International Journal of Automotive Technology, Vol. 6, No. 6, pp. 579−584 (2005)

1229−9138/2005/024−04

NUMERICAL ANALYSIS OF FUEL INJECTION IN INTAKE MANIFOLD AND INTAKE PROCESS OF A MPI NATURAL GAS ENGINE

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(Received 12 April 2005; Revised 26 October 2005)

ABSTRACT−Unsteady state free natural gas jets injected from several types of injectors were numerically simulated. Simulations showed good agreements with the schlieren experimental results. Moreover, injections of natural gas in intake manifolds of a single-valve engine and a double-valve engine were predicted as well. Predictions revealed that large volumetric injections of natural gas in intake manifolds led to strong impingement of natural gas with the intake valves, which as a result, gave rise to pronounced backward reflection of natural gas towards the inlets of intake manifolds, together with significant increase in pressure in intake manifold. Based on our simulations, we speculated that for engines with short intake manifolds, reflections of the mixture of natural gas and air were likely to approach the inlets of intake manifolds and subsequently be inbreathed into other cylinders, resulting in non-uniform mixture distributions between the cylinders. For engines with long intake manifolds, inasmuch as the degrees of intake interferences between the cylinders were not identical in light of the ignition sequences, non-uniform intake charge distributions between the cylinders would occur.

KEY WORDS : Multidimensional numerical simulation, Natural gas jet, MPI engine, Intake interference

1. INTRODUCTION

There are tens of thousands of natural gas or LPG vehicles currently used in China (Deng and Zeng, 1998; Zhang, 2002). However, most of them cannot meet the increasingly strict govement emission regulations (Yang, 2001; Hao *et al.*, 2000) due to their out-of-date mechanical or electronic-control-compensating-air mixers, which are thought to have many natural drawbacks, such as low precision of air-fuel ratio control and long response times to variation in engine working states. To enable continued use of these vehicles, replacing the mixers by MPI (Multi-port Injection) fuel supply equipment seems to be a simple and effective solution (Matsuki *et al.*, 1996; Brault *et al.*, 1998; Lee *et al.*, 2001). Nevertheless, it was found that in practice if a gasoline injector is replaced by a natural gas injector without altering the location of the injector on the intake manifold wall, the modified natural gas engine will suffer from non-uniform mixture distributions between the different cylinders (Japan Gas Association, 1996). This was probably attributable to two phenomena: 1) non-identity of intake airflows entering different cylinders, due to large volumetric gas injections in intake manifolds and 2) reflection of injected natural gas towards intake manifold inlets. However, so far, few studies that emphasis these problems have been performed (Liang and Xu, 2003). Bench tests on a modified MPI natural gas engine (Matsuura *et al.*, 1995) suggested that to achieve the best integrated performance with respect to the output power and the low emission of engine, it was necessary to control the entire natural gas injecting duration to fall into the time range from the moment of the top dead center of the compression stroke to the moment of the beginning of the intake stroke. That is to say, during the injecting of natural gas, the intake valve should be in a closed state.

The purpose of this paper is to quantitatively investigate the influence of natural gas injections in intake manifolds on the fuel-air mixture distributions between different cylinders, using a multidimensional numerical method. The main content of this paper is fourfold: (1) unsteady state natural gas jets injected from a single-nozzle and a double-nozzle injector were numerically simulated by the use of a commercial CFD software AVL Fire 8.3; (2) to examine the computations, two sets of

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visible photographs of natural gas jets were taken with a schlieren apparatus for the two types of injectors, respectively; (3) injections of natural gas in intake manifolds were simulated for a single-intake valve engine and a double-intake valve engine, respectively; (4) for quantitatively investigating the intake interferences between the cylinders of a MPI natural gas engine, the intake process of a Honda F22B4 gasoline engine was computed, based on a precise measurement-based construction of the computational model.

