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Advanced combustion for low emissions and high efficiency: a literature review of HCCI combustion concepts

Advanced combustion for low emissions and high efficiency: a literature review of HCCI combustion concepts

A review of current scientific knowledge and technology status, prepared for the CONCAWE Fuels Quality and Emissions Management Group by its Special Task Force FE/STF-26:

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ABSTRACT

This report reviews an extensive published literature covering a class of advanced combustion concepts for low emissions and high efficiency in internal combustion engines. For diesel engines, these combustion concepts are generically called Homogeneous Charge Compression Ignition (HCCI) and, for gasoline engines, Controlled Auto-Ignition (CAI). The market drivers for exploring these advanced concepts as well as the fundamentals of how they work to enhance performance and reduce engine-out emissions are described using examples from recent literature. Following this introduction, various engine hardware options are reviewed that have been reported to enhance performance and extend the practical speed and load range for advanced combustion modes. Finally, the impact of fuel properties are described for enabling advanced combustion, including the impact of the fuel's ignition resistance (cetane number), volatility, and fuel composition on engine performance, emissions, and noise.

KEYWORDS

Internal combustion engine technology, diesel, gasoline, fuel, combustion, Homogeneous Charge Compression Ignition (HCCI), Controlled Auto-Ignition (CAI), exhaust emissions, cetane number, octane number, RON, MON, biofuels, alternative fuels

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SUMMARY

This report reviews an extensive published literature covering a class of advanced combustion concepts for low emissions and high efficiency in internal combustion engines. For diesel engines, these combustion concepts are commonly called Homogeneous Charge Compression Ignition (HCCI) and, for gasoline engines, Controlled Auto-Ignition (CAI). In fact, these terms are quite generic and describe a broad range of advanced combustion concepts that combine the best features of spark- and compression-ignition engine technologies.

The market drivers for these technology developments are the usual ones – cost-effective and simultaneous improvements in exhaust emissions, fuel consumption, and vehicle performance. In general, advanced combustion for low emissions and high efficiency offers different advantages depending upon whether the starting point is a spark- or a compression-ignition engine. For spark engines, CAI combustion can reduce fuel consumption over the portion of the speed and load curve where it can be successfully applied. For diesel engines, HCCI combustion can reduce engine-out emissions and thereby reduce the performance requirements and costs of the vehicle's aftertreatment system.

These advanced combustion technologies depend on the fuel being sufficiently premixed with air before spontaneous combustion begins inside the engine cylinder. The primary advantage of this mode is that the fuel charge is more completely combusted under conditions in which emissions, especially NOx and particulates, are reduced. In this report, the characteristics and requirements of these advanced combustion concepts are analyzed based on an extensive and growing literature from automakers, suppliers, government labs, and academic groups. The fundamentals of advanced combustion are also explained beginning with the critical importance of controlling the ignition delay through hardware options and fuel properties. Because some of these hardware options individually improve performance in both conventional and advanced combustion modes, they are expected to become standard practice on most modern engines.

Much of the existing literature on advanced diesel combustion has focused on the ways in which fuel properties can enhance performance and extend the operability range, especially by making the fuel more resistant to ignition and more volatile compared to today's diesel fuels. The impact of fuel properties is discussed in detail, including the preferred ranges for cetane number, volatility, and fuel composition. Other properties, such as deposit-forming tendency, lubricity, and the future potential for bio-containing and alternative fuels, are also reviewed.

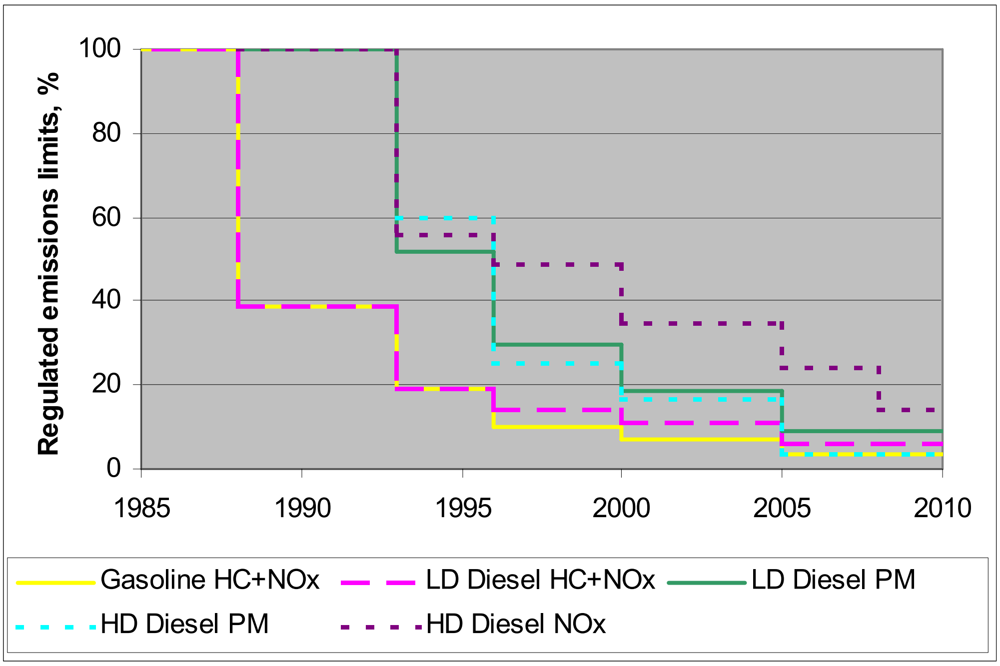
The overall advantages of these combustion concepts, and the costs that can be avoided, depend on the portion of the speed and load range where the approach can be routinely applied. Currently, full-time HCCI does not appear possible over the entire engine operating range so that engines utilizing advanced combustion will still need to revert to conventional diesel or gasoline engine performance under some operating conditions. When this occurs, the engine will require fuel properties that are also fairly conventional. Although some fuel properties may help to extend the range of advanced combustion performance, the available literature does not suggest that an "HCCI only" fuel will be required at the service station in the near future.

1. INTRODUCTION

1.1. EUROPEAN ROAD TRANSPORT

The volume of road traffic in the developed world has increased dramatically over the past 40 years, and this same growth is now being seen in other developing countries, especially China and India. This growth has brought challenges for the vehicle and fuel industries, in particular the need to control vehicle pollutant emissions (**Figure 1**). The success of the vehicle industry to improve emissions performance has been impressive, and emissions from today's road vehicles are many times lower than the levels prevailing before emission regulations were introduced.

Figure 1 Change in European Regulated Emission Limits from Vehicles



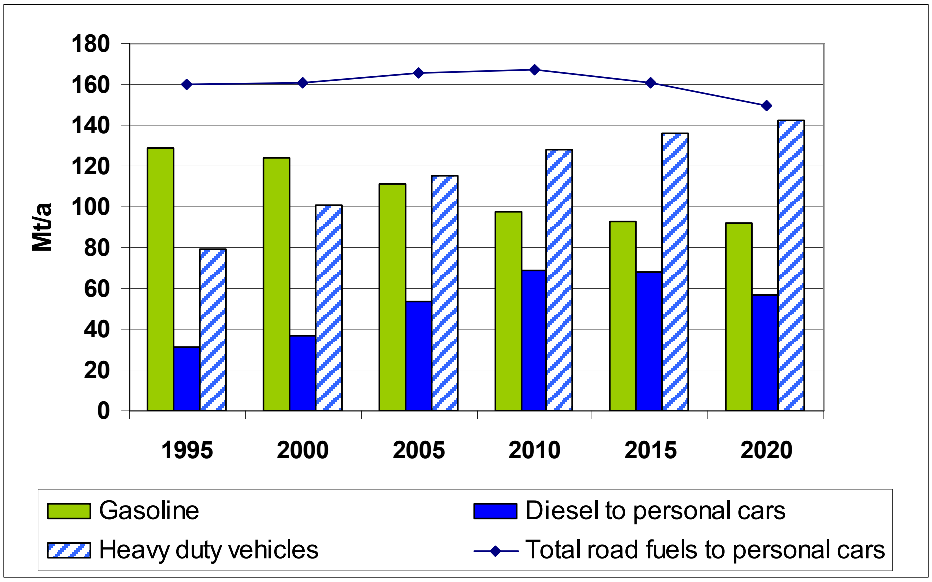
More recently, attention has focussed on vehicle efficiency, driven by concerns over long term energy supplies and greenhouse gas (GHG) emissions, particularly CO_2 which is directly linked to fossil fuel use. Regulations to limit the total fuel consumption of the vehicle fleet were first introduced in the USA through the 1975 Corporate Average Fuel Economy (CAFE) regulations. Similar requirements have been introduced in Japan and other countries while, in Europe, energy use has been cast in terms of CO_2 emissions. In the 1990's, the European vehicle manufacturers entered into voluntary agreements to limit the fleet emissions of new passenger cars to 140 g/km by 2008-2009. Although this voluntary target was not met, this initiative was partially successful in encouraging the development and marketing of more efficient vehicles, but customer preferences for larger, heavier, and more powerful vehicles have largely eliminated these improvements. Some of the increased vehicle weight was also due to mandated safety features and other customer preferences. For these reasons, the light-duty overall fleet energy use and CO_2 emissions have continued to rise.

With each new hurdle in emissions performance, innovation by the vehicle manufacturers and their suppliers has allowed the Internal Combustion Engine (ICE) to successfully meet each new technical challenge. As a result, the ICE remains today almost the universal choice for power plant in road vehicles. Worldwide, gasoline spark-ignition vehicles dominate the passenger car market and most modern gasoline vehicles use stoichiometric combustion coupled with a Three-Way Catalyst (TWC). The engine-aftertreatment combination achieves very low levels of tailpipe emissions, although with some sacrifice in fuel efficiency.

Diesel engines also compete strongly in the passenger car and light-duty vehicle market. Modern diesel engines provide good power and torque as well as low fuel consumption, but with some sacrifice in smoothness and higher emissions compared to a gasoline engine. A diesel engine consumes up to 30% less fuel on a volume basis (slightly less on an energy basis) compared to a gasoline engine but they also emit higher quantities of particulates and NOx in spite of significant technology improvements. Although diesel cars are available in many areas of the world, it is in Europe where they have been most successful, with over 70% of new cars being diesel powered in some countries. Vehicle and fuel taxation rules have also encouraged the purchase of more fuel efficient, and hence low CO_2 emission vehicles. In the medium- and heavy-duty vehicle categories, almost all are powered by diesel engines. In fact, about half of the road transport fuel in Europe today is consumed by goods vehicles.

As shown in **Figure 2**, the total road transport fuel demand by EU-25 passenger vehicles is expected to decline beyond 2010 in spite of a growth in vehicle kilometres. This is because fleet average fuel consumption is expected to improve as diesels and more efficient gasoline cars replace older vehicles in the fleet. In contrast, there are fewer opportunities for efficiency improvement in heavy-duty diesel vehicles, and overall fuel use for these vehicles is expected to grow, driven by increased goods transport, particularly in those countries that have recently joined the European Union.

Figure 2 Projected Road Transport Fuel Demand for EU25[1]



[1] Data are from CONCAWE, based on a multi-client study by Wood Mackenzie

Looking to the future, alternative power plants, such as fuel cells, still face many research and development challenges and most projections show them making relatively little impact on the market before 2030. ICEs are therefore expected to provide practically all of the power plant needs for road vehicles in the immediate future. For this reason, significant research efforts are being made to further improve engine performance, with the goal of producing vehicles that simultaneously achieve low emissions and low fuel consumption. The objective is to achieve these improvements within the engine, minimising the need for exhaust aftertreatment, which is becoming increasingly complex and can itself increase fuel consumption in some cases.

1.2. ADVANCED COMBUSTION TECHNOLOGIES

These advanced combustion technologies are the subject of this report. In the technical literature, the terms 'HCCI' (Homogeneous Charge Compression Ignition) and 'CAI' (Controlled Auto Ignition) are often used to describe these concepts. However, these terms have meanings that are very specific to the way that combustion takes place and they are not the only alternative combustion approaches that are currently being considered. In fact, after much more development, it is likely that practical engine systems will use combustion schemes that cannot strictly be called either HCCI or CAI. For this reason, we prefer the more generic term 'Advanced Combustion for Low Emissions and High Efficiency', defined by reference to the performance of the combustion system rather than the way in which the combustion is performed.

In this report, we will use the generic term 'HCCI' in a broad sense to cover all such advanced systems that demonstrate:

- low engine-out emissions (of CO, HC, NOx, PM etc);
- low fuel consumption (comparable to or better than conventional diesel); and
- stable operation over a wide load range.

1.3. EVOLUTION IN FUELS

Fuel developments have also played a significant role in improving vehicle performance. Perhaps the most significant change was the removal of lead additives from gasoline, thus allowing the application of very effective TWC aftertreatment systems to reduce emissions. Lead additives had been used to boost octane since the 1930s but the lead was removed from US gasolines around 1975, with Europe following in 1989. Japan removed lead from regular gasoline in 1975 and from premium gasoline in 1987. More recently, a concerted effort has been made through the United Nations Environment Programme (UNEP) to phase out lead in gasoline from the African continent and considerable progress has been made since 2000. Today, only 18 countries worldwide are still using lead in motor gasoline.

Throughout the 1980's and 1990's, other changes were made to fuel specifications to provide lower emissions in conjunction with more stringent vehicle emissions standards. The increasing reliance on exhaust aftertreatment resulted in fuel sulphur levels receiving the most attention in specification discussions. Fuel sulphur is a much less serious problem for catalysts than lead but it does compete for active catalyst sites and can lower catalyst efficiency. The lower levels of sulphur agreed for Europe, USA, and Japan have been motivated in part to enable new exhaust catalyst technologies. In some cases, sulphur levels lower than are strictly necessary have been regulated to enable additional emissions improvements. The levels to be introduced in Europe by 2009 (<10ppm S) mean that gasoline and diesel will be effectively sulphur-free by that date. A recent paper [1] concluded that developing countries do not necessarily need to apply the same ultralow sulphur levels as in Europe and the USA to enjoy the benefits of low emission vehicles.

The emission reductions that can be achieved by advanced vehicle technology in combination with lower fuel sulphur levels are significant and far outweigh emission reductions that can be achieved through other fuel changes. CONCAWE's technical view on future fuel quality has therefore been that:

- Sulphur-free fuels meet the needs of all vehicle technologies expected in the market over the next ten years;
- Changes to other fuel properties offer little additional air quality benefit and significant potential to increase refinery CO_2 emissions.

Although this is our view, CONCAWE also believes that alternative combustion systems, such as HCCI and CAI, need more study. Indeed, the way in which combustion takes place in these new systems is so different from the conventional Spark Ignition (SI) or Compression Ignition (CI) engines, that some have suggested that an engine that combines the benefits of gasoline and diesel engines might require a fuel that is also unique and different.

For such an optimisation to occur, alternative combustion would need to be sustained over the whole engine operating envelope. While this may be feasible in the long term, most alternative combustion systems, at their present state of development, can only be operated over a portion of the light load range. At high loads or, in some cases, at very low loads, they must revert to conventional SI or CI combustion modes. In such cases, the fuel would need to retain characteristics enabling it to operate as a conventional gasoline or diesel fuel, thus limiting the extent of possible fuel adaptation. As this report will show, the engine configurations that will eventually be successful and commercialized are not yet clear. We can already see some trends in fuel response in these engines but detailed recommendations must wait until the best performing and most cost-effective engine configurations are more clearly defined.

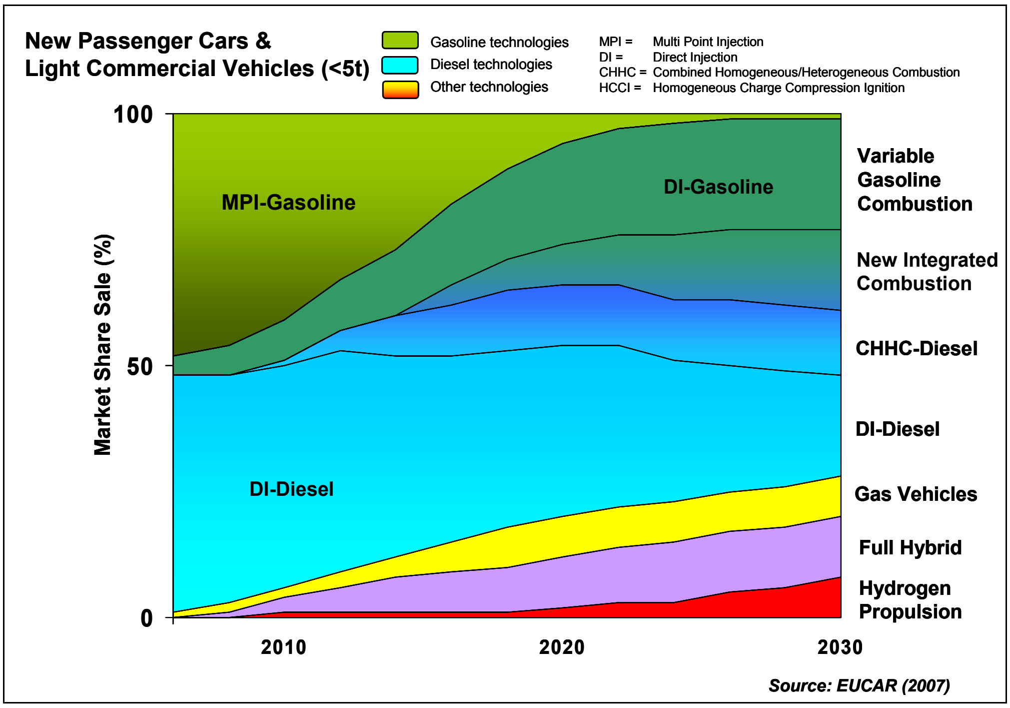
2. TECHNOLOGY/DEMAND OUTLOOK

2.1. VEHICLE TECHNOLOGY

The ICE is by far the dominant power plant for road vehicles today. The majority of today's vehicles are fuelled by gasoline or diesel with some contributions coming from Liquefied Petroleum Gas (LPG) and Compressed Natural Gas (CNG). Direct electric power has had only a limited impact so far, most significantly for local delivery vehicles and for trams in some cities. In the future, electric power will find some use in the form of hybrid electric or plug-in hybrid vehicles which reduce energy consumption by more efficiently using the energy produced by the on-board engine. In the near-term, this energy will continue to be produced by an on-board ICE. In fact, the high cost of hybrid electric vehicles may limit their penetration, particularly in Europe where they must compete with efficient and customer-accepted diesel vehicles. The lower use of home garages in Europe may also limit the penetration of plug-in hybrid vehicles unless fast and convenient re-charging facilities are made available.

For the longer term, fuel cells potentially offer higher efficiency than ICEs, but many technical and cost hurdles remain before they can be commercially ready. In addition, the fuel cells favoured for vehicle use (using polymer electrolyte membranes (PEMs)) must be fuelled with hydrogen and the energy needed to produce the hydrogen, either at a central plant or on-board the vehicle, can eliminate some of the efficiency advantages [2]. Forecasts of future vehicle technology (**Figure 3**) predict only a small impact of fuel cells before 2030, with improved ICE vehicles continuing to provide the main source of power.

Figure 3 Evolution of light-duty vehicle technology in Europe



2.2. THE ROLE OF HCCI

A wide range of HCCI technologies is being explored to address the challenges of improving efficiency for spark engines and reducing emissions from diesel engines. In broad terms, the tools that are available to engine developers to meet these challenges can be categorised as:

- combustion chamber design changes,
- air management improvements including variable valve systems,
- fuel injection developments to improve the air-fuel mixture formation,
- exhaust gas recirculation (EGR) to control the combustion environment, and
- engine management and combustion control systems, perhaps including closed loop combustion control using in-cylinder pressure measurements.

These technologies are being used to move the engine to higher load operating points through engine downsizing or cylinder deactivation. All of these technologies can also be used with conventional SI and CI combustion. Whether the additional step of moving to an advanced combustion system is ultimately worthwhile will depend on the incremental benefit in emissions and performance compared with conventional combustion using the new hardware.

For spark engines, the primary goal is to improve fuel efficiency, so the advantage may appear as a few percentage points, which could perhaps be achieved in other ways using conventional combustion. In principle, it may not be necessary to achieve HCCI combustion over the whole load range, since the TWC would ensure that emissions remain low when the engine reverts to normal combustion. If this were the strategy, the efficiency benefits would be proportionately reduced, however. For diesel engines, the prize is higher since the successful reduction of engine-out emissions could significantly reduce the cost of exhaust aftertreatment. The hurdle is also higher for diesel engines, however, because low emissions must be preserved over a wide engine operating range.

The fuel needs of conventional combustion systems are well understood. This report does not attempt to review conventional engine developments, except as a means of gauging the likely success of HCCI options. The focus is instead on alternative combustion strategies where the optimum fuel needs may be different.

2.3. FUEL DEVELOPMENTS

Alternative ICE fuels such as Compressed Natural Gas (CNG), Liquefied Petroleum Gas (LPG), and methanol have so far had only limited success in the market as a whole, although in some specific applications they can have a significant impact. In Europe, LPG represents a well established niche market, used mainly in passenger cars that are often retrofitted with aftermarket kits. Its use is also encouraged by lower fuel taxes that lower the fuel price at the pump. Even with substantial price advantages, however, LPG use has been limited representing only around 1% of European road transport use. Although a large amount of LPG is produced by refineries, this production is already committed to other uses such as for heating or as a chemical feedstock and local production is often supplemented by imports. The automotive LPG market is generally considered to be mature and significant further growth is not expected.

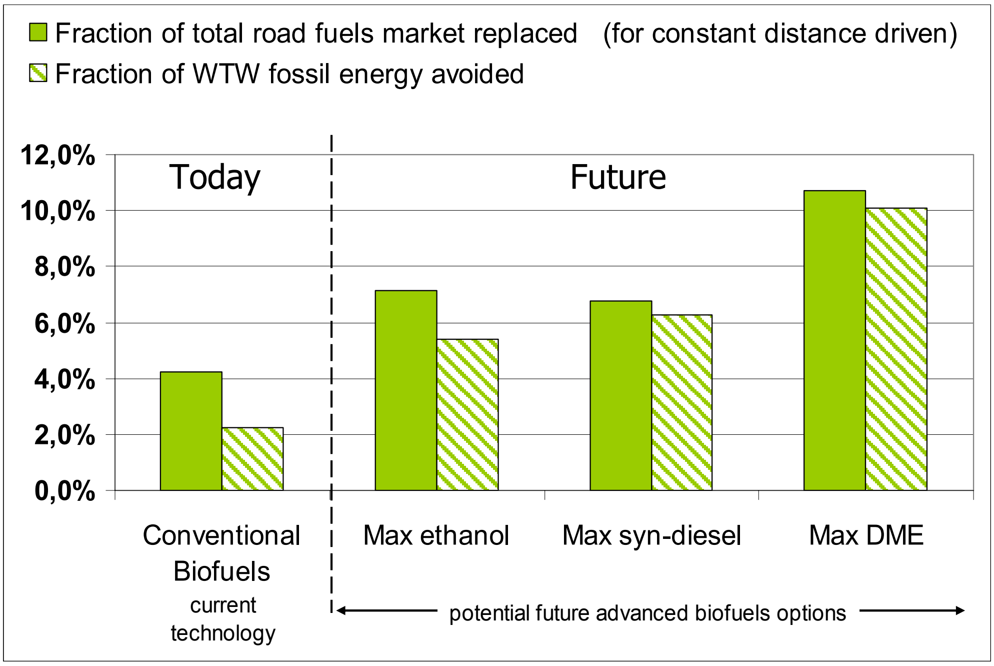
CNG has primarily been used in heavy-duty applications, especially in captive fleets such as urban buses and delivery trucks. In these applications, the SI engine's advantages of low exhaust and noise emissions are a benefit particularly if older diesel vehicles are replaced by CNG vehicles. This trend is expected to continue although CNG faces increased competition from modern low emission diesel engines that also provide better fuel efficiency. Overall, the ready availability, convenience, and performance of liquid gasoline and diesel fuels have made it difficult for these alternatives to find wider application.

The increased interest in energy substitution and GHG emissions continues to drive the search for alternatives to conventional gasoline and diesel fuels, however. Some researchers in advanced combustion have investigated new fuels that could facilitate the development of these systems. It is therefore important to look at the likely trends in fuel availability and the options available for future automotive fuels. To be successful, a future fuel will need to be readily available and have acceptable efficiency with respect to primary energy use and GHG emissions. Well-to-Wheels (WTW) analysis has become an established modelling tool to address these questions. In Europe, the most comprehensive and authoritative source is the JEC WTW study [2], which was jointly authored and regularly updated by the European Commission's Joint Research Centre, EUCAR, and CONCAWE.

Production of liquid fuels from biomass is being encouraged in Europe through the Renewable Energy Directive, with similar initiatives in place in many other countries. **Figure 4** shows projections from the JEC WTW Study of the percentage of road fuel that could potentially be substituted by biofuels from European resources. This study estimates that the percentage of road fuel that could be substituted by biofuels produced from European resources is around 4% using today's technology, rising to perhaps 10% for advanced 'Biomass to Liquids' (BTL) concepts. If non-conventional fuels such as hydrogen and DME are also considered, potential yields are higher but still limited to about 20%.[2] The resource availability calculated on a world basis is similarly limited, and has been estimated to be about 2% of overall liquid fuel demand by 2030 [3].

