AN APPROACH TO COMPUTATIONAL FLUID DYNAMIC AIR-FLOW SIMULATION IN THE INTERNAL COMBUSTION ENGINE INTAKE MANIFOLD

by

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Original scientific paper
https://doi.org/10.2298/TSCI180707063G

The subject of this paper is modelling of an intake manifold of a four-stroke internal combustion engine using contemporary software tools. Virtual 3-D CAD model of an intake manifold was designed based on a real intake manifold of a four-stroke internal combustion engine. Based on the CAD model a 3-D CFD model of the intake manifold was created. The modelling has been done with the purpose of simulation of the air-flow inside the intake manifold in order to monitor values of the internal pressure during several seconds of the engine operation in three different operating points. Also, an experiment was conducted, which included measurements of intake manifold pressure in the same engine operating points in the course of a time interval of approximately the same duration. The results of both the simulation and the experimental measurements have been shown in the paper proving that the created model was good enough for the intended purpose.

Key words: *intake manifold pressure, internal combustion engine, CFD simulation, air-flow, STAR CCM+*

Introduction

For more than 150 years people have been trying to make a system which would transform various forms of energy to mechanical energy as effectively as possible, and thus decrease physically demanding efforts imposed on humans. Starting with the steam engine whose efficiency was on a very low level, people have in time created systems which increase efficiency to a much higher level. The appearance of the internal combustion (IC) engine represents the beginning of a new era of converting chemical energy of fuels to mechanical power. The major division of IC engines in spark ignition (SI) and compression ignition (CI) IC engines was established at the beginning of the 20th century and has remained almost the same today. For SI engines, the mixture is ignited with the help of an energy source coming from the outside, which produces a spark sufficient for ignition of the mixture, as well as its further combustion. With CI engine, the process is similar, but due to the chemical nature of the fuel and higher pressure inside the cylinder of the engine, the ignition of the mixture occurs without the need for an external source of energy, the purpose of which is to generate the initial spark [1]. The IC engines represent generators of mechanical energy which still have a relatively low level of efficiency, in spite of the great influence of modern technology. The research in IC engines mostly

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relies on finding new, innovative solutions for the construction, thermodynamic processes and technology which would lead to an increase in the working efficiency [2, 3].

Thanks to modern software tools, the process of analysing new varieties and the concepts of functioning of IC engines is significantly shorter. For that reason, the influence of computer simulations of certain processes plays an important part in designing an IC engine, and, with the improvement of PC it represents an almost inevitable part of the production and the analysis process. Apart from the process inside the IC engine, another unavoidable part is considering the elements which contribute to the work of the engine, such as the intake manifold, exhaust manifold, injectors for providing the fuel, *etc.* Each of these elements influences the proper functioning of the IC engine and its efficiency to a great extent.

As far as the modelling of IC engine intake systems is concerned, the models could generally be classified into two groups. The first one involves models with the aim to improve the intake system design and hence the IC engine performance. These models are mostly based on 1-D and 3-D CFD simulations. In [4], a 3-D CFD simulation has been used to optimize automotive air intake system in the sense of better air-flow to the engine and hence better fuel economy. Chalet and Cheese [5] developed a new model of the unsteady air-flow through a throttle valve using CFD analysis, with the throttle valve modelled as a function of flow characteristics and geometrical parameters. In [6] CFD simulations, using STAR CCM+, were carried out for estimating the flow losses, mass-flow distribution between the engine cylinders, and swirls inside the intake manifold. Also, an innovative boundary condition method for the CFD simulations was suggested for improving the CFD simulation accuracy. Holkar *et al.* [7] performed a study to predict and analyze the air-flow through intake manifold and inlet ports using CFD. They used pressure boundary conditions to define the fluid pressure at the inlet and outlet of air intake system. Sedlacek and Skovajsa [8] performed a CFD analysis of the intake system of a motorbike engine to find a solution with the best possible distribution of air to the individual cylinders of the engine. Among other things, Kale and Ganesan [9] conducted a computational study of steady flow through intake manifold. They predicted flow field details in the identified regions in the manifold using the CFD.