2. MATHEMATICAL MODEL AND NUMERICAL METHOD

2.1. Governing Equations

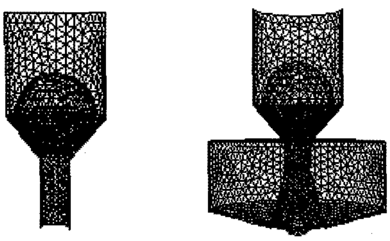
The basic equations consisted of a mass conservation equation, a momentum equation and an energy equation. In addition, a multi-component transport equation was included, to account for the conduction and diffusion of natural gas in air. The compressibility of gas was taken into account by modifying gas density according to a gas state equation. With respect to the turbulence model, the realizable κ-ε model proposed by T. H. Shih (Shih *et al.*, 1995) was employed. For the purpose of reasonable treatment of near wall turbulent flow, the Logarithmic Law Non-Equilibrium Wall Function (Kim *et al.*, 1995) was adopted in the present computations.

2.2. Numerical Method

The FVM (Finite Volume Method) was employed to transform the differential governing equations into numerically solvable discrete equations. Discretization was performed according to a double precision upstream scheme. The SIMPLE (semi-implicit method for pressure-linked equation) method was employed, to enable a coupled solution of pressure and velocity (Partaker, 1981). Relaxation factors have been used during calculations to facilitate achievement of numerical convergence; their values are not fixed but changeable during the progress of computation. The initial values for them can be found in Table 1.

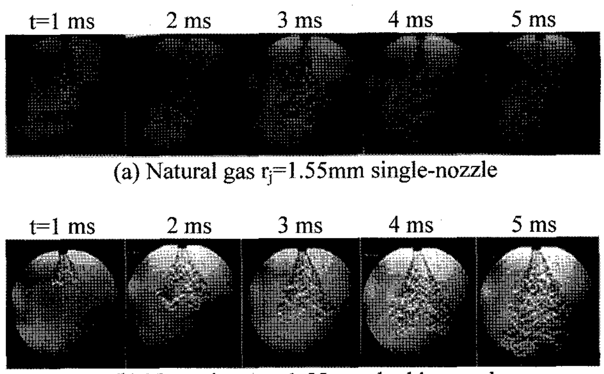
Table 1. Initial values of relaxation factors.

p	u	k	ε	v	T	CH_4
0.1	0.5	0.4	0.4	0.4	0.4	0.5



(a) Single-nozzle (b) Double nozzle

Figure 1. Mesh models of injector heads.



(a) Natural gas r_j=1.55mm single-nozzle

(b) Natural gas r_j=1.55mm double-nozzle

Figure 2. Schlieren photographs of the natural gas jet at different injection times (p=0.3 MPa).

3. NUMERICAL SIMULATIONS

3.1. Free Natural Gas Jet

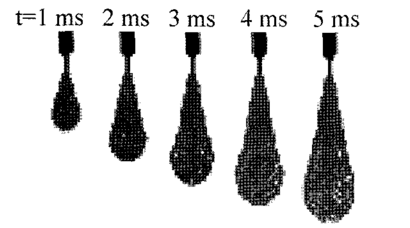
3.1.1. Natural gas injectors

Computational mesh models of two typical natural gas injectors are illustrated in Figure 1, one is a single-nozzle injector with a nozzle radius (r_j) of 1.55 mm, another a double-nozzle injector with a nozzle radius of 1.15 mm. Given that the volumetric flow rate of a gasoline injector being 245 cm^3/min, in the present study, to maintain identical calorific value of natural gas injection with that of the gasoline, the volumetric flow rates of the natural gas injectors were both set at 170.4 L/min with a driving pressure of 0.3 MPa.

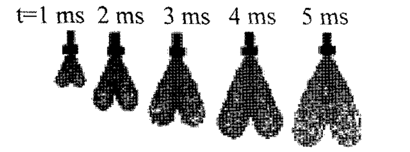
3.1.2. Numerical simulations and comparisons with schlieren experimental results

In order to examine the feasibility of the present numerical simulation method, schlieren experiment was performed (Hosch *et al.*, 1977). In the experiment, schlieren photographs taken by a high speed CCD camera with a shutter duration of 1/10000 s were delivered to an image processor, for investigating the shapes and quantifying the injection angles and penetration lengths of the natural gas jets. Figure 2 shows two sets of schlieren photographs, taken for a single-nozzle injector and a double-nozzle injector at intervals of 1ms in each case.

