# EFFECT OF GAS FLOW ON COMBUSTION AND EXHAUST EMISSIONS IN A DUAL FUEL NATURAL GAS ENGINE

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***Abstract***

*In recent years, natural gas is attracted attention as alternative fuel for oil. This study focused on the dual fuel combustion system as a method to use natural gas for cogeneration engine. The purpose is construction of high power, high efficiency and low emission cogeneration engine by using lean burn and supercharged system. One of important parameters that affects on combustion is gas flow in cylinder. In this study, shrouds that are equipped near the intake valves were used to change the gas flow in cylinder. Combustion process was also observed by visualization. The engine performance and exhaust characteristics were compared to investigate effects of gas flow, that is, swirl, tumble and normal. This test engine has a single cylinder with bore of 96mm and stroke of 108mm. The compression ratio is 16:1. The engine was operated at 1000 rpm. Compressed natural gas was induced from an intake port and diesel fuel was injected into the cylinder near the compression TDC. After the diesel fuel with natural gas and air mixture is auto-ignited, the premixed turbulent flame develops from some ignition parts. The main results are as follows: The combustion process, engine performance and exhaust emissions are different, when gas flows like swirl or tumble exist in cylinder. In the case of swirl flow, unburned hydrocarbons emissions increase and output performance and thermal efficiency decrease. In the case of tumble flow, the combustion duration is shortened. And in leaner condition, the thermal efficiency and output power increase and unburned hydrocarbons are slightly reduced.*

**Keywords:** *Alternative Fuel, Dual Fuel Combustion System, Natural Gas, Supercharged System, Lean Burn, Gas Flow, Swirl, Tumble*

## 1. Introduction

Natural gas is considered to be an environmental friendly energy because of lower carbon dioxide emission at burning and one of alternative energy sources of crude oil. Natural gas engine is considered to be a technology for decreasing an environmental impact among emerging problems of pollutions in global scale. To increase the output efficiency of generator, lean burn gas engines are mainly used.

Dual fuel type with a pilot injection of diesel fuel has been studied by many researchers [1-13]. Karim et al. have studied on dual fuel engine utilizing gaseous fuel and diesel fuel in detail for many years [1-5]. The dual fuel gas engine with diesel fuel injection has a feature of multipoint ignition sources, leading to stable combustion and cleaner exhaust emissions. However, gas engines have some problems, for example, knock at heavy load and misfire at low load [4]. In lower load, hydrocarbons increases because of incomplete combustion in too lean mixture. Therefore, a lot of works have been done to solve these problems, and hot or cooled exhaust gas recirculation (EGR) [6, 7], preheating [7], etc. have been tried. The characteristics of ignition delay of diesel fuel with gaseous fuel and air mixture was investigated [5, 6].

There are several types of ignition systems for gas engines. The spark ignition systems with openchamber or prechamber are used for many engines. However, both systems have disadvantages such as lower ignition energy and the prechamber system has a complex structure of the combustion chamber. Recently, the gas engine which has a very few amount of diesel fuel injection system for ignition has been developed. This system is practical because parts converted from a diesel engine are able to be reduced.

In the previous study, a single cylinder supercharged gas engine with direct injection of diesel fuel and port injection of natural gas and electrical control system was used [13]. Lean burn contributed to restraint of knock phenomena and to achievement of low NOx, while supercharged air increases the engine power. The effects of number of nozzle at injector and quantity of diesel fuel on exhaust emissions and engine performance in changing injection timing and pressure in the intake manifold were investigated. Furthermore, initial combustion was observed through a quartz window installed in the piston.

In this study, the effect of gas flow motion, such as swirl and tumble, on the combustion and exhaust emissions was investigated. It is said that turbulence promotes turbulent combustion in a spark ignition engine cylinder because flame front of the premixed combustion is affected by the turbulence. After performing a simulation of the gas flow, three kinds of cylinder heads were prepared.

