

Research and Development of Controlled Auto-Ignition (CAI) Combustion in a 4-Stroke Multi-Cylinder Gasoline Engine

Jian Li, Hua Zhao, Nicos Ladommatos
Brunel University UK

Tom Ma
Ford Motor Company UK

ABSTRACT

Controlled Auto-Ignition (CAI) combustion has been achieved in a production type 4-stroke multi-cylinder gasoline engine. The engine was based on a Ford 1.7L Zetec-SE 16V engine with a compression ratio of 10.3, using substantially standard components modified only in design dimensions to control the gas exchange process in order to significantly increase the trapped residuals. The engine was also equipped with Variable Cam Timing (VCT) on both the intake and exhaust camshafts. It was found that the largely increased trapped residuals alone were sufficient to achieve CAI in this engine and with VCT, a range of loads between 0.5 and 4 bar BMEP and engine speeds between 1000 and 3500 rpm were mapped for CAI fuel consumption and exhaust emissions. The measured CAI results were compared with those of Spark Ignition (SI) combustion in the same engine but with standard camshafts at the same speeds and loads. The comparison showed more than 30% reduction in BSFC and up to 99% reduction in NO_x at low loads.

INTRODUCTION

Controlled Auto Ignition (CAI) combustion, also known as Homogeneous Charge Compression Ignition (HCCI), since it was first proposed in late 1970s [1,2], has been intensively studied in recent few years. It has been found that CAI combustion has low cyclic variations [3], better efficiency and fuel economy, and much less NO_x emissions [4,5] comparing with SI combustion. While gasoline has been the fuel used in the majority of the published papers since the start of the studies of CAI combustion and would be the fuel in the future CAI combustion engine, other fuels such as methanol [6], natural gas [7, 8] and primary reference fuels [5] were also used in fundamental CAI combustion investigations. Recently diesel CAI combustion was studied [9, 10], to hopefully take advantage of the ultra-low NO_x emissions of CAI combustion and reduce or even eliminate the problem of high particulate emissions associated with conventional diesel CI engines.

CAI combustion is a new combustion process different from both gasoline SI and diesel CI combustion. During CAI combustion premixed fuel and air mixture is compressed to a temperature so high (usually 1000 K or more) that auto ignition happens. Although of the similar auto ignition behaviour, CAI combustion is different from the knocking combustion that is characterised by violent heat releases and pressure rise rates. Its reaction rate is controlled and much slower thanks to the diluent (extra air and/or burnt gas) present in the mixture.

Once the auto ignition is initiated the reaction rate is not difficult to control, because the required amount of diluent (extra air and/or burnt gas) can be easily added in the charge during the mixture preparation process. However, with a fixed, low compression ratio and ambient inlet conditions, it is difficult to compress the charge to the very high auto ignition temperature in conventional SI engines, unless a large amount of hot residuals are trapped in the cylinders. In fact, to reach the high auto ignition temperature and initiate the CAI combustion is one of the major difficulties for the application of CAI combustion in a normal gasoline engine.

There are several methods to achieve the high auto ignition temperature. One of them is using hot residual gas in the cylinder. Mixing the hot residual gas with fresh charge can increase the charge temperature, so the charge can be compressed to a temperature high enough to initiate the auto ignition. Also the residual gas is a very effective diluent to reduce the reaction rate once the combustion is initiated. This method has been successfully used in 2-stroke engines [11,12] because it is not very difficult to trap a large amount of hot residual gas in the cylinder in 2-stroke engines due to their intrinsic scavenging characteristics.

There are two important factors when the hot residual gas is used to initiate CAI combustion [4]: (1) there must be a large amount of residual gas and;(2) the residuals must be hot enough (cold external EGR is not helpful for the auto ignition initiation). Otherwise the high auto ignition temperature will not be achieved under subsequent compression. However, in conventional 4-stroke engines the maximum amount of internal residuals is usually about

10-20% which is only a small fraction of the required amount to initiate CAI. It is extremely difficult to trap larger amount of hot internal residuals in a conventional 4-stroke engine with standard camshafts. Therefore, other methods have been employed to initiate CAI combustion in 4-stroke gasoline engines. One of the most popular ways is to heat the intake air. Thring [13] heated the intake air up to 400° C at a compression ratio of 8 and obtained a CAI combustion range at different EGR rates and air fuel ratios. Increasing compression ratio is a useful method and Christensen et al [14] found that the preheated charge temperature required for CAI can be reduced when the compression ratio increases. They even obtained CAI combustion without heating the intake air (at 25° C) at compression ratio 22.5. Aoyama et al [15] found supercharging is also helpful and managed to achieve CAI combustion in a engine with a compression ratio of 17.6 at 29° C intake temperature and about 1.28 bar intake pressure.

Unfortunately the above methods such as heating intake air and substantially increasing compression ratio are extremely difficult to be used in a practical engine. Exhaust gas through heat exchanger alone may not be able to heat the intake air up to the required temperature. And the electric heater not only increases the cost but also reduces the total efficiency as extra electric power is needed to drive the heater. Also heat inertia of the intake system may cause problems at transitions between CAI with intake heating and ambient inlet SI. High compression ratio is favourable to initiate CAI combustion but it may cause knock when the engine runs SI at high loads.

