

# 1

## Fuels and Combustion

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### 1.1

#### Introduction

All internal combustion engines use fuel as the source for heat driving the thermodynamic process that will eventually yield mechanical power. The fuel properties are crucial for the combustion process. Some combustion processes require a fuel that is very prone to ignition, and some have just the opposite requirement. Often, there is a discussion on what is the optimum. This optimum can be based on the fuel or the combustion process. We can formulate two questions:

- What is the best possible fuel for combustion process x?
- What is the best possible combustion process for fuel y?

Both questions are relevant and deserve some discussion, but it is very seldom that the fuel can be selected without any considerations, and similarly, there is only a limited selection of combustion processes to choose from. This brief introduction discusses the combustion processes and the link to the fuel properties that are suitable for them. Thus, it is more in the line of the first question of the aforementioned two.

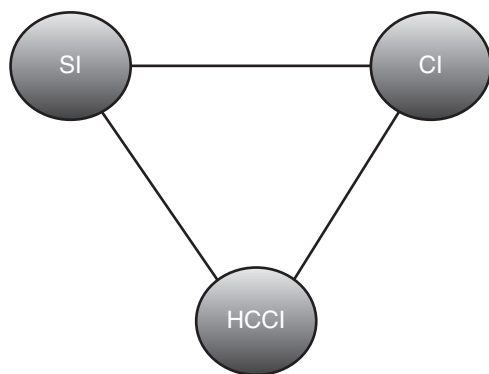
### 1.2

#### The Options

For internal combustion engines, there are three major combustion processes:

- Spark ignition (SI) with premixed flame propagation
- Compression ignition (CI) with nonpremixed (diffusion) flame
- Homogeneous charge compression ignition, HCCI with bulk autoignition of a premixed charge.

These three processes can be expressed as the corner points in a triangle according to Figure 1.1. Within this triangle, all practical concepts reside. Some are a combination of SI and HCCI, some a combination of SI and CI, and others a



**Figure 1.1** The three fundamental combustion concepts, SI, CI, and HCCI.

combination of CI and HCCI. We start by describing the basic three concepts and then move on to discuss the variations and the fuel implications that we can get with combined systems using, for instance, SI and CI at the same time.

The combined concepts to be discussed are as follows:

- Spark-assisted compression ignition (SACI)
- Partially premixed combustion (PPC)
- Reactivity controlled compression ignition (RCCI)
- Dual fuel
- Prechamber flame ignition
- Pilot-assisted compression ignition (PACI) or diesel pilot ignition.

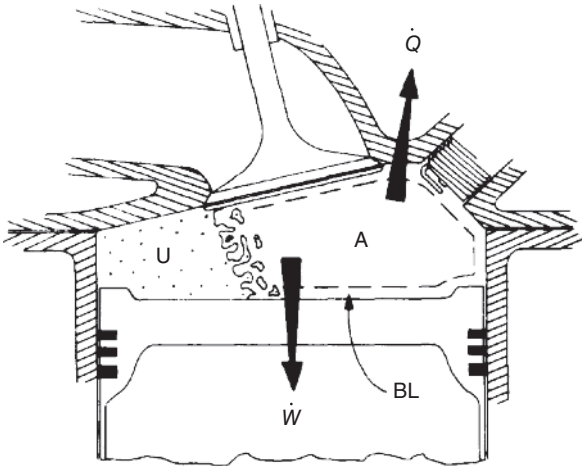
The following section aims to give an introduction to the combustion processes and highlight the fuel requirements. Both chemical and physical properties are discussed.

### 1.3

#### Spark Ignition

The SI process is in principle very simple. Fuel and air are mixed and then the charge is compressed. Close to the piston top dead center, a spark is generated between two electrodes of a spark plug. This results in a locally very hot zone that starts exothermic reactions. Those reactions heat up the vicinity of the reaction zone and thus the reactions start there. The resulting propagating reaction zone is most often called a *flame*. The rate at which this flame propagates depends on the reactivity of the charge and how much the flame is distorted by turbulent eddies. Figure 1.2 illustrates the principle, with a spark plug to the right and a flame some  $2/3$  distance from the spark plug to the back wall of the combustion chamber.

The enhancement of flame speed by turbulence is not much affected by the fuel properties but by the reactivity of the charge. This reactivity is most often expressed as a laminar flame speed. Most hydrocarbon fuels exhibit a laminar



**Figure 1.2** Normal flame propagation [1]. (Reproduced with permission from Heywood [1] of McGraw Hill.)

flame speed at around 0.4 m/s close to that of stoichiometric mixtures, and this then drops to very low numbers as the mixture strength approaches the lean or rich dilution limits. In fact, those *limits* are defined when the laminar flame speed is zero. The major outlier when it comes to laminar flame speed is hydrogen. The laminar flame speed of hydrogen is very much higher than that of hydrocarbons. This means that a hydrogen engine can be expected to undergo a much faster combustion than the one using conventional fuels. Also, the alcohols tend to have a slightly higher flame speed, but here the difference to regular hydrocarbons is much less (<10%).

### 1.3.1

#### Uncontrolled SI Combustion, Knock

The conventional flame propagation with SI is not that fuel-dependent; except for hydrogen, most fuel behaves the same. Instead, the major problem comes when the combustion process is not working according to plan. With SI combustion, the fuel and air are mixed to burnable proportions well before the combustion starts. This means that the reactive mixture will be heated to a rather high temperature for some time. The worst conditions are for the end gas. This is the gas that is furthest away from the spark plug in the combustion chamber. This charge is heated and pressurized during the compression stroke but also by the increased pressure resulting from the combustion of the fuel closer to the spark plug. If the end gas charge is heated too much for too long, it will autoignite. Then the combustion will be very fast and all the end gas fuel/air mixture is converted into combustion products infinitely fast, at least in terms of time scales relevant for the engine. The autoignition is sometimes called knock from the engine noise if operated with autoignition. The noise comes from the pressure fluctuations resulting from the

pressure wave originating at the autoignition site in the cylinder. If a moderate amount of fuel is burned with autoignition, the pressure oscillation amplitude will be a few bars only. This can be heard but will not be directly damaging. But if the autoignition is earlier, more fuel is burned in the autoignition process and the pressure oscillations can directly destruct the engine architecture. The high-amplitude pressure oscillations will also increase heat transfer to the combustion chamber walls by breaking down the thermal boundary layer. This will reduce engine efficiency and also heat the walls. If the walls are heated too much by prolonged knocking, the wall material will eventually reach a temperature where the mechanical strength will no longer be sufficient. When the piston material melts, there will be a permanent damage or even a hole, losing all compression and hence engine performance.

### 1.3.2

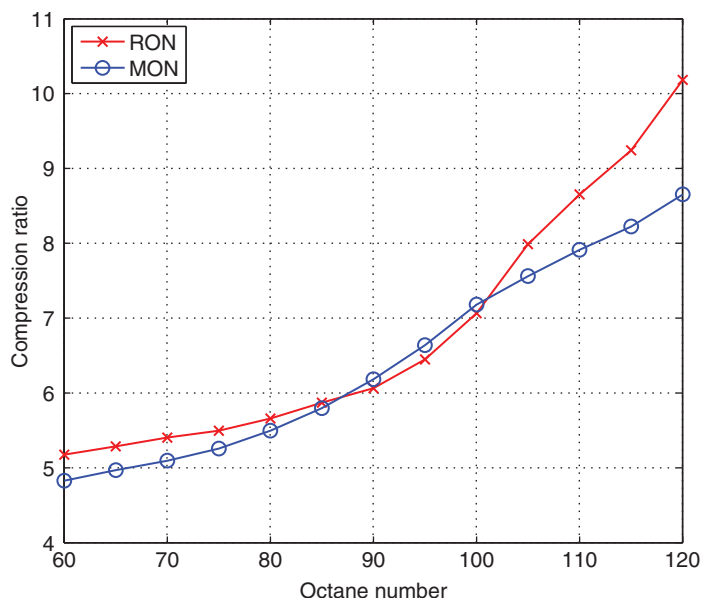
#### Autoignition of SI Engine Fuel

Spark ignition knock is determined by engine parameters such as flame travel length and compression ratio and also by engine operating conditions such as engine speed, load, and spark timing. But it is also fundamentally linked to the autoignition tendency of the fuel/air mixture in the cylinder. This is given by the fuel properties as such but also the fuel/air mixture strength, the amount of hot residual gases in the cylinder, the amount of exhaust gas recirculation (EGR), and other aspects affecting the reactivity of the charge.