## 2. Experimental method

Figure 1 shows the schematic diagram of experimental setup. Table 1 shows specifications of the test engine and operating conditions. The engine tested was a water-cooled, four-stroke cycle and single cylinder. The bore and stroke are 98 mm and 108 mm, respectively. This engine has compression ratio of 16:1, shallow dish chamber and two intake and two exhaust valves.

Natural gas was injected into an intake pipe through a mass flow controller during intake stroke, and then natural gas/air mixture was formed in intake manifold and the mixture was induced into the cylinder. A small amount of diesel fuel was injected by using a common rail system in injection pressure of 40MPa. The amount of diesel fuel was set to 2 mg/cycle, which equivalence ratio was 0.016. The nozzle was three holes of which diameters were 0.1 mm.

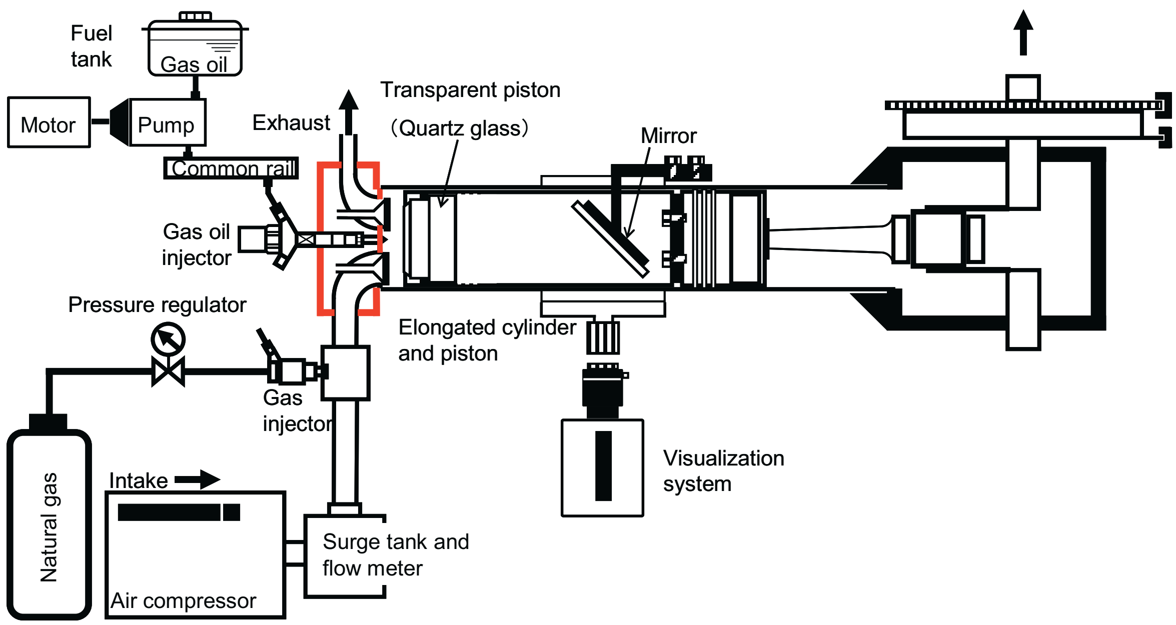


*Fig.1. Schematic diagram of experimental apparatus*

The spray angle of injection was 140 degrees. The equivalence ratio in total was set from 0.59 to 0.62. The pressure in the intake manifold was 200 kPa in absolute pressure by an external air compressor. The engine speed was 1000 rpm and signals of TDC (Top Dead Centre) and crank angle at every 0.5 degree were detected. These signals were sent to the device to operate the injectors of which injection timings and durations of natural gas and diesel fuel were changed variously. Pressure in the cylinder was measured with a piezo electric type sensor (Kistler, type 6052B) attached at the engine head and the rate of heat release was determined. Exhaust gas emissions of NOx, CO and HC were measured in an exhaust pipe. Smoke in the exhaust gas was zero at every operating condition.