The second group of intake system models are intended to be used in control applications. They are mostly based on 1-D adiabatic mean value engine models for the intake manifold filling dynamics. Such models usually describe the time development of the most important measurable engine variables (or states) on time scales a little larger than an engine cycle. Hendricks *et al.* [10] have developed a model of intake manifold filling dynamics, intended to estimate the air mass-flow to an SI engine especially during fast throttle angle transients. Deur *et al.* [11] have developed a three-state polytropic model to predict the temperature in the plenum and the runners of an IC engine during transients. Mattarelli and Valentini [12] analyzed an automotive 3.2 liter, V8, turbocharged SI engine by using integrated computational and experimental methods. In the scope of their research a 1-D fluid dynamics model has been set up for the engine simulation using the Ricardo Wave software.

In this paper the authors intended to combine the above two approaches to modelling of the intake system. In that sense, a 3-D modelling of the intake manifold of a SI engine was carried out by using CAD and CFD software, which simulated air-flow in it at various throttle positions and engine speeds over a time period significantly longer than the duration of one engine cycle. As a parameter for validating the results of the simulation, the intake manifold pressure was used and it was measured on the real engine during the experimental research. That parameter has been chosen since it always demonstrates the engine condition and affects the volumetric efficiency, fuel consumption and performance of IC engines. Also, manifold pressure is well known to be important to engine system stability and performance.

The developed model, with some modifications, could be used in future research as a basis for simulating some faults in the intake manifold (for example: air leakage, EGR valve malfunctions, *etc.*). This would significantly simplify the research of the influence of these faults on the intake manifold pressure value. In this way, possible damage, which may arise during experimental research of the real engine, could be avoided, if the engine is operating with a faulty intake manifold.

The model of an intake manifold

With the application of various software solutions it is possible to make a 3-D virtual model of an intake manifold and after that simulate the behaviour of the fluid inside [6, 7, 13-15]. The intake manifold of an engine is shown in fig. 1. Based on the intake manifold, its 3-D virtual CAD model was created and the model is shown in fig. 2.

Figure 1. Intake manifold of the engine

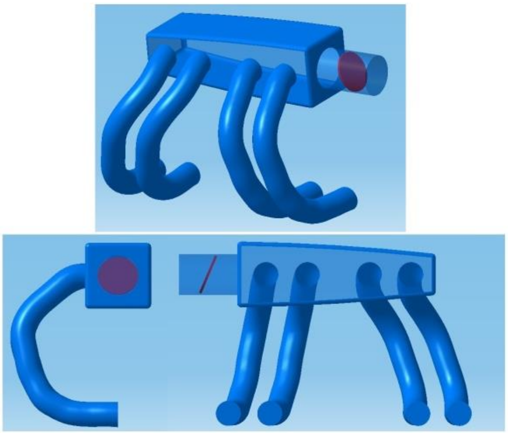

Figure 2. The 3-D model of the intake manifold

The CAD modelling was done in CATIA V5 software with a minor simplification. The model incorporates the manifold and the throttle valve with the possibility to vary the throttle valve angle.

The created CAD model of the intake manifold was imported in CFD software STAR CCM+, with the aim to create a CFD model and to simulate air-flow through it. Firstly, the mesh model was set. For similar intake manifold modelling in reference [7] the authors used generic volume type mesh HyperMesh 8.0 SR1, but in this paper a polyhedral prism shape of a mesh cell was selected. The base size of the cell was set to 2 mm, which resulted in a total of around 470000 cells in a formed volume mesh. The size of the base cell, as well as the total number of cells, which are established in this paper, correspond to the requirements in reference [8]. Figure 3 depicts the appearance of a created mesh model. The CFD model consists of one region, limited with six boundaries. These are wall, entrance, Cylinder 1, Cylinder 2, Cylinder 3, and Cylinder 4, and their positions are shown in fig. 4.