In Figure 2, natural gas jets are observed to maintain a spindle shape at the five typical injecting moments. Numerical results simulated under the same conditions as



(a) Natural gas r_j=1.55mm single-nozzle



(b) Natural gas r_j=1.55mm double-nozzle

Figure 3. Predicted concentration distributions of natural gas jet at different injection times.

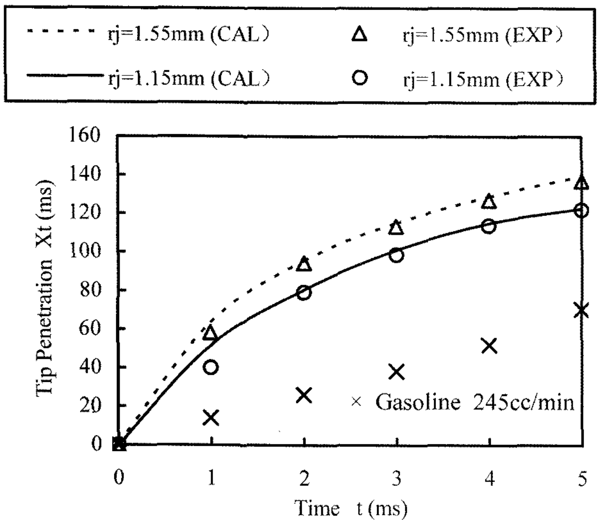
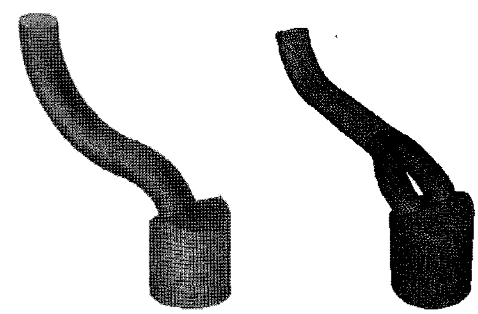


Figure 4. Comparison between the calculating and experimental results for the penetration of the natural gas jet and gasoline spray (p=0.3 MPa).

those in the schlieren experiment are shown in Figure 3, where the shapes of natural gas jets are depicted by the concentration distributions of natural gas in air. Figures 2 and 3 demonstrate reasonable agreement of simulations with experimental results, in terms of the shapes of natural gas jets at each injection moment. Moreover, penetration lengths of natural gas jets at the five typical moments were obtained based on numerical simulations, which compare well with experimental data as shown in Figure 4. Experimentally obtained penetration lengths of gasoline sprays (r_j = 0.35 mm, 245 cm^3/min) are shown in Figure 4 as well. Penetration lengths of natural gas jets are found to be considerably larger than those of gasoline sprays. That is, at t = 5 ms, the penetration length of the natural gas jet (double-nozzle, r_j = 1.15 mm) is about 122 mm, while that of the gasoline spray (double-nozzle, r_j = 0.35 mm) is only 70.1 mm, which is even smaller than that (78 mm) of the natural gas jet at t = 2 ms. In the present study, we assume that the power of a modified natural gas engine is maintained at an approximately identical level with that of the original gasoline engine; this requires a volumetric flow rate of natural gas of about 710 times that of gasoline. Therefore, for a MPI natural gas engine, given that the influence of the manifold wall on the penetration length of natural jet is negligible, the tip of natural gas jet can be expected to arrive at the surface of the intake valve at about t = 2 ms. Thereafter, the natural gas jet will impinge with the intake valve and be partially reflected back towards the inlet of the intake manifold. JARI (Japan Gas Association, 1996) pointed out based on his bench test on natural gas engines that when intake manifold was too short, intake interference resulting from the reflection of natural gas led to a difference in air-fuel ratio of up to 0.4 between cylinders.