*Tab. 1. Test engine specifications and operating conditions*

| | |
|---|---|
| Engine speed | n=1000rpm |
| Injection timing of diesel fuel | $\theta_{inj}$=0~10 deg.BTDC |
| Injection pressure of diesel fuel | $P_{inj}$=40MPa |
| Injection quantity of diesel fuel | $m_{inj}$=2 mg/cycle ($\phi$=0.016) |
| Nozzle | 3holes ($\phi$0.10mm) |
| Intake port pressure | $P_{in}$=200kPa(abs.) |
| Equivalence ratio | $\phi_t$=0.59, 0.6, 0.61, 0.62 |

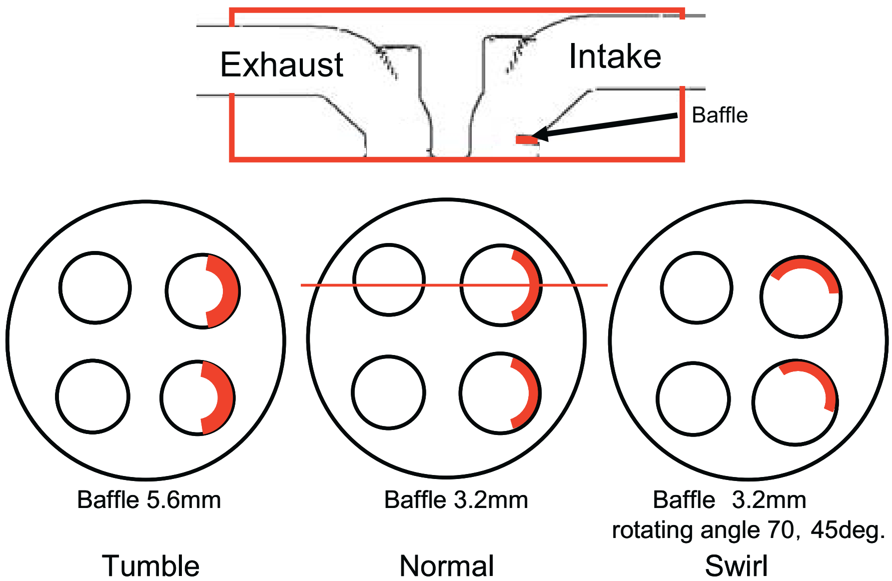


*Fig. 2. Cylinder head*

## 3. Cylinder head and gas flow simulation

For producing different gas flow, three kinds of the cylinder head were prepared. Normal head has two small shrouds of baffle plates in each intake port near the intake valve as shown in Fig. 2(a). Tumble head has two larger shrouds in each intake port as shown in Fig. 2(b). In swirl head shown in Fig. 2(c), each shroud has different angle for the shrouds in the normal head. The angle is decided by the analysis of the gas flow with Star-CD code. The turbulent energy at 30 deg. BTDC was presented for three kinds of cylinder head as shown in Fig. 3. The turbulent energy for the swirl flow was almost the same as that for the normal one. On the other hand, the turbulent energy for the tumble flow was larger than that for the normal head. The tumble motion decays near the compression TDC, producing smaller scale of the eddy and strengthen the turbulence intensity. The turbulent energy for tumble case was about 1.5 times stronger than that for normal case around the middle of the combustion chamber although the turbulent energy was almost the same near the cylinder wall.