Therefore the way used in 2-stroke engines, of using internal residual gas, also seems to be the best way for 4-stroke engines. To substantially increase the amount of trapped residuals, unorthodox camshaft configurations [4] and even electro-hydraulic valves [16] were employed and CAI combustion was successfully achieved in single cylinder 4-stroke engines without any other methods such as heating intake air and increasing compression ratio. In this paper camshaft configurations similar to those in Lavy et al [4] were used in a production type 4-stroke multi cylinder engine equipped with Variable Cam Timing (VCT) systems on both intake and exhaust camshafts. In this engine CAI combustion was achieved by only using special camshafts and VCT system, and a speed and load range for CAI combustion was obtained at lambda 1. The CAI fuel consumption and engine-out emissions were measured and compared with those of SI combustion. Also at some conditions the air fuel ratio was changed and the lean limit and the effects of lambda on CAI combustion were investigated.

ENGINE SETUP

Table 1 Engine specifications

Bore (mm)	80
-----------	----

Stroke (mm)	83.5
Displacement (cm ³)	1679
Compression ratio	10.3

The engine used in this work was a Ford 1.7L Zetec-SE 16V four-cylinder, multi-point port injection production engine. Its specifications are listed in Table 1. The fuel used was a commercial unleaded gasoline of RON 95 complying with the British standard BS EN 228. Apart from the standard VCT system on intake side in this engine, another VCT system was cobbled onto the exhaust side to give the ability to adjust exhaust cam timing when the engine was running, which was proven to be very important to achieve CAI combustion. Another major modification to this engine is that special camshafts were used to restrict the gas exchange process in order to significantly increase the trapped residuals [17]. A Webcon Alpha-Plus ECU system, whose strategies are based on engine speed and throttle angle, was used to control the fuelling and ignition. Since all the tests were running at wide open throttle conditions, a dummy throttle was used and the signal from the dummy potentiometer was fed into the ECU to obtain the required fuelling. Also in this system there are facilities to finely adjust the fuelling and ignition timing manually. A sequential fuelling facility in this system is essential to provide the same fuelling events, hence same fuel preparation, for each of the four cylinders. An UEGO sensor was used to monitor the air fuel ratio and fine fuelling adjustment was made manually if necessary to maintain the required lambda.

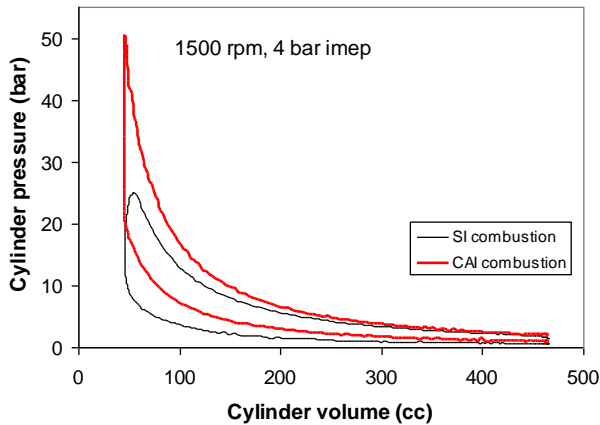
An eddy-current dynamometer was employed to set the engine running at a desired speed, and a load cell was used to obtain the output torque. NOx and unburnt hydrocarbons were measured by using a SIGNAL heated vacuum NOx analyser and a SIGNAL FID total hydrocarbon analyser respectively. CO and CO₂ were measured by an OLIVER K650 MOT analyser.

A pressure transducer was installed in one of the four cylinders to measure the in-cylinder pressure. The signal from this pressure transducer was fed into a PC based data acquisition system which ran a real-time analysis program. The program generated an array of pressure traces against crank angle, and from the pressure records it calculated the heat release data, indicated mean effective pressure (IMEP) and coefficient of variation of IMEP. Four thermocouples were installed in each of the four exhaust ports to monitor the exhaust temperatures and detect misfire should it happen.

A 10 Hz pulse generator, whose pulse width can be continuously varied, was built to control the duty cycle of the solenoid valves in the VCT system, therefore to change the cam timings.

EXPERIMENTAL PRECEDURE

Figure 1 Typical pressure traces of SI and CAI



combustion

Cam timings, which would give as little residuals as possible, were chosen to start the engine by spark ignition. These timings resulted in a relatively small amount of residual gas in the cylinder and caused no problem to start the engine with SI combustion. After one or two minutes the engine was moderately warm and CAI combustion appeared, but it was unstable, i.e. CAI and SI combustion occurred intermittently for several cycles. Misfire was observed occasionally during this period. A few minutes later when the coolant temperature reached about 50 – 60° C CAI combustion stabilised. However to minimise the effect of coolant temperature on CAI combustion all the tests were conducted when the coolant temperature was 90° C or over.

There were two ways to determine if CAI combustion was actually happening in the cylinders. One was to disconnect the ignition coil hence there was no spark in the cylinders. If the ignition coil was disconnected and the engine kept running it proved that the combustion in the cylinders was CAI. The other way was to check the in-cylinder pressure trace. Pressure records of CAI combustion, which usually displayed a distinctively sharp rise and much higher magnitude near TDC, were different from those of SI combustion (Figure 1). The latter way was usually used as the spark was kept on in the tests, and in the case that a CAI misfire happened the spark could ignite the mixture in the subsequent cycles. Also keeping the spark on in CAI mode did not cause any problems. Actually the spark had little effect on CAI combustion, changing the spark timing substantially showed virtually no effects on combustion.