[2] The pace of growth and future availability of many advanced biofuels are highly uncertain and will depend on many factors that are beyond the scope of this report.

Figure 4 Potential European biofuel availability in 2020



Using today's technologies (ethanol from grain or sugar beet and FAME from plant and waste oils) and all available resources, the JEC Study estimates that a volume of conventional biofuels could be produced equivalent to around 4% of the total road fuel demand projected for EU25 in 2020. When the energy used to produce the biofuels is also factored in, the net substitution of fossil energy is around 2%. For the future, alternative biofuel pathways, mostly based on conversion of woody and grassy materials, offer the potential for higher yields. The percentage of fuel available for substitution would increase if these technologies are successful, both because of higher yields from wood and grass and because the fuel production process is more efficient. The JEC Study calculated net energy substitution of up to 6% for liquid fuels that can be used in conventional vehicles. Values up to 10% are possible if other fuels such as DME and hydrogen are also included but these would require the use of dedicated vehicles. Taking all of these factors into consideration, biofuels are likely to be used mainly as blend components in conventional gasoline and diesel because available biofuel volumes will simply not be sufficient to completely replace conventional fossil fuels.

Among fossil fuel resources, coal and natural gas (NG) are more abundant globally than crude oil and are seen by some as potential substitutes for fossil liquids. Exploitation of coal would require substantial processing to produce fuels suitable for vehicle use. Technologies exist for production of Fischer-Tropsch (FT) diesel from both NG and coal and significant development work is under way, particularly in the USA, China and South Africa. However, concerns about high GHG emissions may limit the acceptance of synthetic fuels derived from coal unless the effects can be mitigated, for example by Carbon Capture and Storage (CCS). Coal is therefore best considered as a long term option. A few large scale plants are under construction to produce FT diesel fuels starting from NG. Such plants are based on NG sources which are far from markets and cannot be easily exploited by other means. Although the volume of FT fuel produced by 2015 could be significant, it will remain a small percentage of total diesel fuel demand and will mostly be used as a high-quality diesel blending component except in some niche market applications.

Natural gas is readily available today, and the technology to use it in vehicles is mature, however there are no compelling arguments for a large scale conversion from gasoline/diesel to CNG. The JEC WTW study [2] concluded that the CO_2 savings are small and the cost of CO_2 avoided is relatively high, because CNG requires specific vehicles and a dedicated distribution and refuelling infrastructure. This study proposed that targeted application in fleet markets may be more effective than widespread use in personal cars. The JEC study also notes that a 5% share of the European road fuel market would represent only about 2.5% increase in total gas demand.

The availability of natural gas cannot be taken for granted. The demand for NG for power generation and heating uses in Europe is increasing at the same time as domestic production is declining. Meeting current demands for NG will require significant efforts, and the increasing reliance on imports means that it does not avoid the supply security concerns expressed by some people with respect to crude oil. Growth in demand is expected to continue at around 1.5% per year. Although Europe produces 55% of its own NG requirements today, this is projected to fall to just 15% by 2030 [3], with significant increases in pipeline and LNG imports from outside the EU needed to balance the need.

As a consequence, fuels derived from crude oil are expected to provide the main source of road fuels until at least 2030. Biofuels will provide some additional resource, mainly being blended into gasoline and diesel at levels up to about 10%. Alternative fuels such as LPG, CNG, and high percentage biofuel blends will continue to contribute in local and fleet markets but provide a relatively small part of the total fuel demand. HCCI technology will therefore be most successful if it can use fuels similar to today's gasoline and diesel fuels.

3. ADVANCED COMBUSTION CONCEPTS

3.1. BACKGROUND

The goal of advanced combustion or HCCI concepts is to simultaneously achieve high fuel efficiency and low exhaust emissions.

3.1.1. Fuel efficiency

Efficiency is defined as the useful energy obtained as a percentage of the gross energy input. The thermodynamic efficiency of an engine varies over its load range. Theoretical modelling of a normally aspirated gasoline PFI engine [4] shows that the amount of fuel energy converted to useful work is no more than 35%. Most of the remaining energy appears as losses to the exhaust and coolant and in combustion irreversibilities[3]. In addition, smaller amounts of energy are absorbed in mechanical friction and fluid pumping losses, particularly at light engine loads, where efficiency is much reduced. Significant improvements can be made by optimising the engine's compression ratio and operating at lean conditions, and the same study computes fuel energy conversion up to 43% under these conditions. Turbo-charging was found to have the potential to recover about 10% of the energy in the exhaust, but this amounts only to about 2% improvement in overall engine efficiency. The biggest benefit of turbo-charging therefore comes from the ability to downsize the engine without compromising performance[4].

These potential optimisations represent a 23% improvement compared with a PFI engine, and are very close to the relative performance achieved by today's diesel engines. The higher density of diesel fuel results in about 10% higher volumetric energy content compared with gasoline, and this additional benefit means that volumetric diesel fuel consumption is about 30-35% lower than for a gasoline PFI vehicle.

This range of performance represents the bounds of what can be envisaged with currently known technology. Much higher efficiencies are theoretically possible, but require technologies to capture and use more of the energy currently lost from the system. In [4] it is noted that reducing cylinder heat losses transfers energy mostly to the exhaust, with only a small amount appearing as useful work. This finding is consistent with many previous studies. Past attempts to reduce cylinder heat losses resulted in increased NOx emissions [5]. The energy in exhaust gas flow has already been exploited to a significant extent through turbo-charging. Incremental improvements can be expected as engines develop, but a major change would require new technologies. A brief review of the options is contained in [4].

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3 Combustion irreversibility losses arise when combustion takes place in less than optimal conditions, and is the result of entropy production which reduces the amount of energy available for useful work. For example, when a stoichiometric mixture is burned, combustion raises the temperature. The Carnot equation describes the maximum theoretical efficiency of a cycle working between two temperatures and shows that the higher the temperature, the higher the work that can be extracted (Carnot efficiency = 1-Tmin/Tmax). In a dilute mixture, some of the energy is distributed to the diluent, and because the exhaust temperature is higher than the air intake temperature, some of this energy is lost through the exhaust.

4 Personal communication from the authors of [4].

A practical goal for advanced combustion concepts, therefore, is to match diesel engine efficiency while maintaining low engine-out emissions. Today's diesel engines have a number of features that help achieve high efficiency:

- Unlike current spark ignition engines, they do not need a throttling device, so volumetric efficiency is not impaired at lower loads;
- The overall mixture ratio is lean, leading to good thermal efficiency;
- Provided the injection timing does not need to be over-retarded to reduce emissions, the combustion event can be positioned just after Top Dead Centre (TDC) to maximise use of the chemical energy;
- Compression ratios can be higher than in gasoline engines, leading to better efficiency.

Conventional gasoline PFI engines are constrained to limit the premixed fuel charge using a throttling device: pumping gases past the throttle absorbs energy and reduces efficiency. Secondly the mixture is maintained at stoichiometric conditions to allow operation of the TWC.

In addition, the compression ratio for gasoline engines is generally lower than ideal (limited by knock), and for diesel engines is higher than ideal, because of the need to assure compression ignition particularly on starting.

3.1.2. Exhaust emissions

For conventional gasoline and diesel engines, engine-out emissions are relatively high and exhaust aftertreatment is needed to bring emissions within the most recent regulated limits. For gasoline vehicles, TWCs provide very effective control but at the cost of maintaining a stoichiometric air-fuel ratio which precludes the efficiency gains that could come from lean operation. So far, attempts to introduce lean gasoline vehicles have not been wholly successful, because of disappointing fuel savings and the complexity of controlling NOx emissions under lean conditions.

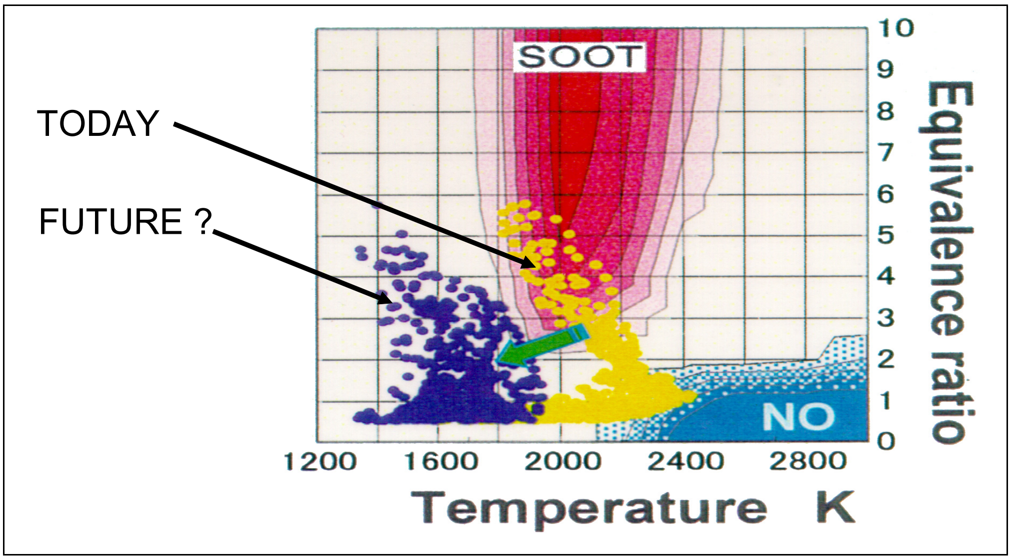
Controlling NOx from diesel vehicles is difficult for the same reasons. In addition, the stratified combustion of the diesel engine leads to soot formation. For conventional engines, the approach so far has been through the use of exhaust aftertreatment. Oxidation catalysts are already used on light-duty diesels and particle traps are expected to be widely used to meet Euro V and Euro VI standards. NOx traps have been used on some lean gasoline engines, and are also being developed for heavy-duty diesel vehicles. However, an alternative technology, Selective Catalytic Reduction (SCR), using urea injected into the exhaust system, is already used in European heavy-duty vehicles and may prove to be the more reliable and cost-effective approach for the future. Some manufacturers have announced plans to have urea canisters on light-duty vehicles which would be replaced at service intervals. An effective NOx aftertreatment allows the engine to be tuned for best fuel efficiency, but the aftertreatment may itself result in some fuel penalty. Engine developments such as HCCI, if they can reduce emissions at their source, could reduce the investment needed in complex exhaust aftertreatment.

Engine emissions[5] can be categorized as:

- Oxides of Nitrogen (NOx),
- Particulate Matter (PM), consisting of soot plus condensed hydrocarbons and other species,
- Unburned Hydrocarbons (HC),
- Carbon Monoxide (CO).

NOx and PM have proved to be the most intractable of these emissions. Their origin can be seen when the combustion conditions are plotted on a ϕ-T diagram. ϕ (=1/λ) is the equivalence ratio, defined as the air-fuel ratio, normalised by the stoichiometric value. Hence ϕ=1 under stoichiometric conditions, ϕ>1 signifies rich conditions and ϕ<1 represents lean conditions[6]. T is the temperature in degrees K, and points on the plot refer to the local condition where combustion occurs. These conditions may vary across the combustion chamber and during the combustion, particularly where conditions are non-homogeneous, as for example in the diesel engine. The ϕ-T diagram (**Figure 5**) was first presented in detail in [6] and is now widely used to describe advanced combustion concepts.

Figure 5 The ϕ-T diagram, illustrating the potential areas of low emission combustion [6]



[5] CO_2 is also an exhaust emission, but in terms of engine design is a function purely of fuel efficiency. Attention here is focussed on those emissions that have been regulated to improve air quality.

[6] Stoichiometric conditions are when the amount of fuel available is exactly balanced by the oxygen available in the air, so both are completely consumed in complete combustion. For a gasoline vehicle, the stoichiometric air-fuel ratio is about 14.7:1 (measured in mass terms). Diluting the charge with more air moves the mixture to leaner conditions (ϕ<1 or λ>1). Exhaust Gas Recirculation may also be used to change mixture conditions. Under fixed operating conditions, replacing part of the intake air with inert exhaust gas will change the air-fuel ratio.

Soot is formed in regions of rich mixture and fairly high temperature, whereas NO can be formed in rich or lean mixtures wherever the temperature is sufficiently high[7]. Normal diesel combustion takes place over a wide range of conditions, resulting in both soot and NOx production. Gasoline engines generally use premixed mixtures and so generally[8] avoid the very rich areas where soot formation occurs, but temperatures are high, so NO is produced. However, significant soot emissions can occur in Direct Injection (DI) gasoline engines if combustion occurs before the fuel is fully mixed with the surrounding air.

Conditions in an engine will follow a trajectory through the ϕ-T plane in the course of each combustion cycle. Soot or NO will be formed even if this course touches the critical areas even for a brief time. The objective of HCCI in its broadest sense is to move the area of combustion to cooler and/or leaner areas where neither soot nor NO formation occurs. The diversity of engine concepts arises from the variety of strategies chosen to achieve this.

3.2. THE HCCI CONCEPT

In its purest form, HCCI combustion is a concept to pre-mix the air-fuel mixture, either outside the cylinder or by early injection of fuel into the cylinder, and then to compress it to the point where ignition will simultaneously occur at multiple points in the cylinder to ensure a rapid and complete heat release.

HCCI concepts have been demonstrated to work under laboratory conditions, and low emissions and good efficiency can be achieved at least under steady state engine conditions. Extending this performance to transient operating conditions remains a challenge, and engine designers face a number of hurdles to develop practical systems:

- For conventional engines, the start of combustion can be controlled directly, through spark timing in the SI engine, and by fuel injection in the diesel engine. In true HCCI combustion, neither of these controls is available. The temperature and pressure at the desired ignition point are the sole determinants of combustion, and these are governed by events (intake temperature and composition, compression ratio etc) defined well before the combustion starts. Controlling the start of combustion is therefore challenging, particularly for vehicle engines where operating conditions can vary rapidly second by second.
- The goal of HCCI combustion is to ensure good thermal efficiency through a rapid and complete heat release. This, however, can produce very rapid rates of heat release and peak pressures that produce high noise levels and could lead to mechanical damage, particularly at high engine loads.
- Since HCCI engines are throttle-less, control of engine power is achieved by adjusting the charge conditions. Complete combustion can be difficult to achieve, especially at the very lean air-fuel ratios needed for light load operation. While this cool, homogeneous combustion produces very low levels

[7] Some of the NO is subsequently converted to NO_2 or other species in the exhaust. In exhaust emission measurements the term NOx is used to denote the total sum of NO and NO_2.

[8] Although particulate emissions are much lower from stoichiometric gasoline vehicles than from diesels, they may not be insignificant. Some US studies have indicated that emissions from older or badly maintained gasoline vehicles can provide a significant proportion of total road PM emissions.

of soot and NO, there can be significant emissions of HC and CO. Catalytic aftertreatment will probably be needed to reduce these to acceptable levels.

The early injection HCCI approach is typified by the Narrow Angle Direct Injection (NADI™) concept developed by IFP [7,8,9,10]. The fuel injectors and combustion chamber are optimised to produce homogeneous combustion, which can be sustained over a power range equivalent to the European NEDC driving cycle. At higher loads, the engine reverts to normal diesel combustion. The narrow cone angle of the fuel injection spray directs the fuel towards the rising piston, in contrast to conventional injectors that direct the spray more widely across the combustion chamber. The narrow angle fuel injector avoids the problem of wall wetting at lower loads, but at high loads may increase the likelihood of fuel impingement on the piston.

Because of the difficulty of controlling the combustion and extending operation to higher loads, many researchers believe that fully mixed combustion will not be successful. A number of alternative engine strategies are being investigated, using later fuel injection coupled with control of charge air temperature and pressure, as well as EGR, to modify the combustion conditions. Such approaches provide less complete mixing, however, the ϕ-T diagram offers fairly large areas of low emission combustion. Provided that combustion temperatures can be kept low, areas of NOx and soot formation can be avoided over a wide range of equivalence ratios. Equally, at higher temperatures, soot formation can remain low even at slightly rich equivalence ratios, so complete mixing is not necessary. Terms such as PCCI (Premixed Charge Compression Ignition) and 'mixed-enough combustion' have been coined to describe these systems.

The term HCCI strictly refers only to the early injection, pre-mixed approach. However, for convenience, in this report we use the term in a broader sense to encompass these other advanced combustion strategies that are being explored, including some that have their own specific acronyms. Two pioneering approaches are:

- The Nissan MK (Modulated Kinetics) system uses high swirl ratio (~3-5), relatively high EGR (~30%) and retarded injection timing (from ~7° ATDC to ~3° BTDC) when the engine operates at light load. Under these conditions nearly complete mixing (equivalence ratio <2) is achieved before combustion. The lower efficiency from retarded fuel injection is mitigated through reduced heat transfer losses. The engine runs in MK mode at low load and switches to regular diesel operation at high loads. [11,12].
- Toyota's Low Temperature Diesel Combustion takes a different approach [6,13], to retain the high efficiency of a regular diesel engine while keeping the combustion temperature below 1600K, which is below the soot-forming region. This combustion process relies on extremely high cooled EGR levels in excess of 60% to keep the temperature inside the diesel jet low enough so that no PM can form. The need for high EGR limits the range of operation to low power conditions.

Both of these approaches are limited so far to light load conditions. As load increases, temperatures rise, taking the engine into the regions of soot and NOx formation, or if combustion is retarded to counter this, then efficiency decreases [14]. Equally, as load increases more fuel must be injected, reducing the degree of pre-mixing before combustion begins. If injection timing is advanced to improve mixing, there is a risk that fuel will wet the cylinder walls unless the injectors and combustion chamber are designed to avoid this.

- Controlled Auto-Ignition (CAI) is a generic term used for applications based on gasoline engines, where auto-ignition with a lean mixture is encouraged to provide a fast and clean combustion. It is limited to low loads, so the engine reverts to stoichiometric spark ignition at higher loads.

Although HCCI combustion shares some of the characteristics of both diesel and gasoline combustion, practical engines need to start from the basis of either a compression ignition or spark ignition design. The goal of an engine that uses HCCI combustion throughout the load range is a long way off, and for the near term, practical engines will use part-time HCCI combustion. For the diesel engine, the benefits of HCCI are to reduce engine-out emissions, particularly of PM and NOx. Even if this does not remove the need for exhaust aftertreatment, the cost and complexity may be reduced if emissions from the engine are lower. The diesel engine also provides a good starting point for engine development because of its robust construction, and the availability of high pressure in-cylinder fuel injection. For light-duty applications, part-time HCCI engines may be based on either diesel or gasoline technology. For gasoline engines, exhaust emissions are already very low, and the benefit of HCCI comes from the improved fuel consumption. Significant reductions in fuel consumption can be achieved at low loads, but the effects may be diluted over driving cycles including higher speeds and loads such as the NEDC (New European Driving Cycle). Gasoline HCCI must also compete with other developments of conventional gasoline engines that offer similar fuel consumption improvements and cost may be the determining factor.

The type of fuel used is one element in the success of advanced combustion, and some researchers have considered new fuels or even dual-fuelling in the search for successful HCCI combustion. Whether this eventually proves to be practical depends in part on whether HCCI operation can be sustained up to full load. Where an engine needs to revert to conventional diesel or gasoline operation at high load, the fuels will need to retain the essential characteristics of diesel or gasoline, so the scope for optimization will be more limited.

HCCI operation at part load only might be considered only a partial success, since the benefits are diluted, and the vehicle may still need to carry conventional emission controls for the higher load operating points. There is also a risk that vehicles may be able to meet the emission limits over the regulated cycle through combustion modifications, but at the price of higher emissions outside the cycle limits or during transient operations. Conversely, part load HCCI offers a way for advanced combustion to be introduced progressively into otherwise conventional gasoline and diesel engines. Investments in some hardware improvements that will later be required to enable HCCI operation seem likely to enhance conventional engine performance in the shorter-term. This provides a path for gradual expansion of the area of HCCI combustion as experience is developed and such vehicles will almost certainly be introduced over the next few years. Still, a practical and robust full-time HCCI engine remains a long term objective and seems unlikely to be developed within the next decade. While this is occurring, continuous improvements in conventional engine technology may supersede the performance and cost benefits of HCCI engines.

In the following sections, the status of research to address the key challenges will be described.

3.2.1. Ignition delay and combustion timing

Although reduced engine-out emissions are a main objective of HCCI development, achieving this has proved relatively easy, at least at steady-state conditions on an engine test-bed, and the barriers to practical use of HCCI lie in other directions. The most important problems have been summarised [15] as mixture preparation, control of the start of combustion, and control of the reaction rate, which governs the maximum rate of pressure rise. To a large extent, these factors are all linked through the ignition delay, and so this aspect is considered first and in some detail.

Ignition delay is conveniently measured in degrees crank angle, and can be generically defined as:

$$ID = SOC - SOI \qquad \text{(Equation 1)}$$

Where: ID is the Ignition Delay

SOC is the crank angle (CA) for the start of combustion

SOI is the crank angle (CA) for start of fuel injection

The exact start of combustion can be difficult to measure, so some pragmatic approach is needed to define it. For example, [16] used CA10, the crank angle for 10% of total heat release, as a surrogate for SOC while [17] used CA50, the crank angle for 50% of total heat release. Such measurements will be internally consistent but care should be taken if comparisons are to be made between different studies.

A long ignition delay is necessary to avoid overlap of fuel injection and heat release, the importance of which for soot emissions is discussed in **Section 3.4.2**. A longer ignition delay also allows more time for mixing, thus avoiding combustion in the hot, rich conditions where soot and NOx are formed, and influences the rate of heat release, discussed in a following section. Understanding the ignition delay, together with the injection timing, also allows the timing of the peak heat release to be controlled, which is important for engine efficiency.

Ignition delay is also a key property of different fuels, so while fuel impacts are discussed in more detail in the appropriate section, some mention will be necessary in this discussion of ignition delay. Conventional diesel engines use fuels which readily auto-ignite under the high compression pressures and temperatures of the diesel engine. Ignition delays in these engines are short, and combustion generally starts before fuel injection is completed. Gasoline engines use fuels which are resistant to auto-ignition, together with relatively low compression pressures and temperatures, so auto-ignition does not occur under most circumstances. Instead, combustion is initiated by a spark, and characterised by a flame front which progresses through the pre-mixed air-fuel mixture.

In true HCCI combustion [7], fuel is injected very early in the cycle, before cylinder pressure and temperature are sufficient to initiate combustion. In this case, therefore, the ignition delay is long, with adequate time for fuel mixing, but there is little control over when ignition will take place, and this limits operation to fairly low loads. For those systems that use later fuel injection (more like a conventional diesel engine), similar difficulties arise when diesel fuels are used - the ignition delays are too short, and except at lower loads combustion reverts to conventional diffusion burning with its attendant high NOx and soot emissions. Combustion can be delayed by measures such as lower compression ratio and high EGR levels, however fuels with more ignition resistance (lower cetane number) have also been investigated. Some researchers believe that gasoline-like fuels running in diesel-based HCCI engines are the most promising avenue [15,17,18], and that HCCI engines running on diesel fuel will be limited to light loads [15].

Ignition delay depends on the physical conditions as well as the chemistry of the air-fuel mixture. The rates of the chemical reactions that initiate the main combustion generally increase at higher temperature, because the frequency of molecular collisions and the kinetic energy associated with them increases, making it easier to overcome the 'activation energy' needed for reaction. The temperature relationship may be modelled by the exponential Arrhenius equation. Generally, higher pressure also increases reaction rate, so ignition delay generally follows an equation of the type:

$$\tau_{id} = A\ p^{-n}\ e^{E/RT} \qquad \text{(Equation 2)}$$

Where: τ_{id} is the ignition delay time, here measured in milliseconds:

A and n are constants,

p is the prevailing pressure,

T is the temperature in Kelvin,

E/R is the activation energy for the reaction.

In the case of a single molecular reaction, the activation energy can be uniquely defined. Reaction of a mixture of molecules, however, is the result of a very complex set of reactions, so E/R is best described as an apparent activation energy. As with all of the other parameters in this equation, E/R needs to be determined empirically in this case[9].

Tests on a wide range of engine conditions in [16] confirmed that the expected relationship was followed in HCCI combustion except that to describe the changes in ignition delay with EGR, an additional term related to the oxygen concentration in the combustion chamber had to be included. The equation they derived was:

$$\tau_{id} = A\ p^{-1}\ X_{ox}^{-1.2}\ e^{E/RT} \qquad \text{(Equation 3)}$$

Here, X_{ox} is the mole fraction of oxygen in the intake gases. This study used a US certification diesel fuel. The equation shows that longer ignition delays are associated with lower pressures and temperatures, and with lower oxygen concentrations, i.e. more EGR[10]. Fuel properties will also influence ignition delay, and this is discussed further in a following section.

Although the ignition delay may be long under some circumstances, this does not mean that chemical reactions are not taking place. Combustion is initiated through

[9] This global equation is a description of a very complex set of reactions and inevitably introduces a degree of over-simplification. The real situation is more complex, for example there are regions that exhibit negative temperature coefficient behaviour.