To avoid SI engine knock, a fuel with a high resistance to autoignition should be selected. The number(s) most often used to quantify the knock tendency of a fuel is called the *octane number*. To make it a bit more complex, there are two, Research Octane Number (RON) and Motor Octane Number (MON). Both are extracted by operating a standardized engine, called a *corporate fuel research (CFR)*, but with two sets of conditions. The first, RON, uses an engine speed of 600 rpm and an inlet temperature of 49 °C (150 °F) and fixed spark timing. For the MON, the engine speed is increased to 900 rpm, the inlet temperature is set to 149 °C (300 °F), and spark timing is adjusted for maximum knock.

When evaluating the octane number, the engine compression ratio is increased until a predefined level of knock is detected with a standardized microphone in the cylinder. The compression ratio is then compared to mixtures of two reference fuels: *n*-heptane, which is prone to autoignition, and *iso*-octane, which is much more resistant to autoignition. The mixture of *n*-heptane and *iso*-octane is adjusted until the same level of autoignition is reached. The *percentage of iso-octane in the reference mixture* is then defined as the tested fuel octane number. Thus, pure *n*-heptane will have an octane number of 0 by definition and pure *iso*-octane will have 100. But it should be noted that the scale between 0 and 100 is not linear. Figure 1.3 shows the compression ratio as a function of RON (and MON).

The required compression ratio as a function of fuel carbon atoms can be seen in Figure 1.4. The *iso*-octane and *n*-heptane are shown but also fuel with an octane



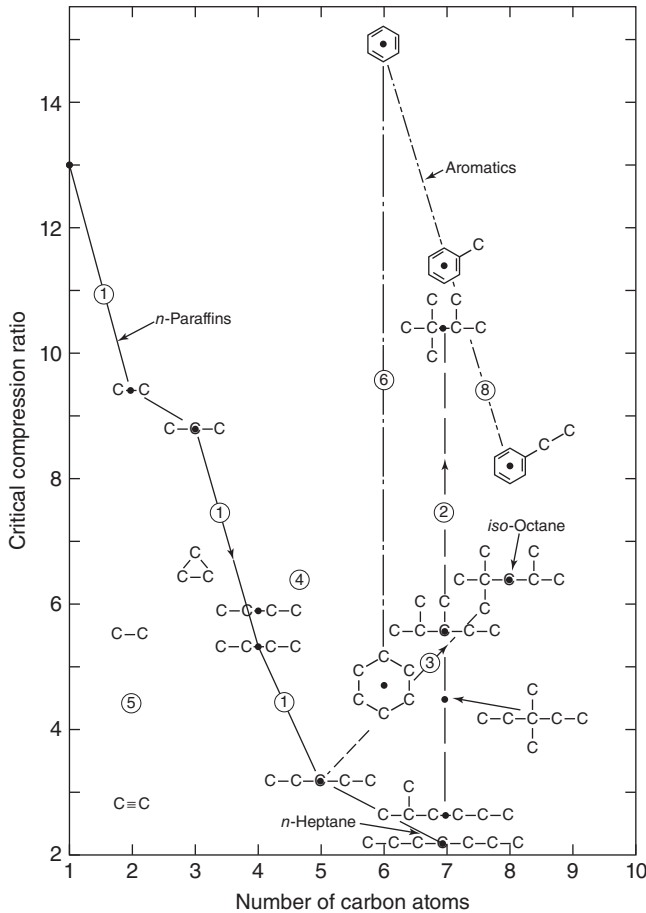
**Figure 1.3** The required compression ratio for knock to be a function of RON and MON.

number higher than that of *iso*-octane. Both methane,  $\text{CH}_4$ , and benzene with the aromatic ring require higher compression ratios.

The RON test was first introduced in 1928 when the effect of fuel autoignition tendency on SI engine performance was first realized. Later, it was argued that the conditions for RON were less suitable and a more severe test was generated. This is called the Motor Octane Number. The same engine is used as with RON, but the inlet temperature is increased from 49 to 149 °C (300 °F) and the engine speed is increased to 900 rpm. There is also a significant difference in how the inlet temperature is measured. For the RON case, it is the air temperature before the fuel is added to the inlet, whereas for the MON case, it is the fuel/air mixture temperature. Thus, if a fuel would have significant heat of vaporization, leading to cooling of the charge as the fuel evaporates, a clear difference between RON and MON is expected.

For most practical hydrocarbons, the RON is higher than MON, often around 5–10 units. This has led to different definitions of octane number worldwide. In the European Union, the RON is used and there are no requirements of MON. In the United States, the average of RON and MON is used and is called *Octane Index*. As MON is most often lower, the numbers found on filling station fuel pumps are different in the European Union and the United States. A fuel rated as 87 in the United States can be equivalent to 92 in the European Union.

There has been some debate about the suitability of using both RON and MON in the United States, especially from Gautam Kalghatgi from Shell (and since a few years, Saudi Aramco) [2]. The argument is that engines of today are using a



**Figure 1.4** Required compression ratio as a function of octane number of RON [1]. (Reproduced with permission from Heywood [1] of McGraw Hill.)

rather high boost pressure and an intercooler. This means that the condition in the cylinder after compression is different from the RON and especially MON cases. With high boost and effective intercooling, the pressure level is high, whereas the temperature is moderate, and hence the  $T/p$  ratio is low. This is in stark contrast to the MON case where the inlet temperature is high and compression ratio moderate, yielding moderate pressure after compression. For the MON case, the  $T/p$  ratio will be very high. The RON condition will be something in between with a moderate  $T/p$  ratio. It is thus argued that for a given RON number, a fuel with a lower MON will ignite easily at high temperature and low pressure (high  $T/p$ -ratio) and, as a consequence, ignite less easily at low temperature and high pressure (low  $T/p$ -ratio). In other words, for a given RON, a fuel with a low MON will be less prone to autoignition at high pressure and low temperature. This is how modern engines are operated.

## 1.3.3

**Physical Properties of SI Engine Fuel**

Apart from the chemical properties, SI engine fuel must also exhibit some suitable physical properties. The most important, for liquid fuel, is the boiling point range. Most SI engines still comprise carburetors or fuel injection systems that add the fuel in the inlet system. Thus, a significant fraction of the fuel is deposited on the inlet walls and needs to be evaporated to enter the cylinder. During cold start, the walls are cold and thus fuel enrichment is needed to get sufficient fuel amount into the cylinder. Here the light fraction of the fuel is very important, as it is this fraction that will find its way to the cylinder. Also, for a heated-up engine, the fuel needs to boil off the inlet walls. Thus, the final boiling point ( $T_{90}$ ) must be reasonably low, around  $180^{\circ}\text{C}$ . The boiling point range for commercial gasoline is specified in EU within the EN228 fuel specification.