3.2. Numerical Simulation of Injection of Natural Gas in Intake Manifolds

In order to investigate the effect of natural gas injection on the performance of a natural gas engine, natural gas injections in two typical intake manifolds have been predicted in this paper. One is a single-intake valve manifold, and another is a double-intake valve manifold. The 3D (three dimensional) computational mesh models of the two intake manifolds are illustrated in Figure 5. In the mesh models, the single-nozzle and double-nozzle injectors are both located 95 mm distal from the intake valves, according to the measurements on an actual gasoline engine. For grid topology and mesh density distribution of the mesh models, we followed two principles: 1) unstructured grid topology was employed to deal with the complex shape of the intake manifold; 2) the density of meshes near the intake valve was enhanced to account for the large flow gradients produced by the



(a) Single-intake valve (b) Two-intake valve

Figure 5. Computational mesh model of automobile engine.

impingement of the natural gas jets with the intake valve. Furthermore, the precision dependency of numerical results on mesh size was studied comparing numerical results predicted based on computational models with different mesh densities. As a result, we concluded that to achieve a mesh independent solution, a mesh number of at least 185100 was necessary for the single-intake valve manifold, while more than 203812 meshes were needed for the double-intake valve manifold.

For a MPI gasoline engine, to prevent possible deterioration in emission performance due to the incompletely vaporized liquid fuel entering the cylinders, the injection duration of gasoline is generally controlled to fall into the time range from 150° CA BTDC to the TDC of the intake stroke (Nakamura *et al.*, 1999) such that injected liquid gasoline can be vaporized on the hot surface of intake valves before entering the cylinder. In the present study, we seek to elucidate the possible problems in direct modifications of MPI gasoline engines into natural gas engines. Therefore, the duration, timing of injection, and the driving pressures of the natural gas injectors will be set identically with those of the gasoline injectors. With this in mind, in the present numerical simulations, the driving pressure of the gas injectors was set at 0.3 Mpa, the duration of injection was set at 5 ms, and the moment when injection starts was set at 150° CA BTDC. The initial pressure in intake manifolds was assumed to be a standard atmospheric pressure.

Simulated concentration distributions of natural gas in two intake manifolds are shown in Figure 6. Figure 6 indicates that at the end of injection (t = 5 ms), due to the large volumetric injection, natural gas jets impinge strongly with the intake valves, leading to pronounced reflections of natural gas back towards the inlets. According to Figure 6(a), at t = 5 ms, natural gas has been reflected backward by 160 mm along the wall of the intake manifold, and the reflection of natural gas will persist until the start of the intake stroke. Based on analysis of the simulated results, we speculate that for a long intake manifold, the reflected natural gas is not likely to approach the inlet. Nevertheless, if a natural gas engine is modified based on a engine whose intake manifold is very short (e.g. a diesel engine), given that the intake duration is long enough (e.g. the operation mode of low speed and large torque), it is very possible that the reflected natural gas will approach the inlet of intake manifold and subsequently be inbreathed into other cylinders, leading to non-uniform distributions of natural gas between cylinders.

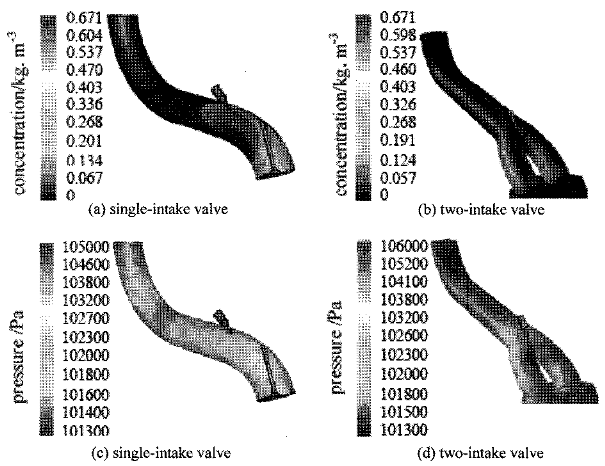


Figure 6. Concentration and pressure distribution of natural gas in intake manifold (t = 5 ms).

3.3. Numerical Analysis of the Intake Process of a Four-cylinder Engine

3.3.1. Computational model

For the purpose of a further quantitative investigation of intake interference in a MPI natural gas engine, a computational model of a four-cylinder engine, as shown in Figure 7, was constructed based on the measurements of a Honda F22B4 gasoline engine. In the model, a double-hole injector is installed on the upper side of the intake manifold wall, with the distance from the injector to the intake valve set at 95 mm. The inclination angle of the centerline of the natural gas injector to the surface of the intake valve is kept identical with that of the original gasoline engine. In consideration of the complex geometric structure of the engine model, the topology of the computational mesh model has employed an unstructured grid scheme with mesh density distributions properly controlled in different regions. As a result of mesh independent analysis, we found that at least 626871 mesh cells would be needed to permit the achievement of a mesh independent numerical solution.

