Chapter 1
Introduction

Abbreviations and Symbols

BDC	Bottom dead center
BEV	Battery electric vehicle
CA	Crank angle
CAD	Crank angle degree
CI	Compression ignition
CNG	Compression natural gas
CO	Carbon monoxide
CRDI	Common rail direct injection
DDFS	Dual direct injection fuel stratification
DI	Direct injection
DISI	Direct injection spark ignition
DME	Dimethyl ether
DPF	Diesel particulate filter
ECE	External combustion engine
ECU	Electronic control unit
EGR	Exhaust gas recirculation
EMS	Engine management system
FMEP	Friction mean effective pressure
GCI	Gasoline compression ignition
GDCI	Gasoline direct injection compression ignition
GDI	Gasoline direct injection
HC	Unburned hydrocarbon
HCCI	Homogeneous charge compression ignition
HEV	Hybrid electric vehicle
HTC	High-temperature combustion
IC	Internal combustion
ICE	Internal combustion engine

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IMEP	Indicated mean effective pressure
IT	Ignition timing
LPG	Liquefied petroleum gas
LTC	Low-temperature combustion
MBT	Maximum brake torque
N	Engine speed
NO_x	Nitrogen oxides (NO and NO_2)
OBD	Onboard diagnostics
OEM	Original equipment manufacturer
PCCI	Premixed charge compression ignition
PFI	Port fuel injection
PM	Particulate matter
PMEP	Pumping mean effective pressure
PPC	Partially premixed combustion
RCCI	Reactivity controlled compression ignition
SACI	Spark-assisted compression ignition
SCCI	Stratified charge compression ignition
SCR	Selective catalytic reduction
SI	Spark ignition
SN	Swirl number
SOC	Start of combustion
TBI	Throttle body injection
TDC	Top dead center
VCR	Variable compression ratio
V_d	Displacement volume
VGT	Variable geometry turbocharger
VVT	Variable valve timing
η	Efficiency
ρ	Density
φ	Equivalence ratio

1.1 Introduction to Reciprocating Engines

Modern society is to a large extent built on the transportation of both people and goods. Transport is almost entirely (>99.9%) powered by internal combustion engines (ICEs), which typically burn fossil petroleum-based liquid fuels. Reciprocating internal combustion engines mainly perform land and marine transportation, and air transport is mainly powered by jet engines [1]. Several factors are responsible for wide-range utilization of liquid fuels for transportation including (1) high energy density, (2) easy transportation and storage, and (3) large global infrastructure developed over time. Reciprocating ICEs are well accepted and the most significant source of energy since the last century due to their superior performance, robustness, controllability, durability, and absence of other viable alternatives.

The electric vehicle and fuel cell-operated vehicles are considered as an alternative to internal combustion engines for automotive applications. Presently, there is much interest in electric vehicles, and several governments are supporting this initiative. However, battery electric vehicles (BEVs) seem to have zero local pollution, but they can actually have higher total greenhouse gas emission (CO_2) than similar reciprocating engine-operated vehicle [1–3]. Thus, promoting BEVs can be counterproductive until the electricity production is sufficiently decarbonized [2]. The hybrid electric vehicle (HEV) is a better option in terms of CO_2 emission reduction rather instead of the BEVs [1, 3]. Even if electricity generation is decarbonized, the BEVs have a very significant impact on human toxicity, freshwater eco-toxicity, and freshwater eutrophication, mainly caused by the production of metals required for batteries [1, 4].

Worldwide several initiatives are being developed and proposed for implementation including alternative fuels and alternative combustion mode in reciprocating engines, and BEVs with green power alternatives. However, the importance of a particular alternative is different at different times and in a different country or region. Kalghatgi [1] proposed that the evolution of transportation energy is dependent on the complex interplay between several drivers for the change, which is illustrated in Fig. 1.1. Various factors affecting the energy policy include the energy security and local pollution concerns, climate change concerns, support to farmers and increase in rural employment, and aspiration for leadership in newer technologies. Transport policy should be based on a balanced approach using all available technologies, considering local and global environmental and greenhouse gas impacts, security of supply, and social, economic, political, and ethical impacts [1]. Reliable projections indicate that even by 2040 about 90% of transportation energy will come from combustion engines operated on petroleum fuels. The reciprocating combustion engines will continue to power transport (especially