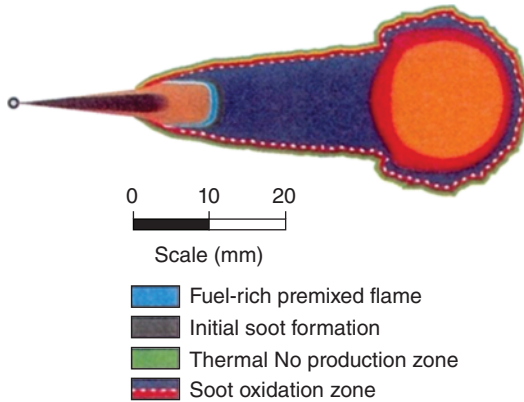
Another important factor for SI engine fuels is the heat of vaporization. Most gasoline exhibits significant heat of vaporization, but some fuels such as ethanol and especially methanol exhibit much more. The heat of vaporization can be used to increase engine performance. The cooling effect of the fuel evaporating can be used to get more air into the cylinder and hence burn more fuel. But it can also be used as an “internal intercooler” reducing charge temperature and hence reducing the risk of knock.

It can be noted that most SI engines are using low-pressure fuel injection and thus can also be adapted to gaseous fuels without major problems. With gaseous fuels, the fuel is mixed with air in the inlet, or it can be injected into the cylinder very early after inlet valve closing if additional engine performance is required.

## 1.4

**Compression Ignition**

The combustion process in CI engines is totally different from that in SI engines. Only air is compressed, and close to top dead centre, TDC, a high-pressure fuel is sprayed into the cylinder. After some initial mixing, the combustion starts with a rather uniform, very rich ( $\lambda=0.2$ ) autoignition. The mixture is so rich that the reaction rate is not extremely fast. After the initial homogeneous, rich autoignition, diffusion-controlled combustion process occurs. Here the fuel is first mixed with air up to the lift-off position where a pre-mixed rich flame is formed. After the flame formation, additional reactions including soot formation take place inside the spray plume and then a thin zone close to stoichiometric reaction zone is formed in the outer boundaries of the plume. Here most of the  $\text{NO}_x$  is formed and much oxidation of soot results. Figure 1.5 shows the conceptual model presented by John Dec in 1997 [3].



**Figure 1.5** Conceptual model of diesel combustion by John Dec. (Reproduced with permission from Dec [3] of SAE International.)

#### 1.4.1

##### Autoignition of CI Engine Fuel

The fuel used in a CI engine is vastly different from that in an SI engine. In an SI engine, the fuel should sustain high pressure and temperature without autoignition. The exact opposite is true for the CI engine. Here the fuel should autoignite as fast as possible after injection has started. This will reduce the amount of fuel burned in the rich premixed region and thus also reduce the first initial spike in the rate of heat release curve. CI engine fuel is rated in the same way as the octane scale is used for SI combustion. The same CFR engine is used but modified to have a direct injection (DI)-type fuel injection system with moderate pressure. The fuel is injected at 13 CAD before TDC and then the compression ratio is adjusted to start autoignition at TDC. Hence, the engine is operated with 13 CAD of ignition delay. The reference fuels are not the same as for the octane number. For CI operation, it is cetane (now called *hexadecane*,  $C_{16}H_{34}$ ) that is given a number of 100 and  $\alpha$ -methyl naphthalene is given 0.

An ignition delay of 13° is more than that of modern CI engines. In combination with an injection pressure of only 300 bar, almost an order of magnitude less than that of modern CI engines, it is argued that fuel ratings in the CFR is less representative of modern engines. Thus, an alternative way of characterizing CI engine fuel has been introduced. It is called derived cetane number (DCN) and extracted from a device called ignition quality tester (IQT). This is a constant-volume chamber heated to 575 °C. The fuel is injected under this constant condition, and the resulting ignition delay is then used as a direct measure of the cetane number (CN). Thus, it is in principle enough to inject once to the IQT to obtain a reading and hence much less fuel is needed to obtain the DCN and the CN from the CFR engine. There are also indirect ways to estimate the fuel autoignition tendency by measuring the fuel density and boiling point range. With density and boiling point in four points, a cetane number, CI, can be extracted. This does not take any



ignition improver additives into account and will not be a direct measurement of autoignition.

#### 1.4.2

##### Physical Properties of CI Engine Fuel

Also, for the physical properties, the requirements of fuel for CI and SI engines are in stark contrast. The CI engine works with a very high pressure fuel injection system to get the spray fully atomized and to get fast mixing. A pressure of 2000–3000 bar induces a very high load on the mechanical components in the fuel system and the risk of cavitation is always there. Thus, the fuel must exhibit suitable properties for high-pressure compression. It is important that the fuel exhibits a minimum viscosity; otherwise, it will leak past the fuel pump elements. It should also be incompressible. This is in principle the case of all liquids, but some fuels are rather compressible. A typical case is dimethyl ether (DME). It is very hard to compress DME above 500 bar due to the compressibility of the fuel [4]. As a CI engine fuel pump should be able to compress the fuel to a pressure much higher than the cylinder pressure, all gaseous fuels will result in significant parasitic losses.

In practice, the physical and chemical properties of most fuels are linked in a favorable way. Smaller lighter molecules most often also have a high octane number and are thus suitable for SI engine operation. If we focus on conventional fossil fuels, the lightest hydrocarbon would be methane. This has a high RON (130) and is easy to add to the inlet of an SI engine. Also, ethane, propane, and butane, often called *liquid petroleum gas (LPG)*, have high octane numbers and are gaseous under ambient conditions. Crude oil fractions up to roughly eight carbon atoms per molecule have a boiling point below 180 °C and can be directly used as gasoline components. The breakpoint of 180 °C occurs to coincide with the lowest boiling point of diesel fuel for the CI engine. The heavier fractions tend to have a lower octane number (and hence higher cetane number) and are thus well suited for the CI engine.

A very oversimplified model of how to produce gasoline and diesel fuel is thus as follows:

- 1) Drill a hole in the ground.
- 2) Pump the crude oil out from the hole.
- 3) Pour the crude oil in a distilling pot and heat the pot to 180 °C.
- 4) What boils off is condensed in a separate cooled pot and is called *gasoline*.
- 5) Heat the pot to 350 °C.
- 6) What boils off is condensed in a separate cooled pot and is called *diesel fuel*.
- 7) What stays in the pot is called *heavy fuel oil* or *possibly asphalt*.

#### 1.5

##### Highly Diluted Autoignition, HCCI

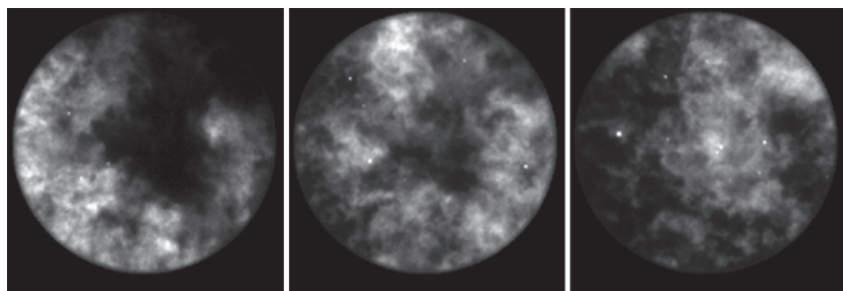
Apart from the dominating SI and CI engines, it is also possible to operate with the third type of combustion: autoignition. This is often called *homogeneous charge*

*compression ignition*. With HCCI, the fuel and air are fully premixed before combustion as in the SI engine, but combustion is started by the increased pressure and temperature during the compression stroke.

The combustion starts in many zones at the same time and can be called a *distributed reaction*. It is incorrect to name it homogeneous combustion though as the combustion process shows significant inhomogeneity with some zones burning much faster than the other zones. Figure 1.6 shows three images of the combustion process from an optical engine operated in HCCI mode. In this particular engine, a Scania 2-l cylinder truck engine, the combustion had a tendency to ignite in the outer regions, and only later, the reactions start in the center [5].