Because the air-flow is directed from the entrance towards the engine cylinders, *i. e.* from the area with atmospheric conditions towards the area of sub-pressure caused by the intake

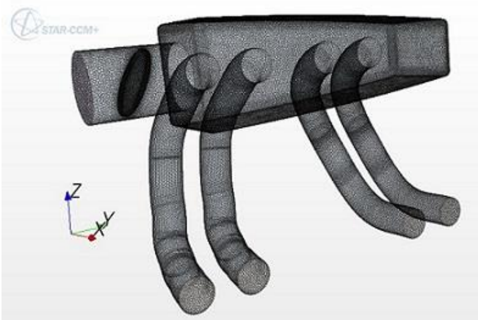

Figure 3. Mesh CFD model of the intake manifold

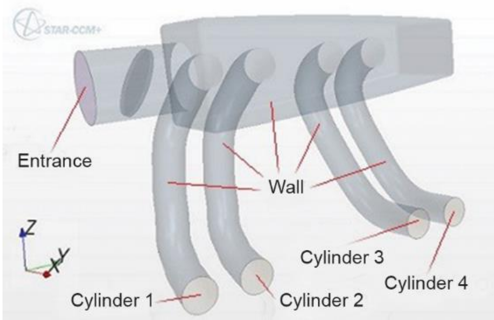

Figure 4. Boundaries and region of the intake manifold CFD model

process, it is necessary to choose the types of boundaries appropriately. For the entrance boundary, stagnation inlet type was selected since this type allows parameter adjustments corresponding to atmospheric conditions (atmospheric pressure, temperature, air density, *etc.*). It was also necessary to add a separate boundary to each of the cylinders because of the effect of the engine's periodical intake. For boundaries Cylinder 1 – Cylinder 4, the mass-flow inlet type was selected with the air-flow in the positive X-direction, fig. 4. In that way, the air-flow through the boundaries Cylinder 1 – Cylinder 4 was in accordance with the real air-flow outlet from the intake manifold. The mass air-flow rate through the boundaries was given as a time function and it is illustrated in fig. 5.

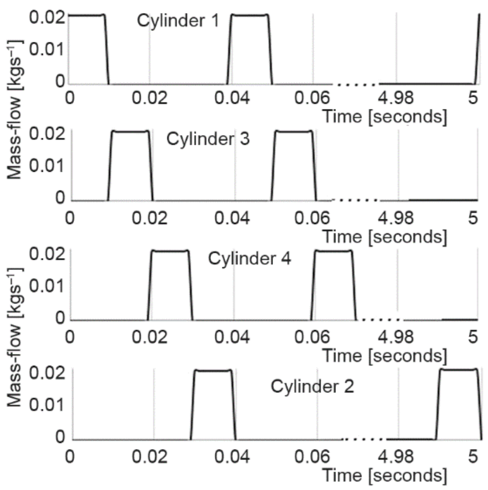

Figure 5. Periodical function of intake processes

As fig. 5 shows, the mass air-flow function is periodical, representing the periodicity of the intake processes in the engine with the firing order 1-3-4-2. The mass air-flow rate is assumed to be constant during the intake process with the value calculated by eq. (1) [16, 17]:

$$\dot{m} = \frac{\eta_V \rho v_T n}{120} \tag{1}$$

The overlapping of the intake processes between cylinders which appears in the real IC engine is also neglected. This assumption, as well as the one of a constant mass air-flow rate during the intake process, is acceptable here, since the aim of the paper is monitoring of the intake manifold pressure during a time interval that is much longer than a single engine cycle.

For setting parameters of the physics model in STAR CCM+ the time condition of flow was chosen as implicit unsteady according to periodical intake processes. Time parameters for simulations were chosen: 5 seconds for the total duration, 5 ms for the time step and 10 for the number of maximum inner iterations. Turbulent flow as the type of flow was chosen, because that type of flow is appropriate to the real intake manifold air streamlines, in the form of spreading and velocity. Reynolds number for the air-flow calculated by eq. (2) in the intake manifold is between 7.7×10^4 and 1.8×10^5 [18].

$$\mathrm{Re} = \frac{\rho v_\infty l}{\mu} \tag{2}$$

Due to the recommendations in [6, 19, 20], we established k-ε type of turbulent flow, because in multiple simulations it proved to be the most valid form. For the flow calculation, mass conservation and momentum conservation laws were used in the paper. The integral form of the mass conservation law is presented in eq. (3) [21]. The momentum conservation law is represented by eq. (4) [21]. The finite volume method, which uses the integral form of the conservation eq. (5) [21], was used for the CFD calculation:

$$\frac{\partial}{\partial t}\int_\Omega \rho \mathrm{d}\Omega + \int_S \rho vn \,\mathrm{d}S = 0 \tag{3}$$

$$\frac{\partial}{\partial t}\int_\Omega \rho v \mathrm{d}\Omega + \int_S \rho vvn \,\mathrm{d}S = \sum f \tag{4}$$

$$\int_S \rho \Phi vn \,\mathrm{d}S = \int_S \Gamma \,\mathrm{grad}\, \Phi n \,\mathrm{d}S + \int_\Omega q_\Phi \mathrm{d}\Omega \tag{5}$$

