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Review of Controlled Auto Ignition (CAI) Combustion Systems

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Abstract

The total name of CAI is Controlled Auto-Ignition, which belongs to a new type of combustion that is different from traditional engines. So far, this technology has been studied by experts from all over the past 30 years. They have experimented and simulated from different aspects of combustion, and those processes and conclusions have finally published in SAE or other website and journals. Therefore, the main purpose of this paper is to read a large number of thesis that related to CAI and HCCI petrol engine, and clearly grasp CAI specific research aspects of the past 20 years, such as different experimental engine, valve strategy, operating technology, load range, emission pollution and so on. Finally, gathering their results in this paper.

There will be different engine operation and software simulation steps during different CAI combustion experiments. Even with the same engine and software, different experts and researchers will operate and explore different contents and processes. This article finds that what the difference among them when so many people are studying CAI's combustion. Different people under the study will get what kinds of results though with the same engine or simulation software (software models have already been built in enterprise).

Finally, the focus of this paper is to emphasize CAI combustion technology which is only applicable on petrol engines, while HCCI is only suitable for diesel engines. The differences between them are mainly due to the realization of HCCI on diesel engine is necessary to remove the water

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vapor that exists in intake and residual gas by a heater, but CAI does not need, which is also the nature of diesel and petrol determine the different.

Keywords: *Controlled auto ignition (CAI)*

I. INTRODUCTION

1.1 Background of CAI and HCCI

From the last century until now, this technology has been studied and updated many times, and therefore, during the long time, CAI as a name corresponds to different engine strokes and there are many different abbreviations as shown in table 1.1 and 1.2.

Table 1.1 CAI's Abbreviations on Two-stroke Engine [12]

Name	References
Active Thermo-Atmosphere Combustion (ATAC)	[1]
Toyota-Soken Combustion (TS)	[2]
Activated Radical Combustion (AR)	[5]
Compressed Air Assisted Fuel Injection Process (IAPAC)	[6]

Table 1.2 CAI's Abbreviations on Four-stroke Engine [12]

Name	References
Compression-Ignited Homogenous Charge (CIHI)	[3]
Homogeneous Charge Compression Ignition (HCCI)	[4]
Premixed-Charge Compression Ignition (PCCI/CAI)	[7]
Diesel Combustion under Uniform Higher-Dispersed Mixture Formation (UNIBUS)	[8]
Active Combustion (AC)	[10]

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1.2 The Definition of CAI

CAI, also known as controlled auto-ignition, is a form of combustion that automatically ignites the homogeneous fuel/air mixture by controlling the amount of residual exhaust gas inside the cylinder. Since the temperature in cylinder is greater than the spontaneous combustion temperature in general chemical state of the reactants at the end of compression stroke, the mixture is compressed to cause its ignition during the combustion process, and this ignition condition is carried out at multiple locations of the mixture, so that the overall reaction temperature and pressure are uniform to a certain extent, illustrated in figure 1.1.

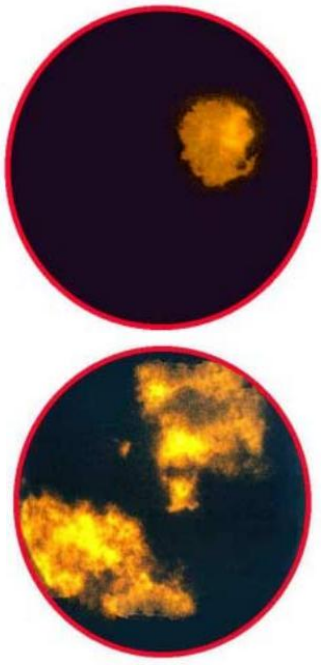


Figure 1.1 Single Ignition of Spark Plug and Multi-Point Ignition [9]

Compared with SI and CI engines, this kind of combustion can drop fuel consumption, significantly reduce NOX and HC, but also slightly decrease CO and CO2, which is more in line with the advocacy of environmental protection and energy conservation conditions, and also is the future trend of energy combustion technology.

CAI combustion can be achieved by the crucial point is the ability to trap residual exhaust gas. The residual exhaust gas, whether its own maintained temperature or internal active chemical substances contained could initiate and promote the combustion and make it controllable. Law et al. used Lotus engine for the first time in 2000 to propose two methods for trapping residual exhaust gas: sequential and simultaneous, which are showed in figure 1.2 and 1.3 respectively.

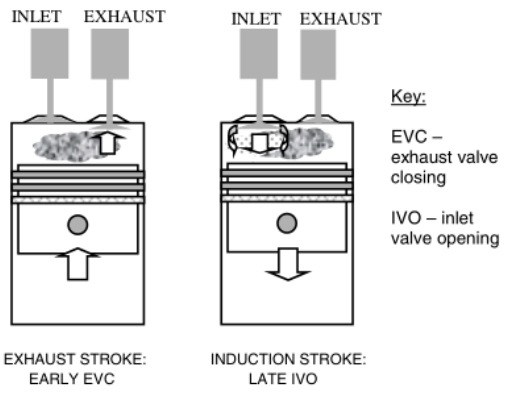


Figure 1.2 The Operating Principle of Method 1. [11]

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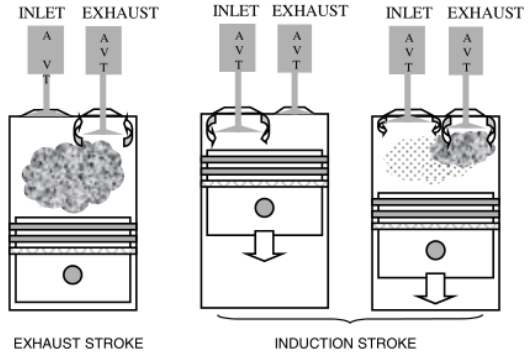


Figure 1.3 The Operating Principle of Method 2. [11]

The first strategy is to close the exhaust valve relatively early in the exhaust process and then open the intake valve later when in the suck stroke, which could capture a certain amount of exhaust gas. The second one is that when the piston reaches the bottom dead center (BDP) in the power stroke, the exhaust valves open, and all the exhaust gas is discharged to outside of cylinder and then close them. At the beginning of next intake stroke, the exhaust valves are opened at the same time with the intake valves, the fresh air and exhaust gas are mixed into the cylinder. This strategy is later known as negative overlap (NVO) technique. These two methods don't require preheating intake air, and the operation cycle is relatively simple and is generally applicable to four-stroke engines.

1.3 The Definition of HCCI

HCCI, the full name is homogeneous compression charge ignition, which is first time mentioned by Thring in the article using the diesel engine test. The fuel is injected through carburetor or inlet jet so that evenly mixed with air before into combustion chamber. It is a new combination of premixed and low temperature combustion. It is pointed out that there are two important preconditions to initiate HCCI combustion: high EGR rate (13%-33%) and intake temperature (370℃).

EGR, is the exhaust gas recirculation, the entire gas cycle process as shown in figure 1.4.

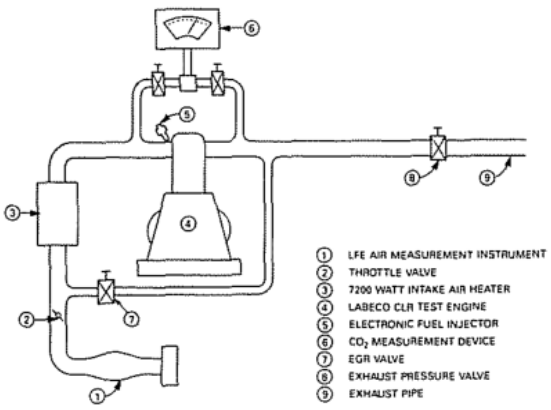


Figure 1.4 Exhaust Gas Recirculation [4]

The combustion of diesel engine is by compressing mixture, when the injector

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injects diesel into the cylinder, the fuel is acted as some liquid columns which are similar to that of the thin wires. However, the diesel fuel which is sprayed firstly due to the high temperature and pressure within the cylinder, is more easily vaporizing in advance than those close to the injector nozzle.

In other words, the first half of the liquid column sprayed will form water vapor while the second half is diesel, and coupled with the combustion process will produce a lot of water vapor and other gas and particles. After the exhaust gas is discharged outside from cylinder, part of which is recycled back into intake manifold and mixed with fresh air. The newly formed mixture is passed through a heater with the aim of heating them to reach the reaction temperature, followed by filtering the vapor of exhaust gas and moisture of the air to make the charge purer and also reduce the misfire problem of combustion. And meanwhile, due to the intake and exhaust will cause a certain amount of pumping losses, the use of EGR could reduce this loss to a certain extent.

Thring [1] stresses that the traditional spark ignition engine has a limited equivalent ratio and EGR, because if the EGR is too high or the equivalent ratio is too low, which also means that the concentration of mixed reactant is too thin. Firstly, it may lead to the inability to support the combustion flame front propagation and resulting in stopping combustion, and secondly, the energy generated by spark cannot provide the temperature and time requirements of the flame core during spark gap. However, neither of these causes restriction on HCCI, since the combustion of HCCI doesn't start from the spark plug, and there isn't only one flame front provided by the spark plug, but is burned evenly throughout the combustion chamber, the heat of combustion gas can be transferred from one point to the other. Even if the flame front is extinguished, but exothermic and temperature of cylinder gas is enough to make HCCI combustion continues.

1.4 The Definition FVVT System

The key to manipulate this combustion is the use of fully flexible valve (FVVT) system, which is primarily the active valve train (AVT) system. It is well known that the traditional sense of engine valve is carried by the larger torque of the camshaft, according to the valve timing to guide its opening and

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closing. Law et al. in the 2001 article pointed out that for CAI combustion of a four-stroke engine, due to the need to trap a certain amount of exhaust gas, the camshaft owns fixed profile limit and cannot be well grasped the time point of intake and exhaust, and ether cannot use the exhaust gas preheat fuel/air mixture. However, the AVT system could solve these problems, which not only with more functionality and flexibility, this technology has also largely studied advanced combustion and valve operation strategies, such as the better control of valve timing and gas mixing and exchange during combustion. This system is used in Lotus engine as shown in figure 1.5. After that, Koopmans and Wallesten published the articles also mentioned the use of Lotus AVT system in Volvo engine to control open and close of valves and thus controlling HCCI combustion.

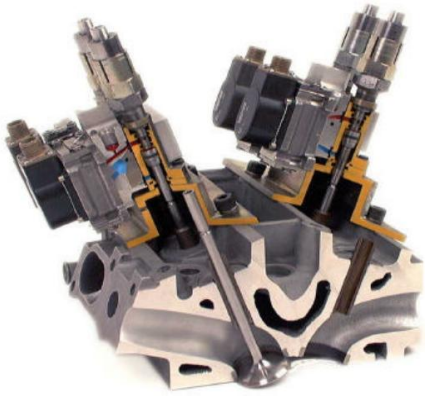


Figure 1.5 Lotus Active Valve Train (AVT) [9]

The principle of this system is to lift the upper and lower surfaces of popped valve through hydraulic pressure and drive intake and exhaust valves opening or closing, which are controlled independently. The entire control process is closed-loop operation, the crank sensor can be used to obtain the signal of crank angle degree, this signal would be transmitted to the linear displacement transducer that connected to the upper end of popped valve, giving instructions to adjust the size of valves' opening and closing, which is to control the number of intake air or residual gas. This part of the information and then feedback to the crank sensor, forming a set of loops. In order to meet the actual profiles of valves coincide with the theoretical position, this process can be designed by software.

1.5 Summarize of the First Part

Although many articles point out that CAI and HCCI are the same, actually, they are not. Indeed, CAI and HCCI are very similar in principle to combustion, both through compress exhaust air and air fuel mixture to achieve spontaneous of multi-point, and then burning in the cylinder and get good results in stable combustion, energy saving and emission reduction. However, many people ignore

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the most important point is that the two modes for the selection and utilization of exhaust gas. As mentioned above, the exhaust gas required for CAI combustion can be said to have not been completely discharged from the cylinder (even if the second method, it is not leave the cylinder too far). Thus, it left in cylinder by the means of trapping. However, the exhaust gas in HCCI mode is recycled and mixed with air before entering cylinder. The reason for the difference between two methods is determined by the self-attribute of the fuel. At the same high temperature, the vaporization property of diesel oil is obviously higher than that of the gasoline. This is why the diesel engine does not need spark plug, only through compression can make the mixture burning and working. This kind of exhaust gas with a large amount of vapor, which is likely to cause the engine fail to activate HCCI combustion if it is recovered directly to the cylinder rather than flow through the heater to filter the water and make it evaporate, but it is possible for gasoline engine. This is also the origin of EGR name: exhaust gas recirculation, that is to say EGR is not equal to exhaust capture.

Secondly, Thring mentioned that, for those engines with spark plugs, some restrictions on equivalence ratio and EGR can affect their normal combustion, but don't limit HCCI combustion. It can also be understood that as HCCI combustion can complete successfully which does not require spark plug assisted, even if there is misfire phenomenon during combustion. But the combustion of CAI will not continue, and many articles also indicate that CAI usually requires spark plug assistance, and CAI-SI-CAI mode is also the trend of many gasoline engines.

In summary, CAI combustion and HCCI combustion are different, which is that CAI is for gasoline engine and HCCI is for diesel one. This paper is mainly concerned with recent articles on CAI gasoline engine and a small amount of HCCI. And in the chronological order to make a literature review, Finally, the effects of CAI combustion on energy dissipation would be studied.

2. LITERATURE REVIEW

Law et al. [10] after the test of Lotus engine at 2000rpm and 2.0-5.0 bar IMEP, the CAI combustion is first defined and referred to as active combustion (AC). Two methods of trap exhaust gas and the use of FVVT mentioned in the text have been

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described above. Law et al. argue that there are three main factors affect CAI combustion: stress, temperature and chemical substances. The first two factors have been confirmed to have effect on the combustion while the third one has not yet been ensured. Thus, there may be four cases exist shown in 2.1 according to the combustion pressure profile shown in figure 2.2.