With HCCI, the charge is very much diluted to make sure the reactivity is moderate. This is needed as the combustion rate would otherwise be extreme, ruining the engine structure possibly in a single cycle with excessive pressure rise rate and peak in-cylinder pressure. This also enables a very efficient thermodynamic cycle and hence HCCI can show much better total fuel efficiency [6] and an SI engine operated with the same fuel in the same engine.

The combustion process in HCCI is very sensitive and can be considered a balance in temperature. The fully premixed charge will break down and combustion start when the temperature and pressure are high enough for a prolonged period. For convenience, it is often expressed as an autoignition temperature; once it is reached, the reaction starts. For *iso*-octane, the autoignition temperature is 1000–1050 K, depending on the air/fuel ratio and other parameters. If we for additional convenience for a moment fix it to 1000 K we have a few options how to reach this temperature. In the first method, we can start with a low inlet temperature and use a very high compression ratio. Another method is to start at a high inlet temperature and use a lower compression ratio. The third method is to trap the hot residual gas (burnt gas from the last cycle). This enables us to use both low inlet temperature and low compression ratio. But independently of the method by which we reach 1000 K, we need to make sure that it is reached at the right time in the cycle. If there is a bit too high inlet temperature, too high compression ratio, or too high amount of hot residual gas, the reactions will start early and combustion will occur early. Early combustion is not desirable as it will yield high



**Figure 1.6** HCCI combustion at 7, 9, and 11 CAD after top dead centre (ATDC) in a single cycle. (Reproduced with permission from Vressner *et al.* [5] of SAE International.)

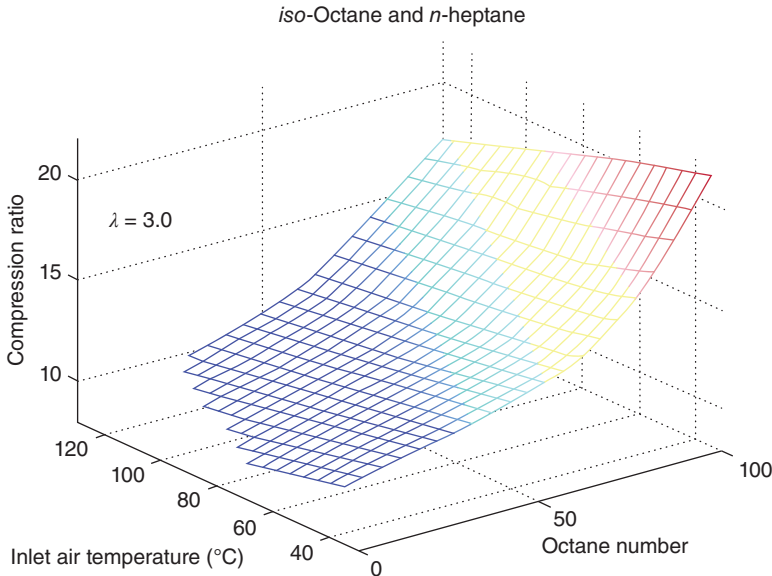
peak pressure, high pressure rise rate, and more  $\text{NO}_x$ . On the other hand, if we have a bit low inlet temperature, low compression ratio, or less amount of residual gas, the combustion will be delayed as the 1000 K temperature is reached very late during the compression stroke. This is also bad, as late combustion will result in much higher emissions of hydrocarbons and carbon monoxide and perhaps even more significantly increased cycle-to-cycle variations. The operating window in terms of combustion phasing is, at lower loads, 5–10 CAD, but as the maximum load for HCCI is reached, it reduces to 1 CAD or less. In this case, there is no room for errors or fluctuations of parameters that affect the combustion onset.

### 1.5.1

#### Autoignition of HCCI Engine Fuel

As the HCCI combustion depends more on combustion phasing, that is, when the cycle of autoignition starts, HCCI becomes very much dependent on the fuel properties. For this reason, it can be argued that HCCI is extremely fuel sensitive.

But it can also be argued that HCCI is not fuel sensitive at all for the same reason. Since the HCCI combustion is so sensitive, an active control of combustion phasing is needed with a powerful enough control variable that can compensate for changes in any uncontrolled variable. Two typical, very powerful control variables are inlet temperature,  $T_{\text{in}}$ , and compression ratio,  $R_c$ . If we have the capability to adjust  $T_{\text{in}}$  or  $R_c$  within a wide range, any fuel with any autoignition quality can be used. Figure 1.7 shows the combinations of  $T_{\text{in}}$  and  $R_c$  that achieve correct



**Figure 1.7** Combinations of fuel octane number inlet temperature and compression ratio that achieve combustion at correct combustion phasing. (From SAE paper 1999-01-3679.)

combustion phasing for mixtures of *iso*-octane and *n*-heptane in a Volvo Truck engine [7]. Those two fuels happen to be the definition of the octane scale, and hence, it is shown that this engine with the control range of inlet temperature 40–120 °C and compression ratio of 10:1–25:1 can use anything from 0- to 100-octane fuels. It can be noted that the matrix is truncated at the lower left. It is not possible to use 120 °C with pure *n*-heptane. The compression ratio in this case needs to be lower than 10:1 to achieve a late enough combustion phasing, that is, outside the control range of that particular control system.

The figure shows a clear link between the octane number and the required compression ratio. This has a clear message in terms of how to operate an HCCI engine to achieve fuel efficiency. A higher compression ratio leads to a better thermodynamic cycle and hence higher efficiency. Hence, it is only with high-octane fuels HCCI can be expected to achieve high efficiency. Diesel-like fuels need very low compression ratio and therefore less efficiency.

This leads to an interesting observation. Most SI engines today are operated with a compression ratio around 9:1–12:1. If we should operate such engines in HCCI mode, the most suitable fuel would be diesel. Diesel engines, on the other hand, use a compression ratio of around 16:1–18:1. If such engines would be operated in HCCI mode, the most suitable fuel would be gasoline with a high octane number. So in short, the most suitable fuel for an SI engine is diesel and the most suitable fuel for a diesel engine is gasoline if they are run in HCCI mode.

The results in Figure 1.7 show what combinations of  $T_{in}$  and  $R_c$  should be used for a given fuel octane number. However, it can also be used in the opposite way. If an engine will have a compression ratio between 10:1 and 25:1 and an inlet temperature between 40 and 120 °C, the figure shows the fuel octane number that is needed for a given combination. Hence, if the  $T_{in}$  and  $R_c$  are uncontrolled variables, the fuel octane number can be used to control combustion phasing. It is perhaps one of the most powerful control variables, provided that the fuel octane number can be changed within a wide range.

### 1.5.2

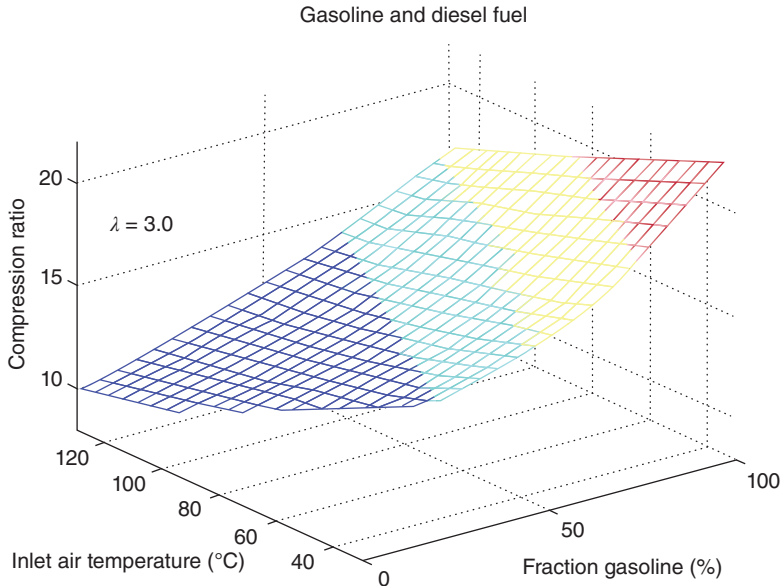
#### Physical Properties of HCCI Engine Fuel

The HCCI combustion process is not directly affected by fuel physical properties, but since it requires a fully homogeneous charge at the time of autoignition, the fuel preparation should be done some time before combustion onset. The simple way to generate a homogeneous charge is to mix the fuel with air at the inlet with a carburetor or port fuel injection system. If such a system is used, the boiling point range of the fuel becomes important. Port fuel injection works well with gasoline with a boiling point up to 180 °C. Also, gaseous fuels can be injected in the inlet without problems. This means that most high-octane fuels can use simple port fuel injection.