3.3.2. Computational conditions

The present study chose cylinder 2 as an example in investigating the effects of natural gas injection in the

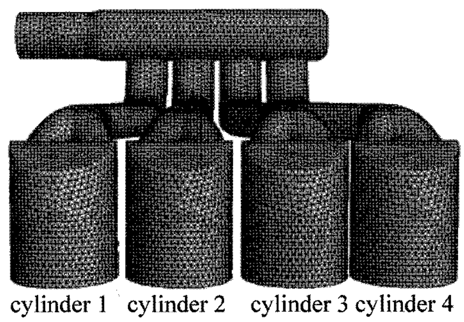


Figure 7. Computational mesh model of automobile engine.

dependent intake manifold on the intake processes of other cylinders. The intake processes of cylinder 1, 3, 4 were controlled by the movements of the intake valves and pistons according to the firing order sequence: 1-3-4-2. The engine was assumed to work in a state of 6000 r/min in speed and 100% in throttle opening degree. The natural gas injector operated at a driving pressure of 0.3 MPa with a fixed injection duration of 5 ms. Moreover, the moment of the start of injection was set at the TDC of the compression stroke according to reference 3 (Matsuura *et al.*, 1995).

3.3.3. Simulated results

Figure 8 shows the concentration contours in the dependent intake manifold of cylinder 2 at an injection moment of 4 ms. In Figure 8, it can be observed that significant impingement between the injected natural gas and the intake valve occurs, giving rise to a pronounced backward reflection of natural gas towards the inlet of the manifold. After injection, there is another 5 ms before the opening of the intake valve, which means that the reflection of natural gas may persist, However, because the injection has stopped, the reflection dominated by the diffusion of natural gas is remarkably slow.

To investigate the variations in intake charges of the other three cylinders under the influence of fuel injection in intake manifold 2, path lines between the four cylinders were computed and illustrated in Figure 9 and 10. Path lines in Figure 9 show that at t = 4 ms when cylinder 2 is at compression stroke and cylinder 3 at intake stroke, the increase in pressure in intake manifold

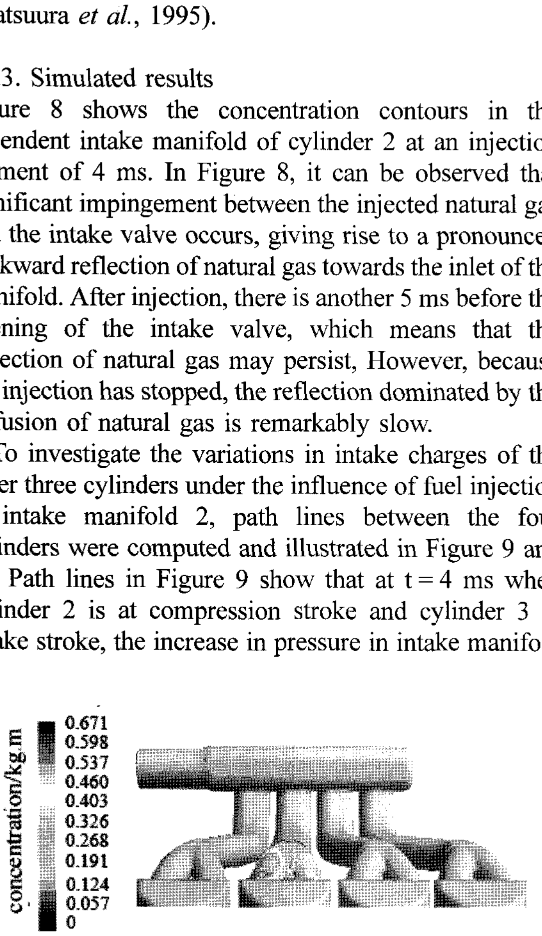


Figure 8. Concentration contours of natural gas in intake manifold (t = 9 ms).