During the tests the throttle was kept at wide open and the air flow was changed by varying the cam timings, which can be continuously changed by up to 40 degrees crank angle.

RESULTS AND DISCUSSIONS

There are several parameters that could be varied in the tests, for example cam timings, air fuel ratio, engine speed

and load etc. Some of parameters were fixed while looking at the effects of other parameters. At first, the cam timings were fixed, which means that the air flow was kept unchanged because the throttle was always at wide open conditions and the cam timings were used to control the air flow, and the effects of air fuel ratio were investigated. By changing the air fuel ratio at a given air flow rate the BMEP was varied. In later tests the air fuel ratio was fixed at lambda 1 and the cam timings were varied to obtain different BMEP at each speed from 1000 to 3500 rpm.

TESTS WITH DIFFERENT AIR FUEL RATIOS – In these tests, the cam timings (IVO at 110 CA after TDC and EVC at 80 CA before TDC) were kept constant and the air fuel ratio was changed. At each speed, the air fuel ratio was set at slightly rich (about lambda 0.98), and then it was gradually increased until the mixture was too lean to sustain CAI combustion (the engine stalled). Figure 2 shows the air fuel ratio range and BMEP at different speeds for CAI combustion. Also showed in this figure are BSFC, exhaust temperatures and emissions in these tests.

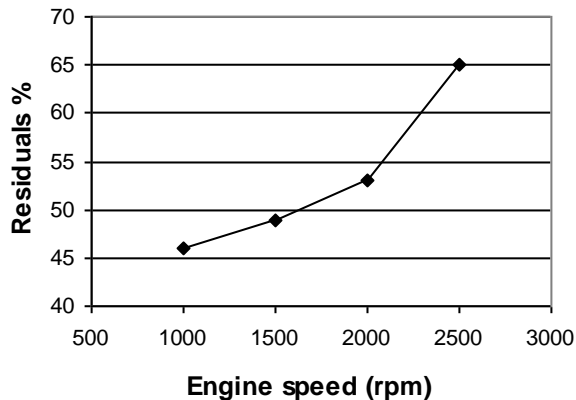
At the lowest speed 1000 rpm, maximum lambda achieved is only about 1.14. At 1500 rpm, it managed to reach about 1.27. As speed increases the maximum lambda decreases. One might think that lambda 1.27 is not very lean compared with the air fuel ratio achieved with CAI by other investigators [5,13]. However, there was a large amount of residuals in the cylinders, therefore the charge was still highly diluted. The rich side of lambda was not fully investigated because it would loss efficiency. Also, it was found in the experiment that CAI combustion is less tolerant with mixture enrichment than SI combustion. At 2000 rpm CAI misfire occurred at lambda 0.95. At 2500 rpm, the mixture could only be as rich as lambda 1. Further enrichment resulted in unstable combustion and misfire. For speeds from 1000 to 1500 rpm, the rich limit was not fully investigated although CAI combustion at these speeds was more tolerant to enrichment than at high speeds. The BMEP was changed while varying air fuel ratio. Because the air flow did not change and the variation range for air fuel ratio is small, the BMEP did not change very much, only from about 1.3 bar to 3.5 bar.

As lambda increases the exhaust temperatures decrease owing to the reduction in fuelling. BSFC decreases a little as the mixture changes from slightly rich to slightly lean. And it then increases with lambda. It can be seen that at slightly lean conditions (about lambda 1.05) BSFC displays minimum values.

The trends of emission components, NO_x, HC and CO, varying with lambda are similar to those of SI combustion. NO_x peaks at about lambda 1.05. As lambda changes from rich to stoichiometric and lean, CO decreases dramatically. It shows little change as lambda further increases and finally it slightly increases near the lean limit. HC emissions increase as the mixture becomes leaner indicating that the combustion becomes worse.

Figure 3 Residual gas percentage at fixed cam timings

Figure 3 shows the residual gas percentage at different speeds for the fixed cam timings. At 1000 rpm the residual is about 46%, the lowest value in the speed range, but it is



still enormously higher than the residual rate in SI engines which is about 20% at most. As the engine speed goes up, the residual rate increases due to less time for gas exchanges. At 2500 rpm the residual rate reaches 65%. The values in Figure 3 are only for these specific cam timings. At different cam timings, i.e. different air flow rate since the air flow rate is controlled by the cam timings, the residual rate is different even for the same speed. In fact, at 2500 rpm the maximum residual rate observed for CAI combustion was as high as 75%.

It can be seen from figures 2 and 3 that at lower speeds, 1000 and 1500 rpm, less residuals were trapped in the cylinders and hence higher BMEP was achieved. However, NO_x emissions at these lower speeds are much higher at near stoichiometric conditions, indicating that the dilution rate was not large enough and higher residual percentage is needed to reduce NO_x emissions. Therefore it is necessary to maintain higher residual rates while increasing the BMEP. Otherwise NO_x emissions will largely increase and one of the major benefits of CAI combustion will be lost. The high load condition with the special camshafts could be achieved by supercharging, while using external EGR, or raising exhaust back pressure hence increasing trapped residuals, could maintain the high dilution rate required for low NO_x productions.