Diesel fuel has a high initial boiling point, in fact close to the end boiling point of gasoline. This means it will not evaporate much, if at all, if port fuel injection is used without inlet air heating. If the inlet air is heated, this will promote

vaporization and a reasonable amount of fuel will be gaseous when it enters the cylinder. In Figure 1.7, results with *iso*-octane and *n*-heptane are shown. They both have a boiling point of 98–100 °C, and thus, there is no issue of using port fuel injection. Figure 1.8 shows the results using mixtures of gasoline and diesel as high- and low-octane fuels. As diesel fuel has an octane number of around 20, the 10 : 1 compression ratio is sufficiently low, and hence, the corner with a combination of low compression ratio and high inlet temperature can be operated. However, the combination of low inlet temperature and low percentage of gasoline is not possible, due to a too low inlet temperature. With a high-boiling-point fuel such as diesel, a minimum inlet temperature of 100 °C is needed. It is actually possible to operate the engine at low inlet temperature, but then soot emissions will increase sharply when the fuel is poured into the cylinder, forming a pool on the piston top. The resulting pool fire does not burn in a homogeneous way.

From the results in Figure 1.8, it can be concluded that high-boiling-point fuels such as diesel should not use port fuel injection but rather direct injection with a diesel-like injection system to achieve proper atomization. The injection should be rather early to form the homogeneous charge at TDC. The compression ratio must be reduced with low-octane fuels to prevent very early combustion. The combination of early injection and low compression ratio means that the pressure in the cylinder will be low at the time of fuel injection. This must be taken into consideration when the fuel injection pressure is selected. A too high injection pressure will lead to excessive wall contact of the spray and hence less homogeneous charge.



**Figure 1.8** Combinations of inlet temperature and compression ratio that achieve combustion at correct combustion phasing with mixtures of diesel fuel and gasoline. (From SAE paper 1999-01-3679.)

## 1.5.3

**HCCI Fuel Rating**

The SI engine has two octane numbers to indicate the autoignition quality of the fuel, RON and MON, and the diesel (CI) engine has one, the cetane number. There is a lack of an equivalent number for HCCI. Thus, a procedure was proposed by Johansson and coworkers for extracting an HCCI number [8]. It was proposed to use the same engine as for the RON/MON and CN, but the engine should be operated in HCCI mode with a fixed lean mixture ( $\lambda = 3$ ). The inlet temperature was the same as that of the RON or MON cases and an engine speed of 600 and 900 rpm was used. Hence, four combinations were used, high/low inlet temperature and high/low engine speed. The engine compression ratio was adjusted to achieve combustion phasing, CA50, at 3 CAD ATDC for a test fuel, and then this was compared to the compression ratio needed for mixtures of *iso*-octane and *n*-heptane. The percentage *iso*-octane in the mixture that gave the same compression ratio will then determine the HCCI number.

With CN, RON, and HCCI number, we have the fuel autoignition behavior with rich, close to stoichiometric and lean mixtures. Thus, if a fuel will not change its behavior with stoichiometry, the three numbers will be very much the same, but with a highly  $\phi$ -sensitive fuel, the number will be quite different. Having all three, it is also possible to estimate the fuel behavior in mixed modes within the triangle shown in Figure 1.1.

## 1.6

**Other Combustion Concepts**

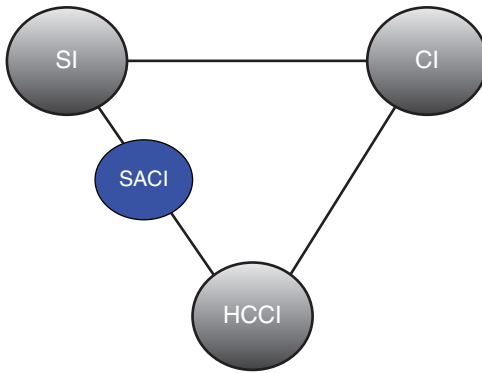
Apart from the three fundamental combustion processes with SI, CI, and HCCI, there are also a few combined or intermediate concepts. Those presented here are discussed in terms of the requirements of fuel properties. The chemical and physical properties are discussed separately.

## 1.6.1

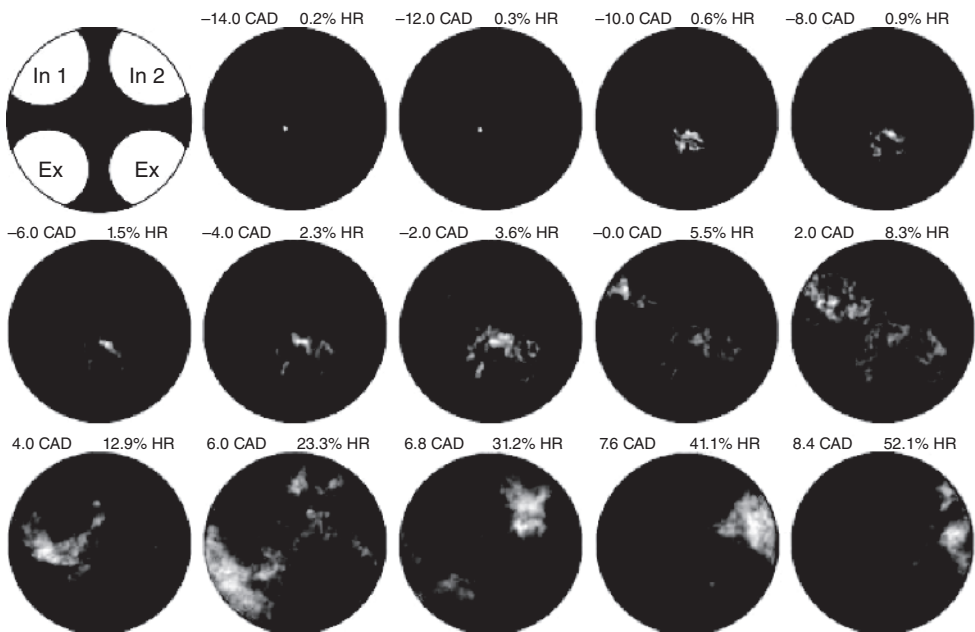
**Spark-Assisted Compression Ignition, SACI**

SACI is a combined process, which starts with a flame propagation and ends with HCCI. Figure 1.9 shows the location in the triangle between SI, CI, and HCCI as an intermediate between SI and HCCI. Figure 1.10 shows a typical behavior with initial flame propagation from the centrally located spark plug. At 2 CAD before Top dead centre (BTDC), the autoignition starts and then quickly consumes the charge in the HCCI mode [9].