Boundary conditions for the CFD model were chosen as follows. Pressure and temperature values at the boundary entrance were set as atmospheric conditions (1 bar, 300 K). As far as wall boundary conditions are concerned, the thermal specification condition was set to environment with constant ambient temperature, constant heat transfer coefficient and constant thermal resistance. The values for the enumerated quantities were set according to the recommendations in reference [22]: thermal resistance for aluminium walls of the intake manifold was set to 1.873×10^{-5} m^{2}K/W and the heat transfer coefficient for air was set to 35 W/m^{2}K. Furthermore, wall surface specification was set to smooth, *i. e.* flow friction is neglected.

For the described CFD model settings, several simulations were performed in order to monitor the intake manifold pressure at different operating points of the engine. Thereby, three operating points, defined with three throttle valve openings and three engine speeds, were chosen. The values of the parameters, together with the corresponding values of mass air-flow rate, are given in tab. 1.

Table 1. Some input data for the three simulations performed

Simulation case	Throttle opening	Engine speed	Mass-flow rate of air
I	8%	2000 rpm	0.013669 kg/s
II	15%	3000 rpm	0.022213 kg/s
III	20%	3200 rpm	0.027339 kg/s

Experimental set-up

The experimental part of the research was conducted in the Laboratory for engines and vehicles at the Faculty of Technical Sciences in Novi Sad, Serbia. The experimental set-up that is used in this research is shown in fig. 6 [23].

The main components of the experimental set-up are an automotive SI engine – 1 and the dynamometer SCHENCK W230 – 2 with the control board – 3, which enable the setting up the engine operating points desired. The basic engine data are given in tab. 2.

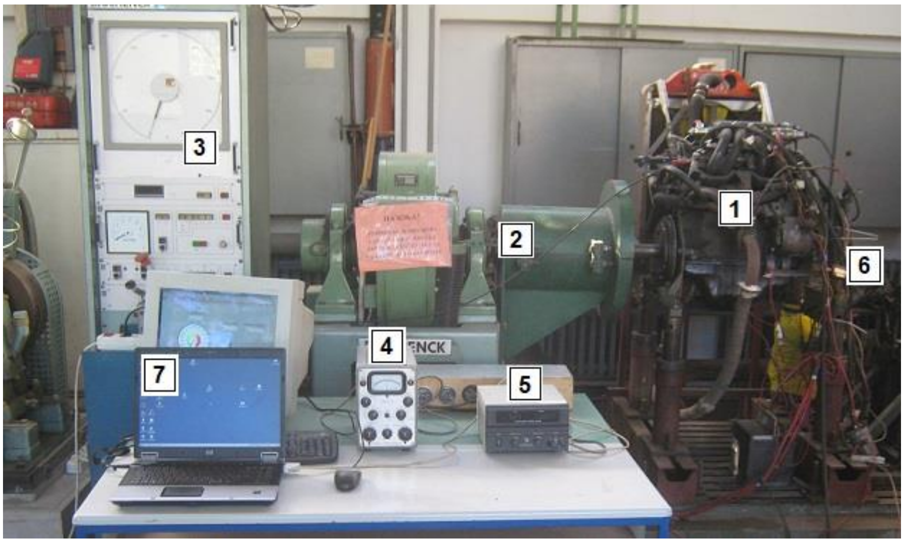

Figure 6. Experimental set-up [8]

Table 2. Basic engine data

Aspiration	Naturally aspirated
Cooling	liquid cooled
Engine cycle	four-stroke
Engine displacement	1116 cm^3
Number of cylinders	4
Number of valves per cylinder	2
Engine power	45 kW at 5600 rpm
Engine torque	85 Nm at 3800 rpm
Injection type	Multi-point intake port injection

Figure 7. The MAP sensor mounted on the engine [23]

In addition to the main components of the experimental set-up, there are also some very important components that served for parameters measurement and data acquisition. These include all the sensors, the signal amplifier – 4 with its power supply – 5, as well as versatile engine management system VEMS32 V3.6 – 6. The values of important parameters were monitored and recorded on the computer – 7. For the intake manifold pressure measurement the MAP sensor VEMS MPXH6400, fig. 7, was used. The plug-in location of the sensor was at the front side of the intake manifold between the second and third runners. The average sample rate of the sensor is 37 ms and it was acceptable considering the aim of the research.