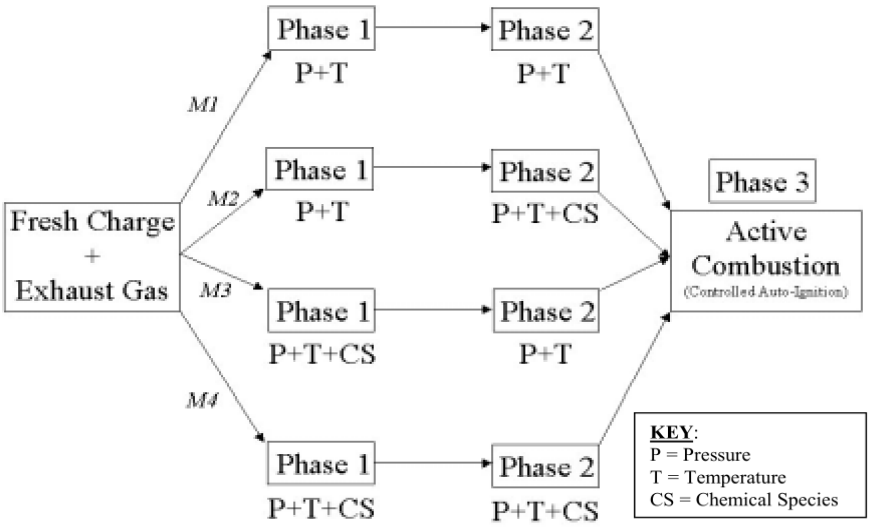


Figure 2.1 Four Possible Cases in CAI [10]

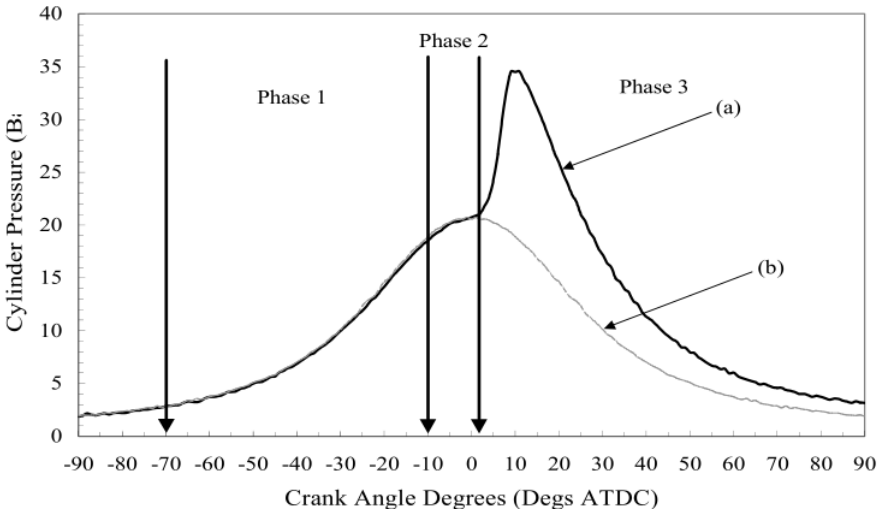


Figure 2.2 AC Pressure Curves [10]

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The first case: the overall process of phase 1 is driven by pressure and temperature, but they are significantly higher in the second stage than the first one, which finally induce the AC occur. The second case is that the first phase is also driven by temperature and pressure, but as the increase of them in phase 2, the exiting mixture molecules are split and react, or form new chemical species (CS), and then initiate AC combustion. The third one is the first stage driven by pressure, temperature and CS, but CS is in residual exhaust gas, and at phase 2 may cause the delay to induce AC occur for some reason. The last case is the combination of the second and third ones, CS both in residual exhaust gas and in the form of molecular fragmentation to re-composition, and finally burn. These are just assumptions, Law et al. will give proof in the following articles.

Law et al. [11] mention in the article that even if the two methods obtain exhaust gas are different, but both by the end of the experiment get similar heat release curve and cylinder pressure profile. Especially the cylinder pressure curve, both of them in two methods increase firstly by compression and followed by a significant delay stage before CAI combustion begins. This also shows that the two strategies have no dramatic difference in CAI combustion. After that, authors make comparisons of SI and CAI modes in NOX, heat release and combustion stability (shown in figure 2.3) and some other aspects. In below picture, SI mode has one load peak with more fluctuates while CAI has two peak load, one is combustion peak, the other one is for trapping gas, both are more smoothly than SI.

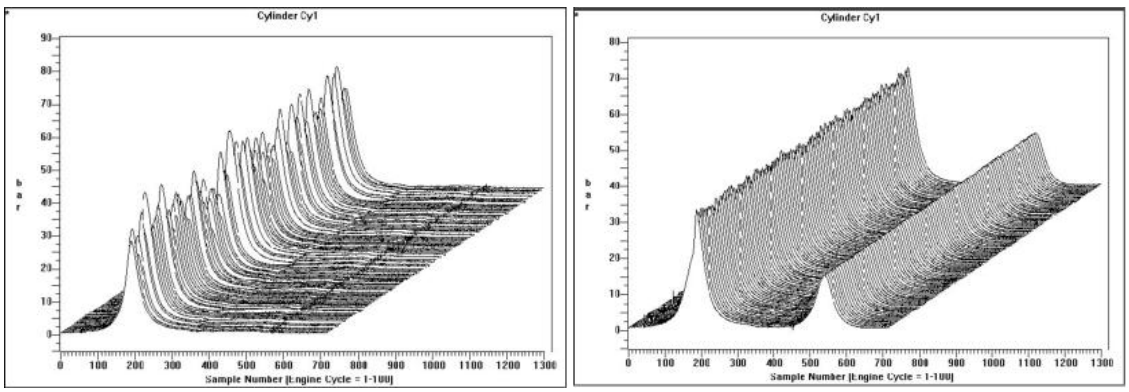


Figure 2.3 Combustion Stability of SI and CAI [11]

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It is concluded that the AVT system and the amount of trapped exhaust gas are the key to the CAI controllable combustion process. When the number of exhaust gas in-cylinder increases, the cylinder pressure curve indicates that CAI combustion starts ahead, and vice versa. The AVT system is more flexible to control the valve open and close, and with which would no longer need to preheat the intake air. Moreover, during the combustion process under condition of 2000rpm, 3.5bar IMEP and 14.6:1 AFR, the pressure generated by SI mode fluctuates greatly and CAI is relatively gentle, and the amount of NOx finally produced by CAI is much lower than that of SI because internal EGR is diluted and cause concentration of oxygen decrease, thus influence combustion rate, resulting in peak pressure in-cylinder and temperature drop down. Finally, author believes that CAI combustion is apply to four-stroke engines.

Oakley et al. [13] combined experimental and theoretical study to investigate the effect of air/fuel ratio and EGR on CAI operation range and their influence of combustion rate, ignition timing, emissions and so on. Results show that the dilution of air and EGR has great effect on the CAI starting, ignition timing and combustion duration: CAI combustion begins when intake air temperature reaches 320℃, air/fuel ratio greater than 80:1 and EGR rate is about 60%. When EGR rate is less than 45%, the influence of EGR dilution is greater than the air; when EGR rate is less than 30%, combustion duration is mainly affected by air/fuel ratio rather than EGR rate. If both air and EGR dilution are added simultaneously, coefficient of variation in IMEP (COVimep) will increase, and even if CAI combustion is generally fuel efficient, it will also reduce combustion efficiency and increase ISFC. They also refer to three boundary range of CAI operation: firstly, due to the lower temperature in the case of lean air/fuel ratio resulting in partial burn boundary, followed by the misfire boundary generated by large amounts of water vapor and CO2 cause ignition delay and large combustion durations, finally the knocking limit caused by the increase pressure in-cylinder due to exotherm rate. As for those exhaust emissions, it proves that the introduction of a small amount of EGR in intake charge can reduce HC, NOx and CO emission more in high and low load respectively. At last, they conclude that optimizing the fuel for CAI combustion

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under partial loads and maintaining critical fuel characteristics that permit conventional full load operation is a future area for development.

Law and Allen [14] in 2002 published CAI combustion to obtain the speed load and control strategy of an article stressed that in order to achieve all the advanced strategy, it is necessary to carry out a comprehensive control of valve system to complete gas mixing and exchange during combustion process, and thus make sure that FVVT technology is the key to help CAI performance smoothly. In this paper, direct injection is regarded as the trend of future development, and in order to maximize the accuracy of fuel injection, which cooperated with CAI combustion, the stratified operation should be combined, and these requirements are also the basis of acquire speed load relationship. In the experiment, the FVVT has the ability to modify the valves to run the periodic contour of the strategy to control engine, and change combustion mode from SI to CAI. The first test is change the load by installing throttle valve in intake manifold of SI mode and trapping the amount of exhaust gas in CAI to obtain speed load image. Comparing the differences of fuel consumption, NOx, HC and CO in the two burn modes. The results show that CAI combustion strategy is about 30% lower than that of SI in these aspects and CAI is more suitable for combustion at part-load. The second experiment is carry out hybrid SI-CAI combustion, it is proved that CAI combustion is achieved at high internal exhaust gas recirculation (IEGR) and high speed and low load. The fuel consumption is increased to 32%, NOx is reduced by 97% and CO is also significantly reduced, while the HC is only about 50% lower on the images. The two experiments show that under the condition of equipped with FVVT, CAI combustion strategy can be applied to actual engines.

Law and Allen [15] through experiments to verify the speculation in previous articles of the existence of chemical species and whether they can promote CAI combustion. And using the FVVT institutions with nitrogen and hot air instead of EGR for testing. Nitrogen is not used for ignition timing, but can dilute the mixture and lead to lower heat release rate because of the inert gas itself, while the addition of oxygen can advance the ignition timing but can’t reduce heat release since it is combustion reactant. It is mentioned that the effect of EGR may

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have two aspects on CAI combustion: thermal and chemical effect. Enriching nitrogen can't make mixture auto-ignition unless the spark plug assisted and the temperature should be raised to 400℃, which possible to delay start time of combustion and to stabilize the reaction rate. Thus, the effect of nitrogen on combustion only acts as thermal effect. If charge hot air of 120℃ into cylinder, spontaneous combustion would begin without spark plug, advance ignition timing and finally occur uncontrollable knock. Therefore, oxygen plays a chemical effect. From the two experiences, it means that EGR plays both effects as it contains a lot of nitrogen and oxygen, which promote combustion and can also make it controllable.

The theory is more specific according to the research and experiments of CAI combustion by Law et al., and most of the subsequent articles are based on these preconditions for further discussion and comparison.

Koopmans et al. [16] explain in detail of the conversion of SI-HCCI and HCCI-SI on single and multi-cylinder Volvo engine equipped with FVVT system. They mention that the average amount of exhaust gas in the case of 40-60% can achieve the temperature of automatic ignition. However, due to the need of trap exhaust gas and dilute the mixture, the whole process cannot start from HCCI, and the entire process of HCCI can only be part of the load in the operation. Figure 2.4 shows haw the engine is switched from SI to HCCI at 2.8 bar and 2000 rpm.

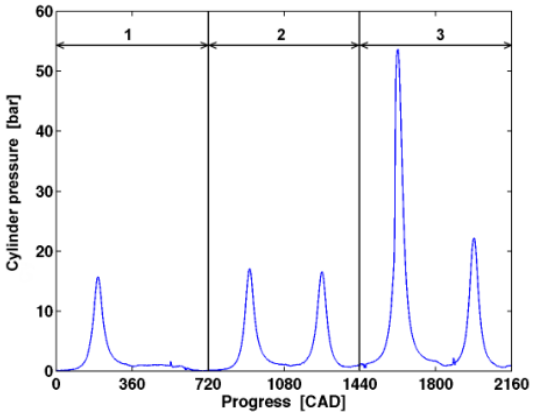


Figure 2.4 Mode from SI to HCCI [16]

The first part is the traditional SI mode of previous cycle, the application is also conventional valve timing, the second part is a new round of SI mode, during which a certain amount of exhaust gas is trapped and finally results to HCCI mode in the third part. Figure 2.5 illustrates the change of cylinder pressure from HCCI to SI mode at 2 bar BMEP and 2000 rpm.

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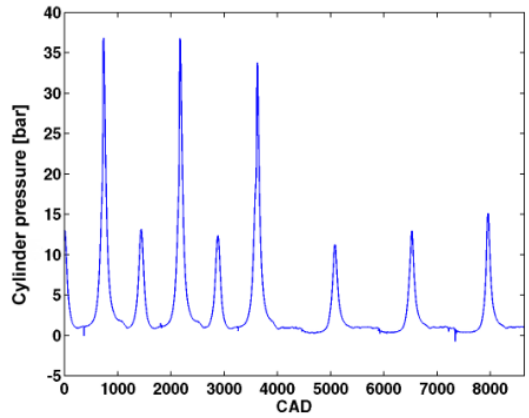


Figure 2.5 Mode from HCCI to SI [16]

The authors conclude that SI-HCCI is more difficult than HCCI-SI because HCCI starts with more factors than SI mode, such as the conversion of fuel supply and valve timing. And there are many problems that need to be noted during the conversion, for example, HCCI is part-load and the engine is operated at low to medium speed while SI is the higher loads with high speed of engine operate. These are in the form of control parameters to convert, if it could not set parameters well, then the conversion process will make combustion unstable.