For this process to work, the mixture strength must be sufficient to enable flame propagation but at the same time diluted enough to produce a reasonably slow autoignition. With air dilution alone, the lean limit ( $\lambda$ ) for the flame propagation is 1.6–1.8. At the same time, the rich limit ( $\lambda$ ) for HCCI is often 2.2–2.5. Under



**Figure 1.9** Spark-assisted compression ignition in between SI and HCCI.



**Figure 1.10** Initial flame propagation and subsequent autoignition in SACI in a single cycle. (Reproduced with permission from Persson *et al.* [9] of SAE International.)

these conditions, SACI would not be possible. What has to be done is to operate the engine at a temperature higher than the normal temperature. This extends the lean limit for SI, and at the same time, the rich limit of HCCI is moved in the right direction. Most often, SACI is better utilized with residual gas or EGR instead of air dilution. Thus, a dilution window with stable flame propagation and reasonably smooth autoignition can be achieved.

### 1.6.1.1 Chemical Properties

The autoignition properties of SACI do not differ much from those of pure HCCI. What could be beneficial would be a fuel with a higher laminar flame speed but moderate octane number. This would improve the lean limit of SI flame propagation and make HCCI trigger early. There is no hard evidence that such fuels can be used or their effects demonstrated though.

### 1.6.1.2 Physical Properties

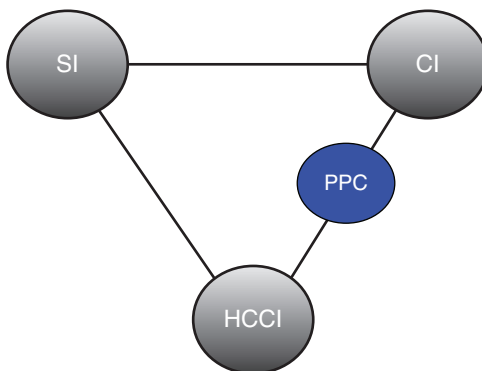
SACI is most often implemented in SI based engines with direct fuel injection. Hence, the requirements for fuel physical properties would be less or the same as those of a conventional direct-injected SI engine.

## 1.6.2

### Partially Premixed Combustion, PPC

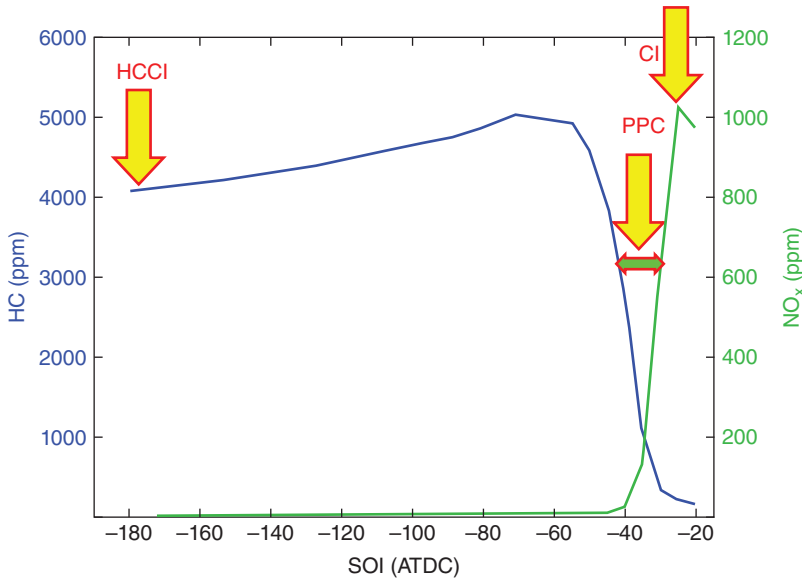
PPC is something between HCCI (fully mixed) and CI (very limited mixing before combustion) as shown in Figure 1.11. Hence, the combustion will take place in a stratified charge, but it will not be diffusion-controlled, spray-driven combustion as in the conventional diesel engine. Figure 1.12 shows the gradual change of behavior from the fully premixed HCCI combustion with very early fuel injection in the cylinder via PPC to the classical diesel with late injection and very limited delay from fuel injection onset to combustion start [10].

The concept is based on injection rather late in the compression stroke and operation with a significant ignition delay. A crude definition of PPC is that all the fuel should be in the cylinder at the time of ignition. This means that start of combustion (SOC) should be after end of injection (EOI). Most often, multiple injections are used with PPC to generate a suitable stratification of fuel/air in the cylinder at the time of ignition [11].



**Figure 1.11** Partially premixed combustion, PPC.





**Figure 1.12** Gradual change of unburned fuel (hydrocarbon, HC) and  $\text{NO}_x$  with later start of injection (SOI) and indication of combustion mode. (Reproduced with permission from Nordgren *et al.* [10] of SAE International.)

#### 1.6.2.1 Chemical Properties

The PPC concept can be implemented with diesel fuel [12] as well as high-octane fuels [13]. The PPC is well suited for diesel fuel at low loads and gasoline-like fuels at higher loads, but an ideal PPC fuel will change the octane number as a function of load. There is a very simple rule of thumb for optimum fuel octane number as a function of load. The optimum octane number is five times the load in indicated mean effective pressure (IMEP). This means that in idle mode, the fuel can be very similar to diesel fuel and at full load, an octane number of 100 or higher is good.

But even if the ideal fuel is different at low and high loads, it is possible to counteract the effect of load with engine measures. At a low load, a higher inlet temperature, higher effective compression ratio, and high amount of residual gas aid in the process [14]. At a high load, a low inlet temperature, reduced effective compression ratio with variable valve timing (VVT), and possibly retarded combustion all can aid. It has been shown that it is possible to operate PPC all the way from idle mode to 26 bar load with one single fuel provided that it is with suitable octane number [15]. With RON70, this was possible using a low fixed inlet temperature. With higher octane numbers, it is required to use VVT with negative valve overlap or rebreathing to go down in load all the way to idle [16].

It should be noted that PPC is rather simple to implement in conventional diesel engines at part load. Here the diesel fuel will have sufficient ignition delay to get the fuel in before combustion starts by applying large amounts of EGR. This is also a part of the engine calibration for many light-duty diesel engine manufacturers.

But with gasoline, the load range is extended, and hence, very high efficiency can be reached – up to 57% is reported.

### 1.6.2.2 Physical Properties

The physical properties of PPC fuel are mainly linked to the fuel injection system used and the potential wall wetting with early fuel injection. Most often, PPC uses a CI engine architecture and hence its high-pressure fuel pump. This can work but normally the injection system is not adapted to gasoline or gasoline-like fuels having lower viscosity and much lower initial boiling points. For instance, tests with the standard Delphi injection system in a Volvo truck engine revealed that the maximum injection pressure was only 1200 bar using diesel in comparison to the 2400 bar with diesel fuel. That system used unit injectors that build up the fuel pressure just before the injectors open. With high gasoline internal leakage, the pressure could not build up. The problem is much less with common-rail type of systems as the time available for pressure buildup is longer. Even so, the parasitic losses can increase much with more pumping work by the fuel pump.

PPC with diesel fuel does not have any issues with the fuel pump or injectors as the system is designed for that fuel. A problem with PPC and diesel fuel is, instead, the potential wall wetting with early fuel injections. As the pressure in the cylinder is very low early in the compression stroke, the fuel can impinge on the wall. If much fuel is deposited on the wall, the lubricating oil can be diluted and the piston rings can seize. Too much fuel into the top land region, between the piston and the cylinder above the piston ring, can push it into the crankcase. This will also dilute the lubricating oil and overfill the crankcase. If gasoline finds its way to the crankcase, most of it will evaporate and be returned to the inlet via the closed crankcase ventilation system. That is not so simple with diesel fuel having such high boiling point range.