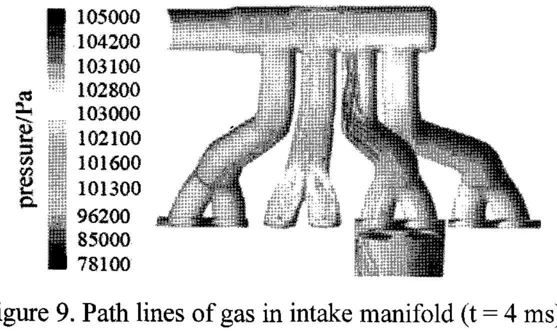


Figure 9. Path lines of gas in intake manifold (t = 4 ms).

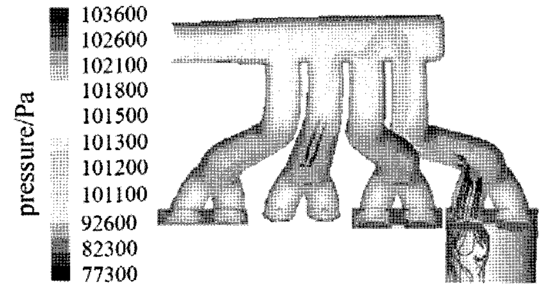


Figure 10. Path lines of gas in intake manifold (t = 9 ms).

2 caused by natural gas injection significantly enhances the flow directed from intake manifold 2 to 3, which indicates the existence of significant intake interference between cylinders 2 and 3. Figure 10 shows that at t = 9 ms, when injection in intake manifold 2 has ceased and cylinder 4 is at intake stroke, intake interference (indicated by the flow directed from intake manifold 2 to 4), takes place between cylinders 2 and 4. However the intensity of the interference is remarkably weak when compared with that between cylinders 2 and 3. Based on these simulated results, we consider that for engines with short intake manifolds, it is very possible that the reflection of injected natural gas could be resulting so strong that it would approach the region near the inlet of intake manifold and subsequently be inbreathed into other cylinders, resulting in non-uniform mixture distributions between the cylinders.

From Figures 9 and 10, we find that the intake charges of cylinders 3 and 4 are both affected by the natural gas injection in intake manifold 2; whereas, the degrees of the intake interferences are significantly different, due to the differences in ignition sequences and their geometric locations. It is easy to speculate according to the firing order sequence of 1-3-4-2 that cylinders 2 and 3 may suffer from significant influences from the natural gas injections in the dependent neighboring intake manifolds; whereas, cylinders 1 and 4 may be free from large influences arising from fuel injections in other intake manifolds, due to the lack of direct interferences between them and their neighboring cylinders. That is to say, even without the interference of natural gas, the degrees of the air intake interferences between the four cylinders are not identical still implies the occurrence of non-uniform air distributions and consequently non-uniform air-fuel ratios between the four cylinders.

To find realistically ways to reduce the effects of the injection of natural gas in intake manifolds on the intake charge distributions between cylinders, this paper suggests that one promising method may be letting partial natural gas be injected after the opening of intake valve by properly delaying the injection timing, to reduce the reflection of natural gas in the intake manifolds. Moreover, application of the lean-burn technique that is currently employed in gasoline engines to natural gas engines might provide another effective approach. These two methods have been included in our undergoing studies.

4. CONCLUSIONS

(1) The transient natural gas jets were reasonably simulated, using of the present multidimensional numerical method.
(2) Numerical predictions indicated that injection of natural gas in intake manifolds resulted in a remarkable backward reflection of injected natural gas towards the inlets of intake manifolds accompanied by a significant increase in pressures in intake manifold, which was considered to be an important cause of the non-uniform mixture distributions between cylinders of a MPI natural gas engine. For engines with short intake manifolds, we speculated that based on our simulations it was very possible that the reflected natural gas would approach the regions near the inlet of intake manifold and subsequently be inbreathed into other cylinders, resulting in non-uniform mixture distributions between cylinders.
(3) We quantitatively investigated the intake interferences between the four cylinders of a MPI natural gas engine that was modified based on a Honda F22B4 gasoline engine. As a result, it was found that the intake interference degree undergone by individual cylinder was not identical, implying the existence of non-uniform air distributions between the four cylinders.

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