[10] Recirculation of exhaust gas to the cylinder reduces the amount of air inducted, and hence lowers the oxygen concentration. The oxygen concentration is 21% where no EGR is used. A very high EGR rate of 65% reduces this to about 10% [17]

chain-branching reactions that begin slowly before culminating in an exponential growth of activity as the main heat release occurs. Depending on the conditions and the fuel properties, these Low Temperature Heat Release (LTHR) reactions may be apparent as a small heat release a few degrees of crank angle before the main combustion begins. Low temperature reactions are important because by forming active radicals they determine the temperature at which the main heat release will start and hence the timing of combustion [15,19]. LTHR has been extensively studied, and [19] provides a good overview. The low temperature reactions occur around 420-450°C [20], so depend on the fuel and air being present together during the compression stroke. As compression continues and temperatures further increase, these reactions cease, but the radicals formed may be preserved and contribute to the main ignition.

The type of fuel also affects the presence of LTHR. In tests using diesel secondary reference fuels, [19] found no discernable LTHR at cetane numbers of 34 or 19, but it was present with blends at 48 and 62 cetane number. It is not clear whether this variation is a true cetane effect or related to the composition of the fuel. In [19], LTHR is attributed to normal-paraffins, with iso-paraffins also contributing to some extent, but since paraffins exist in the lower CN blends tested other components must play a part. Fuel effects have been studied in depth in [89], using a Rapid Compression Machine to evaluate pure hydrocarbon mixtures. These authors found that the presence of LTHR was associated with shorter ignition delay times. LTHR was related to the fuel composition, with hydrocarbons containing the structure -CH2-CH2-CH2- showing two stage ignition with relatively short ignition delays. There seems no consensus on whether LTHR is desirable or undesirable, although [15] suggests that less is better. However, HCCI combustion has been demonstrated with both types of fuel and understanding of the effect is important to develop effective engine control systems.

In [20], evidence of chemical reactions was seen under some circumstances at even lower temperatures. These are related to reactions initiated in recirculated exhaust gas and it was shown (by comparing with nitrogen dilution) that the effect must be chemical rather than thermal. The effect is again to promote the main heat release. They propose that high levels of internal EGR could be effective to stabilise low load and idle operation.[11]

It has been shown [21,22] that spark ignition can help, particularly at light loads, to assist ignition. Even if a flame is not initiated at the spark as in a normal gasoline engine, the heat input could be sufficient to start cool flame reactions, accelerate the main heat release and stabilise the combustion.

3.3. CONTROLLING THE IGNITION DELAY

The nature of the challenge to optimise ignition delay depends on the starting point. Gasoline engines are normally designed to avoid uncontrolled ignition, so where an engine is modified for HCCI (or CAI) combustion, measures are needed to encourage auto-ignition through, for example, increased intake temperatures, higher compression ratio or lower octane fuels. Where the starting point is a diesel engine the problem is reversed, and ways must be found to slow the reaction processes.

[11] Conversely, [48] showed that CO_2 addition to the charge delayed both low LTHR and the main combustion, whereas N_2 addition had no effect. They attributed the effect of CO_2 to the change in specific heat ratio.

These options include:

- increasing the compression ratio,
- controlling the temperature of the intake air,
- increasing exhaust gas recirculation,
- introducing charge inhomogeneities,
- changing fuel properties, or
- increasing the air-fuel ratio

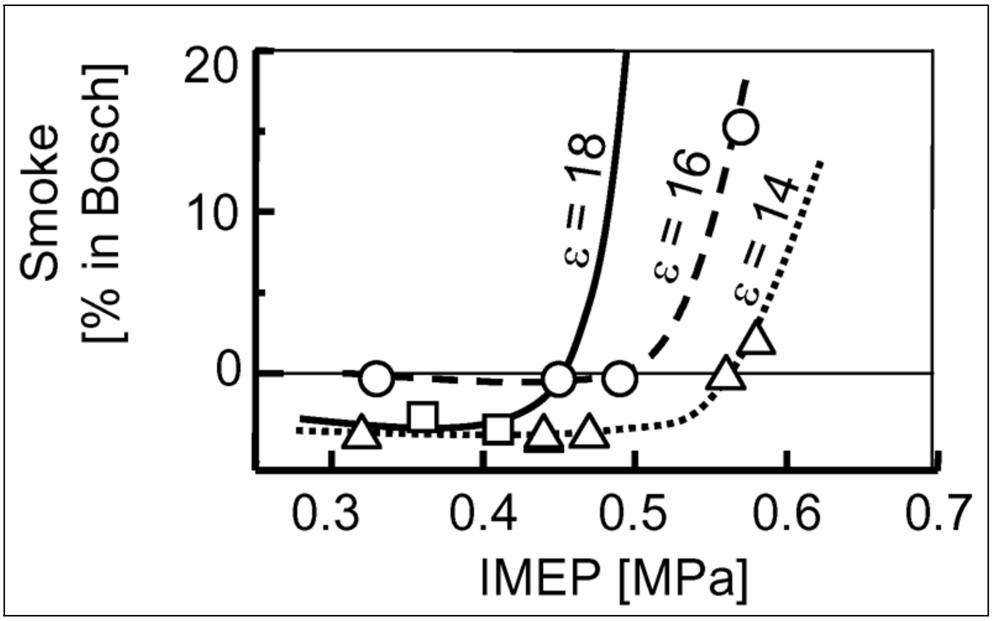
3.3.1. Compression Ratio

Increasing the compression ratio (CR)[12] improves the thermodynamic efficiency of the engine up to a certain point, beyond which pumping and mechanical friction losses negate further improvements. In practice, spark ignition engines are limited to ratios lower than the optimum to avoid knocking, while diesel engines need ratios higher than the theoretical optimum to assure good cold operation. Gasoline engines typically use CRs of 7-10. DI diesel engines are typically in the range 16-18, while even higher CRs are needed by indirect injection diesel engines (20-23) to compensate for the higher heat losses caused by the large surface area inside the combustion chamber. The eventual optimum is unclear, but probably lies in the range 14-15, a level that requires a robust engine construction. This factor, together with the need for direct injection, may make the diesel engine a better basis for future developments than the gasoline engine.

A single cylinder engine test program [23] evaluated performance in a diesel engine where the compression ratio was changed by fitting pistons with different bowl volumes. They found that the range of smokeless operation with very high EGR (62%) was improved at compression ratios lower than the standard 18, although at the cost of slightly lower thermal efficiency (**Figure 6**). The benefits of lower compression ratio are attributed to better mixing, because of the longer ignition delay, and lower in-cylinder temperatures.

[12] Compression ratio means the geometric ratio between the cylinder/combustion chamber volumes when the piston is at its lowest and highest points. The compression pressure will in addition be influenced by other factors, such as heat losses and valve timing.

Figure 6 Lower Compression Ratio allows higher load operation [23]



A similar effect has been seen in HCCI engines based on early injection; [24] found that reducing compression ratio led to an increase in the area of HCCI combustion, but noted an increase in fuel consumption and some impacts on cold start performance.

Variable Compression Ratio (VCR) engines have been proposed, and could enable the engine to be optimised at each point of the operating range, rather than for the most critical point (high load for gasoline, cold start for diesel) as at present [25]. A number of design approaches are outlined in [26], and their impact on gasoline engine performance evaluated, using a design based on eccentric mounting of the crankshaft. A similar design has been proposed for heavy-duty diesel engines [27], and a review of potential options is given in [28]. Several alternative approaches have achieved low-scale production, however, the inevitable increase in complexity means that the benefits would need to be substantial before VCR is widely used.

An alternative approach may be implemented through variable valve timing (VVT). The geometric compression ratio of an engine is usually fixed, but the effective compression ratio can be adjusted by controlling the timing and lift of the intake and exhaust valves. In simple terms, if a high geometric compression ratio is chosen, the effective compression ratio can be reduced by reducing the amount of the intake charge, usually by very early or very late closing of the intake valve.

3.3.2. Intake cooling/heating

Depending on the specific engine conditions, temperature control may be considered as a way either to speed up or slow down the rate of ignition and heat release. For example, if the starting point is a gasoline engine with relatively low compression ratio, achieving compression ignition may be difficult, particularly with gasoline which has a high resistance to auto-ignition, and heating the fresh charge may help to ignite it. Conversely, an adaptation of a diesel engine, with high compression ratio and using diesel fuel may result in premature ignition. In this case, cooling the intake charge may help control the combustion timing. External or internal Exhaust Gas Recirculation (EGR) can provide a source of hot intake gas. In the case of external EGR, there is also the opportunity to cool the gas before it is reintroduced into the cylinder, if needed.

Tests on a 2-litre single cylinder engine, typical of a heavy truck engine, running in HCCI mode with a standard US diesel fuel were reported in [18]. At intake temperatures of 33°C and 43°C, combustion was maintained just after TDC. When the temperature was increased to 53°C, the cool flame combustion was advanced, and this in turn caused the main heat release to be advanced by about 5 degrees CA.

In [19], tests were carried out on an engine without EGR, using only intake temperature to control the start of combustion. If the intake temperature was too low, misfire occurred, and there was also an upper limit of temperature beyond which the rate of pressure rise increased to unacceptable levels. For high cetane fuels (Secondary Reference Fuel blends 62-76 CN), HCCI operation could be achieved with intake temperatures in the range 120-170°C, whereas fuels of 19 or 34 CN required temperatures up to 280°C.[13]

3.3.3. Exhaust Gas Recirculation

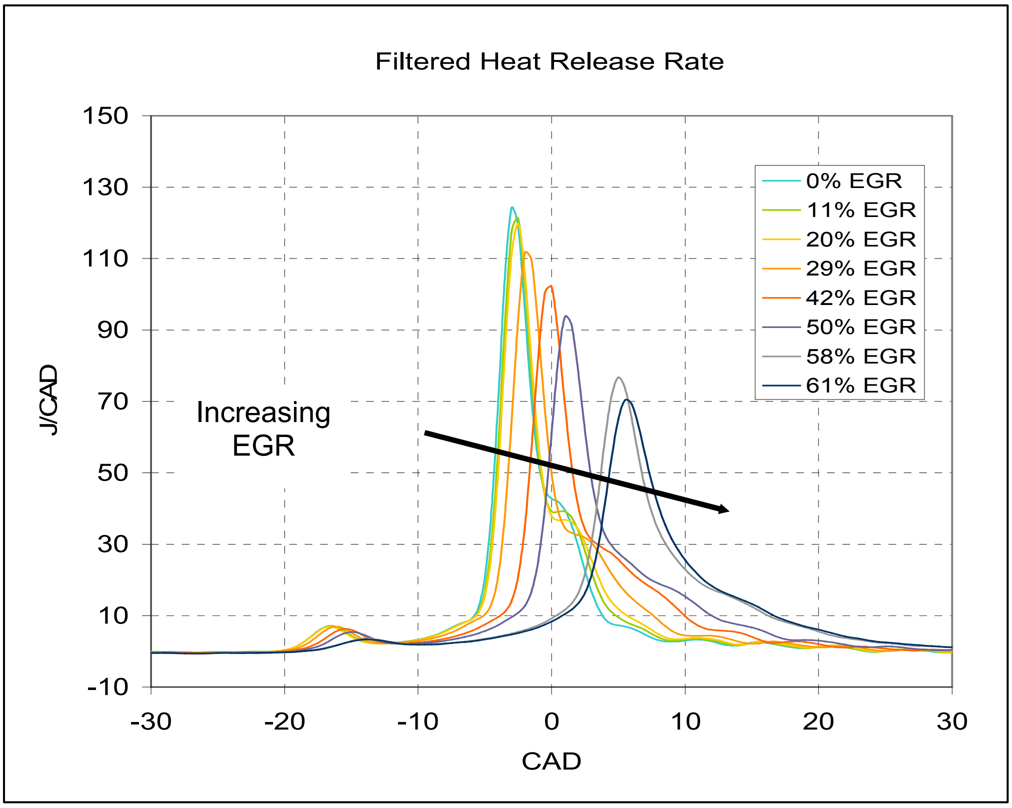
Exhaust Gas Recirculation (EGR) affects combustion in different ways. First, there is a thermal effect. Since exhaust gas is usually hotter than the fresh intake air, the charge temperature is increased, which tends to accelerate the combustion process. EGR also acts as an inert diluent which is particularly effective because of the high heat capacity of the CO_2 contained in the exhaust. This reduces the rate of combustion and the temperature at which combustion occurs[14]. It is this temperature reduction that makes EGR so effective in reducing emissions of NOx from conventional engines. These are the two dominant mechanisms, however as noted above, EGR may also participate in the low temperature reactions that can 'condition' the air-fuel mixture, and so influence the start of the main combustion event. This latter mechanism seems only to be important at very light load conditions.

The effect of EGR in lengthening ignition delay has been discussed above, and is illustrated in **Figure 7**, taken from [15].

[13] As noted above, the latter two fuels did not exhibit LTHR, which could explain why they were more difficult to ignite. Caution should be exercised in extrapolating from the very specific conditions in this test programme to engines using EGR.

[14] See discussion in 'fuel effects' for more detail.

Figure 7 Increasing EGR lengthens the ignition delay [15]



CAD: Crank Angle Degree **J/CAD**: Joules/Crank Angle Degree

In tests on a single cylinder engine with CR of 18.7, ignition delay increased by a factor of about 2.5 as EGR increased from zero up to 65% [16]. However, if the EGR rate is too high, the combustion can deteriorate, and efficiency may suffer at higher loads [23], so there may be practical limits on the amount of EGR that can be used.[15]

External recirculation of exhaust gas is not the only way to achieve charge dilution. By closing the exhaust valve early, some of the burnt gas can be trapped in the cylinder to be recompressed and dilute the charge for the subsequent firing stroke. An alternative strategy is to open the exhaust valve during the intake stroke so that exhaust gas is drawn back into the cylinder. These two approaches were studied in a DI engine with CR 11.3 [21] using a high octane gasoline fuel, conditions which make auto-ignition difficult. The study concluded that recompression with early fuel injection gave the best efficiency, as well as being easier to implement. Rebreathing using valve opening during the intake stroke led to heterogeneity in the mixture, with high HC emissions and incomplete combustion. These features may, however, have some benefits where the engine and fuel conditions are more favourable to auto-ignition. In tests on a gasoline engine of CR 7.9 [29], the intake air had to be heated to at least 400K to enable auto-ignition without EGR. However, use of negative valve overlap allowed up to 60% internal EGR, and HCCI combustion was achieved without heating the intake air.

[15] The figure of 60-65% EGR is frequently cited in HCCI research and is significantly higher than the levels of EGR used in today's engines.

3.3.4. Charge inhomogeneity

While the above parameters influence the length of the ignition delay, they do not necessarily assure consistent cycle to cycle performance. Maintaining a low Coefficient of Variation (COV) between successive cycles is an equally important challenge; a target of COV<5% IMEP has generally been recommended by HCCI researchers. The idealized HCCI concept, fully premixed compression ignition, provides the engine designer with virtually no control once the fuel has been injected, and small cycle to cycle differences can translate into large combustion variations. In contrast, for conventional engines, gasoline spark ignition and continuation of diesel fuel injection after combustion starts provide good control over the combustion timing. To overcome the problems of a completely homogeneous charge, researchers have investigated different ways of introducing inhomogeneity into the system, to provide a more repeatable combustion, moderate the rate of heat release and allow higher load operation. Inhomogeneity can take the form of variations in temperature, fuel mixture or turbulence.

The cooling effects of the cylinder walls provide some natural thermal stratification within the combustion chamber. Sjöberg and Dec [30,31] describe this in terms of a 'thermal width' or range of temperature across the combustion chamber. The thermal width is about 20K for current engines, and they compute that increasing it to 40K would give considerable benefits. Possible approaches suggested in [32] are increased swirl (for radial stratification), or increasing valve overlap for one intake valve (for lateral stratification). It was found that combustion started in the hot region and progressed to cooler regions. As a consequence, the duration of heat release was extended, reducing the rate of heat release and allowing operation at higher load. In a study on a 0.5-litre optical engine [33], modelling coupled with Laser Induced Fluorescence (LIF) measurements showed that temperature inhomogeneity shortened the ignition delay time, and that HCCI combustion would not be initiated without some temperature inhomogeneity. They commented that spontaneous ignition under these conditions sometimes exhibits high temperature gradients and therefore has the characteristics of a flame propagation. Sequential ignition of different regions has also been observed in [30]. In [34], injecting two fuels with different temperatures was found to cause progressive combustion from hotter to cooler regions.

In [31], computations showed that doubling the 'thermal width' to 40K decreases the maximum rate of pressure rise (ROPR) by 50%. Achieving this in practice appears to be difficult, however. A previous reference showed that about 35% reduction in ROPR could be obtained by decreasing the coolant temperature to 50°C and increasing swirl. Further improvements could be achieved if the colder boundary air could be mixed into the main combustion region; it is suggested that a combination of hot residual gases and cooled intake air could be effective, particularly if direct injection is used to spread the fuel over the hot and cold regions.

In [33], the effects of turbulence were generally to increase the ignition delay. This is perhaps surprising at first sight, since turbulence generally increases reaction speed in conventional combustion. However, in HCCI combustion, turbulence can smear out the temperature variations and hence slow reaction. The researchers note some exceptions where, with high temperature inhomogeneity, turbulence can accelerate reaction by transporting hot gases into cooler regions. In tests on a gasoline engine of CR 7.9 [29], the use of internal EGR created inhomogeneity in the combustion, but at the highest EGR rates the spatial pattern of combustion varied from cycle to cycle. The same study investigated the effect of fuel mixture stratification by comparing a fully premixed charge with one where the gasoline was injected onto the back of an open intake valve. Although tests were conducted only without EGR, changes in the combustion pattern were clearly seen.

Variable Valve Actuation (VVA) potentially offers a great deal of scope for engine optimisation, and such systems have been extensively reviewed in [35]. Flexible valve operation allows internal EGR to be achieved either by trapping gas in the cylinder or by re-breathing gas from the exhaust. In this way the recycled exhaust gas can be kept separate from the incoming air charge and help provide thermal stratification.

Gasoline PFI engines are conventionally considered to use homogeneous mixtures, and because the challenge here is to encourage rather than moderate combustion, inhomogeneity has received less attention. However, laser diagnostics reveal some inhomogeneity even in PFI engines [36], compared with a air-fuel charge prepared in a mixing tank.

3.3.5. Fuel effects on ignition delay

The choice of fuel is clearly a factor in control of ignition delay. Tests using diesel fuels of different cetane number [37] showed that the amount of EGR needed to maintain the desired combustion timing changed from 25% for a 40 CN fuel to 45% for a 63 CN fuel. A number of studies have compared gasoline and diesel performance in the same engine [17,18,38] and in some cases mixtures of gasoline and diesel have been evaluated [39]. Tests in an engine with CR of 14 showed that gasoline gave much longer ignition delays even than a diesel of 30 CN [17]. They were able to obtain lower smoke and NOx as well as operate at higher engine power using gasoline rather than diesel fuel. A wide range of fuels have also been evaluated in a single cylinder 2.44-litre engine representative of a large heavy-duty engine [18,27,38,40]. Good performance was obtained with gasoline, kerosene and diesel fuels, and performance was equivalent regardless of volatility. However, the engine calibration must be matched to the fuel used, with a lower compression ratio needed for higher cetane fuels. For high load operation, their preference was for a fuel of 25-35 CN or 60-80RON, and in [40] it was found that fuels in this range, intermediate between gasoline and diesel, provided the widest range of HCCI operation at a moderate compression ratio. A further comprehensive series of studies [17,41,42,43,44,45,46] has also included extensive studies of fuel effects. They conclude [45] that gasoline is an attractive fuel if HCCI can be sustained over the whole operating region, and that where the engine needs to revert to diesel operation at high load, the fuel should be as much like gasoline as possible consistent with operation in a diesel engine. Blends of diesel and gasoline were tested [39] in HCCI combustion using a diesel engine with minimum engine modification. Successful operation was obtained over a wide range of blends, although PM emissions were high in the engine configuration studied.

Several studies have evaluated the performance of dual-fuelling where two fuels are carried on the vehicle. In principle, the blend proportion could then be varied to provide suitable ignition performance for different engine operating conditions. Tests in an IDI diesel engine [47] found that iso-octane was difficult to ignite, but that introducing ethanol as a blend component made ignition easier. Conversely, ethanol failed to ignite at high engine speeds (perhaps through over-mixing or insufficient residence time), and the authors concluded that intermediate blends could mitigate both problems. A study investigating how EGR affects combustion [48] showed that CO_2 addition to the intake charge delayed both low temperature heat release (LTHR) and high temperature heat release (HTHR). In contrast, adding nitrogen had no effect. Kinetic modelling confirmed that CO_2 acts as an inert diluent, by changing the specific heat ratio. Adding hydrogen retarded heat release even more than CO_2, and it was found that in this case there was a chemical effect, with hydrogen suppressing the formation of OH radicals during LTHR, thus delaying the start of the main combustion.

Ignition improver additives, such as 2-ethylhexyl nitrate (2EHN), at ppm levels have been suggested as another way of changing fuel ignition performance. Di-tertiary-butyl-peroxide (DTBP), sometimes used as an alternative to 2EHN in diesel fuel, was also shown to be effective in gasoline [15]. However, ignition suppressants for diesel fuel seem unlikely to be developed, since all known candidates are metallic. Diesel-water emulsions have been used to control emissions from some in-service vehicles and give some reduction of PM and NOx emissions. However the emission reductions targeted by HCCI are much greater and one study [49] indicated that impractically high water concentrations would be needed to obtain significant gains.

Because of its natural abundance, natural gas has generated significant interest as a potential vehicle fuel, although HCCI combustion is challenging. Examples have been demonstrated, but required high intake temperatures to achieve combustion. A recent study [51] found that NG exhibited no low temperature heat release, and the long ignition delay led to very rapid combustion with high knock levels. The authors considered that NG was not successful as an HCCI fuel because of high NOx emissions and low efficiency.

Carrying two different fuels on board the vehicle introduces a significant complexity. An alternative is to produce different fuels on the vehicle by processing the single fuel carried in the tank, provided this can be done efficiently [50]. Although such an approach is also complex, the burden of this complexity is borne by the vehicle manufacturer rather than the customer, so might have more chance of acceptance. Most work in this area is still at the research stage, although Delphi is reported to have a reformer available. Reformer gas, comprising largely CO and H_2 can be produced from a range of different fuels, either by using internal EGR as a source of reaction heat, or in an external reactor[16]. Simulated reformer gas was added to the intake charge in a CFR engine fuelled by NG [51]. Some improvement was seen in the operating range of the engine, but specific NOx emissions remained high. Since reformed gas further delays heat release in what is already a very ignition-resistant fuel, applying this technique to NG does not seem promising. A second study [48] started with methanol, a fuel that is among the easiest to reform. They noted that DME, an easily ignitable fuel, can be formed from methanol by dehydration, while a different reactor could produce a reformer gas. They further consider using the water gas shift reaction to increase the hydrogen content of the reformer gas. Although complex, this approach has the merit of producing both a highly reactive fuel (DME) and an ignition resistant fuel (reformer gas). Adjusting the proportion of these fuels could provide a way of adapting the fuel to changing engine conditions.

3.3.6. Air-fuel ratio

Increasing the air-fuel ratio is another way of slowing ignition and combustion, so many HCCI concepts use very lean mixture ratios. This in turn impacts on the maximum power that can be produced, since lower air-fuel ratios are needed as engine power increases. Turbo-charging has the potential to restore the lost power, but the increased temperatures will also accelerate combustion.

[16] See references [48] for a brief review of previous studies

3.4. EXHAUST EMISSIONS

3.4.1. NOx

Current gasoline and diesel engines both produce high levels of NOx at the engine outlet. Stoichiometric gasoline engines can successfully reduce these emissions using three-way catalysts, but catalysts that operate under lean conditions remain a challenge. NOx formation in conventional engines is dominated by the Zeldovich mechanism, where nitrogen and oxygen from the air combine to form NO at high temperatures. Although the reaction is reversible, it is extremely sensitive to temperature, and as the combustion gases cool, any NO present is 'frozen' and carried through to the exhaust, from where it is eventually converted to NO_2 in the atmosphere. On modern vehicles with oxidation catalysts, much of the NO can be converted to NO_2 over the catalyst, and this has led to increased concern over direct NO_2 emissions from diesel vehicles [52].

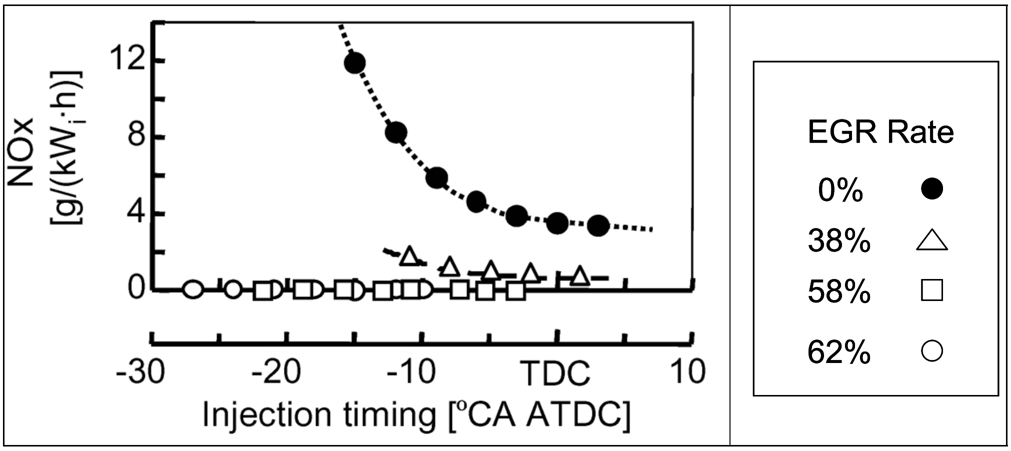
In HCCI combustion, very low exhaust NO levels can be achieved, and under these conditions other NO formation pathways may become important. In [53] it is estimated that half of the NO under these conditions came from an alternative reaction pathway involving N_2O. Although in this work 'prompt'[17] nitrogen did not play a part, very low NOx emission levels could lead to closer scrutiny of even low levels of nitrogen in the fuel and lubricant. However, avoidance of thermal NO formation is the main challenge, and as the ϕ-T diagram shows, reducing combustion temperature is the key to lower NOx emissions.