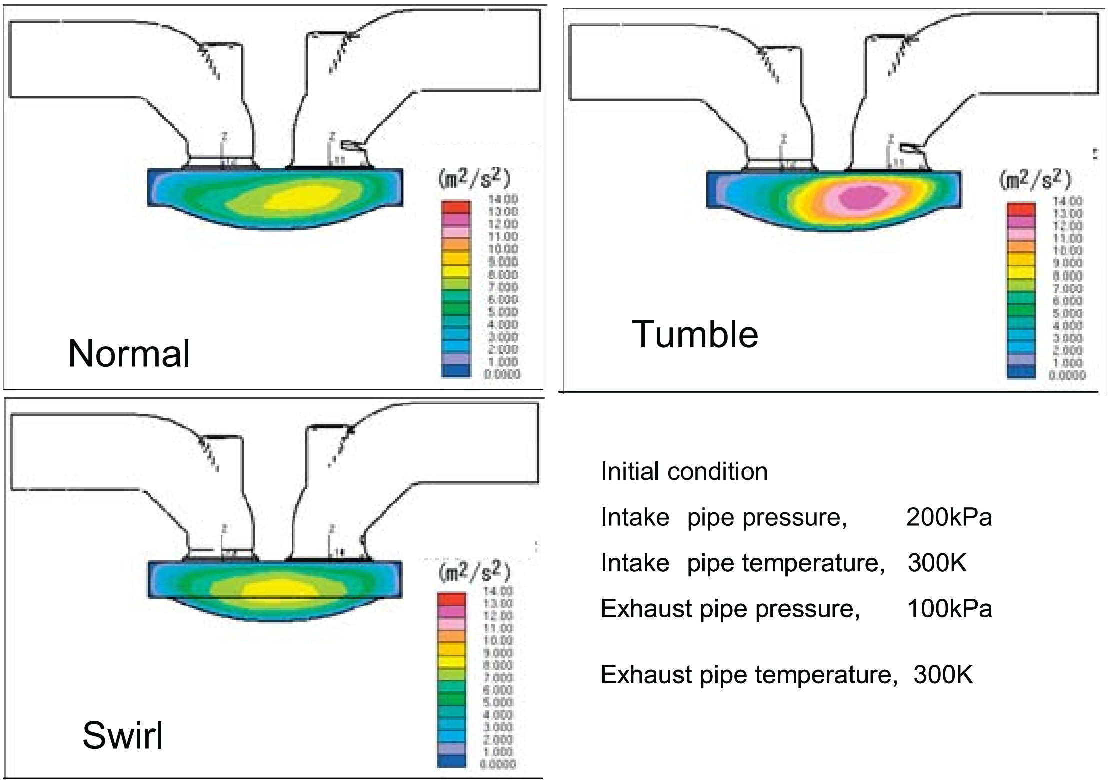


*Fig. 3. Turbulent energy at 30deg.BTDC*

**4. Pressure history and rate of heat release**

Fig. 4 shows the pressure history and the rate of heat release when the cylinder head is changed under the condition of equivalence ratio in total, $\phi_t$, of 0.6 with diesel fuel quantity of 2 mg/cycle. The injection timing was changed from 10 degrees BTDC to TDC. As the injection timing was advanced, the maximum values of pressure and rate of heat release increased for each cylinder head.

Two peaks in the rate of heat release were observed except initial combustion at the injection timing of 10 deg.BTDC for normal case and of 8 deg. BTDC for tumble case. This is not a knock but just auto-ignition at late stage of the combustion because there is no pressure oscillation on the pressure history. Here, initial combustion duration is defined as the duration between the start of injection of diesel fuel and the time when mass fraction burned reaches 10%. Main combustion duration is the time when the mass fraction burned is between 10% and 70%. As shown in Fig. 5, the initial combustion duration became a little bit shorter with advancing injection timing of diesel fuel. The initial combustion duration was almost the same for three kinds of gas flow. The gas flow motion did not affect ignition delay very much. However, the main combustion duration became shorter with advancing the injection timing of diesel fuel. The main combustion duration for swirl case became a little bit shorter than that for normal case. It is considered that the swirl flow promotes the combustion near the cylinder wall. In tumble case, the main combustion duration became shorter than that in normal case at 8 deg.BTDC because the effect of small size of the turbulence decayed from tumble motion remains more strongly.