Koopmans et al. [17] propose the position of auto-ignition of HCCI under two modes, both of which spray fuel into combustion chamber by direct injection. They take the temperature of automatic ignition by heating the intake air to achieve the desired temperature for HCCI combustion as the first mode, also known as preheat HCCI mode. This method has a cam lift profile of 8 mm and a duration of 236 crank angle degree (CAD). Trapping hot exhaust gas by the way of negative valve overlap (NVO) to achieve HCCI occurrence is considered as the second mode, also known as NVO HCCI mode, which has 3 mm and 130 CAD lift and duration respectively. In order to investigate the distribution of local auto-ignition at HCCI combustion, the authors use the optical engine and calculate the distribution of local mixture temperature, residual exhaust fraction and equivalent ratio at the end of the compression stroke. It is concluded that the local ignition timing of the fluid element in combustion chamber plays a main role in stable combustion. The overall heat release depends not only on the distribution of local ignition timing but also by the influence of local combustion rate of individual elements. Author also mention that the first ignition of the Preheat HCCI mode is even more uniform than the NVO HCCI ignition, since the NVO type occurs in half of the combustion chamber. Thus, the NVO method produces a greater degree of non-uniform than Preheat HCCI.

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Cairns and Blaxill [18] summarize the effect of internal and external EGR on the CAI combustion. The internal EGR is the two methods of capturing exhaust gas mentioned by Law et al, while external EGR is a method that Thring refers to by reclaiming the exhaust gas into intake manifold and then into cylinder. The test uses a direct injection engine and EGR valve is mounted on the rear of the cylinder head, allowing the coolant pass through the valve to make external EGR cool before entering cylinder. Reasonable control of the cam profile of intake and exhaust valve and well limited of the duration. The final experimental results show that the introduction of the cooled external EGR delays the ignition and combustion and also extends the knocking limit. But it could increase the output power of the engine which is in lower speed to get better results. However, as for fuel consumption, the internal EGR consumes less fuel and improves accuracy. If using internal and external EGR together, it will increase the peak load and the transmission from SI to CAI could be more smoothly.

After Cairns and Blaxill [19] studied the effects of internal and external EGR on CAI, they continue to research the lean boost and external EGR on CAI combustion at high loads by experiments. This paper briefly mentions an opinion, which different from previous articles, is that the four-stroke engine has two conditions to make the charge heated and achieve auto-ignition, one is to use the combination of high or variable compression ratio and assisted intake heating/exhaust recirculate to achieve automatic ignition of mixture, which also known as HCCI. Another method is to use the compression of traditional SI engine, trapping a large number of recirculating combustion gas mixed with fresh charge to get auto-ignition point, which will perform a relatively high level of stratification between the burned and unburned gas. This method has also named CAI. Thus, authors believe that CAI and HCCI are the different technologies, which echo the content at the beginning of this paper. In order to improve the peak output of natural intake, the small fixed exhaust turbocharger is used to raise the achievable load of petrol controlled auto-ignition burning. However, the highest load generated by this method is affected by the increasing of in-cylinder pressure and burning noise and dropping of the gas exchange efficiency. But the advantage is that the load range of CAI

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operation is increased and no lean post-advantage of NOX is required.

Martin and Zhao [20] report for the first time that CAI combustion strategy is implemented on a Ford engine with port fuel injection, and conclude four main standards are needed to achieve CAI on a four-stroke engine: intake air temperature; compression ratio; spontaneous fuel and EGR. Authors use the Twin Independent Variable Cam Timing (Ti-VCT) strategy, which has a smaller lift and short duration mechanical camshaft instead of the valve control method used in previous experiments and execute negative valve overlap (NVO) strategy. This paper mainly analyzes the influence of speed and load on exhaust temperature, fuel consumption, NOx, CO and HC, and conditions in-cylinder on the CAI operation. Most analyzes have the range of λ=1, the speed reaches 2500rpm, change to λ=1.05 and 1.15 respectively, corresponding to 3000rpm and 3500rpm. (λ is air/fuel ratio). Since in this part of conditions, the engine could maintain a stable CAI combustion. The results show that if the exhaust temperature increases, the corresponding minimum load limit will be extended; the load on fuel consumption is more sensitive than the speed; NOx emissions reduce with speed increase and load decrease; CO and HC decrease as the load increased. Finally, the authors conclude that the amount of residual gas in cylinder can control output of engine torque, and two of them are inversely proportional, the load and speed have great effect on CAI combustion, the starting point of CAI is earlier, various temperature and pressure indicators have an upward trend when the load increases, and the impact of speed on CAI is more inclined in the aspects of emissions and fuel consumption.

Martin and Zhao [21] in a paper investigate the effect of turbocharging on CAI combustion and compared with the naturally aspirated one of 2007. Unlike the previous experiments, the newly set camshaft and turbocharger are used to achieve the purpose of forced airflow. A short lift and duration of camshaft on naturally aspirated can't get enough exhaust gas to drive the turbocharger, changing it to a higher lift (4mm) and long duration can result in a higher-level boost. Meanwhile, it still uses NVO approach to get residual gas. The second change is that reducing the compression ratio from 11:1 to 8.8:1, which reduces the likelihood of engine knocking and

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increases the operation range of the engine. The results show that when the engine speed lower than 2500rpm, there is only SI mode; and while it is in the range of 2500 to 3000rpm, CAI would be started if the EGR is over than 40%; after that until 4500rpm, CAI combustion only exit in the engine. In the case of turbocharger, CAI combustion can occur at higher loads, and the lower load limit is higher than the higher load restriction of natural intake, and extends the load on CAI range, and also increases the speed. The upper and lower limits can be changed by a series of adjustments, the upper one can be increased by cooling EGR or intake gas, and the lower limit of load can be extended by a slight increase of compression ratio or reduced the boost pressure of valves.

Brewster et al. [22] describe the use of high-pressure air-assisted spray injection directly into combustion chamber. The improved injector, as shown in figure 2.6, allows for more precise control of fuel and air (eg. fuel temperature, the turbulence in air and combustion chamber, and fuel stratified injection) to change the injection parameters, and then make assessment on the impact of CAI combustion. However, the previous study of CAI on this area is almost very little.

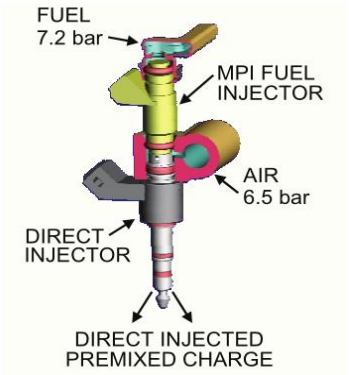


Figure 2.6 Air-assisted Injection Equipment [22]

The authors tested the single cylinder engine with a compression ratio of 10.4:1 and implemented a NVO strategy with a camshaft with low lift and short duration to ensure the required temperature condition for automatic ignition. During the combustion of 50% mixture, the appropriate injection strategy can simultaneously obtain a maximum pressure at a rate of 300 kPa/degree and a net IMEP load range of 100-700 kPa, and at the same time as the engine speed is 2000 rpm.

Under the same conditions, the change in the spark-free injection mode and parameters can produce combustion phase of 355 to 375 ATDC for the CA50, and the maximum pressure rise rate is in the range of 70 to 450 kPa/degree. This explains to a

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large extent that the setting of injection has a great effect on the operation authority of CAI. As for the condition of given injection, the increase and decrease of the load will change the combustion phase and out of the desire scope. If it increases, the maximum pressure rise rate will also increase and eventually limit the load, and if it decreases, it will cause quenching and fire to fail, which cannot run in CAI mode. The authors also emphasize that spark assist can affect the linear relationship between the burning rate and maximum pressure rise rate, which can also possible to maintain the optimum combustion phase while adjusting the maximum pressure rise rate.

Hunica and Kordos [23] make experiments on comparison with SI and CAI combustion by using indicated specific fuel consumption (ISFC) and emission exhaust. The CAI combustion is also analyzed separately, the control combustion conditions are defined A/F (from stoichiometric to λ=1.2), the scope of indicated mean effective pressure (IMEP) at 0.2 to 0.42MPa, and get the engine's performance and emissions with load changes and rotational speed. The experiment uses an engine of single cylinder with direct injection system and fully variable valves train (FVVT). The authors emphasize that NOx produced by combustion in SI mode is independently of the degree of mixture. At low load, NOx is about 20 times higher than that produced by CAI. Although CAI can reach SI emission level under the condition of λ=1.05 and high load, the subsequent processes can still reduce the emission of substances. When they both are at λ=1.05, the emissions of SI are 4 times than CAI. As for others, when engine speed is 1500 rpm, λ=1.05 and IMEP is 0.36 MPa, the fuel consumption of SI is 328 g / kWh, while the equivalent CAI is 293 g / kWh, which saving about 10%. Finally, they conclude that, due to the characteristic of CAI combustion, even in the relatively high load, cylinder' temperature would not exceed 1800K; injection timing on the impact of combustion is that if it is delayed, the release of heat is reduced at high loads, thereby extending the upper limit of the load.

While under lower load situations, gas in-cylinder can be diluted by exhaust recycle, through the injection timing make auto-ignition advanced and reduce fuel consumption. However, even if CAI consumption does bring a lot of benefits, but due to the rapid increase and decrease of pressure, engine's noise and vibration

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significantly larger than SI, which need to study and improve in the future.

Adomeit et al. [24] summarize the CAI combustion based on previously published articles and simulates combustion using software. The CAI operation range is summarized as shown in figure 2.7.

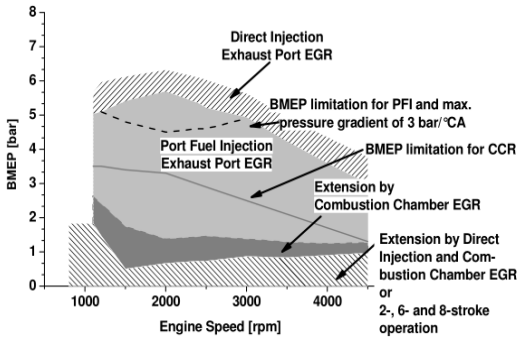


Figure 2.7 CAI Operation Range [24]

The authors describe the two methods of capturing exhaust gas as exhaust port recirculation (EPR) and combustion chamber recirculation (CCR) respectively. The effects of the combustion of the two methods are observed and compared by installing a camera and other equipment on a single cylinder engine which equipped with electromechanical valve train (EMVT) and direct injection system. The experimental premise is that the NMEP is 3 bar and engine speed is 2000 rpm, since at this point both the CCR and EPR can be operated by valve timing. In both cases, the range of flame and contour in cylinder is shown in figure 2.8.

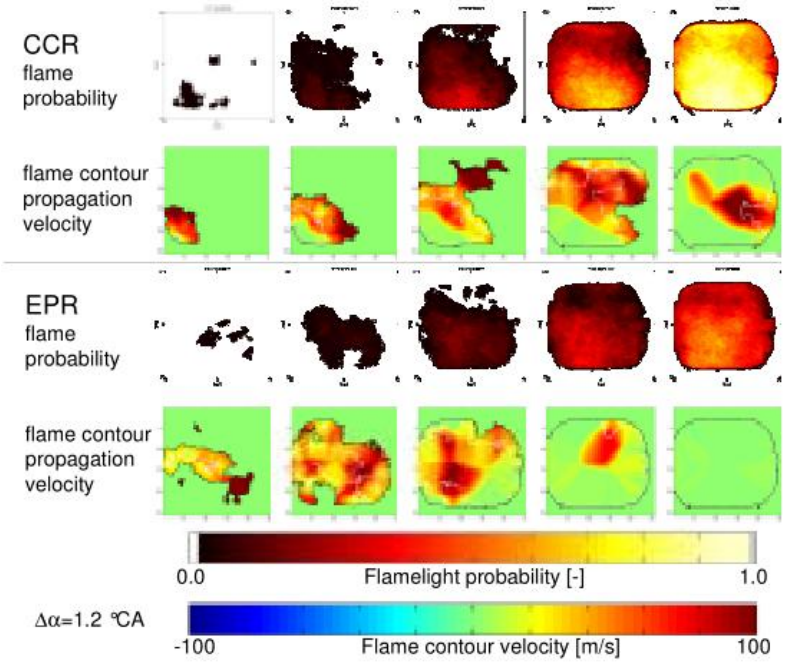


Figure 2.8: The Flame Probability and Contour Velocity [24]

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The image can be seen that the flame probability of CCR filled with entire combustion chamber is from 0 to 0.9 while the EPR is 0 to 0.7. Similarly, the flame propagation velocity of CCR is faster than that of the EPR. In this respect, the authors emphasize that the CAI propagation is about 100m/s, which is much faster than the SI mode under the same case. **Figure 2.9** shows the effects of the proportion of residual exhaust gas in the mixture (RGF) on rest results.

The RGF in CCR mode is typically 42-47% and 49-52% in the EPR. Higher RGF in mixture will cause both pressure and temperature rising and initiate CAI combustion earlier, vice versa. In general, although the RGF in EPR is relatively large, it is still slower than CCR combustion. **Figure 2.10** shows the influence of stratification on the results of the mixture of combustion.