EGR has been used for many years as an effective way of reducing engine-out NOx emissions, and its success can be readily explained by the high temperature sensitivity of the NO formation reactions. Exhaust gases act as an inert diluent, moderating the peak combustion temperatures. Moderate levels of EGR are used in conventional vehicles for this purpose, especially for diesel engines where exhaust aftertreatment is more difficult than for stoichiometric gasoline vehicles. In conventional diesel engines, the amount of EGR that can be used is limited by the increase in PM emissions at higher EGR (the well-known 'PM-NOx trade off'). The effect of EGR on NO formation is mainly thermal; Kook et al [16] calculated that the adiabatic flame temperature dropped from 3700K, with no EGR, to 2000K with 65% EGR, at which point the oxygen content is reduced to around 10%.

The effect of EGR on NOx emissions is shown dramatically in **Figure 8**, adapted from [23]. Tests were performed in a 1.0-litre single cylinder DI engine with CR of 16:1 and using a common rail injection system. NOx levels below 10ppm were achieved with EGR rates of 58 and 62% over a wide range of injection timings.

[17] 'Prompt' nitrogen refers to nitrogen contained in the molecules of the fuel.

Figure 8 EGR reduces NOx emissions [23]



Leaner air-fuel mixtures will also lower temperature and reduce NO formation. In premixed diesel combustion using gaseous fuels [54], NOx emissions decreased by 100 times as the excess air ratio (λ) increased from 2 to 4. However, in a practical engine, such leaning of the mixture means lower power, so the problem of NO emissions will remain at high load. If engine conditions are adjusted to maintain the same power as the air-fuel mixture is leaned, NOx emissions may rise [6].

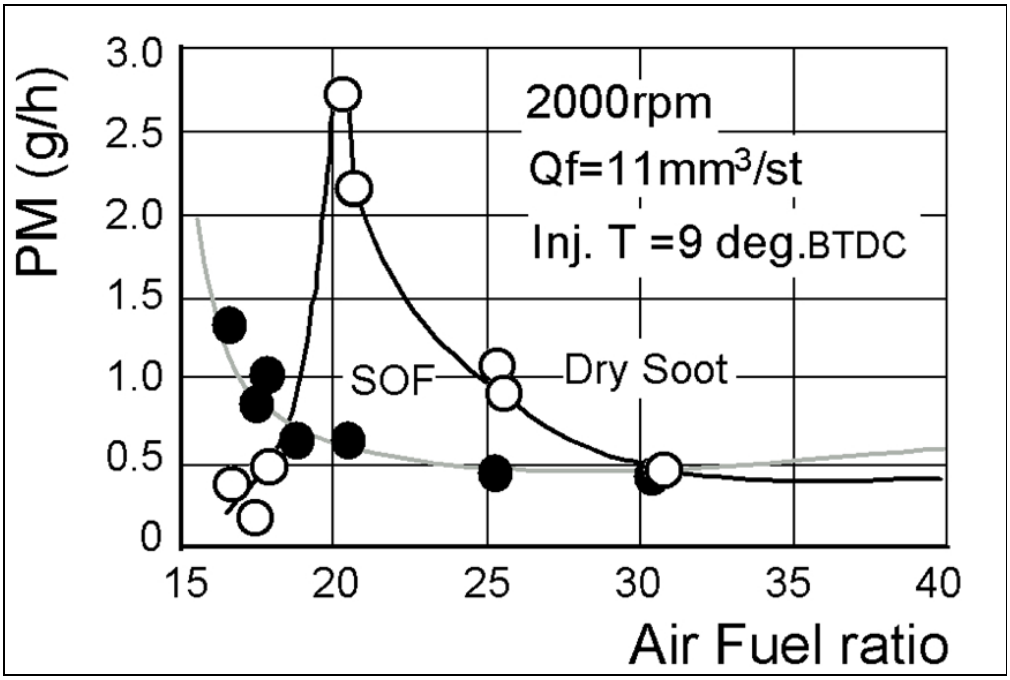
The effect of different compression ratios was investigated in [23]. With a CR of 14 or 16:1, the injection timing had to be advanced in comparison to 18:1 CR to maintain ignition at TDC, and this effect became more marked as the EGR rate increased. However, provided ignition remained at TDC, NOx emissions were the same regardless of compression ratio.

It therefore appears that changing the mixture composition through EGR is the most effective way to control NOx emissions. The effects of leaner air-fuel ratio or lower compression ratio may be negated when the constraint of equal power or constant ignition point are imposed.

3.4.2. Particulates and soot

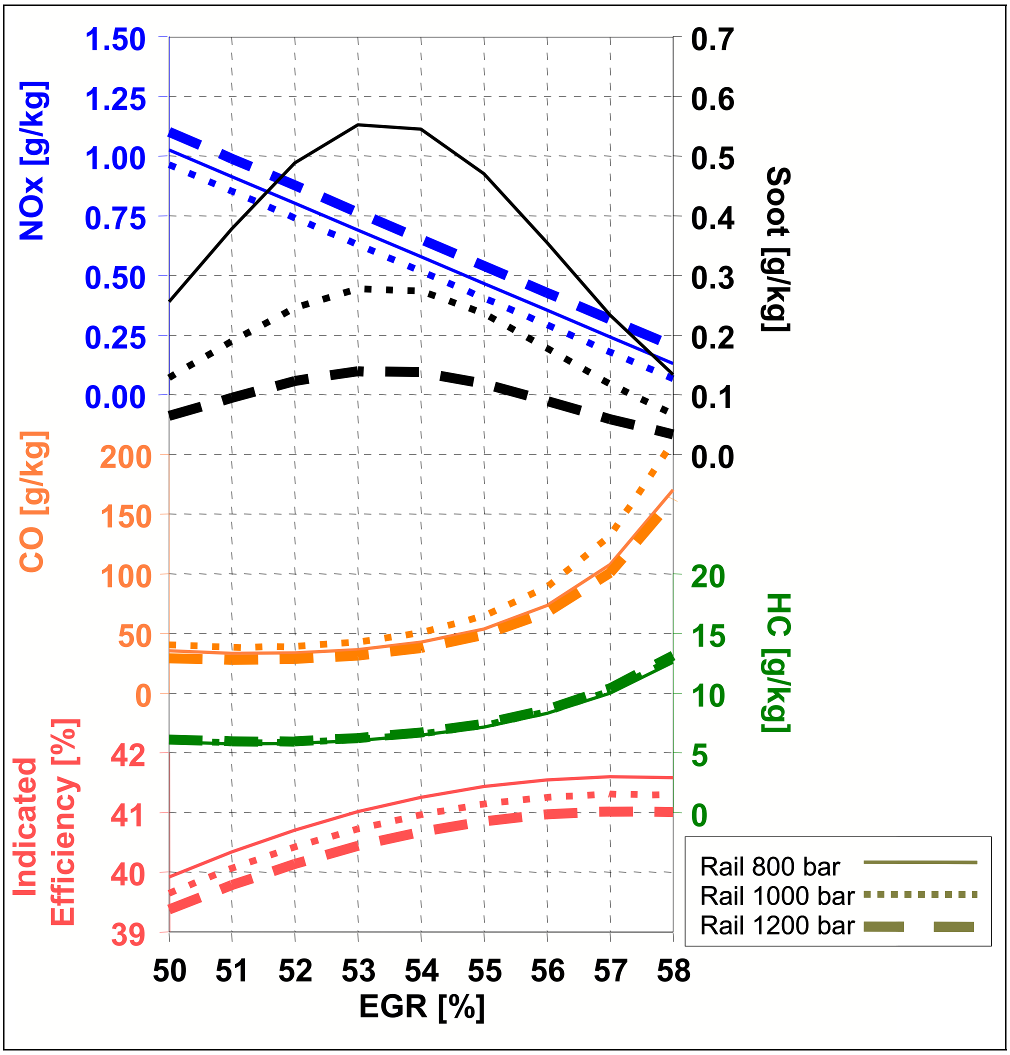
Soot formation in conventional engines is primarily a function of mixture preparation. Hence, in gasoline PFI engines where the fuel and air mixture is almost uniform, soot emissions are very low. In conventional diesel engines, although the overall mixture is lean, fuel distribution is stratified and combustion takes place around the fuel spray where fuel rich conditions exist. Large amounts of soot are formed in the early stages of combustion, however most of this soot is oxidized as the combustion proceeds [55]. Measures that influence either the initial mixture preparation or the subsequent combustion can therefore impact engine-out PM emissions. In conventional diesel engines, higher fuel injection pressure and higher boost pressure are effective ways to reduce soot, but to reach the very low PM levels required by future emission legislation, particle traps will be needed. Higher injection pressures are equally effective in HCCI combustion to reduce soot emissions [23]. In true HCCI combustion, where the fuel is injected very early in the cycle, significant mixing takes place before ignition occurs, avoiding the very rich conditions where soot can form. In conventional engines, EGR tends to increase soot emissions, however several researchers have reported that at very high EGR levels (above 55%), reductions in both NOx and soot can be achieved [23,16,14,6] (**Figure 9**). Because soot oxidation rates are thought to decrease with charge dilution, this effect is generally attributed to reduced soot formation. In [16], it was found that dilution through EGR did not completely suppress soot formation, but emissions were reduced by two orders of magnitude at 65% EGR.

Figure 9 Effect of reducing AFR by increasing EGR on PM emissions [6]



Many proposed HCCI systems take advantage of this effect of high EGR levels to reduce soot, however [14] has shown that as engine load increases the air-fuel ratio at which soot emissions start to decline moves to richer (higher EGR level) conditions that are harder to achieve. The same study also shows that increasing fuel injection pressure dramatically reduced soot formation, as for conventional diesel combustion (**Figure 10**).

Figure 10 Impact of EGR variation at different rail pressures on emissions and efficiency under Homogeneous Charge Late Injection (HCLI) conditions [14]

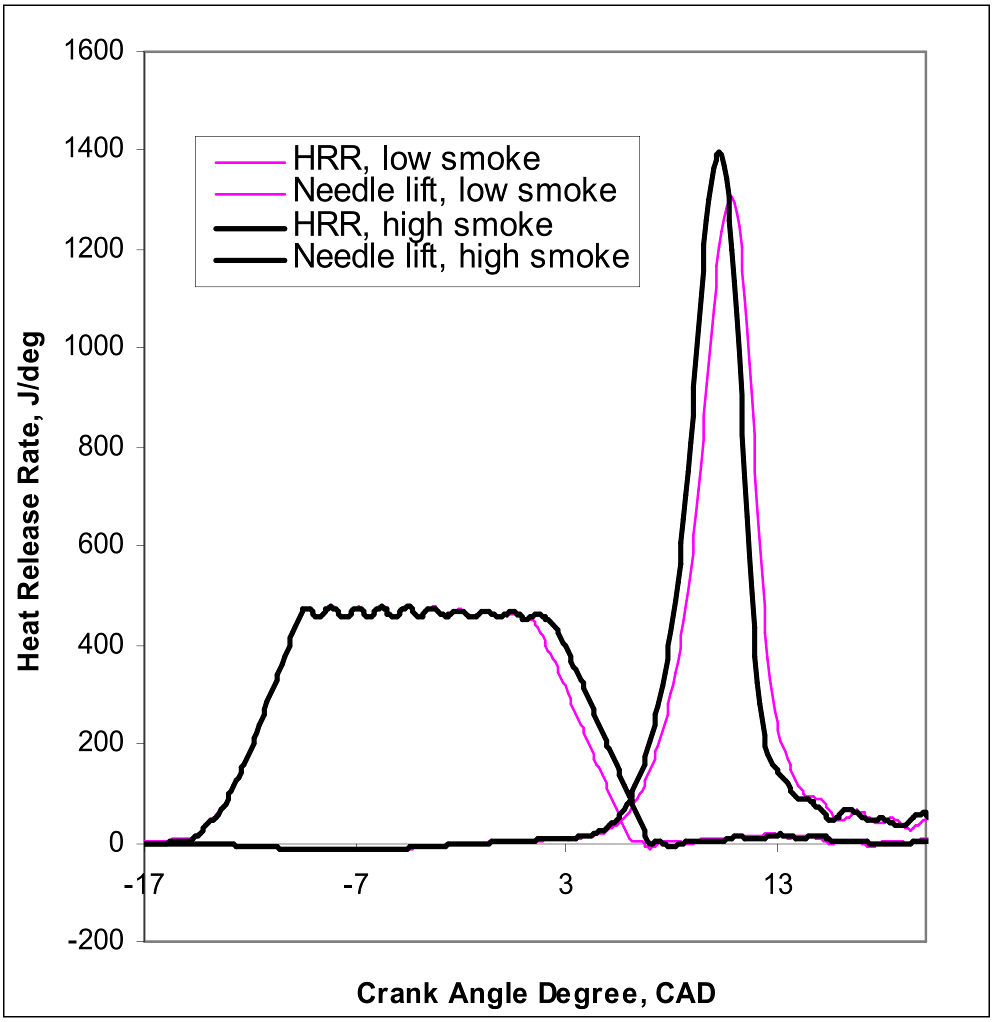


As the ϕ-T diagram shows, soot formation can also be reduced or even suppressed if the temperature is sufficiently low, around 1600K, and this may be one reason why very high EGR levels can benefit PM emissions, however charge dilution through EGR also increases the ignition delay time, and the increased mixing before combustion starts may also contribute. Such low temperatures are difficult to sustain, but even at 2100-2200K both NOx and soot can be suppressed if very rich conditions (ϕ<2) are avoided [23].

In [56], tests were performed in an optical engine with a quiescent combustion chamber and single spray injectors. The authors caution against direct application of the results to real engines, however, they demonstrated that sootless combustion can be achieved in different ways. High EGR rate coupled with a small nozzle hole (50 micron) led to good mixing as well as cooling of the combustion, but the same injector gave sootless combustion even without EGR if the charge temperature was lowered sufficiently. They noted that mixing controlled combustion occurred, but the mixture was lean. Lastly, using a more conventional nozzle with a 180 micron jet, even higher EGR and an oxygenated fuel, sootless combustion again occurred; they believe the temperature was too cool for soot formation (although the oxygenated fuel would also contribute). These tests illustrate the difficulty of separating the different effects, and practical systems are likely to employ a combination of temperature and mixture control.

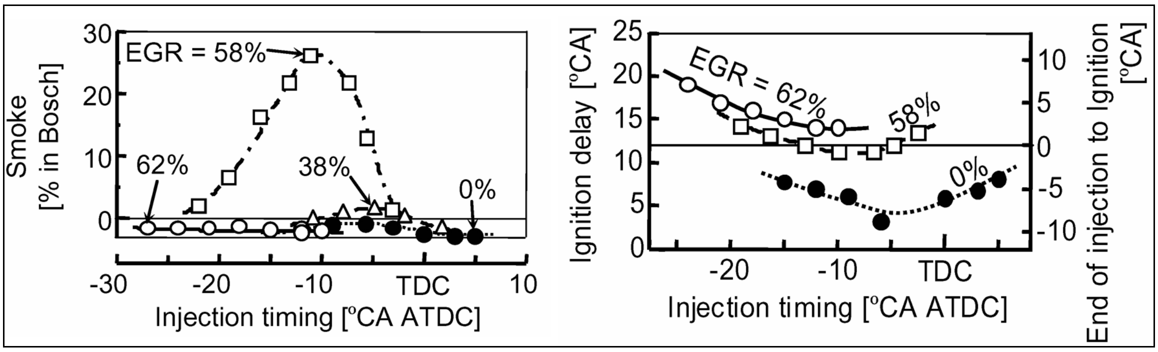
Several researchers have shown that soot can be avoided provided that the main heat release does not occur until fuel injection has been completed [17,23,57,12,58]. Kalghatgi [17] showed that even a small overlap of fuel injection and heat release led to a dramatic increase in soot. The variations in amount and timing of the peak heat release between the two test points illustrated were small, but the smoke increased by a factor of more than 20 **(Figure 11)**.

Figure 11 Heat Release Rate (HRR) and Needle Lift - effects on smoke emissions [17]



Tests in [23] also illustrate this point. The effect of high EGR rates in suppressing NOx emissions have already been shown in **Figure 8**, and the same tests showed low soot formation at the highest EGR rate. The ignition delay increased significantly as EGR rate increased, and it can be seen (**Figure 12**) that sootless combustion occurred at those points where fuel injection was completed before ignition took place.

Figure 12 High EGR reduces soot by increasing ignition delay [23]



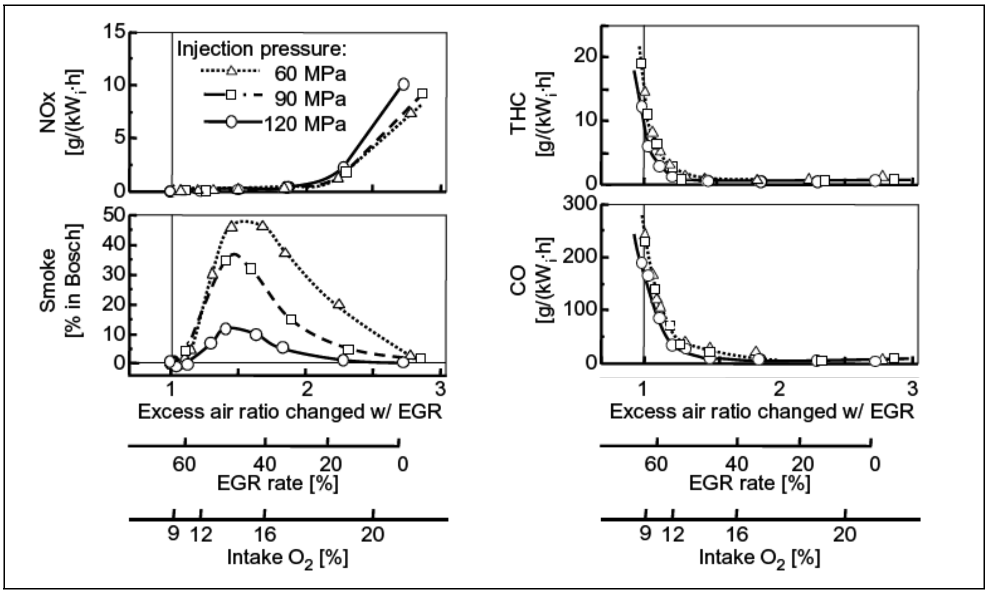
Delaying the main heat release until fuel injection is complete is feasible at lower loads, but becomes increasingly difficult at high load: the duration of fuel injection increases to provide the fuel volume needed for high load, and since engine temperatures tend to increase with load, the ignition point tends to move earlier. Ways of controlling the ignition delay and combustion timing are discussed further in **Section 3.5**.

The influence of fuel injection and boost pressures has already been mentioned. Other engine design features such as combustion chamber geometry and compression ratio can also affect soot formation. Tests in a naturally aspirated diesel engine [23] showed that reducing compression ratio from 18:1 to 14:1 increased the range of smoke-free operation using cold EGR. The lower compression ratio allows more time for mixing, because the start of combustion is delayed; however, this benefit comes at the price of lower thermal efficiency.

The effects of swirl have been studied in [14]. Increasing swirl initially led to smoke reduction and NOx increase, as seen in conventional diesel combustion. When swirl was further increased, NOx emissions reduced, but above a certain point volumetric efficiency decreased, so the amount of swirl that can be used is limited by fuel consumption increase.

While very high EGR rates seem to offer the best opportunity to simultaneously reduce PM and NOx emissions, such high rates can be difficult to achieve at higher loads. EGR reduces the amount of air available in the cylinder, which in turn limits the amount of fuel that can be burned. More modest levels of EGR are required to reduce NOx emissions than to achieve sootless combustion and soot emissions can be reduced by engine modifications such as injection pressure as shown in **Figure 13**, again taken from [23]. Such strategies may be attractive in some cases, particularly where vehicles are already fitted with PM traps.

Figure 13 Impact of EGR and fuel injection pressure on exhaust emissions [23]



3.4.3. HC and CO emissions

In conventional diesel engines, the abundant supply of excess air, coupled with high combustion temperatures result in very low emissions of HC and CO. In HCCI combustion, where excess oxygen is reduced and temperatures lowered to reduce PM and NOx emissions, conditions for complete oxidation of the fuel are much less favourable, and virtually all studies of HCCI combustion have reported significant HC and CO emissions. It seems inevitable that HCCI engines will need oxidation catalysts to remove these emissions from the exhaust gases. Since technology exists to do this, it is not generally considered to be a serious problem. In some cases, however, incomplete combustion can be sufficient to have an impact on the engine's overall efficiency.

Consideration of engine design contributions to HC and CO emissions in [54] concluded that fuel contact with the cylinder walls could be a factor with long ignition delays. They therefore devoted considerable attention to minimising the spray penetration to avoid this, but found that other sources of emissions remained. Modelling studies have shown that the quenching at combustion chamber walls and in crevice volumes contributes to HC emissions in HCCI combustion as in gasoline engines, but that an additional mechanism, termed 'bulk gas quenching' in [20] also plays a part. In simple terms, the combustion gases are not hot enough to carry the reactions through to completion.

In [17], both CO and HC emissions were found to increase as the ignition delay increased (**Figure 14**). As mixing increases, some part of the fuel may be in very lean areas where complete combustion is difficult, and unburned or partially reacted fuel can survive to be emitted through the exhaust. Temperature also plays a part in the completeness of combustion, and the cooler combustion that favours lower NOx and soot is unfavourable for complete oxidation of HC and CO. In conventional engines, reaction continues up to and even after the exhaust valve opening. Where combustion is later, or cylinder temperatures lower, it is more difficult to achieve complete oxidation [19].

Figure 14 CO and HC emissions vary with ignition delay [17]

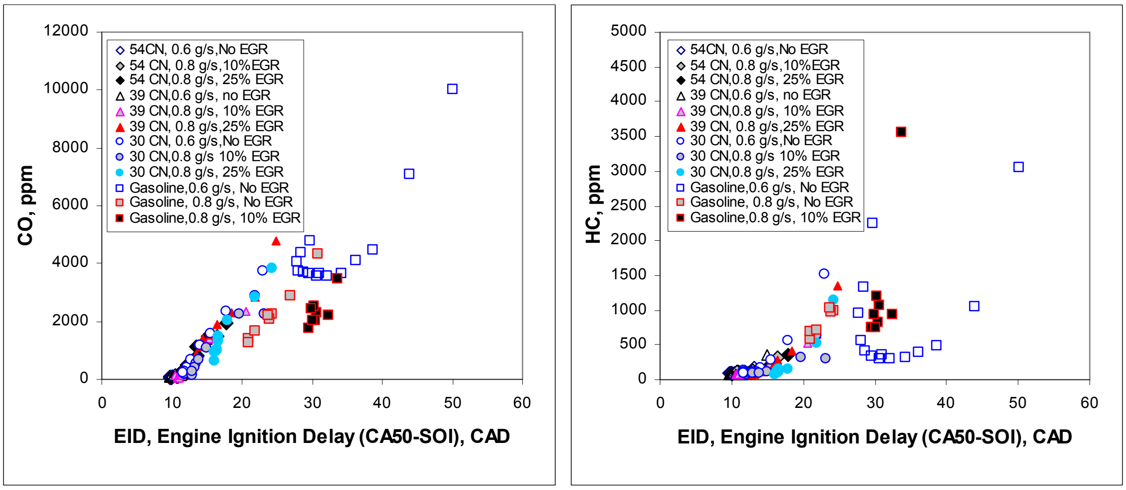
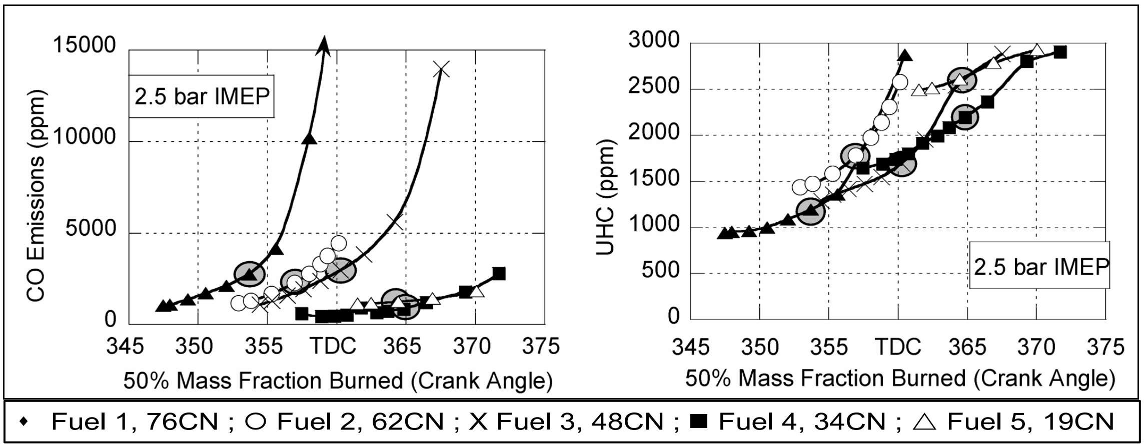


Figure 15 CO and HC emissions vary with combustion timing and fuel type [19]



In [19], emissions increased with combustion retardation for all the fuels (**Figure 15**). However, the lower cetane Secondary Reference Fuels which did not produce LTHR (fuels 4 & 5) produced high HC emissions, but lower CO than the higher cetane fuels that did show LTHR (fuels 1, 2 & 3).