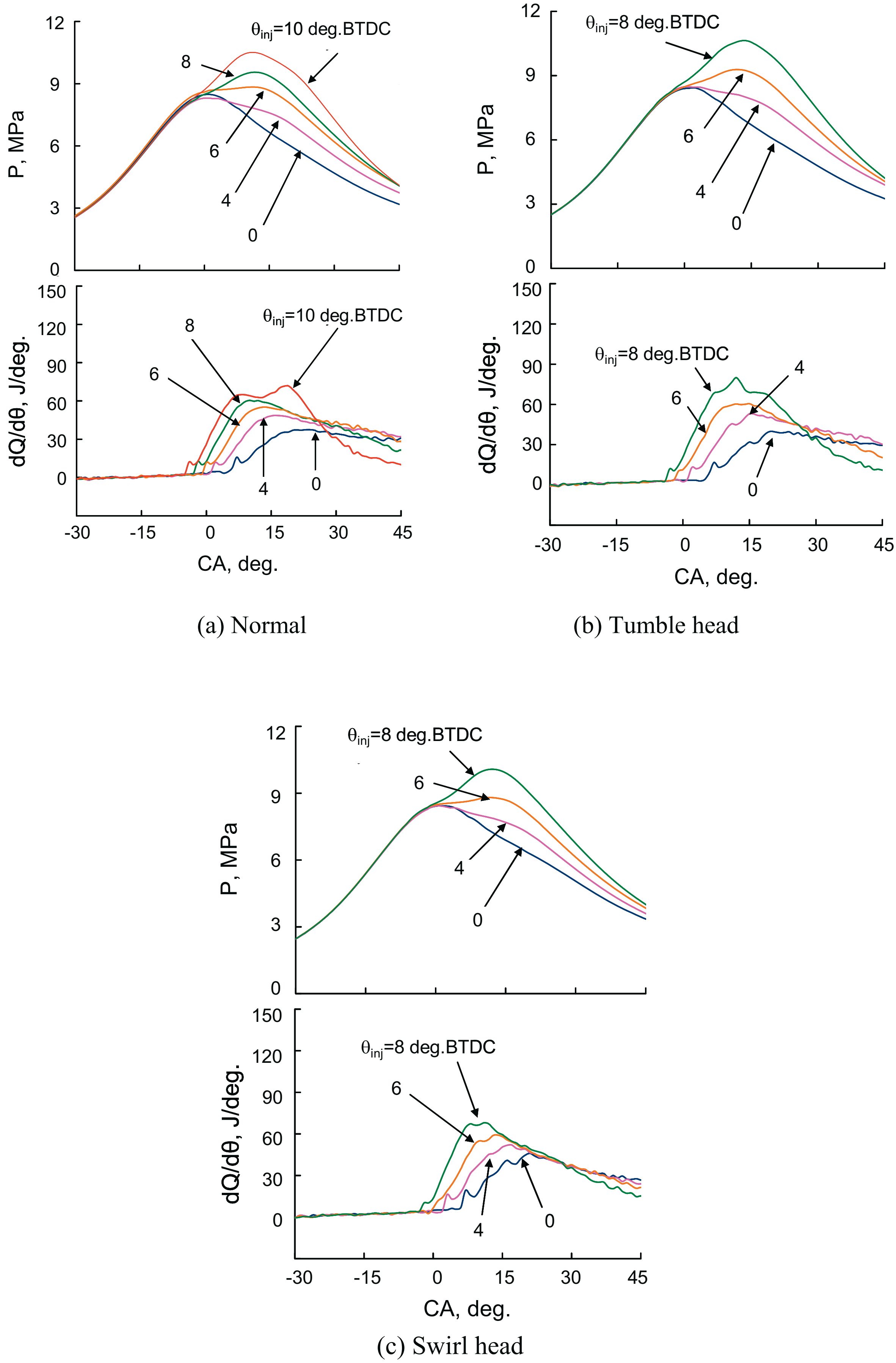


*Fig. 4. Pressure history and rate of heat release*

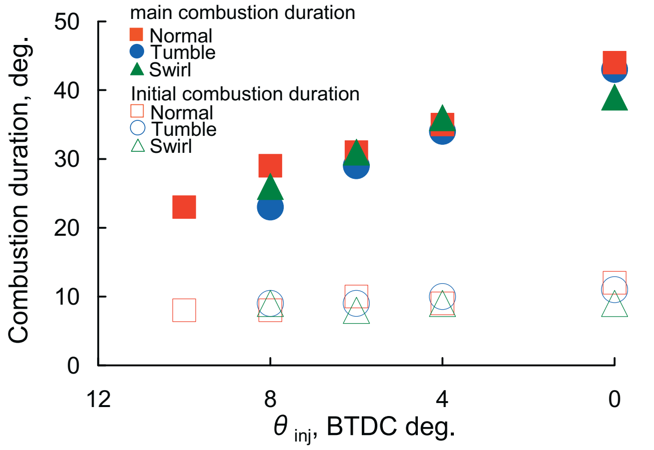


*Fig. 5. Combustion duration (Initial combustion and main combustion durations)*

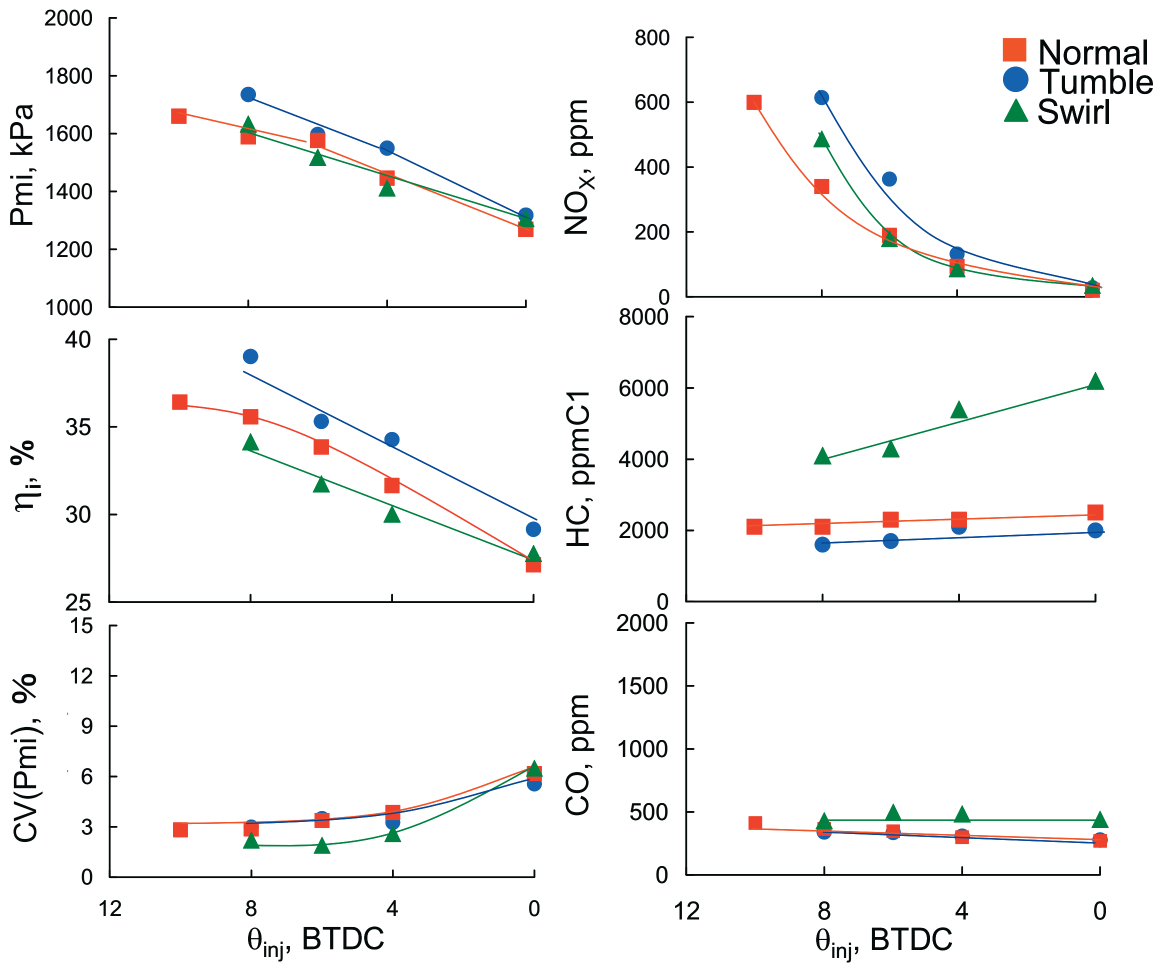


*Fig. 6. Engine performance (indicated mean effective pressure, its coefficient of variance and indicated thermal efficiency) and exhaust emissions (NOx, HC and CO) under the condition of equivalence ratio of 0.6 for normal, tumble and swirl heads*