It can be concluded that slightly lean mixtures at about lambda 1.05 give rise to minimum values of the fuel consumption BSFC and CO and HC emissions, and maximum NO_x emissions. But even the maximum NO_x emissions are only a small fraction of those of SI combustion. At lambda 1, the BSFC, CO and HC emissions are close to the minimum figures, and NO_x emissions are slightly lower than peak values. Also if the engine runs at stoichiometric conditions, all the facilities existing in the current SI engines such as lambda control and three way catalysts can be directly utilised in the CAI engines without modifications. So tests at lambda 1 were conducted to investigate the speed and load range of CAI combustion and emissions.

RESULTS OF CAI COMBUSTION AT LAMBDA 1 – At lambda 1 engine tests were conducted at different engine speeds and loads. For a given speed the load was varied by changing the cam timings. Figure 4 shows the speed and load range within which CAI combustion was achieved in the engine. The lowest speed tested was 1000 rpm. Any speed lower than that was considered not typical for a real engine (unless at idle) hence not investigated. The maximum speed in figure 4 is 3500 rpm. In fact at a speed higher than 3500 rpm CAI was achieved, but only in a tiny load range, therefore it is not included. At each speed there is a maximum and a minimum load. The upper limit is a result of restrictions of gas exchange. At those conditions the air flow rate is the maximum the engine can intake with the special cam timings. Knock did not happen normally even at the upper limit. Only at 1000 rpm suspected knock was observed rarely at the maximum load. The lower load end is limited by misfire. At these conditions there were too much residual gas and its temperature is very low (figure 5). The mixture could neither be auto ignited (temperature too low), nor spark ignited (too much residuals). As speed increases the maximum load reduces due to the fact that at higher speed, there is less time for gas exchange process hence more residuals trapped in the cylinders and less fresh charge is inducted.

Figure 5 shows the exhaust temperature across the CAI speed and load range. The CAI exhaust temperature varies with speed and load similarly as the SI exhaust temperature does. An interesting thing in figure 5 is that no matter what the engine speed is, the exhaust temperature at the low load end is about 300 to 350° C. This suggests that it is the exhaust gas temperature limits the low load and if the exhaust gas is cooler there is not enough energy in the exhaust gas to heat up the mixture to the required auto ignition temperatures.

Figure 6 shows the brake specific fuel consumption across the CAI range. BSFC mainly depends on load and does not change very much with engine speed. For the same speed BSFC reduces as the load increases. But for the same load at different speeds BSFC shows little change, especially at low load conditions.

Figures 7 – 9 give the NO_x, CO and HC emissions of CAI combustion in the speed and load range. For a given speed, as load increases NO_x increases, CO and HC decrease as expected. The variations of each of the three emission components with speeds are different. NO_x does not change very much with speed at low and high speeds, but increases with speed in the mid-speed range. Speed hardly shows effects on CO, especially at high speeds. HC decreases as speed increases suggesting that combustion at high speeds is better and more complete. The reasons why the three emission components behave like this in the speed and load range are not very clear and detailed combustion features of CAI need to be investigated. It is thought that analysis about the residual gas fraction and combustion details would be helpful to better understand the behaviour of BSFC and emissions. The analysis will be conducted in the near future in another paper.

COMPARISON BETWEEN CAI AND SI RESULTS – To study how the performance of a CAI combustion engine is relative to that of an SI engine, the measured CAI results were compared with those of SI combustion in the same engine with standard camshafts at the same speeds and loads

Figure 10 shows the percentage change of CAI BSFC relative to SI BSFC. In the whole CAI speed and load range, there is a BSFC improvement for CAI. At higher loads, the fuel economy improvement is small, only about 5 –10% at 2.5 bar BMEP. As the load decreases the improvement becomes significant and it reaches about 30% at 1 bar BMEP. Further fuel consumption reduction is achieved at lower loads. The better fuel consumption comes from two sources. The first is that the engine is unthrottled with CAI combustion so there is a reduction in pumping work. This effect is much more important at low loads because at these conditions SI engines are usually highly throttled, the reduction in pumping work is enormous for unthrottled CAI combustion. At higher loads the throttling of SI engines is reduced and less improvement in BSFC is observed. The other reason for BSFC improvement is that during CAI combustion the sharp pressure rise occurs near TDC and it makes CAI combustion look like constant-volume combustion, which is the most efficient process. In figure 1 the typical P-V diagram of the power cycles of CAI and SI combustion is shown and the constant-volume like feature can be clearly seen on the CAI curve.

Figure 11 shows the percentage improvement in NOx emissions. In the CAI range 90 – 99% reduction in NOx emissions are observed. That means NOx emissions from CAI are only one hundredth to one tenth of those from SI combustion. This immense reduction is similar to the results of an effective catalyst. It may be argued that after treatment may not be necessary in this range for CAI combustion. The huge reduction in NOx emissions is a result of much reduced combustion temperatures. Due to the large amount of residual gas (especially at low loads) in the cylinders the typical temperatures of CAI combustion are around 1800 K while typical SI combustion temperatures are around 3000 K. In the much cooler CAI combustion there is hardly any NOx formed. This is true at low load conditions during which the readings from NOx analyser were close to figures of ambient air. At higher loads more energy is released during combustion and the temperature of the total mixture is higher, hence NOx is higher. But even at the upper end of the load the NOx from CAI is only about 10% of NOx from SI.