The authors suggest that this is because CO is formed during the LTHR, but that further oxidation is inhibited in the main heat release by the presence of unburned hydrocarbons. An alternative interpretation is presented in [20], where a review of previous work suggests that temperatures of 1400-1500K are needed to convert CO, and that fuels producing LTHR can operate with cooler charge intake, and may not reach these combustion temperatures.

The composition of partial oxidation products from HCCI combustion have been studied in [20]. They noted that the FID measurement used for regulatory tests underestimates some species, particularly oxygenated products, and developed their own analytical technique which allowed them to analyse longer chain aldehydes and carboxylic acids than previous methods. Using these techniques, the authors estimate that emissions of oxygenated compounds from HCCI combustion may be as much as pure HC emissions. Trends for individual pure hydrocarbon species and smaller aldehydes were similar to those for total HC, however some differences were seen for longer chain oxygenated species, with higher emissions as combustion was retarded for a fuel producing LTHR. The broad findings of this work are in line with studies on conventional diesel vehicles [59], where a significant contribution of oxygenated compounds to total organic emissions was also seen. In general, oxidation catalysts are effective at removing both pure hydrocarbons and oxygenated species, so a shift in the detail composition is not necessarily a cause for concern, provided overall emissions can be maintained at low levels.

3.5. CONTROLLING HEAT RELEASE, PEAK PRESSURE AND COMBUSTION NOISE

In conventional engines, fuel is consumed progressively, so at any time only part of the energy in the fuel is being released. In the case of the SI engine, for example, a flame front passes progressively through the pre-mixed charge. In the diesel engine, the initial combustion takes place very rapidly in a portion of essentially pre-mixed fuel, and is then followed by a diffusion burning in the fuel spray as more fuel is injected. If large scale ignition takes place homogeneously across the cylinder (through knocking in a gasoline engine or because of a very long ignition delay in a diesel engine), the high rate of heat release and pressure rise result in high noise and can even lead to engine damage. Since near-homogeneous and rapid combustion is the goal of HCCI, some measure is needed to moderate the rate of combustion and avoid these problems. A high Rate of Heat Release (ROHR) is associated with high Rate of Pressure Rise (ROPR), which is directly related to combustion noise. To avoid engine damage or durability problems, the peak cylinder pressure is the most important variable.

In conventional engines, combustion noise is a feature of uncontrolled combustion. Combustion in Gasoline PFI engines takes the form of a flame front that moves progressively through the pre-mixed air-fuel mixture, so the heat release rate is fairly uniform, and combustion noise is low. It is only under the abnormal conditions of knock, when the remaining mixture ignites explosively that combustion noise may be heard. Combustion noise is a more familiar occurrence in diesel combustion, particularly at low loads and with a cold engine when the ignition delay is long. A longer ignition delay means that more fuel has accumulated and mixed in the combustion chamber before ignition occurs. On ignition, all of this fuel burns very rapidly, producing the familiar diesel knock. Combustion noise in modern diesel engines has been reduced significantly by improved combustion chamber and fuel system design, and in particular the use of pilot injection to reduce the amount of premixed combustion. Since noise depends on ignition delay, the cetane number of the fuel also plays a part; a rule of thumb for conventional diesel engines is that reducing cetane number by 10 increases combustion noise in the cylinder by 3dBA. The overall noise perceived from outside the vehicle depends not just on the noise from combustion, but also on the degree of noise attenuation provided by the basic engine design and any noise shielding installed by the manufacturer, and on the mechanical noise. Combustion noise will only be heard when it exceeds the level of mechanical noise.

Noise is an important factor in engine design that can significantly affect consumer acceptance of the technology. In light-duty applications, the gasoline engine provides a ready comparison, and engines that are significantly noisier would not be acceptable, and noise from heavy-duty vehicles is equally a cause for public concern. A successful HCCI engine should therefore aim to produce noise levels no higher than today’s diesel vehicles, but since a goal of HCCI combustion is precisely to increase ignition delay, there is clearly a potential for high noise levels unless measures are taken to dampen the rate of combustion or ROHR. Meeting this challenge could be more difficult than simply reducing PM and NOx emissions.

We can identify three main variables that influence the ROHR. In the discussion on control of combustion timing, the ignition delay was identified as an important parameter to control combustion timing, with long ignition delays facilitating better mixing for lower emissions. This trend directionally increases the amount of pre-mixed burning and the ROHR. Heat release is faster at high temperatures and pressures, so moving the combustion later after TDC can moderate the ROHR, but potentially with a loss of efficiency. The third variable is the mixture condition: reactions are slower in diluted mixtures. This is readily evident for high air-fuel ratios, however where EGR is used, the slowing effect of dilution by inert gases needs to be balanced against a possible acceleration brought by the hotter exhaust gases, if the EGR is not cooled. Controlling ROHR therefore presents a challenge, since many measures to reduce it would compromise other performance goals. The peak cylinder pressure is closely related to ROHR, since the highest pressure will be obtained where combustion takes place close to the peak compression pressure near TDC, and where the heat release is very rapid and takes place over a short time period.

Increasing the air-fuel ratio at the same combustion timing reduces the ROHR and peak pressure significantly [30]. However, for a given engine, increasing air-fuel ratio implies a reduction in load, so a comparison at the same power could only be made if the engine size or boost pressure were increased. In practice, unless injection timing is adjusted, the combustion timing tends to retard as load decreases [15]. This aspect is discussed further in the section on extending the power range.

At fixed start of injection, increasing the amount of cooled EGR reduces the peak cylinder pressure and ROHR [16]. The longer ignition delay with EGR moves the combustion later in the cycle where the piston has already started to descend and compression pressure is lower. Advancing the injection timing to obtain the same start of combustion with EGR was shown to increase peak pressure back to the same or slightly higher values than were obtained without EGR. The same study considered the effects of EGR on the burn duration, which is of course related to the ROHR. The results are strongly dependent on injection timing, but the minimum time from 10% to 50% heat release (CA50-CA10), which may be considered representative of optimum combustion timing, was increased by a factor of 2 or more for EGR rates over 60%.

The concept of less homogeneous combustion has already been discussed, and in addition to other benefits, could provide a way of prolonging the heat release, thus reducing ROHR and peak pressure. This more pragmatic approach means that systems where the fuel is directly injected into the engine even at a fairly late stage in the cycle can be considered, provided that the fuel injection and combustion events can be separated in time, to avoid the diesel-like diffusion burning that produces soot. Many specific concepts exist, each with their own names, but the more generic terms 'low temperature combustion' and 'PCCI' (Premixed Charge Compression Ignition) are becoming used, to distinguish these more practical systems from 'true' HCCI.

It has been observed that measured rates of pressure rise are lower than predicted by chemical kinetic models [30,31], and this has been attributed to the effects of natural thermal stratification in the combustion chamber. Studies with an optical engine [30] suggested that there is sequential auto-ignition of different regions within the combustion chamber. Potential sources of thermal stratification are hot residual gases, fuel/air mixing, heat transfer to engine surfaces and turbulence. The impact of stratification was found to be greater at more retarded combustion timings. Peak pressure and ROHR were reduced when thermal stratification was increased through cooler walls and increased swirl.

Fuel stratification was investigated in [21] by using an early fuel injection to create a homogeneous mixture, followed by a second injection to generate stratification. Engine conditions were adjusted to maintain the same combustion timing in each case. The second injection led to a less smooth combustion with an increased maximum ROHR.

The effects of turbulence were studied in an optical engine at λ=3, supported by modelling [33]. Where temperature inhomogeneities were low, turbulence delayed ignition and extended combustion duration. However, turbulence also appears to smear-out temperature inhomogeneities, and so diminish the impact of thermal stratification. The role of multiple fuel injections is discussed in [17], including their potential to moderate ROHR.

Fuel effects on the burn rate were studied in a Rapid Compression Machine in [89]. They found that the burn rate decreased as the ignition delay increased, but for the same ignition delay there were differences depending on fuel composition. Generally, fuels containing cyclic structures had lower burn rates than n-heptane/iso-octane PRF blends.

3.6. EXTENDING THE POWER RANGE

3.6.1. Power requirements of modern engines

Engine power is measured on a test bed by operating the engine against a brake, and may be expressed as brake horsepower (bhp) or kW. Engine power can also be expressed in terms of the work released in the engine cylinder. This is defined in terms of the average pressure which, if exerted on the piston over the complete power stroke would produce the power measured by the brake, and is termed the Brake Mean Effective Pressure (BMEP). An alternative measure which may be encountered is the Indicated Mean Effective Pressure (IMEP). This is based solely on measurements of pressure in the cylinder, while brake mean effective pressure (BMEP) is a measure of the practical power available at the brake, after deduction of friction and other losses in the engine. The size of engine needed is linked to BMEP, since BMEP is directly related to the specific power and torque per litre of engine capacity.

There is a trend for the specific power of engines to increase, driven by customer demands for higher performance, and the potential for fuel efficiency gains through use of smaller engines. For conventional gasoline engines, specific power and torque has increased by about 5% during the period 2000-2005, and a further 5% increase is expected in the period through to 2015 [60]. Typical power and torque for gasoline engines are currently 45-60 kW/litre and 87-103 Nm/litre [61] The lower end of this range represents the more basic engines found in smaller cars, while the top end represents more sophisticated and costly technologies employed in luxury vehicles. A similar trend for higher output is seen for light-duty diesel engines, with specific torque doubling from 1985 to 2005. The average power and torque for light-duty diesel engines in the European fleet in 2005 were 46 kW/litre and 150 Nm/litre, however the best performing engines achieved 70 kW/litre and 195 Nm/litre [61].

Diesel engines have already benefited from more attention than gasoline engines, and the scope for further near-term improvement is perhaps limited to a further 5-10%.

The increasing penetration of diesel engines into the light-duty market has played a large role in the motor industry's effort to reduce fleet fuel consumption and CO_2 emissions, however further improvements to the gasoline engine will also be needed if progress is to be maintained. The current trend is towards higher specific power outputs allowing the use of smaller engines without loss of performance, and concepts using variable valve actuation and EGR coupled with high pressure turbo-charging are being developed. If successful, these concepts could come to production around 2012-2015, a similar timeframe as envisaged for HCCI concepts. Up to 40% reduction in engine size is possible, using turbo-charged gasoline engines producing up to 180 Nm/litre torque [60], and fuel efficiency could match that of diesel engines [61]. The good low-speed torque characteristics of these engines would also enable them to match diesel performance. A final consideration is cost. Current diesel engines cost around 75-100% more than a basic PFI gasoline engine: the acceptability of these advanced gasoline engines for smaller, more economical vehicles will depend on how cheaply they can be produced.

These engine trends all imply increases in the maximum BMEP of the engine. An increase in BMEP means that a smaller engine can be used, with potential advantages for engine weight and vehicle efficiency. Hence, if HCCI engines are to be successful, they need to match the power and BMEP capabilities of conventional engines that are improving year by year.

Conventional naturally aspirated gasoline engines typically achieve a maximum BMEP of about 12 bar [60,61]. The best naturally aspirated production engines can manage 13.6 bar, but this requires careful development and specialized components, while some race engines can produce over 15 bar, albeit at very high engine speeds [63]. Turbo-charging opens the possibility for much higher power output, and gasoline engine concepts using BMEP of around 20 bar have been demonstrated. Even higher ratings, up to 28 bar, are under consideration [60,61].

Diesel engines currently used for both light-duty and heavy-duty applications already routinely use turbo-charging and intercooling to maximise power and efficiency, and today's production light-duty engines typically achieve 26 bar BMEP [60]. Further improvement by higher levels of boost is possible, but may be constrained depending on future exhaust emissions requirements.

The range of BMEP needed over the European drive cycle for exhaust emissions depends to some extent on the vehicle and engine sizes. For a typical 1.6-litre gasoline car, powers up to 7 bar BMEP may be used [60], while for a diesel car the figure may be around 9 bar [61]. These figures will vary for other driving cycles, for example the same diesel car would operate up to 16 bar BMEP over the US06 cycle.

For heavy-duty diesel engines, normal operation can involve loads up to 100%, and this is reflected in the regulated emissions test procedures. Typical US heavy-duty engines produce 30-45 kW/litre, with BMEP in the range 15-23 bar, with figures at the higher end of this range expected for future engines [62].

3.6.2. Power capabilities of HCCI engines

Engine power is increased by injecting a larger amount of fuel. For a conventional gasoline engine this is achieved by increasing the amount of mixture inducted at essentially fixed air-fuel ratio. For more advanced engines, where throttling is not the main means of controlling engine power, the air-fuel ratio must be progressively reduced as maximum engine power is approached. The maximum power will be governed by the lowest air-fuel ratio that can be successfully burned. Diesel engines, because of their stratified combustion, need to maintain overall lean conditions even at full load. If the air-fuel mixture becomes too rich, soot emissions increase dramatically, and this limits the maximum power of the engine.

Two limits can be defined for HCCI operation. When the rate of pressure rise becomes too high (the engine requirement exceeds the ignition resistance of the fuel), knocking will occur and this may define the upper load limit [15,41]. Rapid heat release rates also lead to high combustion noise, and this may also be limiting. A second limit can apply at very low loads; if the ignition resistance of the fuel is less than the engine requirement, engine instability can appear, causing large cycle to cycle variations. It is the first of these limits which is of importance for the current discussion. In most cases, the maximum power that can be achieved under HCCI combustion is less than that achievable in normal diesel operation.

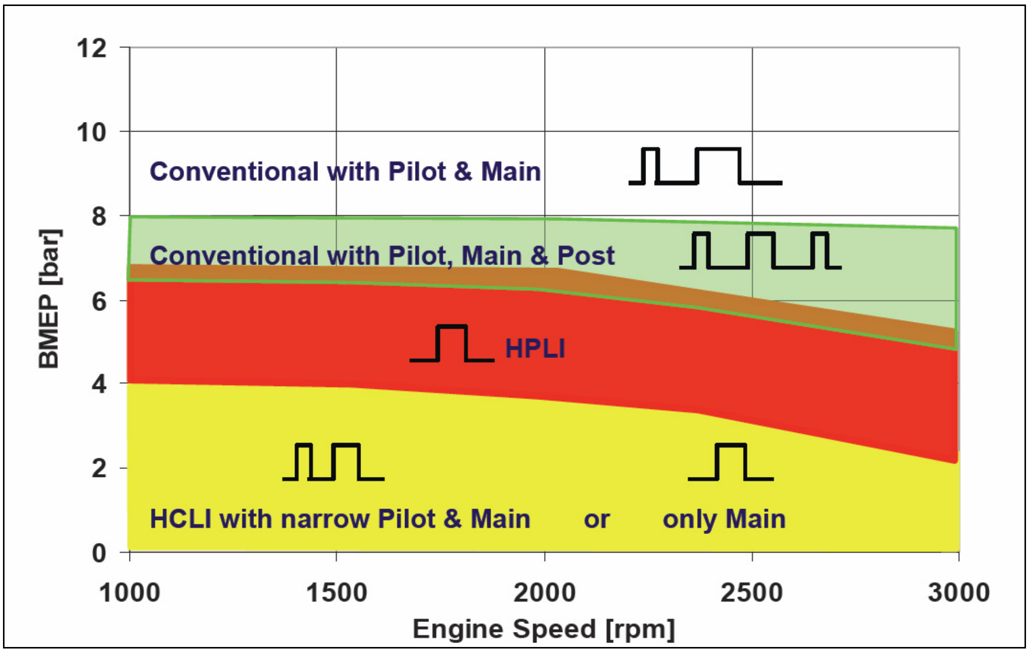
In fully pre-mixed combustion using iso-octane, the knock limit was reached at an equivalence ratio (ϕ) of 0.27 with combustion (CA50) at TDC [30]. Modelling showed that retarding the combustion timing allows higher equivalence ratios to be used, and even a retard of 10°CA is sufficient to make a large difference. However, this change also moves the combustion into an unstable area and the COV of IMEP rises dramatically as combustion timing is retarded.

Many concepts reported in the literature are limited to around 6 bar IMEP maximum, and revert to conventional SI or diesel combustion above this level. Practical engine developments are therefore focussed as much on optimising the non-HCCI areas of the engine map as the lower load HCCI regions. IFP has developed the NADI™ concept (Narrow angle Direct Injection) which uses a combustion chamber and fuel injector configuration specifically to achieve homogeneous combustion [7,8,9,10]. The NADI idea is a ‘dual mode’ concept using highly premixed combustion at low and medium load and conventional combustion at high loads. This concept is based on the use of multiple injection strategies depending on engine speed and load. At low engine load the early injections obtain a good mix between fuel and air before the combustion, with increasing load however the injection strategy changes. The first fuel is injected late, before TDC and a second injection takes place after TDC, which is the so-called ‘TDC split injection’. To minimise wall wetting during early or late injection timing injection cone angles of typically 50º-100º are used (conventional engines typically use 145º-155º). HCCI can be maintained over the range of the European driving cycle, but conventional combustion must be used outside this range.

AVL envisages a similar progression of combustion from pre-mixed HCCI at low load, through partially mixed combustion to conventional diesel operation, but using a late injection approach [14,61] (**Figure 16**). In the intermediate 4-8 bar power range, low soot and NOx emissions can be achieved, but the later combustion timing leads to a loss of about 2% in efficiency, which implies a fuel consumption penalty of 5% [14]. AVL’s approach also illustrates the potential of modern flexible fuel injection systems to optimise the fuel injection using several injection pulses.

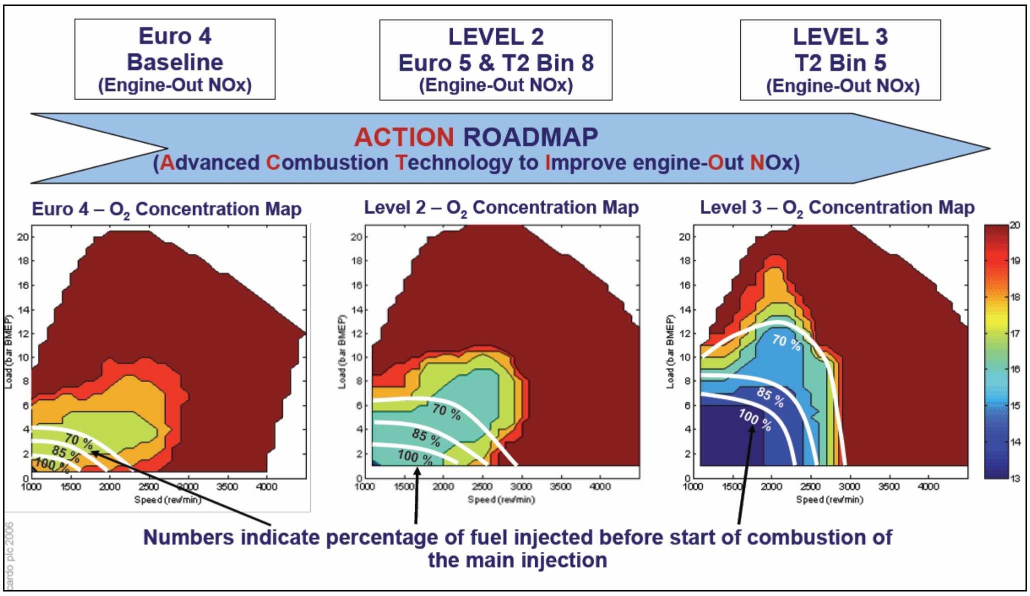
This can provide a way to moderate the rapid heat release that can occur with a single main injection following a long ignition delay.

Figure 16 Different combustion strategies needed according to load [61]



These partial HCCI concepts would allow progressive development to extend the range of HCCI operation, as illustrated by Ricardo's vision below [60] (**Figure 17**).

Figure 17 How development of HCCI might evolve [60]

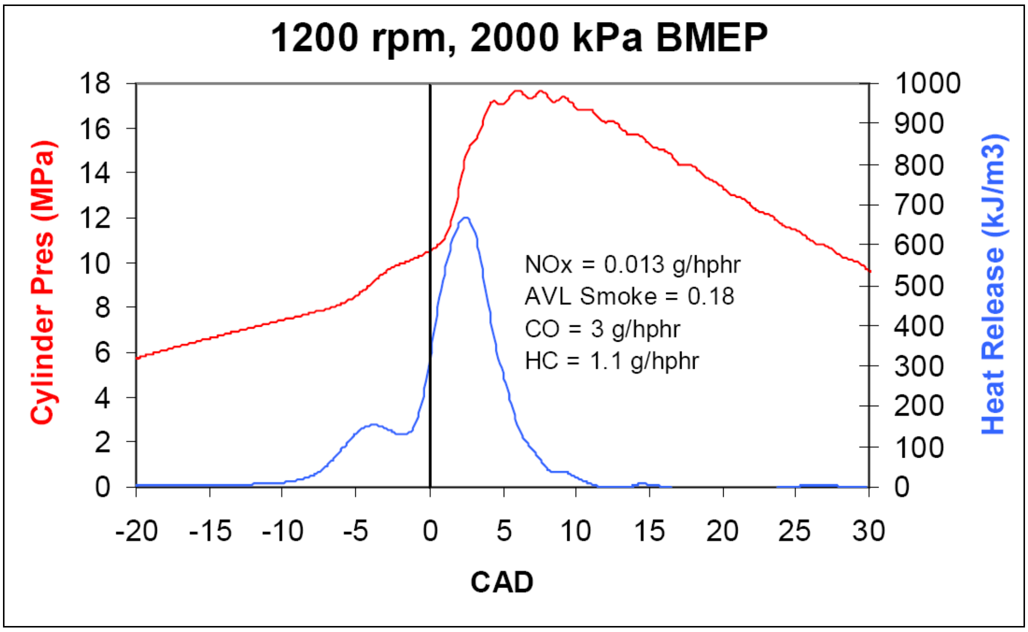


For gasoline engines, CAI combustion has been demonstrated up to around 4 bar BMEP [60], with fuel consumption benefits up to 18% at low load, but diminishing rapidly as load increases. The load limitations of CAI make it difficult to capture the benefits over the full driving cycle. Variable valve lift was found to be a more potent way of improving fuel consumption, so the benefits of CAI diminish as these alternative technologies are introduced. For gasoline engines, other technology options have the potential to improve fuel consumption by 20-25% over the urban drive cycle [64], so gasoline HCCI has received less attention than diesel applications. Nevertheless, VW has announced that a part-time CAI vehicle is nearing production readiness [65]

Fuel properties can impact the range of smokeless combustion. In tests on a 1-litre/cylinder engine [37], reducing cetane number from 63 to 40 increased the range of operation from 5 bar IMEP to 6 bar IMEP. As noted above, the benefits of more ignition resistant fuels have led some researchers to concentrate efforts on gasoline-like fuels.

The above examples represent what could be achieved in the near term, using already existing engine hardware. These concepts are all constrained by the need to revert to conventional gasoline or diesel operation at higher loads, which limits the changes that can be made both to engine hardware and to fuel properties. In the research arena, there is intense activity to develop engines that can achieve the benefits of HCCI combustion at higher loads, including some recent studies where HCCI combustion has been achieved even up to full load. Tests on a 2.44-litre single cylinder engine representative of a heavy-duty diesel engine [18] reported achieving combustion with less than 5ppm NOx and very low smoke at up to 15.5 bar, which is approximately 75% of full load, while later results demonstrated 100% load operation, or about 21 bar [38,40,66]. **Figure 18** illustrates the combustion timing and emissions achieved at 20 bar BMEP. However, the authors note that many challenges remain, including multi-cylinder adaptation, control of combustion phasing, structural reliability and clean-up of HC/CO emissions.