Fig. 6 shows engine performance such as indicated mean effective pressure, its coefficient of variance and indicated thermal efficiency, and exhaust emissions such as NOx, HC and CO under the condition of equivalence ratio of 0.6 for three gas flow cases. At the same injection timing, indicated mean effective pressure and indicated thermal efficiency were the best within three gas flow cases. However, NOx emissions were also highest within the three. The HC emissions in tumble case were a little bit smaller than the normal one. On the other hand, for the swirl case, although the combustion was the most stable, the indicated mean effective pressure and the indicated thermal efficiency was not good. In addition, HC and CO emissions were also higher than other two cases. This is because the heat loss increased due to the swirl flow with higher velocity of the gas flow near the cylinder wall. The flame development near the cylinder wall was weak and the flame may quenches near the wall.

## 5. Discussion - Leaner condition

As indicated in the previous chapter, the swirl flow was not so good that the tumble flowand the normal flow were compared in detail. Fig. 7 shows pressure history and rate of heat release in leaner natural gas-air mixture for the normal and tumble cases. The equivalence ratio, $\phi_t$, was 0.59. And two-stage combustion occurred in advanced injection timing. In the tumble case, the injection timing could be advanced to 10.5 deg. BTDC while the injection timing could be advanced to 10 deg. BTDC for normal flow case. Fig. 8 presents indicated mean effective pressure, $P_{mi}$, indicated thermal efficiency and NOx emissions under the condition of equivalence ratio of 0.59, 0.61 and 0.62. When equivalence ratio decreased, the thermal efficiency increased due to lean combustion. And when the injection timing was advanced, the indicated mean effective pressure, indicated thermal efficiency and NOx increased. The mean effective pressure was almost the same for the tumble and normal cases. In equivalence ratio of 0.62, NOx emissions showed higher in tumble case. However, for leaner mixture, NOx emissions decreased in equivalence ratio of 0.59. Furthermore, the injection timing in tumble case could be advanced more than that in normal case. This means that higher engine performance can be achieved with lower NOx owing to the air flow of tumble. The swirl does not tend to decay soon, so that larger heat loss leads to lower thermal efficiency and larger HC and CO emissions. However, tumble motion decays near the TDC and does not affect the larger heat loss around the end of combustion.