It can be seen from figure 12 that CAI combustion reduces CO emissions by 10 to 40% depending on speed and load. At low speed and high load conditions CO reduction is higher. However, HC emissions from CAI combustion are worse than in SI combustion. In figure 13, a 50% to 160% increase in HC is found for CAI combustion. That means HC emissions from CAI are 1.5 to 2.6 times of those from SI combustion. The reason for the HC increase is not absolutely clear. It might be due to the low CAI combustion temperatures. At lower loads the combustion temperatures are even lower and more HC increases were recorded.

However the actual cause of the HC emission increase needs to be further investigated.

CONCLUSIONS

CAI combustion has been achieved in a production type 4-stroke multi-cylinder gasoline engine without any special methods such as intake air heating and raising compression ratios. CAI combustion was achieved by utilising a large amount of trapped hot residual gas in the cylinders to heat the mixture. Relevant modifications to the engine (special cams and dual VCT) were made to obtain the required amount of residuals. The following conclusions can be drawn from the experimental results.

1. Using hot residuals gas is an effective and feasible way to initiate CAI combustion. The largely increased hot residuals alone, whose rate at one set of fixed cam timings was 46 to 65% in this engine, are sufficient to achieve CAI combustion.
2. In this engine the maximum air fuel ratio achieved for CAI combustion is about λ 1.27. For different speeds the lean limit varies slightly. At slightly lean conditions (λ 1.05) BSFC, CO and HC emissions have minimum values while NOx emissions peak. Rich mixtures are not favourable to CAI combustion at high speeds.
3. At λ 1, a range of speed from 1000 to 3500 rpm and load from 0.5 to 4 bar BMEP was obtained for CAI combustion. In this range, the fuel consumption and exhaust emissions of CAI combustion were measured and mapped. The upper limit of the load is a result of restrictions of gas exchanges. Supercharge plus external EGR or raising exhaust back pressure could extend this limit. The lower end of the load is limited by misfire (residual temperatures too low). To extend the lower limit is more difficult and some other ways may be needed to raise the in-cylinder temperatures.
4. CAI combustion shows a 5 to more than 30% reduction in fuel consumption and 90 to 99% reduction in NOx emissions. CO emissions from CAI are slightly lower than those from SI combustion. But unburnt hydrocarbons from CAI combustion are 1.5 to 2.6 times of those from SI combustion.

ACKNOWLEDGEMENTS

This work is a part of the 4-SPACE project, and the authors would like to thank the European Commissions for its financial support to the 4-SPACE Consortium, and other partners in this Consortium for their collaboration.

REFERENCES

1. Onishi, S., Hong, S., Shoda, K., Do Jo, P., and Kato, S., Active Thermo Atmosphere Combustion (ATAC) – a new combustion process for internal combustion engines, SAE paper 790501, 1979.
2. Noguchi, M., Tanaka, Y., Tanaka, T. and Takeuchi, Y., A study on gasoline engine combustion by observation

- of intermediate reactive products during combustion, SAE paper 790840, 1979.
3. Iida, N., Ichikura, T., Kase, L. and Yoshiteru, E., Self-ignition and combustion stability in a methanol fuelled low heat rejection ceramic ATAC engine - analysis of cyclic variation at high wall temperatures and lean burn operation, JSAE review 18, pp233-240, 1997
 4. Lavy, W., Dabadie, J., Angelberger, C., Duret, P.(IFP), Willand, J., Juretzka, A., Schaflein, J.(Daimler-Chrysler), Ma, T. (Ford), Lendresse, Y., Satre, A. (PSA Peugeot Citroen), Schulz, C., Kramer, H. (PCI – Heidelberg University), Zhao, H., Damiano, L. (Brunel University), Innovative ultra-low NOx controlled auto-ignition combustion process for gasoline engines: the 4-SPACE project, SAE paper 2000-01-1837, 2000
 5. Oakley, A., Zhao, H., Ladommatos, N. and Ma, T., Experimental studies on controlled auto-ignition (CAI) combustion in a 4-stroke gasoline engine, International conference on 21st century emissions technology, C588/019/2000, IMechE, 2001
 6. Iida, N., Combustion analysis of methanol-fuelled active thermo-atmosphere combustion (ATAC) engine using a spectroscopic observation, SAE paper 940684, 1994
 7. Christensen M., Johansson, B. and Einewall, P., Homogeneous charge compression ignition (HCCI) using isooctane, ethanol and natural gas – a comparison with spark ignition operation, SAE paper 972874, 1997
 8. Chen, Z., Konno, M., Oguma, M. and Yanai, T. Experimental study of CI natural gas/DME homogeneous charge engine, SAE paper 2000-01-0329, 2000
 9. Ryan, T. and Callahan, T. Homogeneous charge compression ignition of diesel fuel, SAE paper 961160, 1996
 10. Kimura, S., Aoki, O., Kitahara, Y. and Aigoshizawa, E. Ultra-clean combustion technology combining a low-temperature and premixed combustion concept for meeting future emission standards, SAE paper 2001-01-0200, 2001
 11. Ishibashi, Y and Masahiko, A., Improving the exhaust emissions of 2-stroke engines by applying the activated radical combustion, SAE paper 960742, 1996
 12. Duret, P. and Venturi, S., Automotive calibration of the IAPAC fluid dynamically controlled 2-stroke combustion process, SAE paper 960363
 13. Thring, R., Homogeneous charge compression ignition (HCCI) engines, SAE paper 892068, 1989
 14. Christensen, M. and Hultqvist, A., Demonstrating the multi fuel capability of a homogeneous charge compression ignition engine with variable compression ratio, SAE paper 1999-01-3679, 1999
 15. Aoyama, T., Hattori, Y., Mizuta, J. and Sato, Y., An experimental study on premixed charge compression ignition gasoline engine, SAE paper 960081, 1996
 16. Law, D., Allen, J. Kemp, D. and Williams, P., 4-stroke active combustion (controlled auto-ignition) investigation using a single cylinder engine with Lotus active valve train (AVT), International conference on 21st century emissions technology, C588/006/2000, IMechE, 2000
 17. Duret, P. and Lavy, J., Process for controlling self-ignition in a 4-stroke engine, US patent 6 082 342