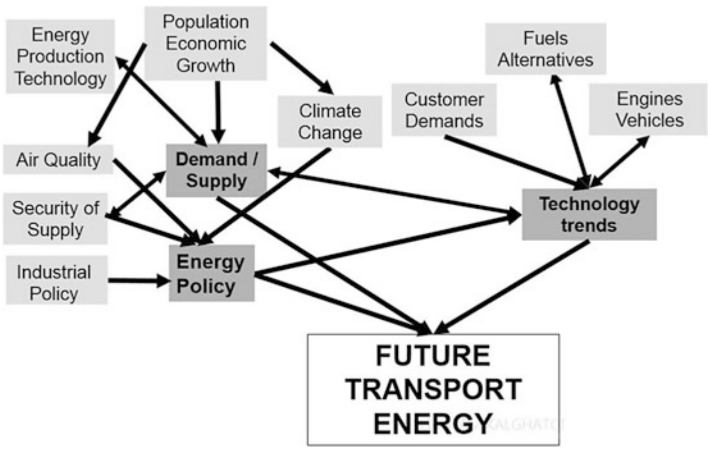


Fig. 1.1 The evolution of future transport energy system [1]

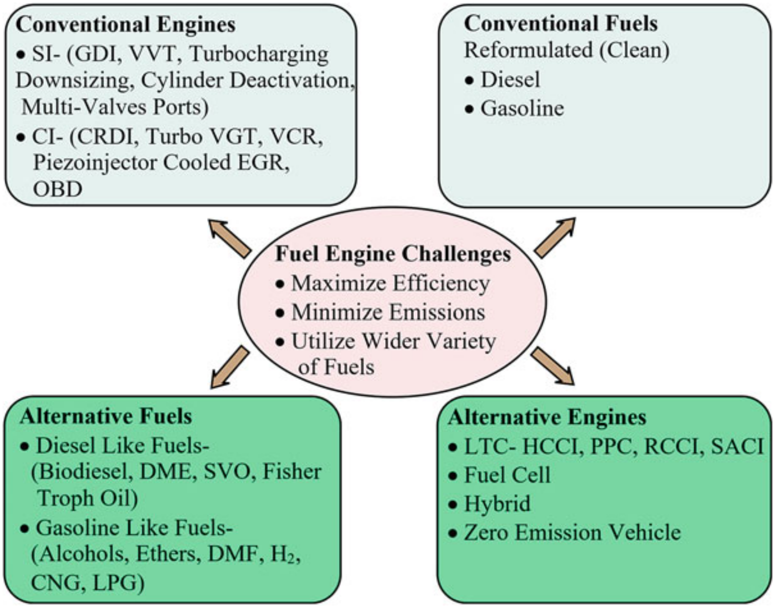


Fig. 1.2 Fuel-engine challenges and possible technological solutions [5]

commercial sector) by a large extent for several decades, and it will also continue to improve [1].

With the increasing concern on achieving a substantial improvement in automotive fuel economy, automotive engineers are striving to develop engines that provide both significant reduction in the brake-specific fuel consumption (BSFC) and compliance with future stringent emission requirements. To meet these requirements, four possible approaches (improving conventional engines and fuels or using alternative fuels and engines) and their combination are proposed. The major fuel-engine challenges are summarized in Fig. 1.2 along with possible technological solutions. The alternative fuels can be used in conventional engines with the latest technologies as well as in advanced low-temperature combustion concepts [5]. There are plenty of options for improving the reciprocating ICEs (Fig. 1.2). Furthermore, to meet the emissions legislation, aftertreatment technologies can be utilized along with improved engines.

At the automotive vehicle level, several efforts are made to operate them as efficiently and cleanly as possible. The major trends are downsizing, hybridization, driver support system, and newer infrastructure development [6]. To meet the emission legislation requirement, downsizing is one of the options. Downsizing includes two approaches: (1) developing smaller and lightweight vehicles leading to lower fuel consumption and (2) developing smaller engines with lower fuel consumption (often using turbocharger). Hybridization of the vehicle is another technique used to improve fuel consumption. Hybrid vehicles use the battery as well as smaller size ICE. Several factors contribute to efficiency improvements in a hybrid vehicle that include avoiding transients, running the engine at optimal conditions, and regenerative braking. Hybrid vehicles are mainly relevant to spark ignition engines in stop/start city driving conditions [1]. Fuel consumption and quantity of emissions are highly governed by the driving of the vehicles. There is a strong interest in developing a system which assists the driver in optimal driving which leads to fuel saving. A driver support system proposes the speed and gear selection to the driver and can also evaluate and educate the driver [6]. The technological developments such as GPS and map database can be beneficial for optimal driving in all circumstances, which include informing traffic conditions, roadside information, weather, etc. These developments can potentially lead to fuel saving and improving the environmental conditions by reducing the exhaust emissions.