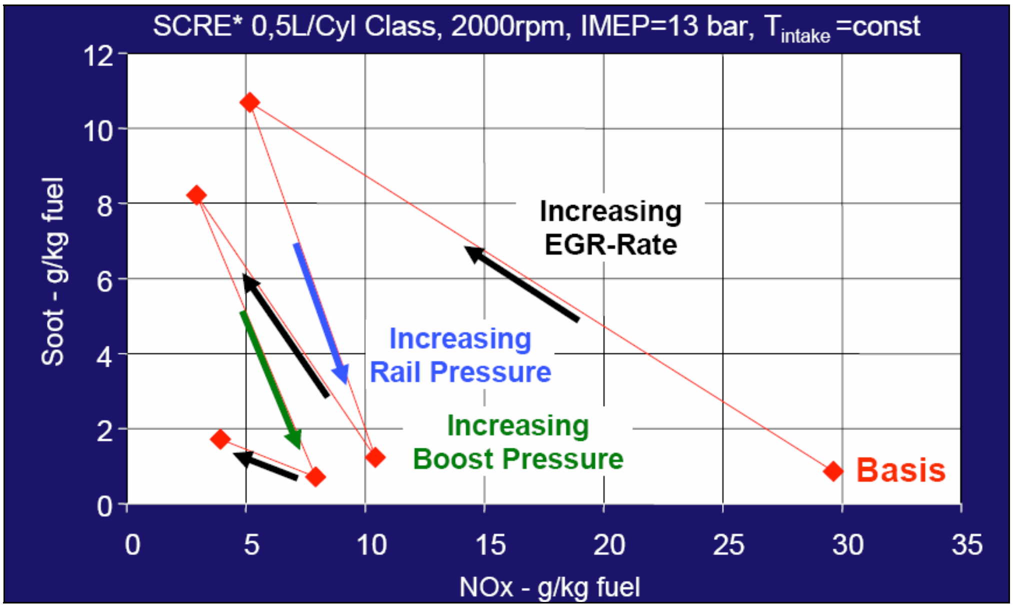
Figure 18 HCCI combustion has been demonstrated at 20 bar BMEP [66]



A separate study, also using a heavy-duty single cylinder engine (2-litre capacity) achieved 15 bar IMEP before excessive smoke emissions occurred, using gasoline as a fuel [17]. NOx emissions were relatively high at this point (1.21g/kWh), but since the EGR rate was only 25% there is scope to reduce emissions by using a higher EGR level. Similar results were found in [40], where operation up to 17 bar was possible with a compression ratio of 12:1 under constraints of maximum 0.27g/kWh NOx and 0.1FSN.

The techniques used at lower loads (EGR, increased fuel injection pressure and turbo-charging), have also been shown to be effective in lowering both PM and NOx at the higher load of 13 bar IMEP [61]. **Figure 19** shows how increasing EGR rate, rail pressure and boost pressure can be used to progressively reduce emissions from the engine.

Figure 19 Effect of engine improvements on NOx and PM emissions [61]



These tests were performed in an engine of 0.5-litre/cylinder which is more typical of light-duty engine configurations. Assuming 200g/kWh fuel consumption, the emission figures achieved translate as 0.8g/kWh NOx and 0.4g/kWh PM.

A further single cylinder HD engine programme using ‘Highly Premixed Combustion’ at higher loads achieved 13 bar BMEP with NOx emissions below 0.3g/kWh, but with PM emissions comparable to a conventional diesel engine [24]. Lowering the compression ratio was found to extend the range of HCCI operation, but with some loss of efficiency and cold-starting capability.

The present status may be summarized as follows:

- HCCI combustion can be achieved up to about 4-7 bar BMEP, depending on the engine configuration.
- At higher loads, ‘true’ HCCI is not generally achieved, but partially mixed and cooler combustion is possible that can produce low NOx and PM emission

levels. Although PM and NOx still increase with load, they do so at much lower rates than for conventional diesel combustion.

- A number of studies have demonstrated the ability to reach 13-15 bar BMEP before high emissions preclude further load increases. One study has demonstrated low emission combustion to 100% load (about 21 bar BMEP).
- Most of these studies have used large engines typical of heavy-duty applications, but similar progress has also been demonstrated in smaller engines.

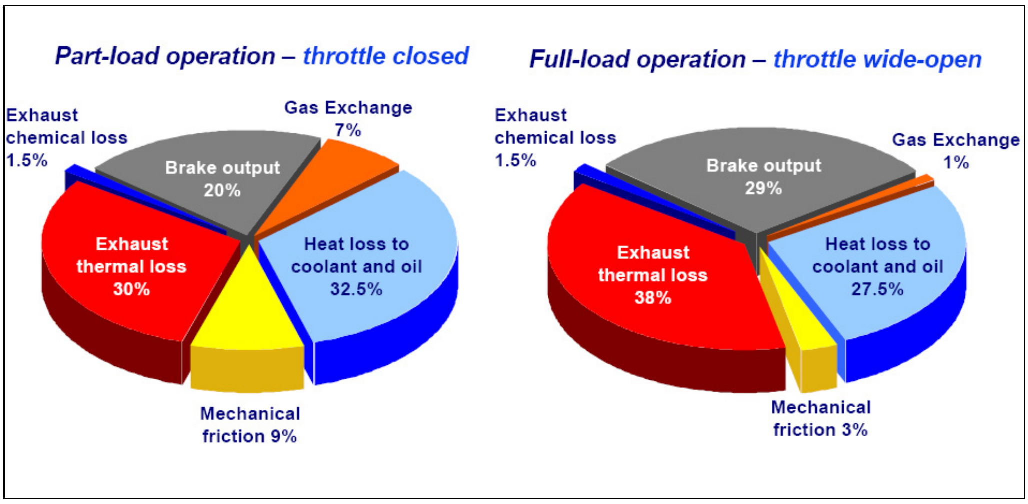
The demonstrated range of low emission combustion seems sufficient, in principle, to cover the range of the light-duty regulated emission tests in Europe, and perhaps also heavy-duty requirements. However, considerable work is still needed before the performance demonstrated in steady state tests can be applied in real vehicle applications: these must be considered as longer-term developments. Short term HCCI developments will need to revert to conventional operation at high load. If not wisely applied, this could mean that real world emissions outside the speeds and loads of the driving cycle could be much higher, leading to air quality problems.

For heavy-duty engines, the normal operating regime is at higher load, so a different strategy may be needed than for light-duty vehicles. Although the trend is for increasing BMEP and smaller engine size even in large engines, the achievement of HCCI operation at loads up to 15-21 bar opens the possibility of full time HCCI operation in those applications where some increase in engine size could be accepted. This may be attractive if HCCI allows a reduction in the complexity of aftertreatment to meet exhaust emissions regulations, and the improvements in efficiency could be sufficient to offset the effects of the higher friction and weight of the larger engine.

3.7. ENGINE EFFICIENCY

A major part of the energy released by combustion of the fuel in an internal combustion engine is lost to the coolant, oil and exhaust, with smaller losses to gas exchanges (pumping losses) and friction. An example for a gasoline engine is shown in **Figure 20**.

Figure 20 Where the energy goes in a gasoline engine [60]



The overall engine efficiency has been subdivided in work by Lund University [67] into four parts.

$$\text{Overall Efficiency} = \eta_c \times \eta_{th} \times \eta_v \times \eta_f, \qquad \text{(Equation 4)}$$

where η_c is combustion efficiency,
η_{th} is thermal efficiency,
η_v is the volumetric or gas exchange efficiency
η_f is the friction efficiency.

Overall efficiency is directly related to BMEP for a given fuel input. Measures of IMEP do not include friction losses, and so are proportional to $\eta_c \times \eta_{th} \times \eta_v$

1. Combustion Efficiency (η_c): Unburned hydrocarbons and carbon monoxide emissions represent an unwanted contribution to air pollution and a loss of fuel energy that has not been captured to produce motive power. For conventional engines, fuel conversion levels are generally high, typically 97-98% in a spark ignition engine, or a little lower for high compression ratios where the crevice volume represents a greater fraction of the combustion space. For HCCI, achieving complete combustion is more difficult, and for today's engine designs, typical figures are in the range 90-92%, and may be even lower at light loads where temperatures are lower and sustaining combustion is more difficult.

2. Thermodynamic Efficiency (η_{th}): This represents the efficiency with which energy released in the combustion chamber is converted into mechanical energy. Typical SI engines have a thermal efficiency of about 30%, with high compression ratio models approaching 40%. The efficiency of HCCI combustion will depend on how successfully the timing of combustion can be controlled, but it could be as high as 50% at higher loads.

   Based on the available studies, fuel consumption under HCCI combustion conditions has the potential to match the fuel consumption of diesel engines and is expected to be much lower than that for conventional spark ignition engines.

3. Gas Exchange (volumetric) Efficiency (η_v): This represents the efficiency with which air and fuel are inducted into the cylinder and exhaust gases are expelled. SI engines, because of their throttling action, suffer pumping losses except at high loads. Efficiency is higher for diesel engines, including HCCI combustion, where there are no throttling losses.

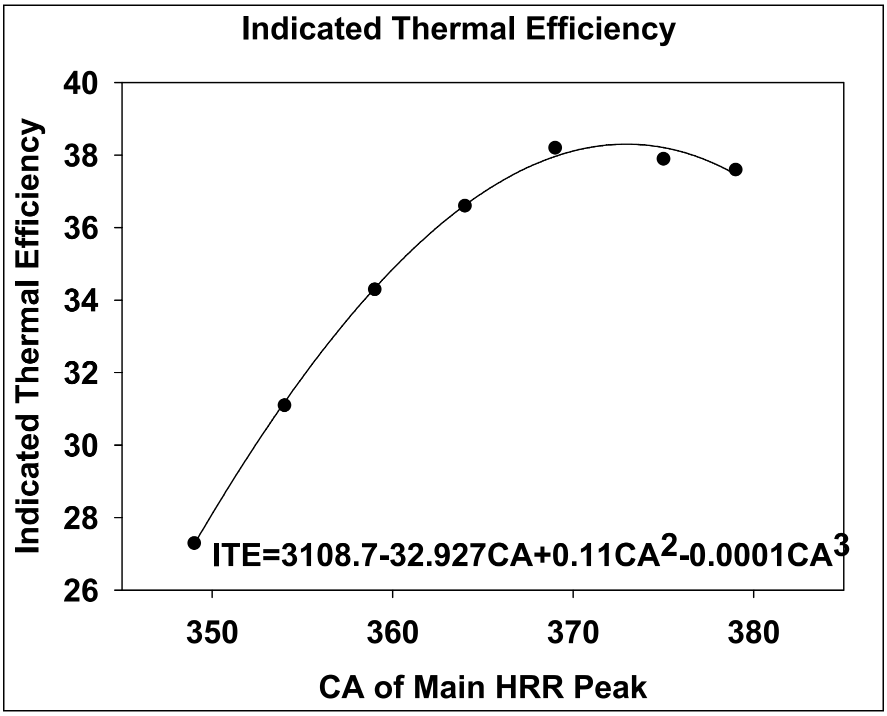
4. Mechanical Efficiency (η_f) = BMEP/IMEP: Mechanical efficiency is the ratio of energy available at the crankshaft to that actually produced in the cylinder. For this reason, mechanical efficiency is related to mechanical friction, which is generally lowest in standard SI engines. Diesel and HCCI engines have a more robust construction in order to handle higher combustion stresses and, for this reason, tend to have somewhat higher friction losses. Additional mechanical complexities in diesel engines, such as variable compression ratio, can further increase friction losses.

In [67], the efficiency of a 2-litre/cylinder HCCI truck engine was compared with that of a 0.5-litre/cylinder HCCI car engine and a small VCR HCCI engine. Overall, the larger engine had the highest brake efficiency, reaching 32.2% at 1000 rpm, but this fell to a level below the 0.5-litre/cyl engine at 2000 rpm. Combustion efficiency for the truck engine was 93-96% at 2 bar BMEP, higher than for the smaller engines which were 90% efficient. This difference is attributed to the larger combustion volume for the larger engine, although earlier combustion timing for the larger engine is also thought to contribute. The thermodynamic efficiency of the three engines was the same at 46%. Heat losses during compression were greater for the smaller engines, but the larger engine's advantage was negated by higher losses during combustion due to the earlier combustion timing. The authors note that retarding combustion in the larger engine would improve thermodynamic efficiency, but worsen combustion efficiency, because of the lower temperatures. It is therefore clear that no single element dominates the overall efficiency. It was concluded, however, that combustion and mechanical efficiency can have a big impact on overall performance at the low loads where HCCI is mainly targeted. In this study, the indicated efficiency achieved in the HCCI load range was much higher than for the standard SI engines, and implied a fuel consumption saving of around 28%, similar to that achieved by diesel engines.

In [16], the effect of high EGR rates was studied, and the overall efficiency of fuel conversion was found to reach a distinct maximum around 14% oxygen content (equivalent to about 55% EGR), with some reduction towards the even higher EGR rates that produce low soot and NOx[18]. Further analysis showed that fuel conversion efficiency deteriorated at the lowest oxygen levels.

Thermodynamic efficiency depends on the timing of combustion. The rapid heat release of HCCI combustion is a positive factor, since the released energy can then operate on the piston over the whole power stroke until the exhaust valve opens. However, if combustion starts before TDC, the initial pressure rise acts to push back on the piston rather than producing positive power. The optimum timing for heat release is therefore after, but close to TDC **(Figure 21)**.

Figure 21 Thermal efficiency is highest when heat release occurs just after TDC [68]



In [68], the optimum timing was found to be when the peak heat release (which corresponds broadly to CA50) is at 12°ATDC. Although rapid heat release was best,

[18] The efficiency also varied strongly with the injection timing; the discussion above relates to the optimum injection timing.

increasing the duration of combustion from 10°CA to 30°CA reduced IMEP and efficiency by only 0.5%.

As long as the timing of combustion can be maintained close to its optimum position, most studies indicate that engine efficiency in HCCI engines can be maintained at levels comparable to conventional diesel engines. However, if combustion timing is moved later at higher loads to increase the ignition delay and maintain low emissions, there is likely to be some efficiency loss, even though low emissions can be maintained [14].

3.8. THE IMPACT OF FUEL PROPERTIES

3.8.1. Background

The characteristics of different fuels have been discussed at various points in the preceding detailed discussion of combustion mechanisms. Here we consider the broader question of how fuels could be adapted to facilitate the effective introduction of advanced combustion technologies.

Vehicle and engine technology is undergoing a period of great change and innovation, driven by the goals of lower emissions and better fuel efficiency. It is therefore worthwhile to reflect on the approach needed to evaluate the future role of fuels in the context of HCCI combustion. In the 1990s, when the US and European Auto-Oil Programmes were conducted, vehicles were evolving in a more evolutionary way, and it was appropriate to carry out programmes that looked at the effects of fuels independently in near-production vehicles, without adapting the vehicles to the fuel. In the case of HCCI combustion, the change in vehicle technology may be considerable, and the fuel needs may be considerably different. In addition, we are considering technologies that in their fully developed form may play a role only 15 years ahead rather than the 5-10 year timeframe considered by the Auto-Oil Programmes. There is time to consider whether a significantly changed fuel is desirable for these vehicles.

There are also uncertainties: while we can be reasonably sure that part-time HCCI combustion will come into the market on vehicles using gasoline or diesel, the eventual success of full time HCCI, that would allow the opportunity for a fuel change, is not clear. Today is a time for research and study rather than jumping to premature conclusions. One reason why clear answers are not available today is the sheer range of engine options being studied. Once part-time HCCI vehicles reach the market in reasonable numbers and prove their effectiveness, more detailed test programmes will be possible to explore incremental changes to gasoline and diesel fuels to enhance their effectiveness. The constraint is that any fuel changes must not compromise the performance of the existing fleet of conventional vehicles.

For full HCCI vehicles, research has clearly shown that the fuel type can have a significant impact on the ability to sustain low emission combustion over a wide operating range. Equally, HCCI engines have demonstrated an ability to operate on a wide range of different fuels. While a particular fuel type might appear in the research laboratory to be optimum, in practice special fuels are best suited to local and fleet use; a requirement for a vehicle to use a special fuel is a barrier to its wide acceptance in the general vehicle fleet. It seems likely therefore, that fuel development will at first be evolutionary, based on the performance of part-time

HCCI vehicles, supported by small scale trials of full HCCI vehicles that will allow a greater understanding of the need for an additional fuel type.

The question of how fuels should be compared in HCCI combustion merits some thought. Practical HCCI vehicles do not yet exist, so there is an excellent opportunity to evaluate fuels as part of the vehicle-fuel system. However, the fact that a wide range of fuels from conventional diesel through to gasoline is being considered, and the big impact the fuel choice can have on optimum engine operating conditions means that comparing widely different fuels at fixed engine calibration will not give meaningful comparisons. In general terms we can say that the performance of the engine should be optimised on each potential fuel, and the overall performance of the optimised engine-fuel systems compared. However the range of performance criteria that must be met and the number of engine parameters that must be optimised make this a complex and potentially subjective process.

3.8.2. Fuel parameters

A Thermodynamic Availability[19] analysis of ICE combustion [4] found that around 20% of the fuel energy available for useful work was consumed in the combustion process, and that this amount was effectively the same for all fuels in the gasoline-diesel boiling range[20]. The conclusion of the study is that changing the fuel composition in the gasoline-diesel boiling range will not affect the combustion efficiency; rather, the importance of fuel is in enabling HCCI combustion.

From the earlier discussion, it is clear that the most important fuel-related parameter influencing HCCI combustion is the ignition delay. This governs both the timing of the combustion event, and the amount of time for mixing. Intuitively, the volatility of the fuel should also be important, since a more volatile fuel will mix more rapidly and so more readily form a near-homogeneous mixture. However, the realisation that completely homogeneous mixtures are perhaps not ideal means that volatility may not play such a large role. Finally, fuel composition, whether of an existing liquid fuel, or as a new, alternative fuel should be considered[21]. Compositional parameters such as aromatics content have been studied for many years because of their perceived role in the combustion process, particularly for formation of soot. However, mixture conditions have much more influence on soot and NOx formation than fuel composition, so the role of fuel composition in the combustion process after ignition is still unclear. The three basic fuel parameters can therefore be clearly ranked in terms of likely importance:

1. Ignition resistance
2. Volatility
3. Compositional factors and alternative fuels

[19] This study investigated 'availability' or 'exergy' which is the second law of thermodynamics' equivalent of enthalpy or energy. Whereas energy is conserved, availability is not; it represents the ability of the energy to do useful work, and is related to the entropy production during the combustion process.

[20] Figures presented are for λ=1, availability loss increases for leaner mixtures, but the availability retained in the cylinder is increased, exhaust gas availability decreases. The impact of this depends on whether energy is recovered from the exhaust, for example by turbo-charging. Gaseous fuels show a small benefit compared with liquid fuels: at λ=1, the combustion irreversibility for methane is about 18%, for hydrogen about 12%.

[21] Fuel composition clearly affects the fuel's ignition behaviour, however this is covered separately; here we are looking for any possible additional effects.

3.8.3. Ignition resistance

In today's compression ignition (diesel) engines, ignition is controlled by the start of fuel injection, using a fuel with high cetane number that has a short ignition delay. SI engines, by contrast, are designed to avoid auto-ignition, using a fuel with high resistance to auto-ignition (low cetane number, high octane number). Under normal SI engine conditions this fuel will not auto-ignite. The two requirements for a successful HCCI engine are control of the ignition event, and adequate mixture preparation. This means that a relatively long delay is needed after fuel injection, to allow sufficient mixing before ignition takes place, so the objective lies somewhere between conventional gasoline and diesel engine behaviour.

Ideally [12,17,23,57,58], fuel injection should be completed before combustion starts, to avoid formation of soot. This becomes more challenging at higher loads, both because the ignition delay tends to decrease with increasing temperature, and because the time required to inject the increased fuel quantity increases. The influence of engine parameters such as EGR and fuel injection pressure has been discussed earlier. Here we consider fuel properties in more detail.

Ignition quality of motor fuels is assessed in engine tests where the ignition delay is directly measured[22]. The standard CFR test engine dates from the 1930s, but in spite of its age fuel ratings based on this method have been found to give surprisingly good correlation with ignition performance in more modern engines. For diesel fuel, a single cetane number scale is used; for gasoline, the variation of knock resistance with engine conditions has long been recognised by the use of two octane numbers. Research Octane Number (RON) is generally considered indicative of light load conditions, while the Motor Octane Number (MON) test is performed at more severe conditions to address high speed knock.

Diesel fuels with cetane numbers in the range 32-55 were tested under HCCI combustion conditions in a single cylinder HD engine [42]. The ignition delay was found to vary according to the cetane number, and there was no additional influence of compositional effects. This finding is consistent with experience in conventional engines where cetane number predicts ignition delay even at conditions differing from those in the CFR engine.

For gasoline fuels, octane number is known to vary with fuel composition. In conventional gasoline, the MON is typically 10 numbers below the RON. This difference is termed the 'sensitivity' of the gasoline. By definition, paraffins are considered to have zero sensitivity; olefins and aromatics have higher sensitivity. These two indices have proved very successful over the years in regulating gasoline ignition quality; in Europe, specifications have been set in terms of both RON and MON, in the USA an index based on (RON+MON)/2 has been used – considered to be what the vehicle sees on the road. A number of studies by Shell and KTH have

[22] The standard reference engine is a single cylinder 'CFR' engine. In practice, other methods are used to estimate octane and cetane numbers, because they are easier to apply in production situations. The Ignition Quality Tester (IQT) measures ignition delay in a fixed chamber, and is now allowed as an alternative to the CFR engine for cetane number measurements. Empirical indices based on more easily measured parameters (density, distillation) are also used to predict diesel cetane number, gasoline composition can be measured through GC analysis and octane number built up from individual components, and empirical relationships based on infrared analysis are also available. The objective of these empirical methods is to <u>predict</u> the ignition delay in the CFR engine, which remains the primary reference.

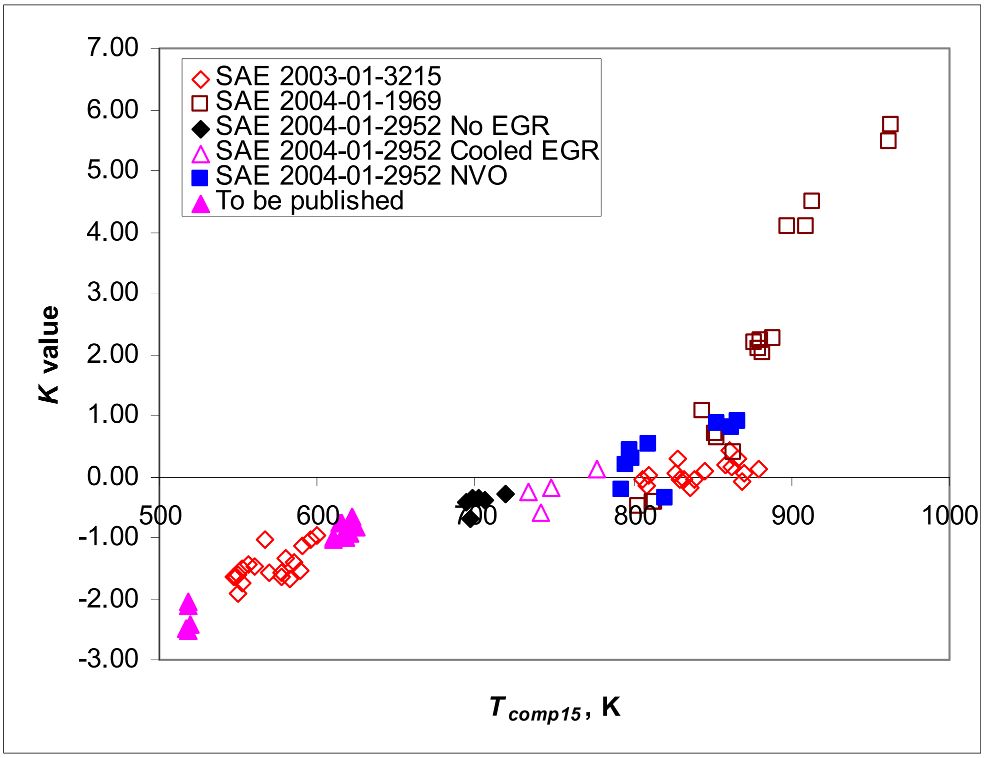
investigated this in depth [43,44,69,70,71], and a more general equation developed to describe an engine's octane needs

Octane Index = (1-k) x (RON) + (k) x (MON) (Equation 5)

where k is a constant depending on engine conditions

The (RON+MON)/2 equation is equivalent to a k value of 0.5; if k=0 then RON is the only influence; if k=1 then MON is the only influence. As vehicles have been adapted to lower emissions and fuel consumption, there has been a change in vehicle requirements, with RON appearing much more important in modern conventional gasoline vehicles than MON. In modern conventional engines the value of k varied with operating conditions, and in some cases was even negative, so that a fuel with lower MON had better ignition resistance than a higher MON fuel [72]. Similar studies have been conducted on gasoline fuels running under HCCI conditions [41,45]. The value of k can vary over a wide range (from -3 to +6) depending on the temperature in the combustion chamber before ignition; differences between fuels also become more pronounced as pressure increases [41]. The implication is that MON becomes less important at the cooler combustion conditions that may be expected in HCCI engines **(Figure 22)**.