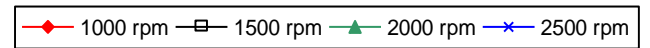
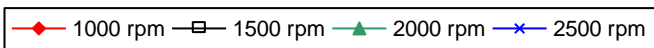
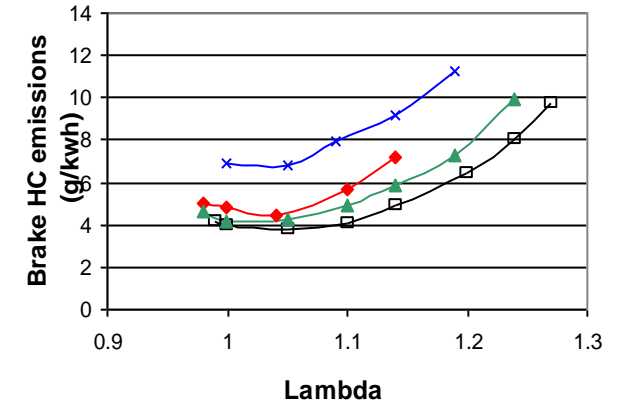
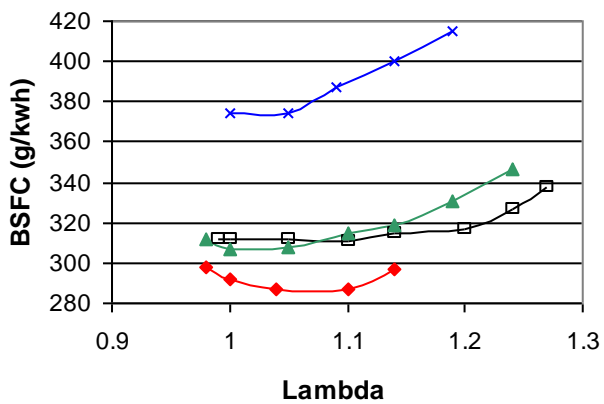
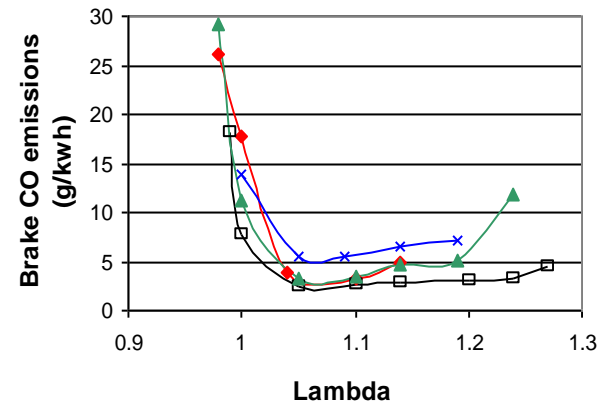
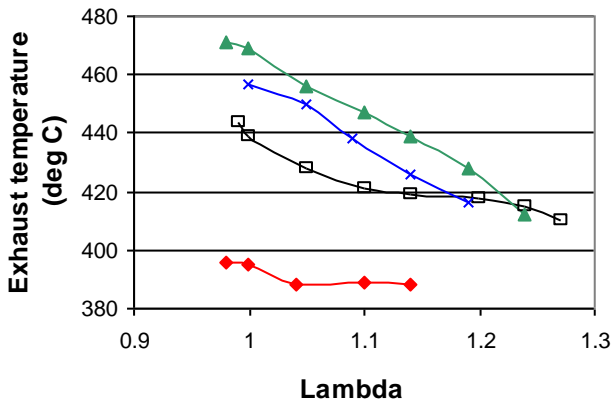
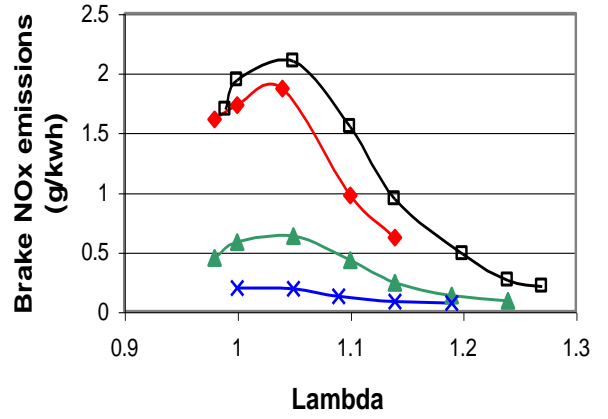
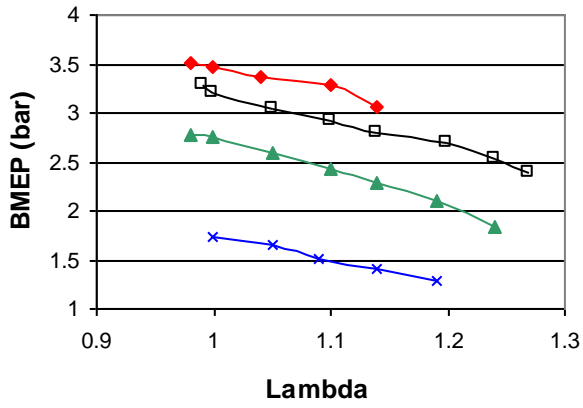