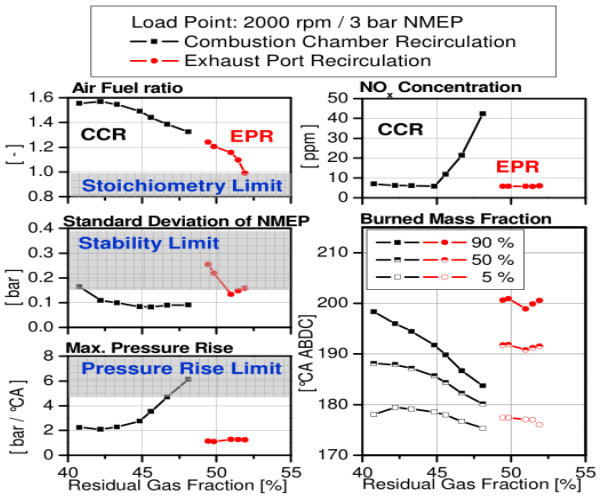


Figure 2.9 Properties of Combustion at 2000 rpm and 3 bar [24]

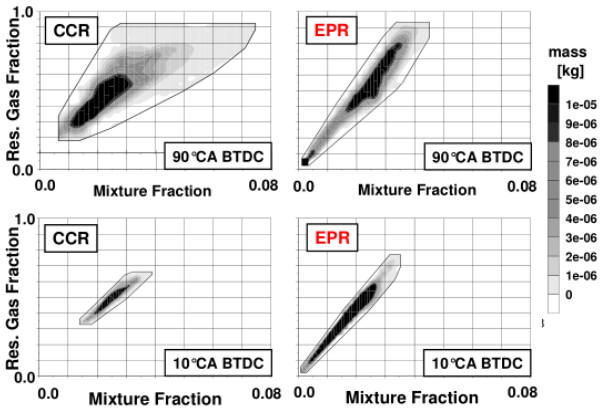


Figure 2.10 Distribution of CCR and EPR in different conditions [24]

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During the compression stroke (90° CA BTDC), CCR has a smaller amount of RGF when it is observed larger zone, whereas the EPR is opposite. When the state of mixture is close to auto ignition (10° CA BTDC), the charge in cylinder is basically homogeneous charge, CCR performance a lower layered structure, that is, the mixture is more uniform, while EPR shows a lower homogeneous distribution. Finally, authors conclude that the stratification of fuel, air and residual gas in combustion chamber have a significant impact on combustion rate and results, and also the control of accurate mode switching is the key to CAI technology.

Zhao et al. [25] compare the SI and CAI cases of 2/4 stroke direct injection petrol engines in different modes, which divided into seven types. The four strokes are including: throttle controlled SI mode (4strTHrSI); intake valve throttled SI type (4strIVLSI); positive valve overlap SI (4strPVOSI); NVO CAI mode (4strNVOCAI); exhaust rebreathe CAI (4strReB). The supercharger should be turned off during operate these modes. As for SI modes, there are only two types: CAI and SI, and the supercharger relates to intake system to promote intake pressure. These modes are achieved by combination of valve timing and duration, shown in figure 2.11, and compare the following aspects.

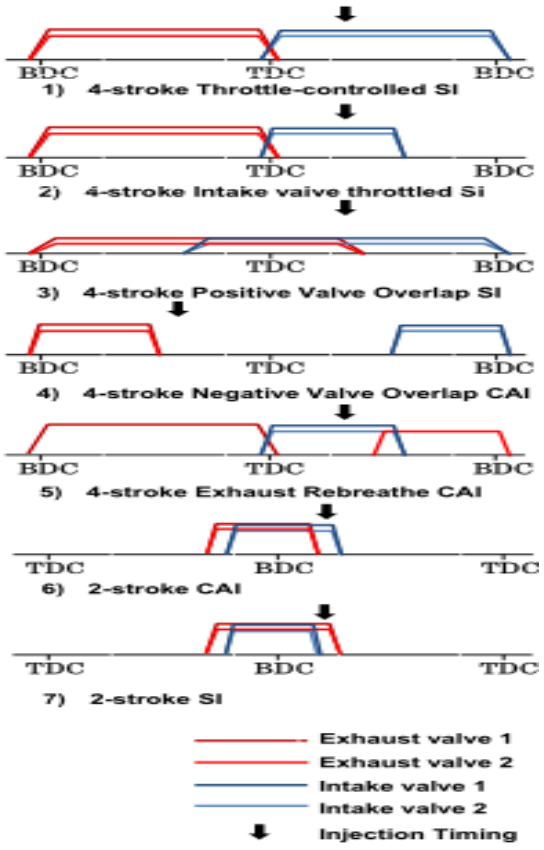


Figure 2.11 Valve and Injection Timing [25]

In the case of gas conversion efficiency is shown in figure 2.12, it is clear that 2-stroke combustion and 4strReB have a 100% exchange efficiency, since the pressure around intake and exhaust valves is substantially the same in 4strReB mode, while the 2-stroke modes don't have pumping loss in gas exchange process, but it still should be calculated the supercharger which provides part of energy caused the loss. In contrast, the efficiency of the other four modes is

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lower, and the traditional SI mode is the lowest. However, if the intake valve instead of throttle to restrict load operation, the gas exchange efficiency will increase about 4%. The 4strNVOCAI is a little lower because there is heat loss during recompression.

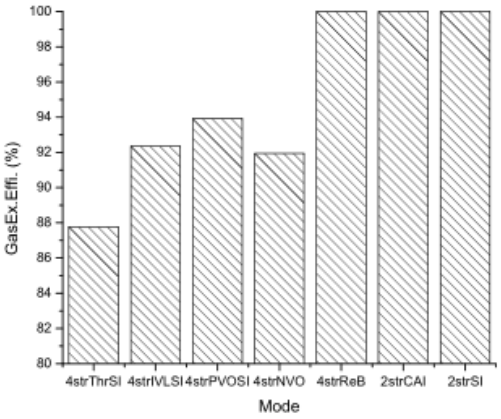


Figure 2.12: Seven Operating Modes of Gas Exchange Efficiency [25]

The comparison in fuel efficiency is described in figure 2.13. 4strNVO shows the highest fuel efficiency is as the fuel has been injected during exhaust process and therefore it has a longer time to atomize and mix than other modes, and the high temperature and recompression of residual gas make it even more homogeneous.

However, the combustion efficiency of 4strReB is slightly lower since the flow and mixing of the air in cylinder could affect combustion. 2strSI has the lowest efficiency because fuel has little time atomizing and mixed with air.

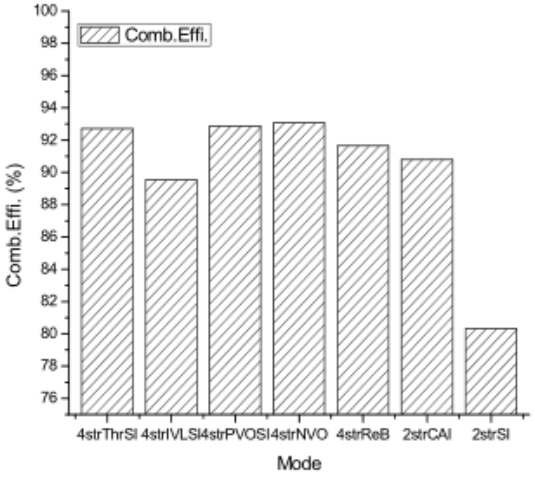


Figure 2.13: Seven Operating Modes of Combustion Efficiency [25]

Figure 2.14 illustrates the comparison of thermodynamic efficiency, and which of 4strTHrSI and 4strIVLSI are basically the same, since the combustion phase and duration are similar. 4strPVOSI owns less efficiency than that two ones because the presence of combustion gases reduces flame propagation speed during burning period. The two CAI modes of 4-stroke show higher thermodynamic efficiency due to their shorter combustion time and lower temperature. The lowest efficiency between the 2-stroke modes is because of the less expansion.

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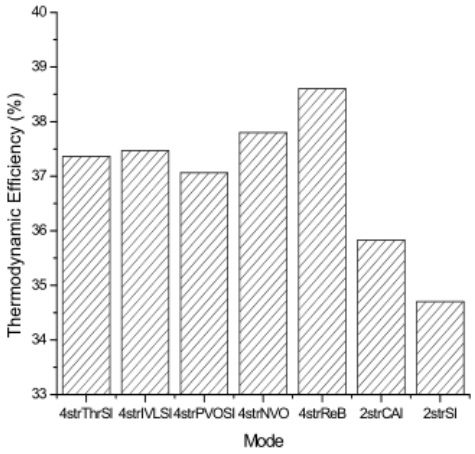


Figure 2.15Seven Operating Modes of Fuel Consumption [25]

Finally, the authors also mention that the 4-stroke SI and 2-stroke modes emission large amount of NOX and CO respectively.

Yu et al. [26] describe how the effects of chemical species on auto-ignition in HCCI combustion of petrol engine and how fuel injection can enhance these effects. The experiments use four-stroke single cylinder engine which equipped with special spark plug, sampling tube and electromagnetic valves. When the engine speed is at 1500 rpm, the composition of in-cylinder intermediate product is studied by different fuel injection types (port fuel injection and combination of PFI&NVO) in compression stroke using Fourier transform infrared (FTIR). The in-cylinder sampling test system is shown in figure 2.16.

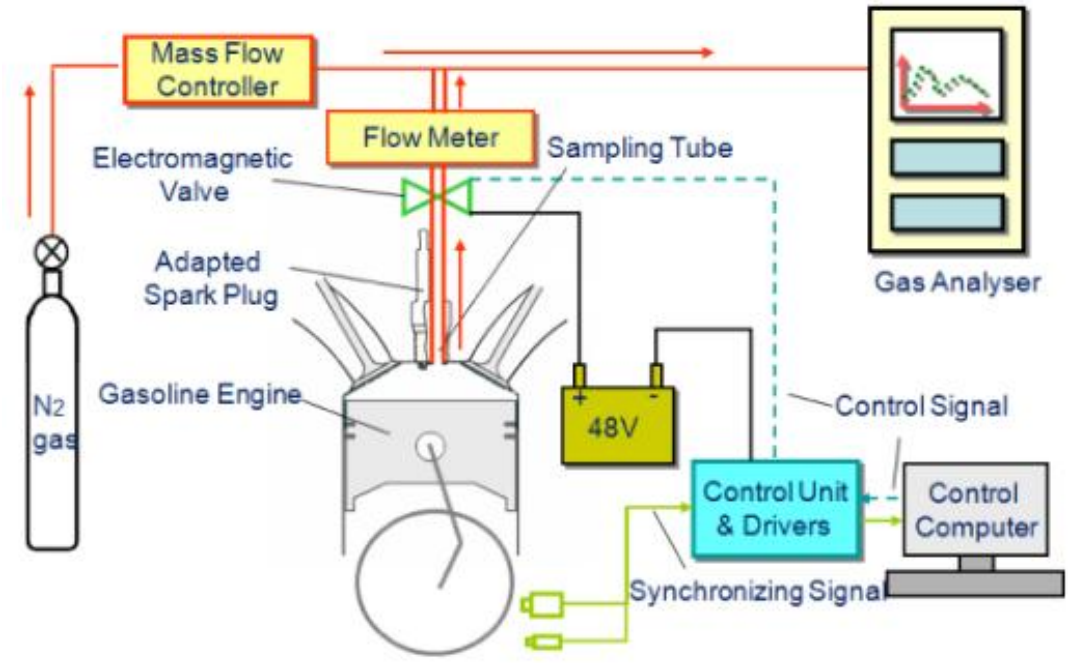


Figure 2.16 Sampling Test System [26]

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The experimental principle is that when the engine is activated, the electromagnetic valve would be opened according to the specific location of crank angle in every cycle of the engine. The gas in-cylinder through the sample dedicated tube, mixed with nitrogen with high pressure and transmitted to the analyzer.The dilution of N2 has three factors on this process. Firstly, since nitrogen is the inert gas and seldom reacts with other substances, thus a large amount ofN2 could reduce temperature of the sample gas and can also eliminate its internal chemical reactions to make sure the composition and concentrationdo not change. Secondly, this kind of dilution can prevent water and fuel from condensing, ensuring that the final concentration analysis is correct. Finally, the dilution of nitrogen stream is under high pressure because the gas that just flow out of cylinder does not have sufficient pressure flow through FTIR, while the increased air flow can cause entire gas sample into analyzer. Generally speaking, the dilution ratio of N2 is 12:1 to 50:1. The graph 2.17 shows the intermediate products in cylinder by two fuel injection types at 60°CA BTDC during recompression. The authors make conclusion and get figure 2.18.

Components	Unit	PFI	PFI & NVO-Inj
C_2H_4	ppm	307.8	561.9
C_3H_6	ppm	132.9	199.1
C_4H_6	ppm	41.8	53.3
C_2H_2	ppm	68.7	156.4
C_2H_6	ppm	36.6	48.2
CH_4	ppm	192.1	567.0
CH_2O	ppm	207.7	236.1
CH_3OH	ppm	39.5	58.0
CH_3CHO	ppm	137.4	99.0

Figure 2.17 Intermediate Products at different Injection Types [26]

SOC/°CA	200PPM	2000PPM
NO UHC	0	0
C_2H_2	-0.9	-6.5
CH_3OH	-0.8	-6.5
CH_2O	-0.7	-5.9
C_2H_4	-0.2	-3.2
C_2H_6	-0.2	-2.8
CH_3CHO	0	0.1
C_3H_6	0	0.5
CH_4	0.1	1.3
C_4H_6	0.9	10

Figure 2.18 The Effects of Product During Ignition Timing [26]

It could be seen in the graphs that C2H2, CH3OH, CH2O, C2H4 and C2H6 have a positive influence on the ignition timing, whereas CH3CHO, C3H6 and CH4 have no effect on HCCI combustion and C4H6 will be bad Impact. In the end, authors argue that the active substances in residual gas do promote HCCI combustion by scientific chemical kinetics, and the production which enhance ignition timing from PFI&NVO are 1.7 times than that of PFI method.

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Wang et al. [27] change the temperature of chamber wall by changing the coolant temperature in the experiment and explore the effect of on the heat release rate. It can be seen from figure 2.19 that the change in coolant temperature has influence on the whole combustion process and that only the second half of the process is affected in figure 2.20, which explains to a large extent that the ability of change in coolant temperature to the cylinder wall is affected by factors, such as reaction conditions, operating capacity and heat flow in-cylinder and other ones. The higher the temperature is, the earlier the ignition time and the greater the peak pressure in cylinder with reducing the duration of combustion.