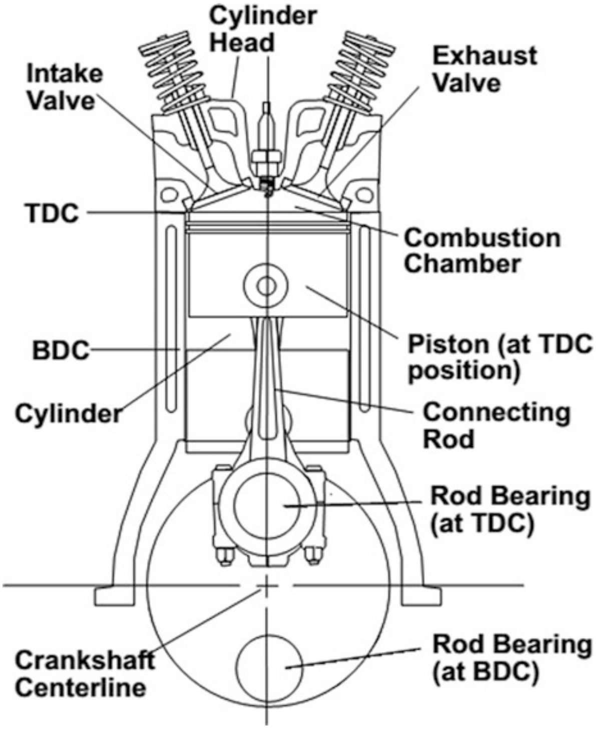
1.1.1 Reciprocating Engine Fundamentals

An engine is a device to convert the fuel energy into mechanical energy (useful work). In engines, chemical energy bounded in fuel is converted into heat energy by combustion, and the heat energy is converted to useful work by the driving mechanism. The driving mechanism can be reciprocating or rotary. There are two basic types of engines: (1) internal combustion engines (ICEs) and (2) external combustion engines (ECEs). In ICEs, the combustion products act as a working fluid, while in ECEs, the combustion products transfer heat (using heat transfer device) to another fluid that act as working fluid. Presently, most of the engines used for transportation are internal combustion type. Reciprocating ICE offers several advantages over the ECE (steam turbines) such as (1) mechanical simplicity (due to absence of heat exchangers (boiler and condenser) in the path of working fluid); (2) works as lower average temperature than the maximum temperature of working fluid (high temperature occurs only of fraction of cycle), leading to possibility of employing higher working fluid temperature for higher efficiency; (3) lower weight-to-power ratio of the engine; and (4) possibility to develop ICE of very small power output [7].

There are four general types of internal combustion engines, namely, (1) four-stroke cycles, (2) two-stroke cycles, (3) rotary engines, and (4) continuous combustion gas turbine engines [8]. Reciprocating piston engines operate on four- or two-stroke combustion cycles. The basic components of the reciprocating piston engine are shown in Fig. 1.3. Reciprocating engine is characterized by a slider-crank mechanism which converts the reciprocating motion of the piston into rotating motion of crankshaft [9].

The piston moves back and forth in a cyclic manner in the engine cylinder and transmits power to the crankshaft through connecting rod. The piston speed becomes zero at topmost position, before its direction changes, and this crankshaft position is known as top dead center (TDC). Similarly, the bottommost position of the piston is known as bottom dead center (BDC). Further details about each of the engine components can be found in any textbook [7, 9, 10].

Fig. 1.3 Cross section of a spark ignition engine [9]