Figure 2 Results of CAI combustion at different lambda

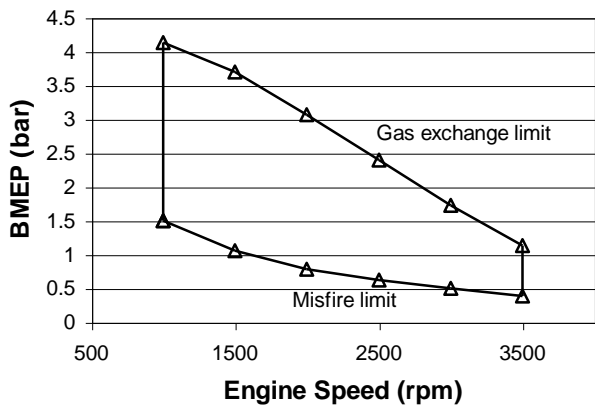


Figure 4 Speed and load range of CAI

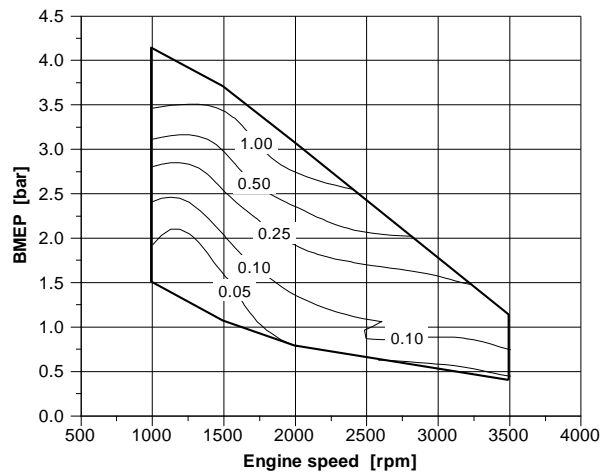


Figure 7 Brake NOx emissions of CAI

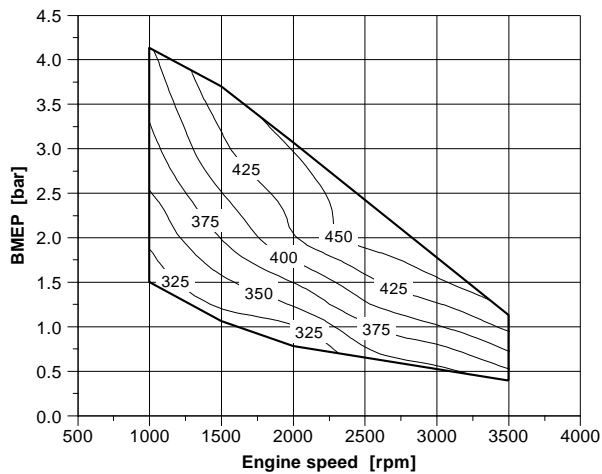


Figure 5 Exhaust temperature of CAI

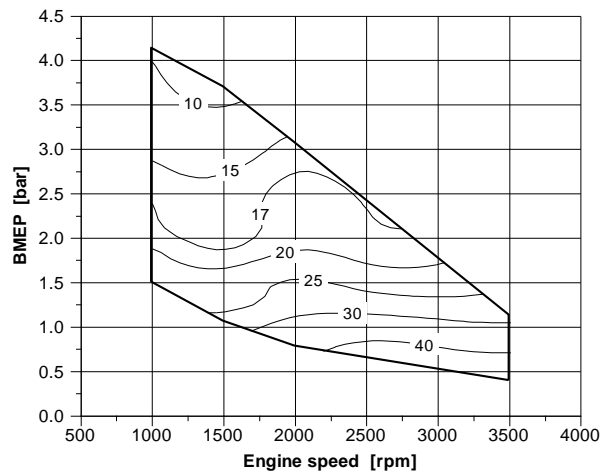


Figure 8 Brake CO emissions of CAI

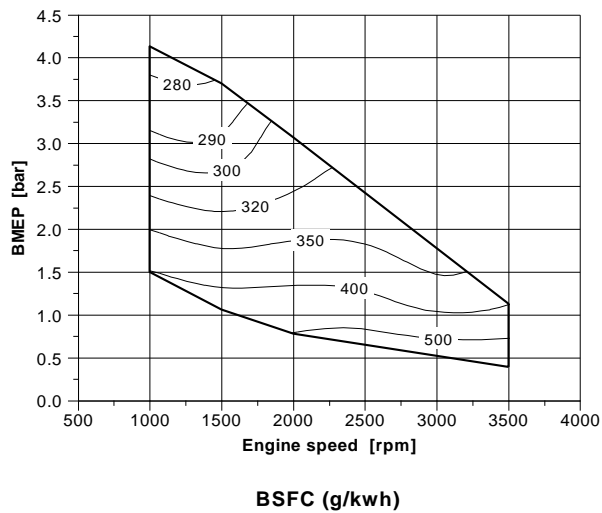


Figure 6 Fuel consumption of CAI

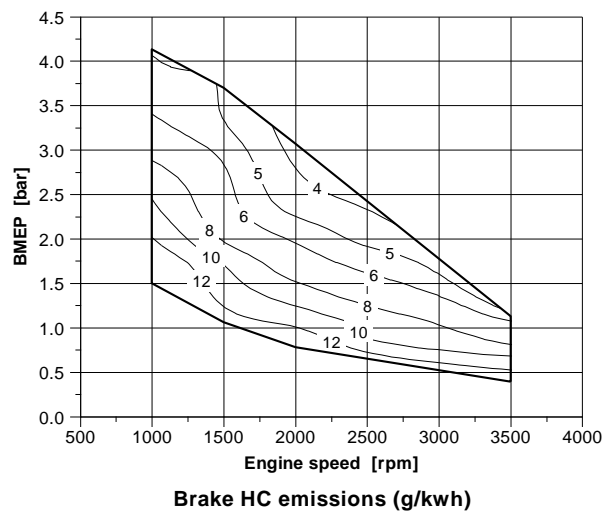
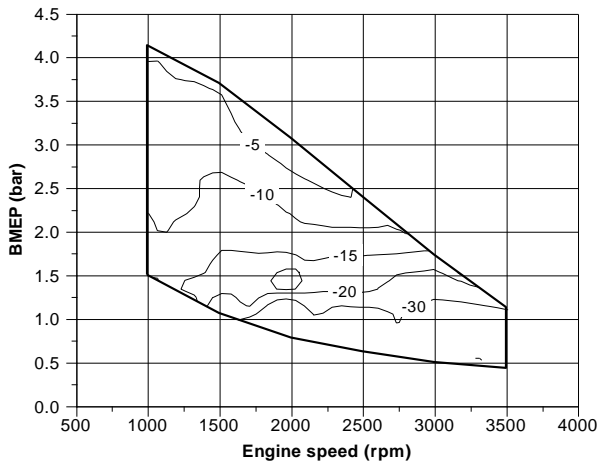