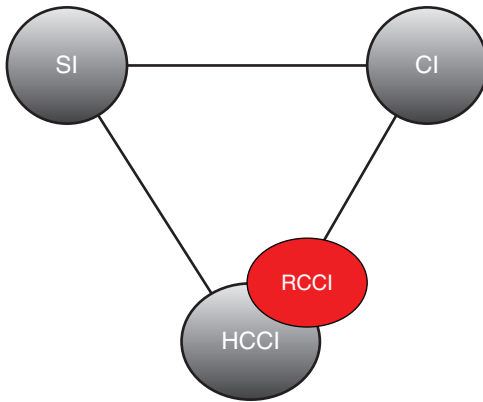
### 1.6.3

#### Reactivity-Controlled Compression Ignition, RCCI

RCCI is a version of HCCI in which dual fuels are used, as described in the HCCI section [17]. The difference in dual-fuel HCCI is that, with RCCI, two-fuel injection systems are used to generate stratification in reactivity in the cylinder. Typically, around 80–90% of the fuel is injected in the inlet port and then 5–10% at 60 CAD BTDC and 5–10% at 30 CAD BTDC [18]. The fuel injected in the inlet port should be with high octane number and the direct-injected fuel with low octane number. The amount of low-octane fuel and its homogeneity at TDC will determine the combustion onset timing and combustion duration. Very high fuel efficiency has been reported with RCCI, up to 60% [19] (Figure 1.13).

#### 1.6.3.1 Chemical Properties

For RCCI to work in a wide operating range, a wide separation in octane number between the two fuels is needed. Typically, the direct-injected fuel is diesel or similar and the port fuel injected is gasoline, ethanol, or natural gas. It is also possible



**Figure 1.13** Reactivity controlled compression ignition, RCCI.

to operate with less separation, but then the load range in which the combustion can be controlled will be limited.

#### 1.6.3.2 Physical Properties

The physical properties needed for RCCI is highly dependent on the injection systems. For the high-octane fuel, conventional port fuel injection is most often used. This means that the requirements are the same for an SI engine or port-fuel-injected HCCI engine. Gasoline is not a problem, but diesel-like fuels are not suitable. Gaseous fuels are very well suited [20].

For the direct-injected portion, RCCI has the same requirements as those of a diesel engine. The fuel injection system is most often from a diesel engine and hence adapted for that type of fuel already.

A clear advantage of RCCI is that it uses the fuel system normally used for the two fuels and hence no major issues should result. There is an obvious drawback with two fuels and two-fuel systems though. The operator needs to fill two-fuel tanks with the right fluid and two-fuel systems are more expensive than single-fuel systems.

#### 1.6.4

##### Dual-Fuel Combustion

Dual-fuel combustion is used in the production of larger ship and genset engines. The concept is rather old and originates from fumigation [21]. With fumigation, part of the diesel fuel was introduced in the inlet and hence a homogeneous charge was formed in the cylinder. By replacing the diesel fuel with a more suitable high-octane fuel such as gasoline [22], ethanol [23, 24], or methanol [25], the performance will be better. However, the most common fuel is natural gas. The large engines for electricity production are often with dual fuel using natural gas and a small diesel injection quantity.

This concept is very similar to RCCI, but the injection strategy and operation are different. The engine uses a premixed charge of air and high-octane fuel as RCCI, but ignition is done with a diesel spray close to TDC. This means that the ignition delay is short and very much shorter than for RCCI. After combustion onset by the diesel spray, the premixed charge is burned with flame propagation as in an SI engine. This means that the engine must be operated to prevent knock. The location in the triangle is shown in Figure 1.14, starting to the right with CI and ending to the left with SI, hopefully not ending up with HCCI (knock) (Figure 1.15).

Dual-fuel combustion has same issues as SACI with balancing the needs of two combustion processes. The pressure and temperature after compression must be sufficient enough to ensure that the diesel pilot ignites when injected close to TDC. This calls for a high compression ratio and not too low inlet temperature. On the other hand, the diesel pilots will ignite the homogeneous charge and then flame propagation will follow. This SI type of combustion has the same issues as

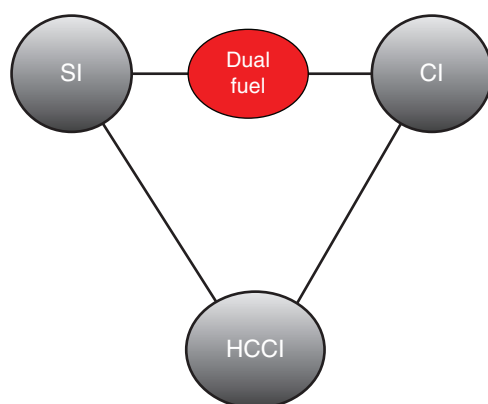


Figure 1.14 Dual-fuel combustion.

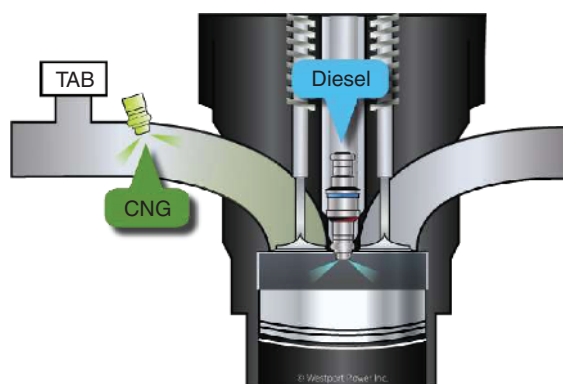


Figure 1.15 Dual fuel with compressed natural gas in the inlet port and diesel injected in the cylinder.

conventional SI combustion. If the end gas is at too high temperature for a longer period, there will be knock. Most often, these types of engines are operated with a very lean mixture, around  $\lambda = 2.0$ , to reduce the risk of knock.

If the engine should be operated with everything from 2% to 100% diesel fuel, there are additional challenges. The same compression ratio is not best suited for 2% and 100% diesel fuel.

#### 1.6.4.1 Chemical Properties

The diesel part of the fuel should ignite as soon as possible, that is, it should have a high cetane number. Unfortunately, that is not often the case for fuels used in large dual-fuel engines often operated on heavy fuel oil.

The premixed part of the fuel should have a high octane number. This is the case for pure methane, but often, natural gas also contains higher hydrocarbons with a lower octane number.

It should be noted that dual fuel with gasoline or fuels with lower octane number are not in commercial use, most likely due to too low separation in autoignition properties from the DI part.

#### 1.6.4.2 Physical Properties

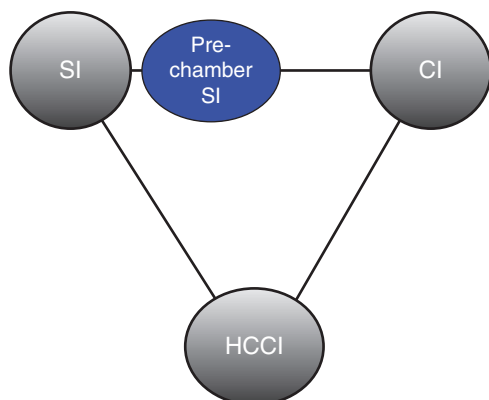
Dual-fuel engines most often are run with gaseous fuels in the inlet, which is not a major problem. The direct injection part can be trickier though. The required fuel amount for igniting the charge can go down to 1–2% of the total fuel. This poses some challenges for the fuel injection system with limited atomization of such small quantities. Sometimes having a third fuel injection system for the diesel pilot only solves this problem.