The purpose of the experimental part of the research was to monitor absolute pressure inside the intake manifold at the engine operating points defined in tab. 1. All the measurements were carried out after both the optimal engine temperature and stationary engine operation had been reached. Among other things, the measured data of interest were: engine speed, throttle valve position, intake manifold absolute pressure and engine coolant temperature. The duration of the experimental measurement in each operating point of the engine was at least 5 seconds and the measurement data were recorded in a log-file using the VEMSTUNE software. Each measurement experiment was repeated several times to confirm the validity of the measured data.

Results and discussion

The results obtained by means of simulations, for all three cases of throttle opening, are depicted in fig. 8. The graph in fig. 8 shows the intake manifold pressure development in the course of 5 seconds, which was the duration time of each simulation. It can be seen that the pressure values oscillate throughout the simulation, as a result of periodic intake processes in the engine cylinders. Also, it is noticeable that in the beginning of the simulations, pressure values intensively decrease until they begin to oscillate around a constant mean value. This pressure drop is the result of the time needed for the simulation to stabilize. It is noted that this time period is shortened with the increase of the throttle opening. This can be explained: larger throttle openings cause higher pressure values, so it takes less time for the pressure to stabilize from the initial value of 1 bar. In fig. 8, a part of the graph is framed by the dotted line indicating the area of stabilized pressure oscillations in all the three engine operating points. For that reason, the corresponding time interval of 3 seconds has been chosen for verification of the simulation results.

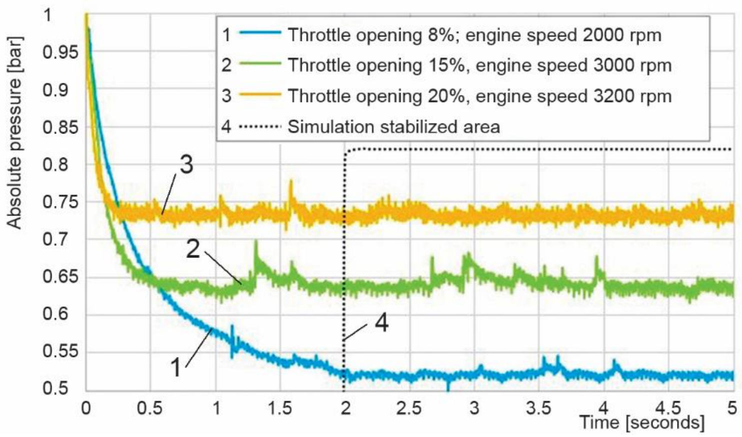

Figure 8. Simulation results

The results obtained by means of experimental measurement on the test stand, for all three cases of throttle opening, are depicted in fig. 9. The graph in fig. 9 shows time development of the intake manifold pressure during the interval of 3 seconds, although the experimental measurements lasted much longer. This shortening of the displayed time interval is done to equalize it with time intervals of the stabilized simulations. In fig. 9, the oscillatory character of the pressure change is observed, which is caused by periodic engine cycles in the cylinders. Also, it can be seen that the value of the pressure decreases with the reduction of the engine load.

Figure 10 shows the intake manifold pressure variations obtained by the simulations and by the experimental measurement. It can be noted that the oscillation frequency of the pressure calculated by the simulation is much higher than in the case of the values obtained by the experiment. This is due to the difference in the sampling rate, which is significantly higher in the case of simulation. However, despite this difference, a very good matching of the results was obtained considering the mean values of pressures during the time period of interest. This is illustrated in tab. 3, showing that in none of the three cases considered, the results of the simulation deviate from the measurement results by more than 2%. This can be considered acceptable in terms of verification of the intake manifold model developed during the research.