Figure 22 MON is less important at cooler combustion temperatures [45]

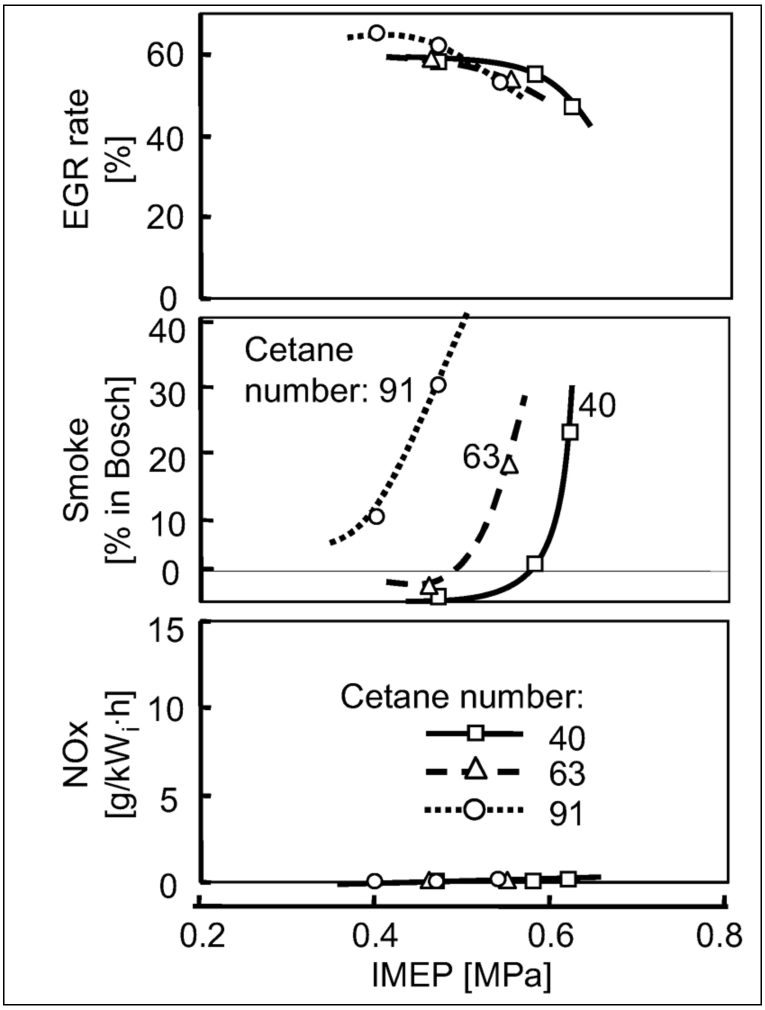


A recent study [73] tested gasoline fuels of widely different composition under HCCI conditions but with both RON and MON maintained essentially the same. They found that at lower load conditions all the fuels could sustain HCCI combustion, while, at higher loads, HCCI combustion was not possible with any of them. At an intermediate condition (1200 rpm, 2 bar), fuel differences did emerge with only three of the fuels able to sustain HCCI combustion. At fixed injection timing, there were variations in ignition delays between the fuels, with paraffinic fuels igniting earlier, followed by aromatic and olefin containing fuels. The same study concluded that efficiency and emission performance was similar for all the fuels when the combustion timing was fixed, confirming ignition delay as the most important fuel parameter.

It therefore appears that the traditional cetane number measurement gives a good prediction of the ignition behaviour of fuels in the diesel boiling range, while for gasoline fuels the situation is more complex: the relative contributions of RON and MON depend on the operating conditions, and there may be residual compositional effects on ignition delay that are not described by RON and MON and need further study.

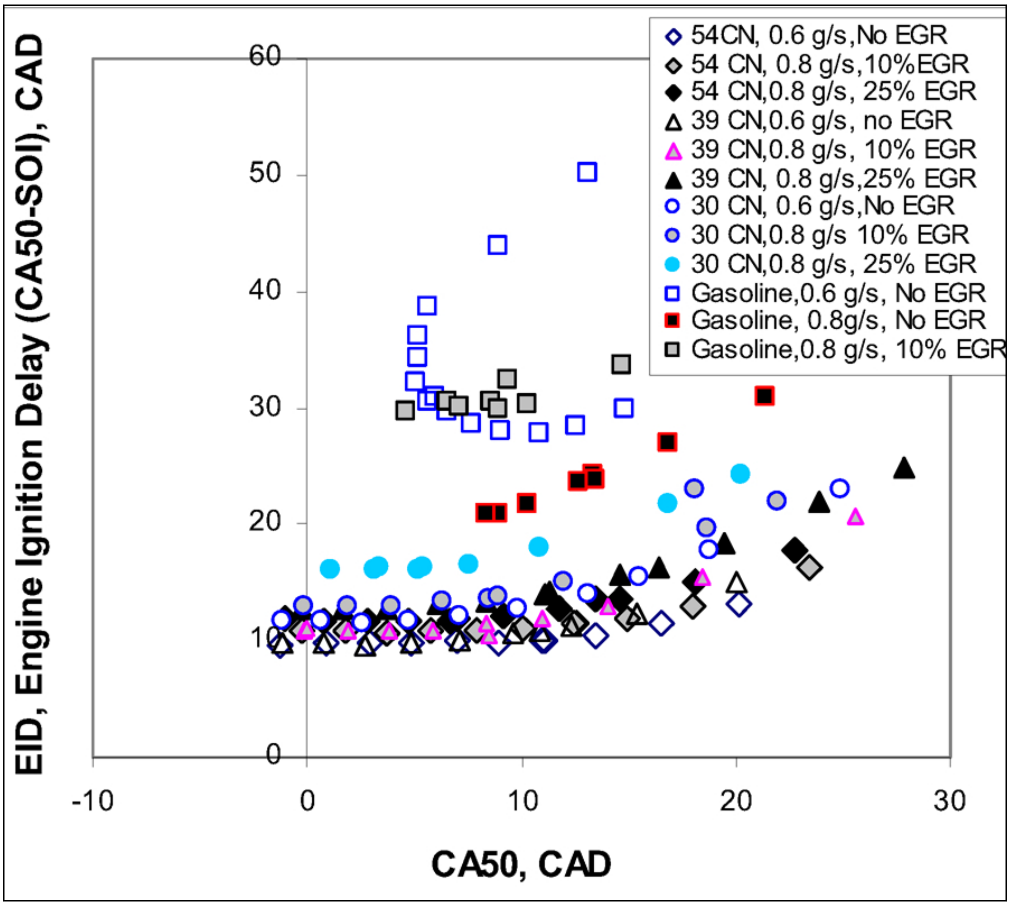
The ignition delay of the fuel may prove to be an important enabler if full HCCI combustion is to be achieved. The role of high levels of EGR in lowering combustion temperatures and lengthening ignition delay has already been discussed. A study using diesel-like fuels with different cetane numbers [37] found that the longer ignition delay of a 40 CN fuel allowed more time for mixing, resulting in more premixed burning, while higher CN fuels produced combustion closer to conventional diesel behaviour. The IMEP limit for low emission combustion was higher for the 40 CN fuels compared with 63 and 91 CN fuels. The 63 CN fuel needed 45% EGR to achieve the same ignition delay as the 40 CN fuel achieved with 25% EGR. Although HCCI combustion is possible with high CN values, the high cetane uses up part of the available EGR resource **(Figure 23)**.

Figure 23 Lower Cetane Number increases the area of low emission operation [37]



Tests using a wider range of fuels were conducted in [17] and showed that gasoline gave much longer ignition delays than even a diesel fuel of 30 CN (**Figure 24**). In fact, this conventional gasoline (with a RON of 95 and estimated CN of -15) proved too resistant to ignition under some conditions, indicating that some intermediate fuel, for example a lower octane gasoline, might be better suited.

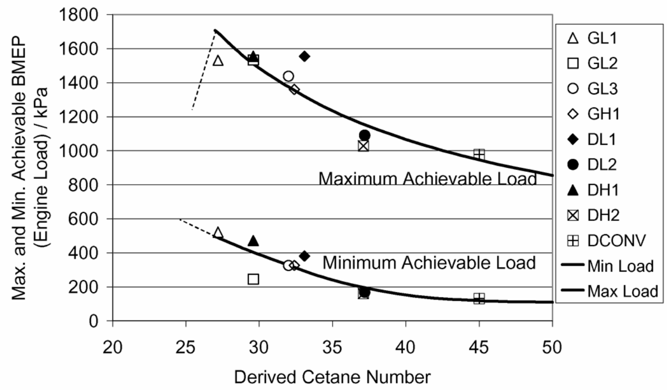
Figure 24 Compression Ignition can be achieved with very low CN fuels [17]



Similar results have been found in other studies. Tests on fuels of 19 CN and 61 CN showed that the lower cetane fuel increases the mixing time and allowed lower smoke levels and NOx levels to be achieved, even without EGR and using late injection [72]. In [38] results close to 100% engine load were shown for HCCI combustion using both gasoline and diesel fuels, however it was noted that the compression ratio of the engine should be matched to the fuel ignition quality, with a lower compression ratio being needed to achieve 100% load with conventional US diesel fuel. The same researchers noted that at fixed compression ratio, increasing CN produced an undesirable advance of combustion timing [74]. Blends of gasoline and diesel have been evaluated in some studies to obtain intermediate properties between the two fuels and as a way of understanding the combustion process [68,39]. However, full time HCCI operation seems more likely to be achieved with gasoline-like rather than diesel-like fuels (i.e. with lower cetane numbers) [15,17,45,68]. The same researchers recommend that where the engine needs to revert to diesel operation at high load the ignition properties of the fuel should be as much like gasoline as possible consistent with maintaining high load diesel operation.

Similar conclusions were reached in another extensive series of studies [18,27,38,40], where lowering the fuel cetane number increased the load range under which HCCI combustion could be maintained. The boiling range of the fuel did not affect the range of operation, and similar results were achieved with fuels covering both the gasoline and diesel boiling ranges. Ignition resistance was confirmed as the most important parameter governing the engine operating range. Large changes in volatility (from diesel to gasoline) and fuel composition (up to 65% aromatics) had little effect on the operating range **(Figure 25)**.

Figure 25 HCCI operating range not affected by fuel volatility or composition [40]



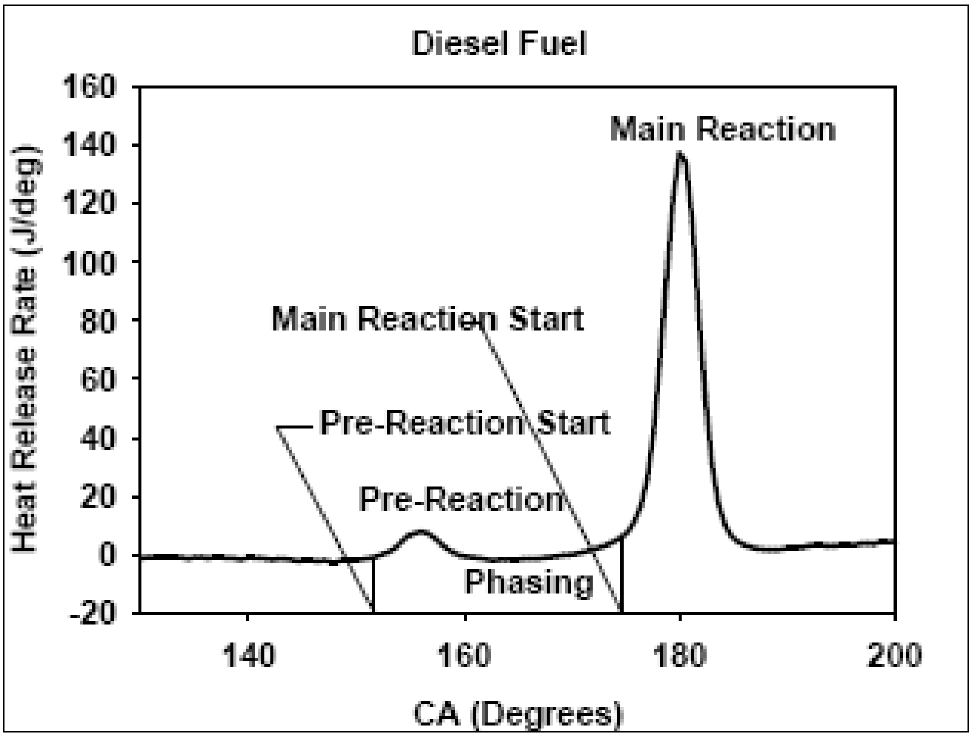
GL1	Gasoline 80MON, 13% aromatics
GL1	Gasoline 73MON, 13% aromatics
GL3	Gasoline 65MON, 13% aromatics
GH1	Gasoline 60MON, 36% aromatics
DL1	Light Diesel, 33CN, 14% aromatics
DL2	Light Diesel, 37CN, 20% aromatics
DH1	Heavy Diesel, 30CN, 65% aromatics
DH2	Heavy Diesel, 37CN, 53% aromatics
Dconv	US diesel, 45CN, 35% aromatics

In these tests, the engine calibration (injection timing, EGR) was optimised for each fuel individually. Fuels with a cetane number below about 35 gave the widest operating range, regardless of whether they were in the gasoline or diesel boiling range. NOx emissions were low for all the results reported. PM emissions were higher for diesel fuels at high load.

Recent studies by VW [65,75,76] advocate a low aromatic fuel of about 45 CN in the kerosene boiling range as a suitable fuel for HCCI. Results are presented for both conventional diesel combustion and for HCCI combustion, and show a dramatically improved soot-NOx trade-off for the proposed fuel, with the fuel comparison made at constant combustion timing.

The chemistry of auto-ignition is complex, and generally includes a low temperature cool flame reaction some time ahead of the main heat release. The Low Temperature Heat Release (LTHR) may be detectable from in-cylinder measurements, but even if it is not, the radicals formed can play an important role in initiating the main combustion. More detailed study is enabled by the use of Laser Induced Fluorescence (LIF) where optical test engines are available. The level of formaldehyde (CH_2O) is often used as a signature for LTHR, whereas the presence of OH signals the onset of the main heat release [33,77].

Figure 26 An example of Low Temperature Heat Release [15]



LTHR is important because it defines the start of reaction temperature and influences the combustion timing [15] **(Figure 26)**. In [77] the proportion of total heat release occurring in the LTHR phase varied from 2-6%, being significantly higher for diesel compared with gasoline. In [19], low NOx was more easily achieved with higher CN fuels that produced LTHR, but at the cost of higher CO emissions[23]. In [77], adding 20% of gasoline to diesel had a big impact, increasing the delay time to almost the same level as pure gasoline. Cool flame behaviour depends on the composition of the fuel; in the gasoline boiling range, paraffins produce cool flames while aromatics exhibit virtually no low temperature reaction. Other fuels, such as methane, may exhibit no LTHR, and combustion takes place through a single step process [77]. In a study on diesel fuels, higher CN fuels (48-76) gave significant LTHR, while lower CN fuels (19-34) gave none. The mechanisms of LTHR are discussed in [73]: the kinetics are related to chemical reactions of low octane fuel components at temperatures around 700-850K[24]. The heat released affects how quickly the mixture reaches the ignition temperature. They also note that if conditions in the early compression stroke are sufficiently hot, reaction can take place early, and suppress some of the LTHR closer to TDC. LTHR therefore depends both on the fuel properties and on the engine conditions. Whether higher levels of LTHR are beneficial seems uncertain; in [15] it is suggested that less LTHR is better.

In summary, many researchers have found that lowering cetane number below the levels found in today's diesel fuel could help control of HCCI engines. Very low

[23] Soot emissions were not reported in this study, also the engine was not equipped with EGR, but varied combustion timing through intake temperature, so equivalence ratio varied to some extent between the test points.

[24] [68] and [73] include references to several earlier technical papers on LTHR.

cetane fuels, such as gasoline could prove to be good HCCI fuels, although if there is an optimum level of ignition quality, it is likely to lie somewhere between gasoline and diesel [40,45,46].

3.8.4. Volatility of liquid fuels

Since time is needed for fuel and air to mix before ignition takes place, the volatility of the fuel might be expected to play a part. More volatile fuels like gasoline (or even gaseous fuels like NG) diffuse and mix much more rapidly than heavier fuels such as diesel. However, a completely homogeneous mixture presents problems of combustion control, and less complete mixing may prove to be a more practical approach.

SWRI [15] have reported that fuels in the gasoline boiling range reduce mixture preparation problems, which is consistent with their goal that all the fuel should be vaporised before combustion starts. In combustion where complete homogeneity is not sought, volatility seems to play a smaller part. HCCI operation has been demonstrated up to 100% load on both gasoline and diesel fuels provided the engine is matched to the ignition quality of the fuel [38]. However, increased volatility in the diesel fuel range generally reduced emissions [74]. In [37], although ignition quality effects were seen, no effect was seen for T50 variation in the range 187-291°C. In other studies where good performance was obtained with gasoline fuels, the benefits of improved mixing were attributed to ignition delay rather than volatility [17]. Even diesel fuels can give good mixture preparation if used in the high pressure injection systems common in today's diesel engines (see **Figure 19**), so the importance of volatility for future engines is unclear. It is certainly possible to demonstrate HCCI combustion using a wide range of fuels of different volatilities, and ignition behaviour seems to be much more important than volatility. Perhaps the most dramatic demonstration that ignition resistance is the dominating fuel parameter is illustrated in **Figure 25** above [40]. Low NOx and PM emissions were maintained for all the results presented, and the engines operating range depended only on the ignition resistance of the fuel, even when volatility varied from gasolines to diesel fuels including one with a very high density of 0.91kg/L. Nevertheless, some benefits of more volatile fuel have been seen, so this question should be kept under review.

3.8.5. Fuel composition

It has already been mentioned that for gasoline fuels, RON and MON do not give a complete description of the ignition delay behaviour under HCCI combustion conditions, and differences in ignition delay have been seen for fuels of different composition at nominally similar octane numbers [73]. This finding may reflect a limitation of the current octane measurement methods rather than an impact of fuel composition on combustion, and a need for greater understanding of the ignition behaviour of gasoline-like fuels. The question to be addressed in this section is whether fuel composition plays a role in fuels which have the same ignition delay and combustion phasing under HCCI conditions.

Aromatics content has long been a subject of investigation in conventional engines, both gasoline and diesel. Early diesel studies were subject to intercorrelations in the fuel matrices where aromatics effects were confused with cetane and density effects. Developments in engine technology have also changed the way in which engines respond to fuel changes. Later studies have clarified these relationships, and the effects of fuel changes have been shown to be real, but generally much smaller than engine effects [59,78,79,80]. In [74], fuels with 28.7% and 44.7% aromatics were compared, and differences in emissions found to be small. In [65,75], benefits were seen for a kerosene-type fuel with very low level of aromatics. However, in [81] a higher aromatic kerosene fuel gave better performance than Swedish Class 1 diesel in a heavy-duty engine under conditions approaching HCCI combustion. The two extensive series of tests already cited indicate that the presence of aromatics does not influence the ability to sustain HCCI combustion. In [45], a gasoline with 29% aromatics was able to operate with low emissions up to an IMEP of 15.95 bar, while the same engine operated on a Swedish Class 1 diesel fuel with 3% aromatics could operate only up to 6.5 bar. Further studies reported in [40] allow the effects of aromatics to be distinguished from the dominating effect of ignition resistance. In **Figure 25** above, the engine operating limit was determined under constraints of maximum 0.27g/kWh NOx and 0.1FSN. The operating envelope was entirely described by the ignition resistance of the fuel. Very wide compositional changes including gasolines with 12.7-35.6% aromatics and diesel fuels with 13.9-65.4% aromatics had no effect on the operational range of the engine, although the gasoline fuels gave lower PM emissions at high load.

Aromatics have been shown in flame studies to have a higher soot formation propensity than paraffins. However, the effect on diesel tailpipe emissions is much less, because 99.5-99.9% of the soot is oxidised before leaving the combustion chamber [55,82]. In [82], although in-cylinder soot formation reduced by half when aromatics were reduced from 25% to near zero, the tailpipe emissions were virtually unchanged. As illustrated by the HCCI work described above, mixture conditions have much greater impact on soot formation. Consider, for example, combustion in homogeneous spark ignition engines where soot production is very low in spite of using fuels containing up to 35% aromatics.

Studies on LTHR show some variations with fuel composition. Three fuels of the same RON and different MON were tested [68] and found to differ in the phasing of the main heat release. The authors concluded that fuel composition affects the timing and magnitude of LTHR. When paraffins were added to a test fuel, the cool flame was advanced, but when toluene was added there was a reduction in cool flame activity and a delay in the main heat release; aromatics were found to have almost no cool flame. It is noted in [83] that although cetane number changes in the range 19-76 affected the amount and timing of LTHR, stable combustion, good efficiency and low emissions could be achieved on all of these fuels. These effects have been further studied in [71]. Tests in a Rapid Compression Machine showed that fuels containing the structure $–CH_2-CH_2-CH_2-$ produced two-stage ignition (LTHR) with relatively short ignition delays, and that for these fuels the ignition delay correlated with the pressure increase during the LTHR. The burn rate was highest for those fuels that produced short ignition delays, although there was some evidence of compositional effects, with fuels containing cyclic structures having lower burn rates. Ignition improver additives (2EHN and DTBP) were shown to influence ignition delay through the same mechanism of promoting LTHR through formation of active radicles.

The main effect of fuel composition seems to be the propensity of paraffins to produce LTHR, and of aromatics to exhibit single stage combustion. There is no clear consensus at present on whether LTHR is a desirable feature, and HCCI combustion has been demonstrated with and without its presence. Similarly, HCCI combustion has been achieved with a wide range of gasoline/diesel fuels and the presence of aromatics does not seem to be a barrier to its implementation. This question needs to be kept under review as the technology develops.

3.8.6. Alternative fuels

Research using dual-fuelling was described earlier. Although this approach can give very good control over combustion timing, the success achieved in establishing HCCI combustion with more easily available fuels makes it unlikely that this approach will be pursued in the foreseeable future. Natural Gas is a globally abundant resource, and although its primary use will be in stationary applications, it is already used to some extent as a vehicle fuel, particularly where local availability makes it attractive. In Europe, the small greenhouse gas savings achievable from NG do not justify its widespread use in transport [2,84], and it seems likely to be used most effectively in fleets, where its unique properties can be exploited while keeping infrastructure costs at a reasonable level. NG has a high octane number[25], and exhibits no LTHR [48,51,68], so it is very resistant to ignition. High inlet temperatures and dilution with EGR or air are needed to establish HCCI combustion, and even so it is difficult to avoid high rates of heat release. The use of on-board reforming to facilitate NG combustion was studied in a CFR engine in [48,51]. They found that the operating range of the engine running on NG alone was very limited: NOx emissions were high, efficiency was low, and there was high cycle variability. Although the engine design may play a part, it appears that achieving HCCI combustion with NG is very difficult. Use of Reformer Gas to initiate combustion improved performance; it is likely that some such approach will be needed if NG is to be used as an HCCI fuel in the future.

The other alternative fuels currently entering the market are biofuels, particularly ethanol and biodiesel (FAME)[26]. These could potentially be available as separate fuels if dual-fuelling concepts proved viable, however they will not be available in sufficient volumes to completely substitute gasoline and diesel, so their most likely use will be as blend components [2]. It is therefore of particular interest to know if addition of ethanol to gasoline or FAME to diesel impacts on HCCI combustion. Ethanol was tested in an HCCI study using an IDI diesel engine [47]. Pure ethanol was found to be easier to ignite than iso-octane, but it performed less well at high engine speeds. Blends of 5% FAME in diesel fuel were included in tests on a 4 cylinder engine [85]. Heat release profiles and start of combustion were similar to fuels without FAME addition. Addition of FAME appeared to give a slight increase in PM emissions, although the effect was small compared with the benefits of moving to HCCI combustion. These results are encouraging and suggest that incorporation of biofuels into gasoline and diesel will not cause significant problems for HCCI engines, however more study will be needed as HCCI engines near production status.

3.9. PRACTICAL ASPECTS

3.9.1. Vehicle complexity and cost

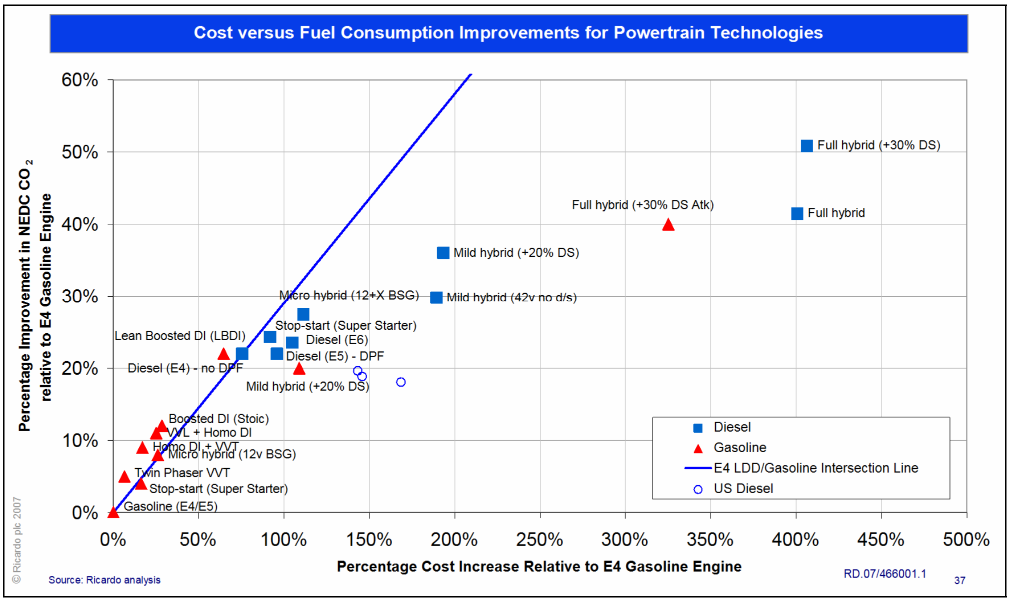
Although HCCI combustion has been demonstrated on the test bench, much work remains to develop these ideas into practical production vehicles. On a purely technical level, challenges remain to extend the operating range and manage transient conditions, but as developments come closer to production, cost will also

[25] The actual value varies with composition, however in Europe, pipeline NG is estimated to assure an MON of at least 115 [2].