### 1.6.5

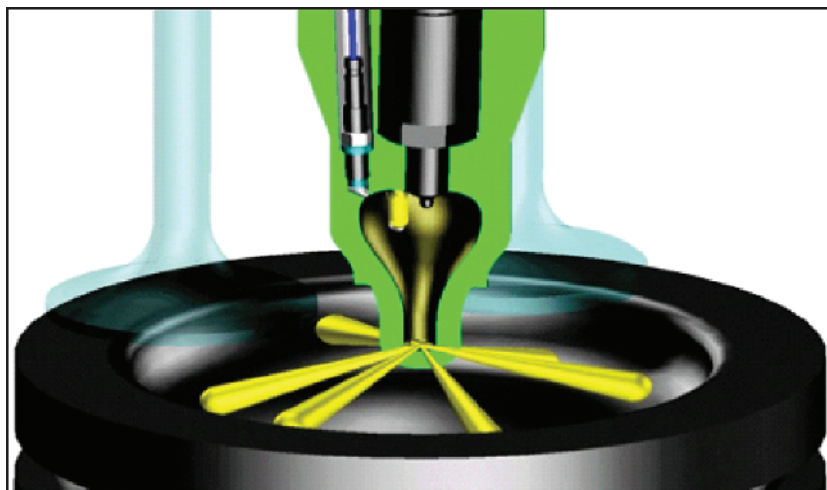
#### Prechamber SI Combustion

A combustion system that is also used in production of large stationary engines for electric power production is a prechamber SI [26]. This is a process that starts with flame propagation, moves to diffusion combustion, and then translates to premixed flame propagation again. Hence, it is located along a line between SI and CI as shown in Figure 1.16. It is more to the left compared to dual fuel, as most of the combustion occurs with flame propagation as in a conventional SI engine.

In these engines, a spark ignites the charge as in a conventional SI engine, but then the similarities end here. In the prechamber engine, a small volume of a richer mixture is used to operate the engine much leaner than what can be done without the prechamber (see Figure 1.17). In the prechamber, a mixture around stoichiometric or even rich is used to achieve a stable ignition and first flame propagation in the prechamber. Then a torch of partially reacted and high-temperature fuel/air mixture shoots out from the chamber, igniting the very lean mixture, around  $\lambda = 2.2$ . Such a lean mixture would not ignite with only a small spark but does so with the powerful torches from the prechamber.



**Figure 1.16** Prechamber SI combustion.



**Figure 1.17** Typical prechamber for large natural gas engine.

Normally, around 1–3% of the compression volume is in the prechamber. The engines using prechamber are rather large with a bore of 340 mm or larger.

In this aspect, the dual fuel and the prechamber SI are similar as a more powerful ignition source compared to a spark plug that is used to ignite the premixed charge. The difference is that the gaseous jets from the prechamber replace the diesel spray from the dual-fuel engine. Hence, prechamber engines are normally operated with only one fuel and hence have less complex and costly fuel systems. The major drawback is the need for spark plugs and hence the need for regular replacements of them, thus limiting the operational hours between services.

#### 1.6.5.1 Chemical Properties

The prechamber engines run very lean to limit the  $\text{NO}_x$  formation. But it is also needed to avoid knock. Prechamber engines are only used for gaseous fuels in production, most often natural gas or similar fuels. That fuel has a very high octane number, but if large fractions of higher hydrocarbons are present in the fuel, heavy knock can occur. There are papers on using prechambers in smaller engines [27]; Honda did produce the CVCC in the 1970s, but since then, it has not been a commercial solution [28]. University of Melbourne presented that hydrogen is especially suitable as a prechamber fuel as the lean limit can be pushed all the way to a value of  $\lambda = 5$  [29]. Its practical use is limited though as hydrogen is scarce and expensive to handle.

#### 1.6.5.2 Physical Properties

Since all prechamber engines use gaseous fuels similar to natural gas, there are not any major issues with the physical properties other than those normally associated with gaseous fuels, such as storage tanks. The prechamber engines can be operated with very low fuel pressure as the fuel to the cylinder is introduced to the inlet and the fuel to the prechamber is fed to it during gas exchange when the in-cylinder pressure is low. A gas compressor that can overcome the inlet pressure, around 3 bar, is hence sufficient.

### 1.6.6

#### Diesel Pilot Combustion

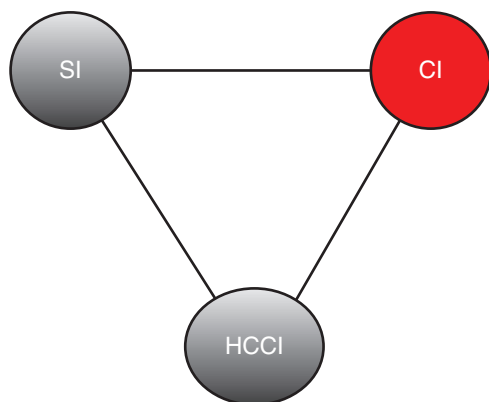
Diesel pilot combustion is a third combustion principle from the large stationary combustion engines [30] but also tried for truck size engines [31]. In this concept, a diesel pilot spray is injected and combustion starts as it normally would in a diesel engine, close to TDC. Then the main fuel, natural gas, is injected into the burning diesel spray. The gas is ignited and then burns with conventional spray-driven diffusion combustion. In a sense, it is not the same though. The fuel is injected in gaseous form, so it is not really a spray but rather a gaseous jet that flows into the combustion chamber. But since mixing and combustion are similar to those in diesel combustion, so are the soot and  $\text{NO}_x$  levels. Figure 1.18 shows the concept as a version of the conventional diesel (CI) process.

#### 1.6.6.1 Chemical Properties

For diesel pilot combustion, the pilot fuel must have a high cetane number but the main fuel does not, as the pilot fuel will ignite it. Hence, the concept can be operated with diesel fuel in the pilot and natural gas as main fuel. The benefit is that the diesel engine fuel efficiency can be maintained as the risk of knock is eliminated and hence a high compression ratio can be used.

#### 1.6.6.2 Physical Properties

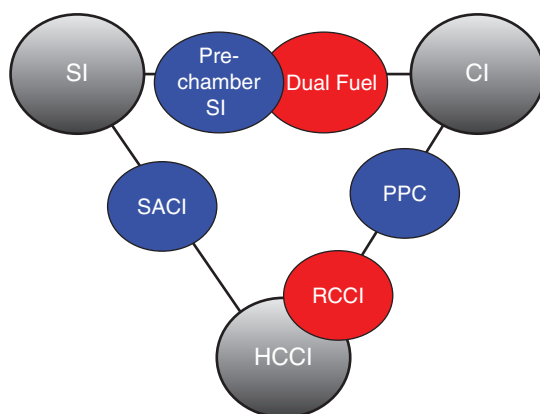
The main problem of diesel pilot combustion is that the fuel, natural gas, is not liquid and hence the parasitic losses for pressurizing the fuel to a level much higher



**Figure 1.18** Diesel pilot ignition.

than the cylinder pressure will be high. With 200 bar in the cylinder during combustion, the fuel pressure must be significantly higher to make the fuel flow in. For a stoichiometric mixture of methane and air, 10% of the cylinder volume is gas. This means that even if the engine is operated at  $\lambda = 2$ , 5% of the cylinder content must be pressurized to a higher level than what will be used during the expansion stroke.

Using high-octane liquid fuels such as alcohols overcomes this major drawback. There have been some trials on replacing natural gas with alcohols [32, 33], and recently (2015), Stena Line in Sweden introduced a diesel/methanol engine for propulsion of a ship [34] and even larger engines are developed for ships [35]. Alcohols have a few benefits: one is that the fuel is much cheaper to pressurize and the second is that no soot is formed with methanol combustion and they are easier to handle compared to gaseous fuels.



**Figure 1.19** The three major combustion modes and five intermediate processes. Blue indicates single fuel and red, dual fuels.



## 1.7

## Summary of Combustion Processes

Figure 1.19 summarizes the different combustion concepts, which are marked within the triangle that spans from SI, CI, and HCCI. For most of the concepts, the marked position is not fixed during the combustion processes. For SACI as an example, it starts as SI and then transitions to HCCI during the process. For most of the processes, the location in the triangle also indicates what to expect in terms of fuel efficiency, emissions of unburned hydrocarbons,  $\text{NO}_x$ , and particulates. This is, however, outside of the scope of this limited introduction.

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