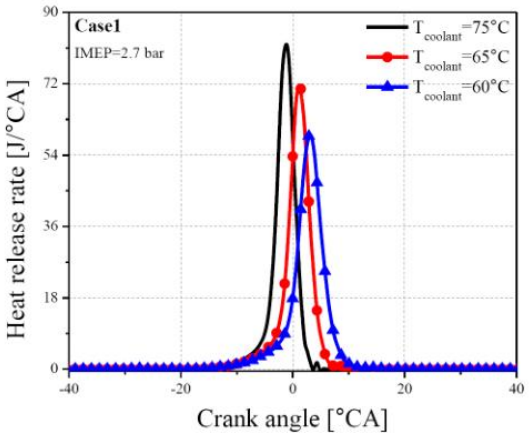


Figure 2.19 Heat Release Rate in Different Coolant Temperature (2.7 bar) [27]

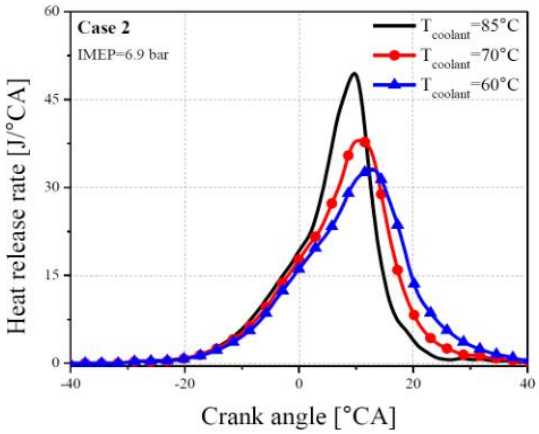


Figure 2.20 Heat Release Rate in Different Coolant Temperature (6.9 bar) [27]

The burning image of 2.7 bar is more like combustion on CAI mode, since the heat of residual gas is sufficient to guide CAI in spite of the spark plug in this state. The condition of 6.9 bar combustion in order to meet the requirements of high load, the amount of residual gas is little so that fresh charge could enter the cylinder. Therefore, the initial conditions in-cylinder of 6.9 bar is very poor compared to that in 2.7 bar. And this situation can dominate the entire combustion process before auto-ignition occur, even if the increase of wall temperature could bring better hot condition in cylinder, which help to improve the occurrence of CAI, but the central part of fresh mixed charge still need spark ignition. By comparison, it can be seen that the auto-ignition mode is more sensitive to the wall temperature than SI mode.

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The experimental results also show that the combustion characteristics of the crank angle at 10%, 50% and 90% (CA10, CA50, CA90) and make comparison among them, which shown in figure 2.21 and 2.22.

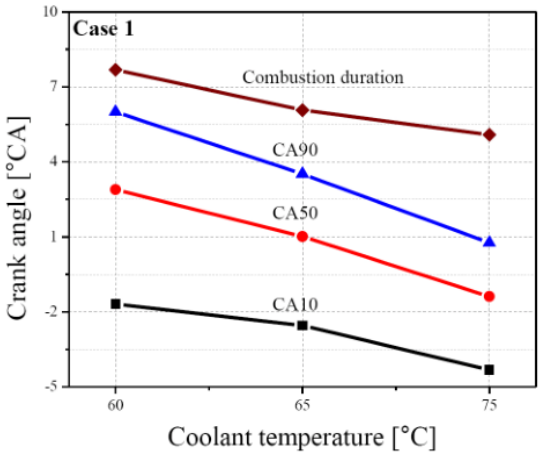


Figure 2.21 The Influence of Coolant Temperature at Different Conditionsduring 2.7 bar [27]

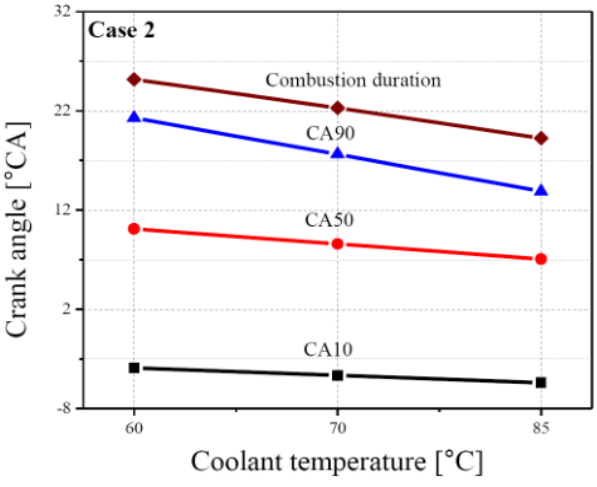


Figure 2.22The Influence of Coolant Temperature at Different Conditionsduring 6.9 bar [27]

In general, the duration of combustion is shortened as the temperature of coolant increases, and the sensitive of CA50 and CA90 to wall temperature is significantly higher than that of CA10. During the period of low load, the cylinder conditions for auto-ignition is very favorable to be created by wall temperature and the reaction is also obvious. When it is in the high load, flame propagation dominates the whole process which mentioned in above, and it is relatively slow to change with the wall temperature, while the late auto-ignition is very sensitive to it, and due to the difference between them eventually lead to a slight change in CA10.

After that, authors use software simulation to analyze a series of changes caused by wall temperature. Firstly, the effects are researched during the process of gas exchange. The in-cylinder pressure show the same under different conditions, which basically similar with the experimental data. And with the increases of wall temperature, even if the thermal conditions in-cylinder raise, but this difference remains basically unchanged before combustion and therefore it could be ignored. When the effect is analyzed during combustion process, the pressure and temperature exhibit large change with wall temperature changes. The obvious

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differences on them are due to different combustion processes caused by different wall temperature. However, the temperature of mixture near the wall is much more sensitive than the central mixture.

To analyze the in-cylinder temperature and residual gas distribution at 670℃ (figure 2.23 and 2.24), they divide combustion chamber into 9 parts, as shown in figure 2.25

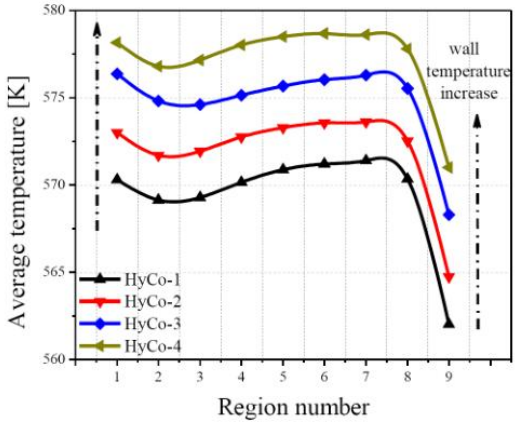


Figure Regions [27]2.23 Temperature in Different

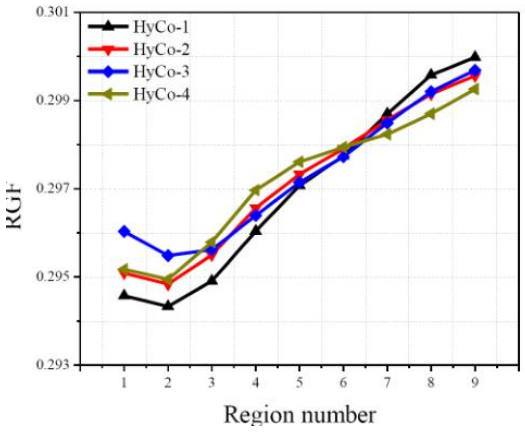


Figure 2.24RGF in Different Regions [27]

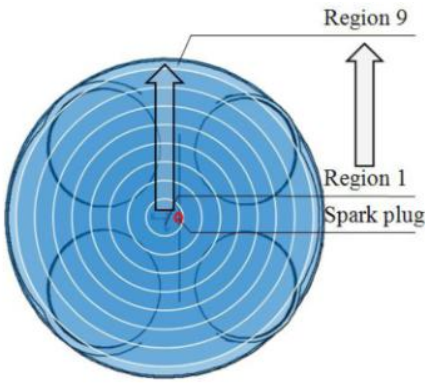


Figure 2.25 Nine Regions Assisted to Analyze [27]

It is obvious that wall temperature has a great effect on overall temperature of cylinder and the maximum temperature difference is the nine area, which also explains that the temperature of mixture gas near wall is sensitive to this change. The distribution of residual gas shows an increase trend and the area6,7 have higher exhaust gas compared with that of 8 and 9. Finally, the analysis of combustion process also confirms the above point of views, and emphasizes that the distribution of auto-ignition and the 2% of the top part of ignition part reacts to the wall temperature in the subsequent combustion process.

Ojapah et al. [28] made experiments with the non-cam single-cylinder direct injection of down size engine under the same load conditions to compare the gaseous and PM produced from CAI and SI modes. It uses an electrohydraulic control system to achieve SI and CAI

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combustion of 2/4-stroke. The down-sizing engine could reduce mechanical losses to some extent, and this type of engine is usually equipped with turbocharge to ensure its normal operation, while installed with DI system, using the advantage of charge cooling under high compression ratio could prevent the occurrence of knocking event. They mentioned that though PFI of the engine can achieve CAI combustion, compared with GDI, it limits the air/fuel ratio and the amount of air into cylinder and ultimately affect the SI combustion. The GDI system has been able to master injection timing and times to control the formation and auto-ignition of the mixture by the technologies of reducing the size of engine, making stratified charge burning and achieving CAI combustion, and finally improve the fuel economy. However, the shortcomings of GDI is also obvious, such as the direct injection process may be affected by airflow and other uncertain factors, there would be too much local fuel or liquid fuel situation, which lead to the formation of HC, CO and PM. And the petrol used in DI system is non-renewable energy, the authors believe that ethanol has high octane number, oxygen content and evaporation, can become alternative fuel of petrol and can also reduce the environmental pollution problems. In many of the previous papers, there were no comparison of PMof CAI and SI emissions. Therefore, after some tests, authors draw some conclusions, which part of them are described in above paragraphs, so only explain to those points that above review haven't summarized.

1. CAI combustion can be achieved in the speed range of 800 rpm to 1300 rpm in this test, and the discharge of DI engine is mainly smaller particles, and the characteristic value of the particle size distribution is between 10mm and 30mm, which shows that the CAI operation also appears with the particulate matter.

2. Spark helps to start CAI and make it burn smoothly with NVO under high load conditions. However, if during the stage of NVO, the pyrolysis of hydrocarbons produced by fuel injection can cause CAI to burn to produce particles with a diameter of less than 5 mm, but the lower combustion temperature cannot oxidize it. While the SI detects fewer particles with a diameter of several nanometers, which are also related to unburned hydrocarbons.

3. Particles with a diameter of 50 to 150 nm, mainly containing carbon element, will decrease as the CAI combustion load

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increases, but the combustion of SI is not defined in this respect.

Zhao et al. [29]explain the effect of flame diffusion on SCHC ignition timing. Since the system begins with spark ignition and then performs auto ignition, the propagation of flame front increases the temperature and pressure of the unburned gas, resulting in spontaneous combustion of the gas mixture. The authors mention that spark assisted in suppression of HCCI combustion around the misfire limit is very helpful, and this experiment through the light engine of chemiluminescence imaging and exothermic analysis of the study.

Figure 2.27 shows the effect of the degree of mixture dilution on auto-ignition timing and flame propagation. It is clear that as the EGR rate increases, the pressure in-cylinder gradually decreases to retard the combustion and heat release. The inflection points on curves mean the beginning of automatic ignition, and when the EGR rates change from 7.58% to 12.2%, there is more ignition timing delay.

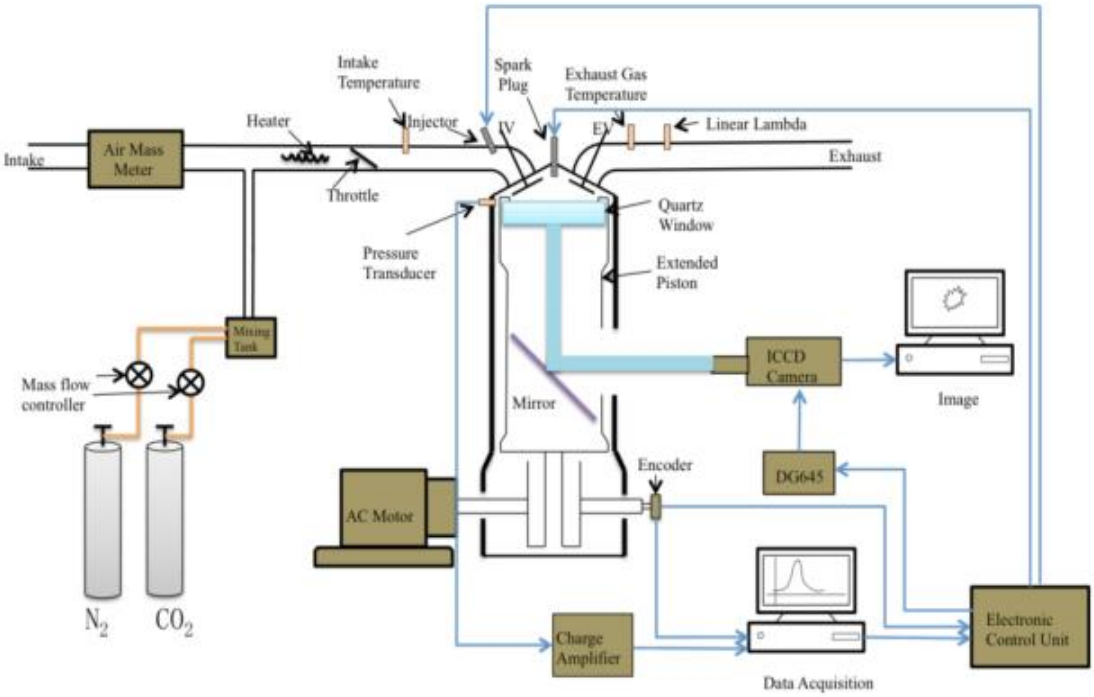


Figure 2.26 Total System of the Experiment [29]

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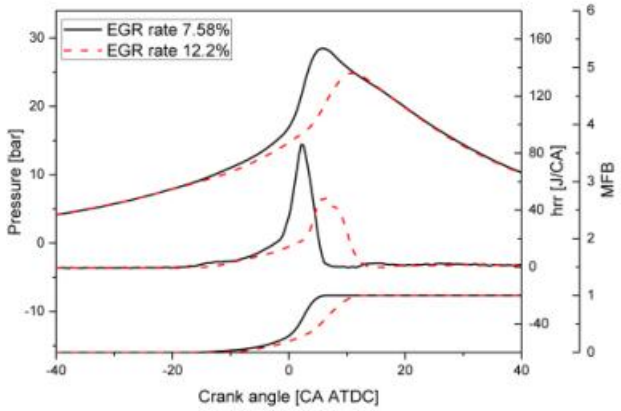


Figure 2.27 The Curve of Pressure, MBF and Heat Release [29]

The authors divide the image into three phases: ignition delay stage, which from spark plug ignites to CA05 (the angle of crank when burning 5% of the gas); flame propagation stage, which from CA05 to CA auto; spontaneous combustion stage, which between CA auto and CA90. Before the auto-ignition, the difference of heat dissipation rate and mass friction burnt can be seen from the slope of curves. In this test phase, authors conclude that the dilution of exhaust gas has a similar influence on flame propagation and automatic ignition, but has a greater impact on ignition delay. The temperature also has effects on these three stages. At the beginning of the combustion, the in-cylinder temperature is low, resulting in a long time of ignition delay and a slower heat release rate during the flame propagation phase, thus leading to a delay in automatic ignition.