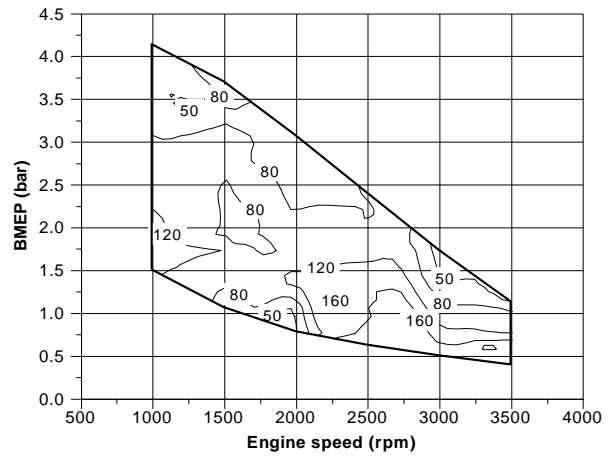


Figure 9 Brake HC emissions of CAI



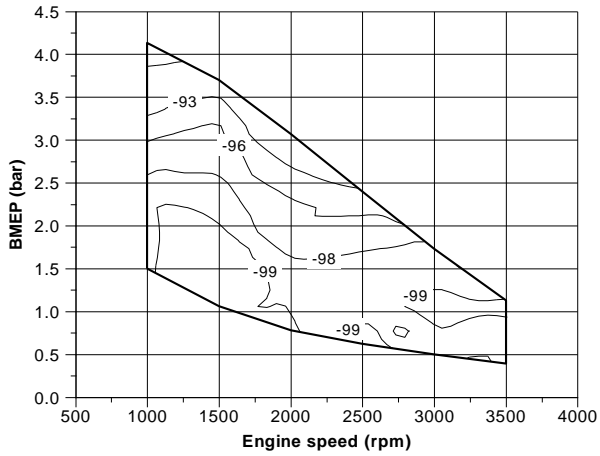
Percentage change of BSFC

Figure 10 Percentage change of BSFC



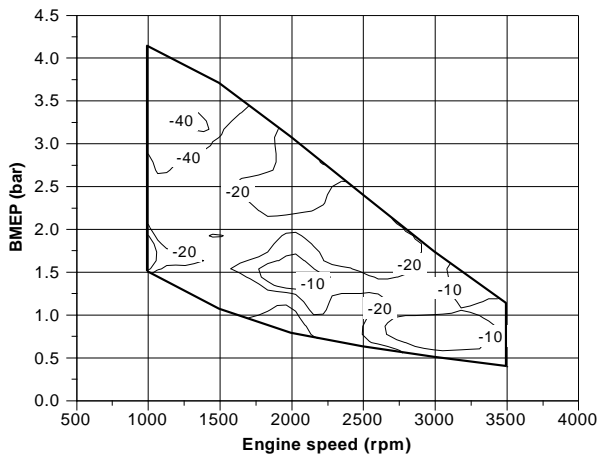
Percentage change of brake HC emissions

Figure 13 Percentage change of brake HC emissions



Percentage change of brake NOx emissions

Figure 11 Percentage change of brake NOx emissions



Percentage change of brake CO emissions

Figure 12 Percentage change of brake CO emissions