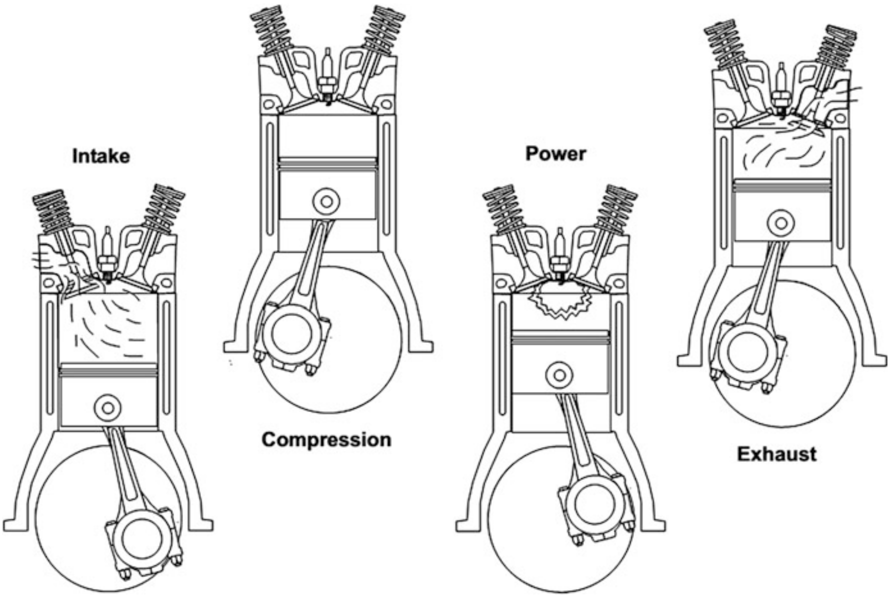


Fig. 1.4 A four-stroke reciprocating internal combustion engine cycle [9]

Presently, most automotive vehicles used for road transport operates on four-stroke cycles. Figure 1.4 illustrates a four-stroke reciprocating internal combustion engine cycle. A stroke consists of piston motion from TDC to BDC position, and during this period, crankshaft travels 180° (half round). Thus, four-stroke cycle completes in two rounds (720°) of crankshaft motion. Four-stroke cycle consists of intake, compression, expansion (power), and exhaust stroke. During the intake stroke, the piston moves from TDC to BDC position and inducts air and fuel (in SI engine) or only air (CI engines) in the cylinder. The intake valve opens slightly before TDC position and closes after BDC position to improve the volumetric efficiency or increase the mass of inducted air [10]. In compression stroke, the piston moves from BDC to TDC positions and increasing the pressure and temperature of the gases in the cylinder. Toward the end of compression stroke, combustion starts in the cylinder either by autoignition (CI engine) or by spark (SI engine). High-pressure and high-temperature gases expand during the power stroke, and useful work is extracted at the crankshaft. In the exhaust stroke, burned gases are pushed out of the cylinder past the exhaust valve as the piston moves from BDC to TDC position. The exhaust valves typically open slightly before the start of exhaust stroke leading to exhaust blowdown (rapid decrease of pressure in the cylinder). This process reduces the work done by the piston during the exhaust stroke. This cycle repeats again, and the piston starts moving again from TDC to BDC position in the next cycle.

The reciprocating combustion engines can be classified in several ways including basic design (reciprocating or rotary), types of ignition and combustion (SI and CI; homogeneous and heterogeneous), working cycle (four- and two-stroke), air intake (boosting) process (naturally aspirated, turbo-charged, super-charged), valve locations (overhead, valve in block, valve in head), position and number of cylinders (single-cylinder, in-line, V engine, opposed cylinder engine, opposed piston engine, radial engine), cooling (air cooled, liquid cooled, water cooled), fuel used (diesel, gasoline, CNG, DME, ethanol, methanol), and method of fuel input (DI, PFI, TBI) [7, 10].

For designing the new engine, four basic questions need to be answered to define its characteristics. The questions are as follows: (1) what is the required displacement volume of engine to produce desired power?, (2) how many cylinders will the displacement be distributed?, (3) what should be the bore and stroke of each cylinder?, and (4) what will be the configuration of the engine?.

The displacement volume required for the engine is calculated by determining the air required corresponding to the maximum power produced. For estimation of air requirement, the full-load curve of the engine along with expected specific fuel consumption needs to be determined. In spark ignition engine, the work output is typically limited by displacement over its entire speed range while in diesel engine displacement limited only at maximum torque condition [9]. Based on the expected maximum power and specific fuel consumption, the mass flow rate of fuel can be computed. Air-fuel ratio of engine operation depends on the type of engine (SI or CI), and the mass flow rate of air is calculated by considering the air-fuel ratio. The displacement volume required for the engine can be calculated using Eq. (1.1) at engine speed (N) corresponding to maximum power output.

$$V_{\mathrm{d}} = \frac{\dot{m}_{\mathrm{air}}}{\rho_{\mathrm{air}} \cdot \left(\frac{N}{X}\right) \cdot \eta_{\mathrm{vol}}} \tag{1.1}$$

where ρ is the air density at the inlet, η_{vol} is the expected volumetric efficiency, and the value of X is 2 for four-stroke engine and 1 for a two-stroke engine. Detailed discussion on displacement volume determination can be found in reference [9].