After that, authors analyze the relationship between parameters of flame propagation and automatic ignition timing. The ignition delay is positively correlated with auto-ignition timing, that is, the shorter the ignition delay time, the more acceleration of heat release of mixture combustion, and thus ahead of auto-ignition. This view shows that flame propagation accelerates auto-ignition, so the two of them present positive correlation trend. The average rate of exotherm at the time of flame propagation is small and negligible for automatic ignition, and it also shows that the heat generated by compression does not act as the only way that flame propagation affects auto-ignition. And if the initial state of cylinder is better, it will help to advance auto-ignition, which is affected by both the initial condition in-cylinder and flame propagation. Finally, authors interpret the evolution of auto-ignition. Auto-ignition timing is a cyclic

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behavior to a certain extent, and it could be explained by the evolution cycle of ignition delay and flame propagation.Due to a series of unstable combustion factors, abnormal circulation occurs in the normal range of circulation.

Seo and Zhao [30] investigate the effect of the addition of Di-methylether (DME) in petrol on CAI combustion by experimental operation. DME has a high cetane number, which is colorless gas or compressed liquid at room temperature, and mixed with air could form explosive mixture. Thus, it is much suitable as a source of ignition to control the premixed or diluted mixture of combustion. DME is more flammable than diesel and the pressure required to achieve atomization effect is small. Therefore, authors use a single-cylinder engine to study two things: intake rebreathe timing, which has significant impact on operating range of CAI due to changes in EGR and intake temperature, and injection time of a variety of separation DME injection ratio on combustion properties and so on.

During the experimental investigation on intake rebreathing timing, the fixed conditions are that: engine speed is 1500 rpm, λ is 1.3, compression ratio is defined as 15 and IMEP is 4.2 bar, the timing of intake and exhaust is shown in figure 2.28. The exhaust valve timing should be fixed at 175°CA when opening and closing at 370°CA.

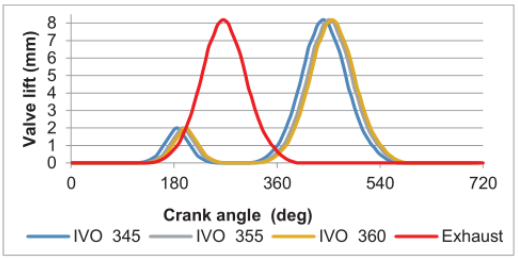


Figure 2.28 Intake and Exhaust Valve Timing [30]

To make comparisonsof indicated efficiency during constant load, the increased injection of DME in the fixed petrol injection amount can satisfy the volumetric efficiency change caused by intake timing variation. In this test, DME spray is carried out twice: the first time is in compression stroke, which is to improve the reactivity of bulk mixture; the second one is to help initiate auto-ignition when the piston reaches near the top dead center. The amount of each injection is in figure 2.29.

Parameters	Value
1st DME SOI timing (CA, Adjusted for each case)	309 ~ 312
2nd DME SOI Timing (CA , Adjusted for each case)	340.5 ~ 342.5
Mass of 1st and 2nd DME (mg/cycle)	1.826 ~ 2.541
Gasoline SOI timing (CA)	344
Mass of gasoline (mg/cycle)	15.225

Fig: 2.29 The amount of DME Injection in Intake Rebreathing Strategy [30]

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They compare the changes among IVO 345, IVO 355 and IVO 360 cases by obtaining various data at the EGR rate, volumetric efficiency, intake air temperature, DME number, CoVIMEP and combustion duration. Finally, authors conclude that as the intake temperature is relatively low, the EGR rate and volumetric efficiency are increasing with the extent of intake valve open timing. Thus, reducing intake air temperature and high concentration of EGR make CoVIMEP increased, which reduced the reaction of spontaneous combustion.

The second experiment is to investigate the influence of DME injection timing on combustion. Once the valve timing is set, the amount of DME injection and duration are the only measurement figure to control the CAI burning. However, the effects of the two kinds of DME injections are different at the beginning of combustion and combustion stage. The first one (1st DME) is that the injection time of DME is less important than injection quantity, and the injection timing is settled as 56°CA at BTDC, while the second time (2nd DME) is opposite to the first one. In the case of the DME/petrol ratio of 1:9, 2nd injection timing at various separation ratios are shown in figure 2.30 and valve timing in 2.31.

Test number	Split DME injection ratio	2nd DME injection timing
1	67%:33%	7^0 CA BTDC ~ 25^0 CA BTDC
2	56%:44%	12.5^0 CA BTDC ~ 24.5^0 CA BTDC
3	33%:67%	14^0 CA BTDC ~ 22^0 CA BTDC
4	0%:100%	16^0 CA BTDC ~ 34^0 CA BTDC

Figure 2.30 Parameters at 2nd DME Injection [30]

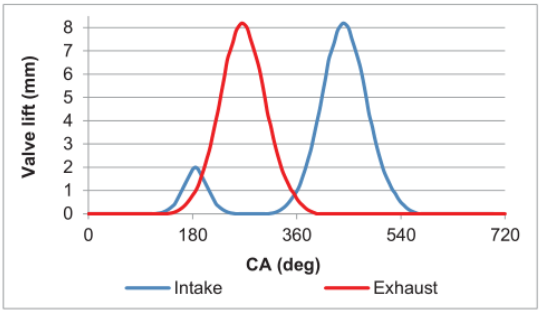


Figure 2.31 Valve Timing in the Second Test [30]

The second test is similar to the first one, authors compare the NET IMEP, CoVIMEP, duration of combustion and so on in four split DME injection ratios. It is concluded that if the injection of DME is too early or late, the output of engine prematurely will be reduced and then increases the emission of CO and HC and reduce net indicated efficiency. If it is earlier, deflagration would be caused and too late could result in incomplete combustion. Increasing the amount of 1st DME injection would delay the 2nd DME injection timing, which would improve the combustion efficiency by reducing CO and HC.

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Matsuura et al. [31] studied the influence of heat release on auto-ignition of end-gas during the flame propagation of CAI combustion. The propagation of flame is from periodic to periodic fluctuations in which the chemical energy can be released in a very short period of time causing the local pressure to rise and thus compress the unburned gas to spontaneously burn.And this end of gas spontaneous combustion is the cause of knocking, which will hinder the improvement of thermal efficiency on spark ignition engine. Although the auto-ignition will occur in each cycle, according to the different surrounding conditions, knocking is not generating cycle to cycle, thus, this instability will also make the flame exothermic mode form fluctuations, and this fluctuation of terminal gas compression and auto-ignition has greater effect than flame propagation. The authors analyze the results of end-gas compression and spontaneous combustion processes in three aspects: burning initiation, duration and flame exothermal centers by detailed chemical reaction calculations. And the results show that the compression rate of end-gas varies according to the change of exothermic mode of the flame, and the ignition delay is affected by compression speed, which in turn affects the timing of auto-ignition. If the burning speed of the flame is increasing, the combustion can be completed before the end-gas is auto-ignited, that is, the terminal gas is burned by the flame propagation instead of using pressure to spontaneously and increase the temperature of the terminal gas, thereby increasing the knocking limit. Depending on the initial conditions such as changes in pressure and air-fuel ratio, the exothermic mode of the flame is changed, and the optimum state during the analysis can effectively avoid knocking and increase the output power of the engine.

Chen et al. [32] study the effects of stratified flame ignition (SFI) and micro flame ignition (MFI) on SI-CAI hybrid combustion (SCHC) of petrol engines. The characterizes of SI-CAI is that when ignition combustion occur under high dilution condition,the cycle becomes large due to the instability of flame.The heat release in SI-CAI has two stages: the first one is the flame propagation during combustion, which plays an important role in the overall exothermic process, and the second one is the heat release at the time of auto-ignition. However, how to control the flame propagation under high dilution situation is a challenge that SCHC needs to face. Therefore, the authors propose two strategies to improve this problem: SFI and MFI. The proposed SFI is focused on

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fuel concentration, which is to improve the flame speed, and at the very beginningto promote the development of flame. This method requires higher design of fuel injector and combustion chamber. The MFI is concluded that using dimethlether (DME) as combustion enhancer under high dilution condition to promote high octane atomization and improve fuel flammability, and thus it focuses on fuel activity. The authors compare air/fuel ratio, exhaust gas temperature, in-cylinder pressure and EGR to analyze the optimal control direction of SFI and MFI. The conclusions are about that the combustion of SI-CAI can be successfully achieved at the condition of relatively balanced load and high dilution concentration. The SFI strategy can improve combustion control and spark timing and adjust the combustion stage, however, because of the large cycle and unpredictable knock, ignition and spark timing are limited in a very narrow range, and it cannot accurately determine the initiating of SI and CAI combustion. Moreover, at higher dilution concentration, there are no better optimization results are shown and therefore, this strategy need to be improved. As for MFI method, because of the addition of combustion aid, which has the roles of auxiliary ignition, inhibit the cyclical change to a certain extent and reduce the impact of spark timing on combustion cycle. These are all better than SFI method. But, because the early heat release could have influence on combustion, it also needs to evolve and improve.

Khosravi et al. [33] analyze the occurrences of the mega-knock and auto-ignition on GTDI engine. The survey uses Ford 1.6L EcoBoost engine to study the effects of abnormal combustion and mega-knock, as well as the causes of this phenomenon. In addition, the influence of residual gas on auto-ignition is analyzed by using the newly developed multi-zone model. The latest multi-zone simulation tool is used to study the effect of residual gas on vehicle ignition, this tool uses chemical reaction kinetics solver to determine the relationship between gas composition and ignition delay as a function of crank angle. It is also possible to model the residual gas time (residence time in the manifold) with the user specified. The main purpose of the experimental study is to determine whether the abnormal combustion in a failure can lead to the next round of knocking. Therefore, itfocuses on three tests: ignition skip, misfire; late spark; short ignition dwell time. The effect of residual gas on knocking in LSPI (low-speed-pre-ignition)

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is studied and it is determined whether the fire or late combustion may lead to LSPI. The researches show that, in any cases, misfire does not cause LPSI. However, in a particular cycle, a late combustion breed will instigate LSPI in the next cycle. In the cause of low engine speeds, late combustion in 1-cylinder and 4-cylinder will instigate the occurrence of 2 and 3-cylinder knock in the next cycle. However, at higher engine speeds, this phenomenon will be caused in the same cylinder. The authors also studied the effect of combustion and number of exhaust gas on spontaneous combustion behavior. The simulation results illustrate that the composition of residual gas, that is, the presence of partially combusted hydrocarbon and radicals play a key role in burning of the vehicle. It has determined that in some conditions, a short ignition dwell time may result in a very late combustion event and lead to LSPI. Although this should not happen, it produces a method of using LSPI with knocking when there is a need. There are many fundamental reasons for generating LSPI and are also fully understood, this process allows users generate LSPI via knock as needed, by which setting a cycle of late spark timing. This method can significantly reduce the test program, because the knock occurs occasionally that few times per thousands. And compared with the engine running for a long time during knocking condition, it is much less harm to the engine. In the future, this process will be used to study the effects of gas dynamics, heat transfer and mega-knock knocking, and the durability of the engine.

APPARATUS AND SIMULATION

During the past many years of the CAI technique has been mentioned, in order to study CAI combustion better and obtain its properties different from the SI mode, and the characteristics of the transition between them. Different engines are used to carry out the test (with different valve timing systems), or different software to make simulation. The benefits of using software simulation arethat the actual situation can be simulated more intuitive and meticulous, all operating process of the enginecould be simulated, and most of the aspects of engine performance can be predicted before creating engine test bed. In a word, it is impossible to test the results of all kinds of conditions in the engine, simulation could reduce the money cost on their development and also provide useful information. If not, there would be spending lots of time and money. For example, Dai et al. [34] used Ford's proprietary CAE tool (GESIM) to make

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model of ethanol or ethanol and petrol mixture and validate it in a 3.6L V6 2 valves engine. Finally, it proves the correct trend and characteristics of ethanol engine. Although it is not mainly study the characteristics of CAI combustion, it also shows the importance of software simulation. However,some of these contents are mentioned in the review above, they are not systematic and complete. Therefore, these are mainly introduced in this chapter.

Experimental Model Used in Previous Study

3.1.1 Lotus Engine Model

The models used in experiments are nothing less than the application of different engines and timing strategies.Law et al. [10] [11] [14] [15] made experiments on Lotus single cylinder engine equipped with fully variable valve train (FVVT) and with the engine specification are shown in figure 3.1 and 3.2.The results of experiments have also been described in detailed.