[26] Advanced fuels from biomass produced by hydrogenation or gasification to FT-diesel are pure hydrocarbons and for this discussion can be considered broadly similar to conventional diesel fuels.

become more important. Most of the technologies needed for HCCI engines already exist today, but in some cases cost could be a significant barrier to their introduction. For example, Ricardo [60] has compared the performance of technologies to improve fuel consumption with the cost **(Figure 27)**.

Figure 27 Engine efficiency improvements need to be balanced against cost [60]

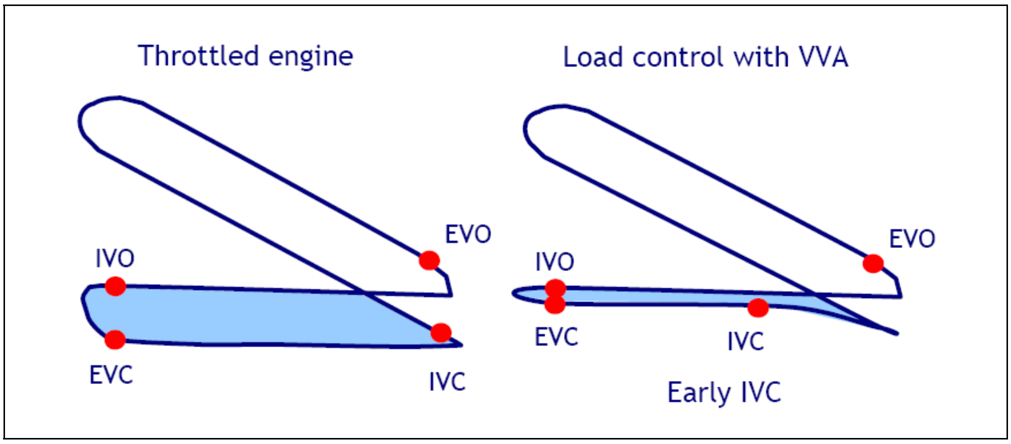


The baseline for comparison is the MPI gasoline powertrain, which provides the lowest cost propulsion system today and provides low exhaust emissions, but at the cost of relatively high fuel consumption. Diesel engines consume about 25% less fuel by mass, which brings a similar reduction in CO_2 emissions. Because of the higher density of diesel fuel, the volumetric fuel consumption reduces by rather more. However, these benefits come at a cost, and a typical light-duty diesel powertrain costs about 75% more than a simple MPI system, due to the need for a more robust engine construction, and particularly the cost of the high pressure fuel injection system. The cost of the diesel engine is expected to prevent its adoption in the smallest vehicles, where the lower cost of the MPI engine gives it a decisive advantage. In addition, more efficient gasoline engines could provide strong competition for diesel and start to displace it in those sectors where usable performance is not a primary consideration. For all new engine technologies, including HCCI, the performance of the diesel engine provides a reference point. Although the example shown relates to fuel consumption, similar considerations apply for technologies to reduce emissions from diesel engines. Advanced aftertreatment such a PM traps and NOx catalysts will further add to the cost of diesel powertrains.

Variable valve actuation is a key technology for HCCI engines. Mechanical cam-phaser systems have been in limited production for several years, and represent the first step towards more flexible valve control. BMW's Valvetronic system is currently the only production system that allows gasoline engine operation without a throttle. By allowing control over the valve lift as well as the timing of valve operation, the amount of charge entering the engine can be controlled, and throttling losses are avoided. The benefits of this can be shown using an engine indicator diagram

(Figure 28). The shaded blue area represents the energy lost in pumping gases into and out of the engine, and is much reduced in throttle-less operation.[27]

Figure 28 An indicator diagram showing the effect of throttling losses [60]



Current VVA systems rely on modulation of the conventional cam-operated valve system, however alternative approaches have been suggested. Electro-mechanical systems use electromagnets to open and close the valves, while electro-hydraulic systems take advantage of the elasticity of hydraulic fluids to provide the high pressures needed to rapidly open and close the valves. These systems provide great flexibility, but their cost has so far prevented their adoption in the market.

One application of VVA that has already been mentioned is the ability to generate internal EGR, either through trapping residual gases in the cylinder, or by allowing gas to flow back from the exhaust during the intake stroke. Such an approach could avoid the complexity of the conventional external EGR system and so offset the cost of VVA. However, one drawback is that retained exhaust gases are hot. This may be an advantage in gasoline HCCI to stimulate auto-ignition, but in diesel applications cooled EGR, which requires an external circuit, may be more effective where lowering the combustion temperature is the goal.

Fuel injection systems are already well developed, and systems exist to meet the demands of HCCI engines. Because of the high pressures involved, diesel injection is the most challenging, but common rail systems are already available up to 1600-1800 bar, and developments of both unit injectors and common rail systems have the potential to reach 2250 bar or higher. Such high pressures, and the desire for multiple injections of very small fuel volumes stretch technology to the limit, and considerable attention is also paid to the fuel injectors themselves.

3.9.2. Control systems

Most HCCI researchers envisage some form of closed loop control system based on combustion sensing, so that the timing of heat release can be accurately controlled. The algorithms needed to operate closed-loop systems have received considerable attention, however the limiting factor is likely to be the availability of a reliable and cost-effective feedback sensor. The requirements for a closed-loop system have

[27] The vertical axis is cylinder pressure, while the horizontal axis is the crank angle position as the engine cycles through 720° on a 4-stroke cycle.

been reviewed in [86]. Because the ignition process is very sensitive, it is very difficult to control it using a pre-determined engine map. Perhaps more critically, the operating point may be unstable, so that without active control HCCI combustion may not be maintained. A number of engine parameters can be used to provide the right conditions for HCCI combustion including intake temperature, compression ratio, valve timing and EGR. Closed-loop control of injection timing could provide a way of regulating the timing of the combustion event. Some researchers have also proposed dual-fuel approaches where the ignition quality of the fuel is controlled cycle by cycle. More challenging is providing the feedback control signal. Measurement of engine speed or torque variations are possible, but may not be sensitive enough to provide the control needed, and most effort is focussed on direct monitoring of events in the engine cylinder.

Cylinder pressure sensors have been used in research engines for many years, and can be used to monitor the timing and progression of ignition and heat release. Over the years, sensors have become increasingly reliable, to the extent that several companies are now developing systems for production vehicles [60,61]. These systems are claimed to be technically close to production readiness, but cost remains a critical issue that will influence vehicle manufacturers' plans. Because of its commercial sensitivity, precise information on production plans is not available. An alternative approach is to monitor the conductivity of the cylinder gases using an ion current sensor. In [53] it is shown that ion current correlates with combustion, and it is proposed that such systems could provide a cheaper alternative to pressure sensors.

3.9.3. Deposit control

Smaller injector holes allow a finer fuel spray, helping mixing and atomization of the fuel. However, accurately machining such small holes is difficult, and two injectors that are nominally the same may perform differently [27]. The surface finish of the nozzle and the shape of the entry also affect the fuel flow and the presence of cavitation. All of these factors can affect the spray formation, and considerable work is taking place to provide ever finer production tolerances.

Since such fine tolerances are needed for engine components including the fuel injectors, control of deposits will be increasingly important. For conventional engines, deposits have always been present, and have been controlled to a large extent by detergent additives in the fuel. DI diesel engines have traditionally been insensitive to the small level of deposits inside the nozzle holes. As nozzle hole sizes reduce, however, even a very thin layer of deposits can have an effect, and the effect has been sufficient to cause power loss in some cases. In the context of HCCI combustion, deposits on the combustion chamber walls have also been studied [87]. It was found that as deposits build up in an engine, the peak heat release increased by 50%, and that the effect was due to reduced heat loss through the chamber walls as deposits increased. Variation in the level of deposits present would clearly present an additional challenge for the engine control system.

3.9.4. Cold starting

For diesel based HCCI systems, the advantages of lower cetane fuels have been described above. For conventional diesel engines, cetane number is most important under light load and cold starting conditions: at higher loads the higher combustion chamber temperatures are sufficient to maintain short ignition delays even with lower cetane fuels. For HCCI systems, many of the concepts proposed would start and warm up the engine using conventional combustion, so these engines face the same challenges as conventional engines. Starting a diesel engine can be readily accomplished, even at low temperatures, using starting aids such as glow plugs or intake heaters which are well developed. Normally, the heating devices switch off when the engine is running, and maintaining smooth combustion immediately after starting and until the engine warms up may prove to be the more difficult challenge.

3.9.5. Lubricity

According to their design, fuel injection systems rely either partly or entirely on fuel to provide lubrication for the high pressure moving parts, with diesel systems being the most critical, because of the higher pressures involved. Historically, the natural lubricity in diesel fuel provided good protection, but when fuel sulphur levels were reduced during the 1990s, it was found that lubricity could deteriorate if measures were not taken to prevent fuel pump wear. A detailed study of diesel fuel lubricity is presented in [88]. General engine lubrication relies heavily on hydrodynamic lubrication, but it was found that for fuel injection systems the role of hydrodynamic lubrication and fuel viscosity is very minor. Instead, because of the high pressures and often low component speeds, wear performance is governed by the surface lubricity properties of the fuel. The loss of lubricity as sulphur levels reduced was attributed not to the sulphur compounds themselves, but to other trace species naturally occurring in the fuel that were removed along with the sulphur in the hydrotreating process. The active components are believed to be hetero-cycle compounds containing nitrogen or oxygen that are effective at very low ppm levels to provide protection for the metal surfaces.

Although it is not possible to replace the natural species that are removed during hydro-desulphurization, fuel additives have been developed that effectively restore lubricity. A specification for lubricity has been introduced into the CEN EN590 diesel fuel specification, and this has proved effective to avoid problems of wear in the general vehicle population. So far, this approach using the HFRR test has been applied to typical diesel fuels. If fuels in the gasoline boiling range are to be used in HCCI engines, its continued effectiveness will need to be verified.

4. DISCUSSION

4.1. GASOLINE- VERSUS DIESEL-BASED APPROACHES

Advanced combustion can in principle be approached starting from either a spark ignition or a diesel engine. To date, the most successful systems are those that provide good control over intake conditions through the application of turbocharging and EGR, and good mixture preparation through high pressure fuel injection. Because of the difficulties of controlling ignition timing, there is currently less emphasis on providing a completely homogeneous mixture, so very early injection is not a prerequisite, and in-cylinder injection, even at fairly late timings, seems suitable.

These requirements of high fuel injection pressure, large amounts of external EGR and high turbocharger boost are not readily available on gasoline engines, and approaches based on diesel engines seem more likely to succeed, even though the fuel they eventually use may be more like gasoline. The diesel approach is more appropriately labelled ‘Pre-mixed Charge Compression Ignition’ (PCCI) since the mixture formed is not purely homogeneous. Compared to gasoline-engine HCCI, higher loads can be achieved because high fuel injection pressures can create the proper degree of air/fuel inhomogeneity and large amounts of external EGR help slow down combustion. EGR also reduces combustion temperatures to produce low NOx emissions.

For heavy-duty applications, engines based on diesel technology seem certain to remain the most successful, provided the durability and reliability of current engines can be matched. Since duty cycles for these vehicles include a large proportion of high load operation, part-time HCCI engines that can provide HCCI combustion only up to moderate power do not seem attractive. However the achievement of HCCI combustion up to 15-21 bar IMEP offers promise if this performance can be maintained in real engines including transient conditions. Such power levels could cover normal engine duty in many applications provided that the engine size is chosen appropriately. This would mean some increase in engine capacity in many cases. This is feasible, but runs counter to current trends for higher power density and smaller engines. The alternative to HCCI is conventional diesel combustion with aftertreatment for PM and NOx. HCCI could provide a way to reduce, if not avoid, the high cost of these sophisticated aftertreatment systems.

For light-duty applications, part time HCCI engines may be a viable option, depending of the load range they can achieve. An IMEP level of 15 bar would cover all the European NEDC cycle plus some higher load operation. However, unless it can be shown that the benefits of HCCI are delivered in real driving conditions, there is a risk of a backlash and accusations of cycle beating. The higher load drive cycles used in the USA, while representing a greater challenge, provide some protection from this dilemma. Light-duty diesel applications of HCCI offer the same benefits as for HD vehicles. Their success will depend on their performance and cost relative to conventional engine with sophisticated aftertreatment.

Approaches based on gasoline engines have lower cost than diesel-based systems, but are limited to relatively low peak loads. Without high-pressure fuel injection, external EGR and turbocharging it is very difficult to control pre-mixed combustion as engine load increases. However, the HCCI benefits of improved fuel consumption are greatest at lower loads and so have the potential to have a big impact in city driving. Again, the success of these approaches will depend on their cost and performance relative to alternative options, particularly whether aftertreatment for NOx control can be avoided. Many of the alternative engine strategies being developed by manufacturers will introduce features such as boosting and flexible valve control that make inclusion of HCCI over part of the operating range a relatively easy task. However, engines for the smallest and most cost sensitive sector of the car market will be constrained to low cost solutions, and this may limit the range of feasible options.

4.1.1. Status of development for HCCI systems

Much of the HCCI literature describes studies on research engines, mainly using steady state testing. Even where successful low emission combustion has been achieved at high loads, authors warn that significant further work is needed to develop these ideas to production readiness. The rapidly changing speed and load conditions in vehicle operation present a particular challenge, and sophisticated control systems will be needed to address this.

While considerable work is reported in the literature on control strategies, many studies propose that closed loop control based on combustion sensors will be needed to benefit fully from HCCI. Several major suppliers have developed sensors based on cylinder pressure or ionization current measurement, and these have been demonstrated in research engine use. It is more difficult to determine when these technologies will reach a stage where the cost and durability are acceptable for production vehicles. Estimates as short as 2 years from today have been mentioned informally, but the subject is commercially sensitive and little firm information is available.

Under steady-state conditions, low emission combustion has been demonstrated at BMEP levels up to 13-15 bar which is sufficient to cover a large portion of vehicle operating conditions, and is similar to the power produced by a naturally aspirated gasoline engine. One study has even achieved full load in a heavy-duty engine. In today’s conventional engines, however, much effort is focussed on increasing specific power output to improve fuel consumption. Diesel engines already achieve BMEP levels around 26 bar, and similar potential is seen for future turbo-charged gasoline engines. It is these more advanced engines with which HCCI must compete; naturally aspirated gasoline engines will continue to find a market where their lower cost outweighs their higher fuel consumption.

An engine running full-time in HCCI mode may not appear before 2015, but versions using HCCI at low load and switching to conventional diesel or SI combustion at high load have already been developed and demonstrated in prototype form. Some of these may be able to use open-loop control, and so avoid the cost and complexity of combustion chamber sensing. Production examples are expected to appear within the next 2-3 years.

The advantage of the part-time HCCI approach is that the area of HCCI combustion can be gradually expanded as experience is gained. The disadvantage is that low emissions may be achievable over the light-duty driving cycle, but increase at higher load conditions that may be seen in real world conditions.

4.2. FUEL NEEDS

Most studies on HCCI have been based on fuels in the gasoline-diesel boiling range. Since ignition delay is a key factor in controlling the degree of mixing, ignition quality has played a major part in many studies. Successful HCCI performance has been demonstrated with a wide range of fuels, however it seems clear that the current European diesel and gasoline grades are not the ideal choice for HCCI. For conventional diesel, the high cetane number leads to too short an ignition delay, while the high octane number of gasoline makes it difficult to achieve ignition. For diesel, the traditional cetane number seems to provide a good description of ignition performance under HCCI conditions, while for gasoline, the situation is more complex. As a minimum, both the RON and MON need to be known to predict performance.

For systems based on the diesel engine, lower cetane number makes it easier to achieve HCCI combustion, and some have suggested that CN values should be in the 40-45 range. Others propose more radical changes, using fuels with ignition properties closer to the gasoline range (CN below 35) giving the best performance. This could have the advantage of using fuel grades already in the market, but would only be feasible if HCCI combustion could be sustained throughout the whole load range, or the engine used spark ignition rather than compression ignition at higher loads. The second factor influencing mixture formation is volatility. If the objective is to create a more homogeneous air-fuel mixture in a diesel engine, then fuels with a lower boiling range should facilitate more rapid evaporation and mixing. The success of gasoline fuel in diesel HCCI is therefore not too surprising. However, the evidence is mixed as to the importance of volatility. High pressure fuel injection systems achieve a good atomisation, and given a sufficient ignition delay time, can deliver a high degree of mixing. In addition, a completely homogeneous mixture is now seen as undesirable, because it gives no control over combustion, and significant research effort is dedicated to introducing non-homogeneity in a controlled way. Opinions are divided as to whether the benefits of gasoline come purely from its very low cetane number, or whether volatility also plays a part.

The third variable considered for gasoline and diesel fuels is composition, and in particular aromatics. Aromatics have long been considered ‘bad actors’ in the fuel, but the mixture conditions and temperature under which combustion occurs may have much more impact on emissions and performance than fuel composition. HCCI combustion has been demonstrated with a wide range of fuels and compositions, and there seems to be little evidence that composition is an important factor in establishing the right conditions. One area where composition does play a role is in the low temperature heat release that occurs during the compression cycle. This may show as a distinct heat release or simply a formation of radicals without a noticeable temperature rise. In either case, the temperature or composition of the cylinder gases is changed, and this can affect the timing of the main heat release. The LTHR reactions come mainly from longer chain normal paraffins in the fuel; they are not seen with gasoline or methane. There is some evidence that the presence of aromatics can inhibit the development of LTHR, even when paraffins are present. There seems no consensus at present whether LTHR is desirable or not. However, it may be important for engine designers to know whether it will occur, and more work to understand this phenomenon is merited.

The literature contains many studies of HCCI combustion using alternative fuels such as ethanol, DME or natural gas. When used as pure fuels, none of these seem to show significant advantages compared with fuels in the gasoline/diesel range. Natural gas is difficult to ignite and ethanol showed advantages in some areas, but problems in others. With the trend for use of biofuels, a more meaningful question is whether the presence of biodiesel or ethanol in a fuel blend causes any problems to HCCI. Current evidence suggests that 5% biodiesel presents no problems, but more work will be needed in this area.

Since ignition delay is such an important parameter, several studies have investigated whether the fuel quality can be adjusted on board to give the desired properties. For example, if DME were carried in a separate tank on a gasoline vehicle, mixing the two fuels would allow the ignition quality to be adapted to changing engine needs. Similar approaches have been suggested using a reformer to produce synthesis gas from the single fuel carried on the vehicle, and mixing in appropriate quantities to respond to engine needs. Such approaches could work in principle, but would need closed loop control to provide the necessary rapid response. The future role of these systems is uncertain, and their application is unlikely if simpler methods can be used to control combustion.

5. CONCLUSIONS

(i) In HCCI prototype engines, very low levels of engine-out PM and NOx emissions have been widely demonstrated under steady-state research conditions. Power levels up to 13-15 bar BMEP have been achieved in some cases, and one study reports HCCI combustion up to full load.

(ii) Low emissions can be achieved over quite a wide range of mixture compositions and temperatures. The earlier approach to HCCI through early fuel injection to achieve good mixing makes it difficult to control the start of combustion. Most recent research is moving towards systems with later injection and high EGR rates that include non-homogeneity and provide more control, while still maintaining low PM and NOx emissions.

(iii) HC and CO emissions in HCCI combustion are higher than for conventional diesel or gasoline engines, and oxidation catalysts will be required to reduce emissions to acceptable levels.

(iv) Diesel engines seem more likely than gasoline engines to provide a basis for full-time HCCI, since they provide high fuel injection pressures, large amounts of external EGR and high turbocharger boost that are not available on gasoline engines.

(v) Much more work is needed to turn the current research results into practical engines that can use HCCI combustion throughout the load range. Increasing the maximum power for HCCI remains a significant challenge, and sophisticated control systems using combustion sensors will be needed to respond to transient operation. If successful, full-time HCCI engines are not expected before 2015 at the earliest.

(vi) In the meantime, part-time HCCI vehicles are expected to enter the market over the next few years. These will use HCCI at low loads, but revert to conventional diesel or gasoline operation at higher loads, so will need fuels similar to today's gasoline and diesel.

(vii) Full-time HCCI has been demonstrated with a wide range of fuels including conventional gasoline and diesel. However, the properties of today's European fuels are not ideal for HCCI; ignition delays are too short using high cetane diesel fuel, and high octane gasoline is too resistant to ignition. Continuing study is needed to define the most appropriate and practical fuel.

(viii) Ignition delay is a key parameter for HCCI combustion, and one where the correct choice of fuel can have an impact. Current evidence suggests that low cetane diesel fuels (below 45 CN) or even gasoline fuels may be the optimum choice for diesel based HCCI. However, radical fuel changes can only be considered if full-time HCCI becomes successful.

(ix) Directionally, lower fuel boiling points should improve mixture formation, but opinions are divided on the extent to which volatility contributes. Ignition quality has a much bigger impact.

(x) There is no evidence that fuel composition is important in maintaining HCCI combustion, apart from its influence on ignition quality. HCCI combustion has been demonstrated using both diesel and gasoline fuels containing high levels of aromatics. One area where composition does play a part is in the extent of low temperature heat release. It is not clear whether LTHR is desirable or not, but more work is needed to understand it.

(xi) Alternative fuels, such as DME, ethanol and natural gas seem to offer no advantages for HCCI when used alone. There is some evidence that low level biofuel blends do not impede the ability to establish HCCI combustion, but more study is needed.

(xii) Significant progress has been made in development of advanced combustion engines, but much more work is needed before full-time HCCI is a reality. A wide range of fuels can be used, but there is clearly potential scope for optimisation. However, much more work is needed to determine the way forward – it is too early to define future fuel specifications.

6. GLOSSARY

ACCP	Advanced Common Rail Combustion Process
ATDC	After Top Dead Centre
BHP	Brake Horsepower
BMEP	Brake Mean Effective Pressure
BTDC	Before Top Dead Centre
BTL	Biomass-to-Liquids
CA	Crank Angle (measured in degrees relative to TDC)
CA10	Crank Angle where 10% of total heat release is observed
CA50	Crank Angle where 50% of total heat release is observed
CAD	Crank Angle Degree
CAFE	Corporate Average Fuel Economy (USA)
CAI	Controlled Auto-Ignition
CCS	Carbon Capture and Storage
CFR	Cooperative Fuel Research
CI	Compression Ignition
CN	Cetane Number
CNG	Compressed Natural Gas
CO	Carbon Monoxide
CO_2	Carbon Dioxide
COV	Coefficient of Variation
CR	Compression Ratio
DI	Direct Injection
DME	Dimethyl Ether
DPF	Diesel Particulate Filter
DTBP	Di-t-butyl peroxide (cetane improver additive)

EGR	Exhaust Gas Recirculation
2EHN	2-ethylhexyl nitrate (cetane improver additive)
EID	Engine Ignition Delay
EU-25	European Union – 25 Member States
EUCAR	European Council for Automotive R&D
FAME	Fatty Acid Methyl Ester
FID	Fast Flame Ionisation Detector
FIE	Fuel Injection Equipment
FSN	Filter Smoke Number
FT	Fischer-Tropsch
GC	Gas Chromatography or Chromatograph
GDI	Gasoline Direct Injection
GHG	Greenhouse Gas
HC	Hydrocarbon
HCCI	Homogeneous Charge Compression Ignition
HCLI	Homogeneous Charge Late Injection
HD	Heavy-duty
HPCC	Highly Premixed Cool Combustion
HPLI	Highly Premixed Late Injection
HRR	Heat Release Rate
HTHR	High Temperature Heat Release
ICE	Internal Combustion Engine
ID	Ignition Delay
IDI	Indirect Injection
IFP	Institut Français du Pétrole
IMEP	Indicated Mean Effective Pressure
IQT	Ignition Quality Test or Tester

JEC	JRC/EUCAR/CONCAWE Collaboration
JRC	Joint Research Centre of the European Commission
kW	Kilowatt
LD	Light-duty
LIF	Laser Induced Fluorescence
LPG	Liquefied Petroleum Gas
LTHR	Low Temperature Heat Release
MK	Modulated Kinetics
MON	Motor Octane Number
MPI	Multi-Point Injection
NADI™	Narrow Angle Direct Injection
NEDC	New European Driving Cycle
NG	Natural Gas
Nm	Newton-meters
NO	Nitric Oxide
NOx	Nitrogen Oxides
OH	Hydroxyl Radicals
PCCI	Premixed Charge Compression Ignition
PEM	Polymer Electrolyte Membrane
PFI	Port Fuel Injected
pHCCI	partial Homogeneous Charge Compression Ignition
PM	Particulate Mass
PRF	Primary Reference Fuel
ROHR	Rate of Heat Release
RON	Research Octane Number
ROPR	Rate of Pressure Rise
RPM	Revolutions per Minute

SCR	Selective Catalytic Reduction
SI	Spark Ignition
SOC	Start Of Combustion
SOI	Start Of Injection
TDC	Top Dead Centre
TTW	Tank-to-Wheels
TWC	Three-Way Catalyst
UNEP	United Nations Environment Programme
VCR	Variable Compression Ratio
VVA	Variable Valve Actuation
WTT	Well-to-Tank
WTW	Well-to-Wheels
ϕ	Equivalence ratio (the air-fuel ratio, normalised by the stoichiometric value
λ	Excess air ratio (=1/ϕ)

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8. FIGURE CREDITS

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