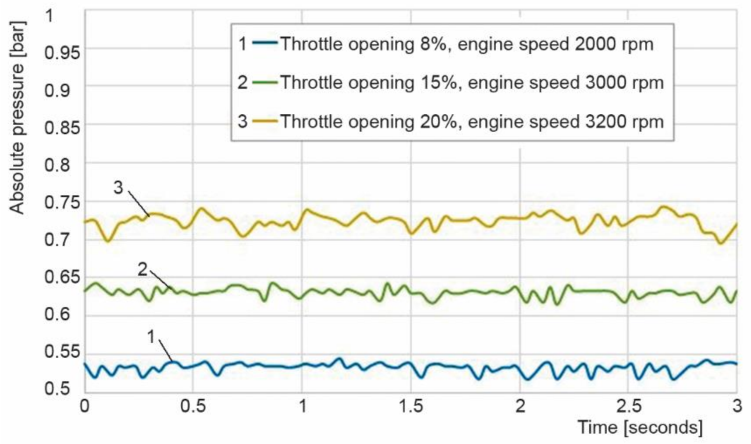

Figure 9. Measurement results

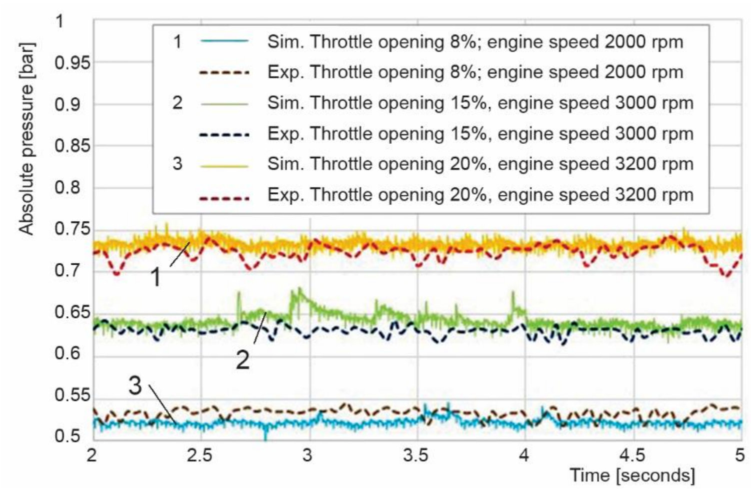

Figure 10. Comparative results

Table 3. Comparative representation of the results obtained

Case	Throttle opening [%]	Average absolute pressure value [bar]		Relative deviation [%]
		Experiment	Simulation	
I	8	0.53	0.52	1.28
II	15	0.63	0.64	1.74
III	20	0.73	0.74	0.70

Conclusions

During the research a CFD model of a SI engine intake manifold has been designed. The aim was to simulate air-flow in order to monitor pressure-time development in the intake manifold at different engine operating points. Validation of the model has been performed by using the data obtained in experimental measurements of intake manifold absolute pressure for three engine speeds and three throttle openings. Comparison of the measured data with the results of the simulations has shown that the deviation of the intake pressure mean value was less than 2%. This means that the model could be successfully used to monitor intake manifold pressure at different engine operating points. With certain modifications, the model could be improved, in order to simulate some malfunctions in the intake path of the engine. In this way, the risks of the experimental research could be avoided, which is the ultimate goal of future research.

Nomenclature

f – forces acting on the control mass, [N]
l – characteristic length, [m]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
n – engine speed, [rpm]
ndS – surface vector, [–]
p_a – absolute pressure, [bar]
Re – Reynolds number, [–]
S – control surface, [m^2]
t – time, [s]
v – air velocity, [ms^{-1}]
v_∞ – free-stream velocity, [ms^{-1}]
v_T – engine displacement, [m^3]

Greek symbols

Γ – diffusivity, [m^{2}s^{-1}]
η_v – volumetric efficiency, [–]
μ – dynamic viscosity, [Pa·s]
ρ – density of air, [kg/m^3]
Φ – control mass, [kg]
Ω – control volume, [m^3]

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Paper submitted: July 7, 2018
Paper revised: February 5, 2019
Paper accepted: February 17, 2019