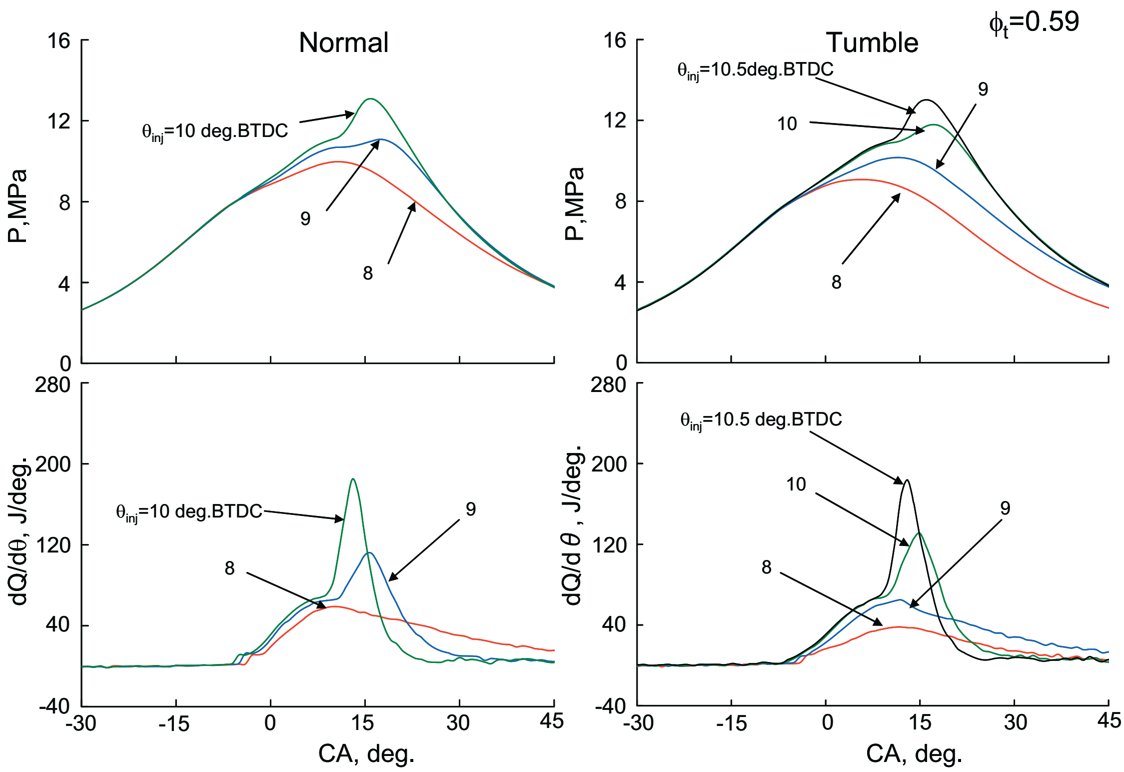


*Fig. 7. Pressure history and rate of heat release for normal and tumble cases ($\phi_t$ = 0.59)*

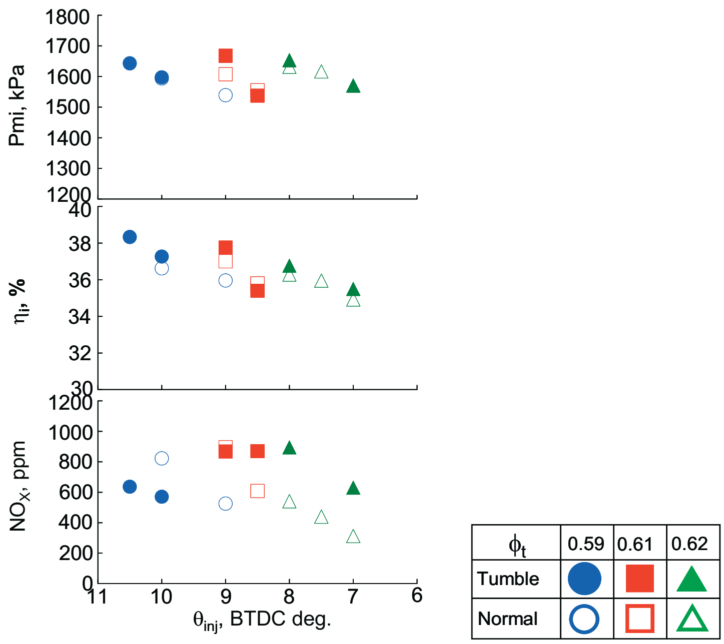


*Fig. 8. Indicated mean effective pressure, Pmi, indicated thermal efficiency and NOx emissions*

## 6. Summary

In this study, the effect of gas flow on the combustion and exhaust emissions for a single cylinder test engine was investigated with lean burn and supercharged conditions. Three kinds of the cylinder heads, normal, tumble and swirl heads, were prepared after the analysis with simulation. In swirl case, the turbulent energy was almost the same as that in the normal flow. The engine showed lower indicated thermal efficiency and higher HC emissions due to heat loss. In tumble case, the turbulent energy near TDC increased. Under leaner condition, the validity of tumble on engine performance and NOx emissions were confirmed. For example, in the equivalence ratio of 0.59, NOx emissions decreased and indicated mean effective pressure and indicated thermal efficiency increased. Under richer fuel-air mixture condition, the effect of the tumble became less.

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