from experimental data are higher for Tehran gas compared to Mashhad); the simulation results and experimental data remain in good agreement for fuels 2 and 3. Therefore, one can conclude that when studying an engine working the natural gases distributed in Iran, it would be sufficient to calibrate the engine comprehensive one-dimensional model with pure methane or each of the gas compositions that exist in Iran. The model predictions will be accurate enough for other gases.
2. The model validity with change in composition of the natural gases in the range that exist in Iran is regardless of engine compression ratio and load.
3. Considering the above-mentioned facts, it seems that the influence of natural gas composition inside Iran, on the gas-fuelled internal combustion engine working parameters, such as engine temperature, is very negligible.

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