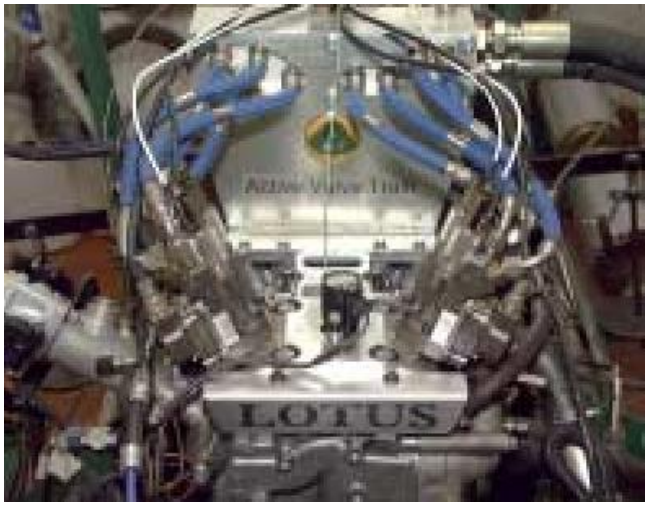


Figure 3.1 Lotus Single Cylinder Engine with FVVT [10]

Engine Type	Converted 1.8L 4-Cylinder into Single Cylinder with Active Valve Train
Bore x Stroke (mm)	80.5 x 88.2
Swept Volume (cm^3)	450
Compression Ratio, Geometric	10.5:1
No. of Valves per Cylinder	4
Valve Control	Electro-Hydraulic Active Valve Train
Cylinder Pressure Measurement	Kistler Type 6121 Pressure Transducer
Intake Air Temperature	25°C
Speed/Load Range Investigated	2000rpm, 2 – 5bar IMEP
Ignition Control	Switchable coil on plug (On or Off)

Figure 3.2 Specification of Lotus 4-Cylinder Engine [10]

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3.1.2 Volvo Engine Model

Persson et al. [37] performed HCCI experiment by using an in-line 6-cylinder 2.9L Volvo engine, which had double overhead camshafts with low-lift and short duration to produce NVO and multi-point fuel injection with natural intake characteristics. The camshaft is equipped with a code plate, which easy to adjust the set valve timing. It is interesting to note that the camshaft phase during operation is not automatically selectable so that the operating range of the engine is individually investigated for each negative overlap, thus the degree of overlap is changed by stopping the engine and manually positioning the camshaft. The other original equipment on the engine remains unchanged and the engine is equipped with heat exchangers for heating and cooling the intake air temperature to simulate the actual ambient temperature during actual operation.

SI operating range of the engine is guaranteed to be greater than the HCCI range and the experimental fuel uses unleaded petrol with a normal octane number of 98. The actual engine parameters are shown in figure 3.3.

Bore	83 mm
Stroke	90 mm
Displaced Volume	2922 cm^3
Compression ratio	10.3:1
Lift	3 mm
Inlet/Exhaust Valve Duration	150 CAD

Figure 3.3 Parameters of 6-cylinder 2.9L Volvo Engine [37]

The Kistler 6053C60 pressure sensor sited in 1 and 4 cylinders detects cylinder pressure, and the resulting data can hold 100 continuous cycles with a distinguishability of 1 CAD. Other aspects that need to be measured have appropriate instrument tracking and data acquisition. Obviously, this experimental instrument for HCCI analysis is indeed different from Lotus engine operation mentioned by Don., this modified camshaft is also an alternative to the timing of the valve generated by the FVVT, but it seems that this change in timing is still stiff and not as fast as FVVT. But on the other hand, a new camshaft is modified by the original material will save more time and effort and even costs.

Koopmans et al. [38] tested anengine with single cylinder and AVL system and a Volvo cylinder head equipped with four valves, and chose the free-valve timing strategy on the control of valve timing, which echoed with Don’s choice. Other experimental processes such as injection,

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ignition, etc. were controlled by the test bed management system. The water-cooled Kistler 6061b sensor with low thermal shock behavior was used to test the cylinder pressure in 150 consecutive engine cycles, and the obtained data was recorded in MATLAB with a resolution of 1 CAD to MATLAB for analysis. The specification of the engine is shown in the figure 3.4.

Valve-lift	8 mm
Bore	83 mm
Stroke	90 mm
Compression ratio	11.5
Displaced volume	487 cm^3
Fuel (CEC legislative fuel)	RF-08-A-85, 97 RON
Inlet temperature	25 °C
Water / Oil temperature	90 °C
Fuel pressure	100 bar

Figure 3.4 Tested Engine Specification [38]

3.1.3 Renault F7P Engine Model

Knop et al. [39] explored the characteristics and effects of CAI combustion by using model tests and software simulations, and finally make comparison among these data. As most off the experiments mentioned, the PFI natural intake engine (derived from the Renault F7P engine series), which is a single-cylinder optical visual one, and the timing system is replaced by FVVT, which is used to implement the NVO strategy.

Some of the engine specifications and schematics are shown in figure 3.5 and 3.6.

Engine Type	Single-cylinder optical engine based on RENAULT F7P
Bore [mm]	82
Stroke [mm]	83.5
Compression ratio	11.5:1
Connecting Rod Length [mm]	144.0

Figure 3.5 Engine Specification of RENAULT F7P [39]

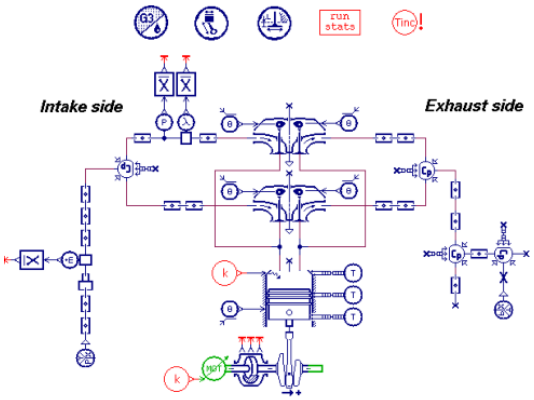


Figure 3.6 The Schematic Diagram of Test Engine Modified from IFP-ENGINE [39]

The detailed description of intake and exhaust lines is to accurately predict the gas exchange process, and the sub-mode of two cylinder headson combustion chamber are to allow the implementation of different valve strategy and the unique pipeline design to each valve.

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3.1.4 Buick Regal Engine Model

Thakkar [40] at Michigan Technological University performed the HCCI experiment by using a modified GM 2.0L Turbo I4 Ecotec LHU engine. The parameters of original engine are presented in figure 3.7.

In order to achieve and detect HCCI process, the main increases in the engine are the intake heater and EGR pipe, and other components are in figure 3.8.

Parameter	Value (units)
Bore	86 mm
Stroke	86 mm
Compression Ratio	9.2:1
Displacement Volume	1998 cc
Valve Configuration	Dual overhead camshafts
Valve Lifter	Hydraulic roller finger follower
Firing order	1-3-4-2
Number of Valves per Cylinder	4
Fuel System	Direct injection
Fuel Type	Premium recommended, Fuel flex capabilty
Rated Power	220hp (164 kW)@ 5300 RPM
Rated Torque	260lb.-ft (353 Nm)@ 2400 RPM
Maximum Engine Speed	6350 RPM

Figure 3.7 Original Parameters of GM 2.0L Turbo I4 Ecotec LHU Engine [40]

Auxiliary Parts	Reason
EGR valve	To control EGR into the system
EGR cooler	To cool the EGR to desired temperature
Thermocouples	To measure temperature at various location
Pressure transducer	To acquire and study pressure data various locations
Lambda sensor	To measure and monitor air-fuel equivalence ratio
Air heater	To increase the intake air temperature to enable HCCI operation even with low compression ratio engine
Crankshaft encoder	To measure crank rotation at specified sample rate
Intercooler	To cool the compressed intake air
Super-charging station	To supply boosted intake air while the original turbocharger is not used

Figure 3.8 Added Parts on GM 2.0L Turbo I4 Ecotec LHU Engine [40]

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Flywheel is made a small modification that connecting the electromotive force through universal joint, fuel injection and intake pipe are also slightly modified to ensure the ability of port injection, and change compression ratio to promote auto-ignition process. The final modified engine concept map is seen in figure 3.9. The engine instrumentation has been designed which containing sensors and actuators.

After the engine is modified, the experimenter used a kind of model named grey box to control exhaust and predict the emissions, which is a new type of emission control mechanism. The main box is composed of physical model and three artificial neural feedback networks. The primary purpose is to detect the six results of HCCI burning: load conditions, exhaust temperature, combustion stage, NOX and CO and the top dead center. This model has no more than 10% difference between the prediction of HCCI combustion and actual result error, and it also could be used on different engines by multiple tests.

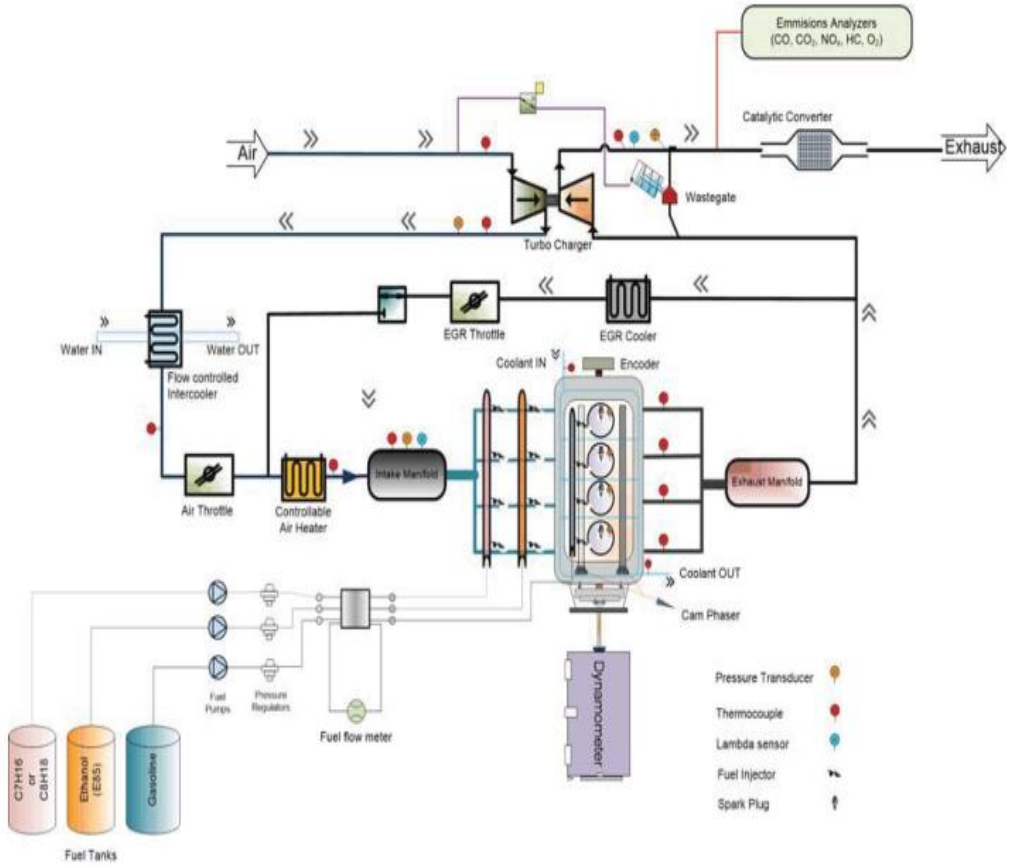


Figure 3.9: Schematic System of Designed Engine [40]

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3.1.5 Ford Engine Model

Khosravi et al. [33] used a 1.6L EcoBoost GTDI Ford engine to study the causes of abnormal combustion and mass explosion during auto-ignition. This paper has already said in above about its objective. The survey focused on the influence of residual gas after detonation on the knocking of multi-cylinder engines, which is whether the abnormal combustion in one cycle will explode in the next round. The engine is modified to a new combination and the specification is in figure 3.10, it is equipped with a gas cylinder and low pressure indication system with compression ratio of 10:1 and is also set up with a SMETEC Combi PI prediction system that explores the beginning of auto-combustion through spark plug's current sign. Therefore, this Ford engine is used to experiment with the knock-related problems.

Engine Type	I4 1.6 L GTDI
Material	Aluminum block and head
Charging	Single scroll turbocharger with 4 - 1 exhaust manifold
Fuel injection system	High pressure common rail
Fuel injector	Central multi-hole
Firing order	1-3-4-2
Valve-train	DOHC, Intake and exhaust cam phasing
Compression ratio	10.0 : 1
Bore x Stroke	79 mm x 81.40 mm
Rated power	125 kW (167 hp) at n = 5500 rpm
Torque	240 Nm (177 lb-ft) at n = 1800 - 3500 rpm

Figure 3.10 Ford Test Engine Specification [33]

3.1.6 Jaguar Engine Model

Xuet al. [41] studied HCCI combustion characteristics (pressure, temperature and substance concentration) and gas exchange modeling by simulating the Jaguar V6 3.0L engine and analysis with corresponding simulation software. The engine parameters are shown in figure 3.11.

Engine type	4 valve per cylinder
Bore	89.0 mm
Stroke	79.5 mm
Compression ratio	11.3
Engine speed	1500 rpm
Intake valve timing	variable
Exhaust valve timing	variable
Intake temperature	300°K
Fuel	95% iso-octane + 5% n-heptane
Air/fuel ratio	14.3

Figure 3.11 Jaguar Engine Specification for Testing [41]

This engine is used the technology of exhaust valve closed early to trap a large amount of residual gas, which not only could promote the occurrence of auto-ignition, but also improve the intake air temperature. This skill has already used by Don. In Lotus engine, while in the Jaguar engine is also smooth application. During the engine speed as 1500 rpm, the compression ratio of engine is improved from PFI version at 10.5 to 11.3 of HCCI.

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Meanwhile, the fuel used was a mixture of 95% isooctane and 5% n-heptane, and initial air temperature is set at 300K, the pressure of intake manifold is 90 KPa, and the stoichiometric amount of air/fuel ratio is 14.3.

3.1.7 Other Sub-Model or Equipment

The following article which written by Epping et al. [36] does not specifically mention which engine is used in experience. It's only illustrated that the experimental work so far mainly has included the use of high-speed four-cylinder engines in cars and single-cylinder in heavy trucks, at the same time, demonstrated that the use of electro-hydraulic VVT systems researched by Stanford University to introduce exhaust gas and HCCI combustion was achieved with a low compression ratio of 10:1 and high octane fuel. The valve timing system is basically similar to the FVVT, which previously mentioned the articles used by Don., both of them are used to allow intake and exhaust valves to have any lifting curve, the concept picture is shown in figure 3.11.

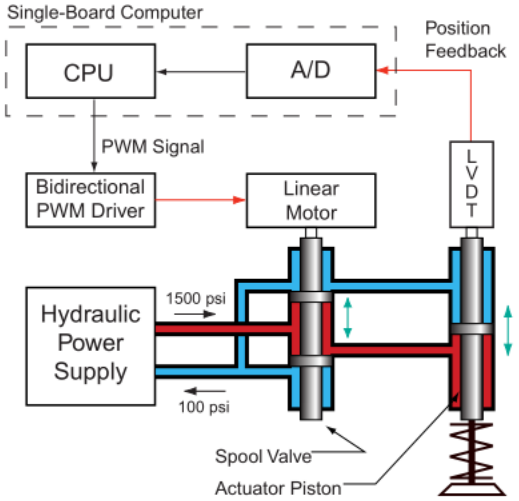


Figure 3.12 VVT System Developed by Stanford University [36]

Actually, no matter it is to achieve CAI or HCCI, the valve timing system is indeed different from that the traditional engine equippedonly to achieve SI mode. The valve strategy can expand operation range of HCCI, which is also used in conjunction with DI system to demonstrate different types of fuel injection and combustion are integrated into HCCI's multimode internal combustion engines. This study is intended for using in conventional metal-made engines and is also should be used in optical engines. This new approach can be used to investigate the main parameters of direct venting of HCCI by exhaust gas re-directing, and these experimental data will help researchers quantitatively predict the performance of new hybrid engines and design the best system performance process.

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3.2 Gathering of Simulation Software Used in Previous Study

3.2.1 Ricardo Wave

Don. [35] uses software called WAVE to simulate the operation of 2.0L PI and 1.6L DI engines from 1000 rpm to 5000 rpm, and the models are shown in figure 3.12 and 3.13.

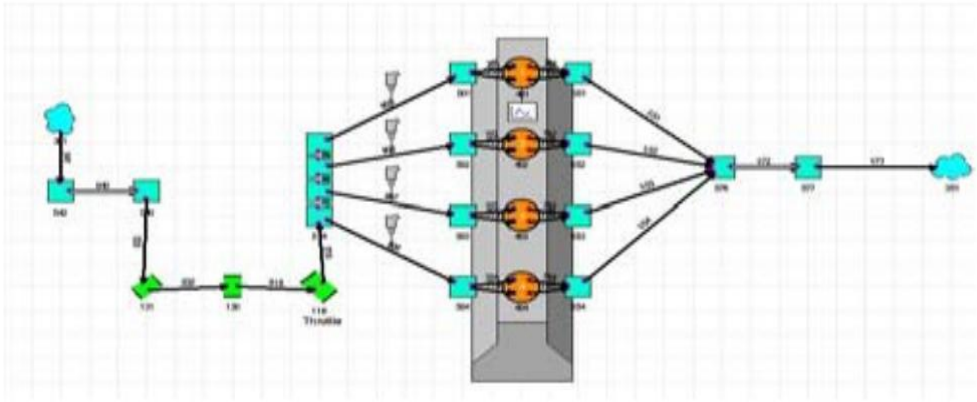


Figure 3.132.0L Port Injection Engine Mode [35]

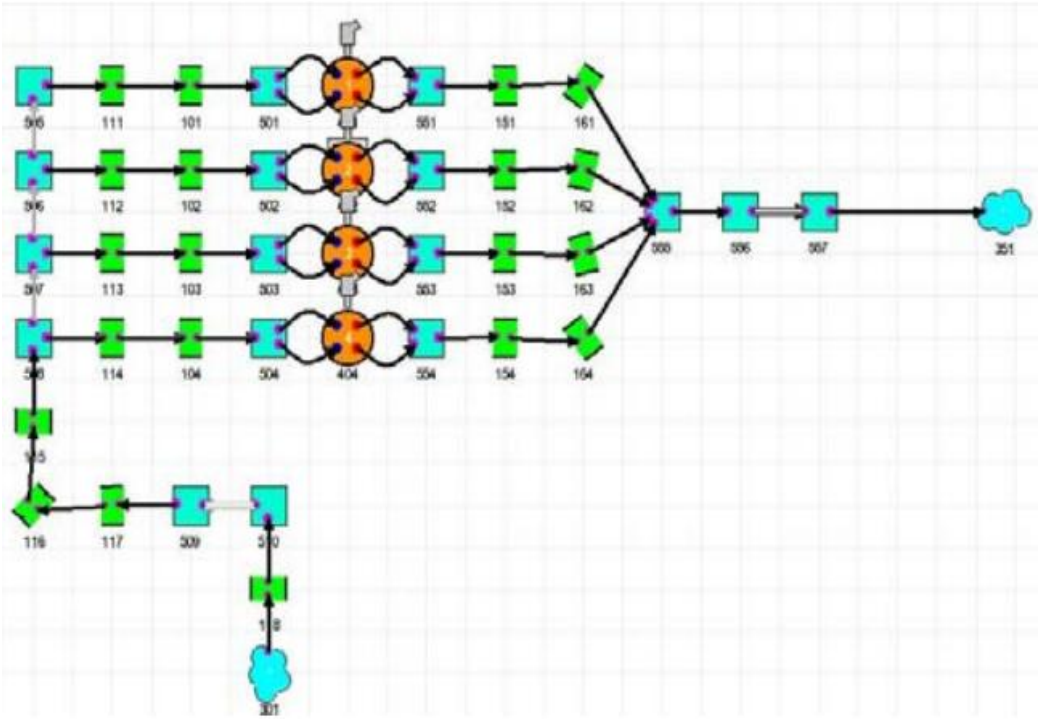


Figure 3.141.6L Direct Injection Engine Mode [35]

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The program represents the engine performance and 1D-2D gas dynamics, focusing on the combustion switchbetween SI and CAI mode and the key data on combustion of the valve opening duration: pressure, power, temperature and emissions. In analog conversion between SI and CAI, set 2000 rpm as the initiate point since CAI is easily achieve at here, no matter the test on 1.6L or 2.0L engines, between various switch makes SI combustion of unimodal and CAI combustion shows bimodal characteristics can be truly appear in the image, while CAI changed to SI is tested in the 5 Volvo engine cylinder, and the simulation is consistent with the result display.

3.2.2 Matlab

Koopmans et al. [38] said that the previous papers had showed the original model was evolved from the simulation code of engine cycles, the modeling software is the simulation code for AVL BOOSTTM, which is belonging to Matlab. Now this test needs to modify the SENKIN code in CHEMKIN (ie add a new term) to implement the piston motion and heat transfer model and use it as calculation of high pressure cycle. CHEMKIN is a kind of code which including the collection of databases and subprogram that used to study chemical kinetics, balance and transport characteristics, which combined with the corresponding procedures that can be used to research pressure in cylinder and temperature changes in the engine.[41] In order to cooperate with different combustion strategies to study HCCI combustion means, add new terminology in the SENKIN program so that it is not only applicable to port injection but only for direct injection, the influence of direct injection is mainly due to the effect of temperature and the descent of combustion before the piston reaches the top dead center when negative valve overlap occurs.

The application of two codes duringthe test as shown in figure 3.14.

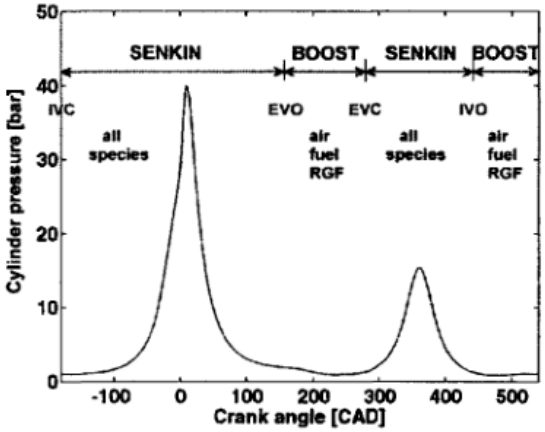


Figure 3.15 BOOSTTM -SENKIN Model for Typical HCCI Cycle [38]

BoostTM is mainly used to calculate and distinguish the residuals, air and fuel during gas exchange. SENKIN is used to

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evaluate the closing interval of intake and exhaust valves and to recalculate the fractional mechanism of all the substances in reaction at the beginning of the program calculation. BOOSTTM -SENKIN program eventually calculated the results obtained through a single zone combustion, and the results of calculation and experiment are much more similar, that is to say that using Matlab numerical simulation to evaluate the experimental figure is proved to be feasible.

Knop et al. [39] using 1D engine simulation and 3D combustion code. In order to correctly simulate all engine elements and be able to maintain a short time in calculations ofmultiple cycles, digital simulation analysis of 1D and 3D CFD have to be combined. 1D is used to simulate the gas exchange process and the necessary parts are entered into 3D CFD for calculation, and the range of the 3D CFD pair is limited only in the combustion chamber in order to describe the determined automatic ignition, combustion and contaminant formation process. The resulting data is transferred again to the 1D analog code for recalculation of the compression and work strokes. The 2D mesh is included in the 3D CFD to simulate the temperature of the intake air and other intake composition (i.e., the condition of the reaction). And the final experiment is similar to the calculated result.

GESIM is used in Ford SI engine simulation program, mainly to simulate the exhaust, combustion and heat transfer on the engine and so on. Actually, we can see that there are many simulation programs on study HCCI/CAI engine combustion properties, but they are used in different ways to detect different nature, and also associated with each other. This simulation program was originally used to design the SI engine, so if it is used to simulate the HCCI engine, there should make some adjustments accordingly, for example, setting the ignition timing before piston reaches top dead center of about some degrees (according to the specific circumstances and since auto-ignition of HCCI should be controlled near the TDC); adjusting the control of burning rate calibration constant and so on. CHEMKIN calculates the ignition timing, pressure and other parameters and retains. GESIM has a communication iteration relationship with CHEMKIN, and GESIM feeds the results back into CHEMKIN by calculating and predicting the pressure and temperature variations of the entire engine cycle. After that, make the comparison among these results, if the convergence of standard to

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meet the results of the model and then, make subsequent comparison of experimental data, if it is not meet with the requirements, the relevant adjustment should be done. This method of analyzing auto-ignition combustion is presented in the form of code, which is simpler than analog on physical engine simulation.

3.3 Summarize of the Third Part

This part is mainly summary the CAI/HCCI combustion equipment used in simulation and the classification of simulation software in recent years. In this section, we can see that there are many scientists in the world to study this new type of combustion every year, using different experimental equipment, which just to test the different characteristic of combustion in different aspects and equipped with different valve timing system. Firstly, the combustion results of the experiment will not result in different burning properties due to different equipment. Secondly, there are indeed many simulation software to simulate CAI burning in fact, such as Ricardo WAVE, Simulink that have mentioned above, and Hypermesh, GTPower, etc that not said in this article. Different software does the different aspects of dynamic simulation on engine work, which compared with actual operation, this way can be continuously debugged until optimal results could be obtained, while saving time and effort and could greatly improve the work efficiency.

CONCLUSION

In summary, this thesis is mainly studied the direction of research and progress on the field of CAI after Don's paper. The concepts and principles of CAI are described in detail at the beginning of presentation, thus we can also know that this type of combustion is indeed quite different from SI and CI. However, in the literature review section, it can be seen that the combustion of CAI is not carried out in the whole cycle of engine, and more experiments have taken SI-CAI-SI as the steps of the experiment. This is because CAI combustion cannot be filled with the entire combustion cycle of engine, and the burning conditions are harsher than SI mode, which is means that the engine could not always meet CAI condition during the operation. Therefore, CAI combustion has to be interspersed in SI mode, which can effectively reduce the same situation under the pure SI engine generated pollutants, and improve the fuel economy efficiency. Although many countries are advocating energy-saving emission reduction now, and vigorously called for the development of new types of vehicles and fuels, even if the new car

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research, development, production, manufacturing and finally put into use, but it still need to invest in market for testing. Therefore, my personal view is that the practicality of the engine cannot be completely replaced in a short time, even in a hybrid car, the engine is still the core of the region, that is, CAI research is valuable. The study has been shown for about 30 years, and its benefits are obvious, but, there is some difficulty in the promotion of globalization, first of all, the technical level of each country is uneven, and CAI burning occurred need high environmental requirements, which is why more experts are more inclined to study and develop the hybrid cars.

The second focus point of this paper is to emphasize the distinction between CAI and HCCI. These two technologies are similar, the difference is that CAI can only be used in the gasoline engine, and HCCI is used in the diesel engine. Both technologies require exhaust emissions from the engine after one round of the cycle, and CAI keeps the gas in the tank through internal exhaust cycle, and the HCCI's exhaust is outside into pipe, flows through the heater and finally into the cylinder. This is the biggest difference between them, which directly leads to the difference is the nature of the fuel. If the exhaust gas produced by the combustion of diesel does not go directly into the cylinder without passing through the heater, the large amount of water vapor in the air mass will cause the engine turn off, and will not cause HCCI combustion to occur. But in many literature, HCCI, CAI is often thought common, but in fact the two are essentially different.

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