Typically, when estimated displacement volume exceeds the 500 cm^3 for high-speed engines, the multicylinder engines are preferred. In multicylinder engines, the number of cylinders and their configuration is two major parameters from a design perspective. Several factors affect the selection of the number of cylinders including cost and complexity, engine balancing and vibration, surface-to-volume ratio of the combustion chamber, engine speed requirements (mean piston speed), and engine breathing characteristics (pumping losses) [9]. The configuration of the engine is governed by balancing of forces and packaging consideration of engine. Typically, the in-line configuration is used up to four cylinders, and V configuration is used for greater than four cylinders.

After deciding the displacement volume per cylinder, the bore-to-stroke ratio is an important parameter that also defines the shape of the combustion chamber. The major factors affected by bore-to-stroke ratio are heat transfer losses (surface-to-volume ratio near TDC position), piston speed, and valve flow area (decides the size of valves). These factors need to be optimized for determination of bore-to-stroke ratio [9]. Figure 1.5 illustrates the trade-offs associated with the three parameters for the determination of optimal bore-to-stroke ratio (please note the scales of the figure). Engine bore must be made sufficiently large to maintain the mean piston speed lower than the design target and to minimize pressure drop across the valves.

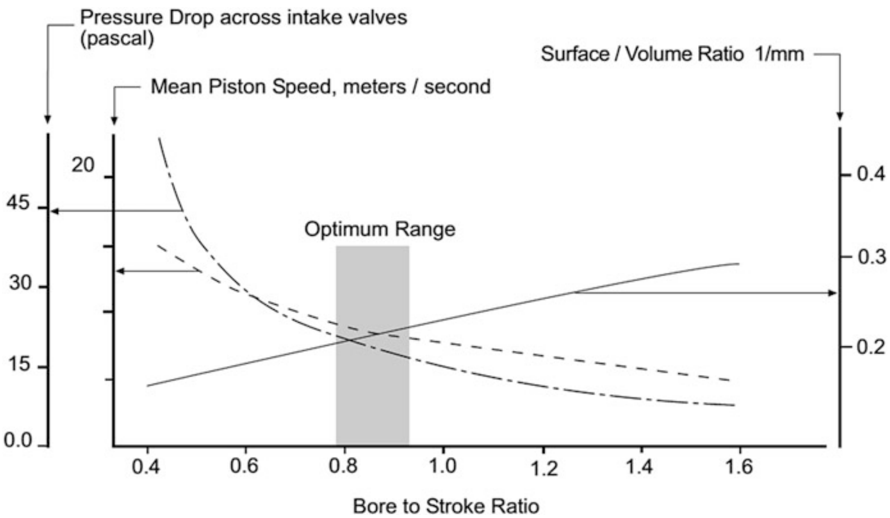


Fig. 1.5 Estimation of optimum bore-to-stroke ratio for a particular engine [9]

The resulting bore-to-stroke ratio is dependent on the design criteria for the specific engine. The bore-to-stroke ratio is typically just below unity to provide the best balance of performance in automotive applications where fuel efficiency and emission requirements dominate. In high-performance applications such as racing engine (racing rules often limit displacement), the bore-to-stroke ratio tends to be larger to maximize breathing and operate at acceptable piston speeds during engine operation at high speeds.

1.1.2 Spark Ignition Engine

The spark ignition (SI) engines are used in applications where higher engine speed and the lightweight engine are required. The SI engines operate at high engine speeds, and thus, the reciprocating components are made lightweight to reduce the inertia forces at high speeds. The SI engines can be designed for two-stroke as well as four-stroke cycles. Due to the urban air pollution concerns, four-stroke engines are largely used for small and large power requirements.

In conventional SI engines, the premixed homogeneous mixture of fuel and air is ignited by a positive source of ignition. The positive source of ignition is often an electric discharge spark plug although alternatives such as laser-induced ignition, corona ignition, etc. are also existing [10]. For the preparation of premixed fuel-air mixture, fuel is introduced in the intake manifold with a carburetor or fuel injection system. Ideally, the stoichiometric air-fuel ratio is maintained in conventional SI engines at all the engine load conditions. Thus, engine load is varied by changing the amount of air using the throttle in the inlet manifold. The spark initiates a flame kernel, which grows with time, and a turbulent flame propagates in the combustion chamber and consumes the entire charge. The flame reaches the cylinder walls by consuming all the charge and extinguishes. Typically, this process is called normal spark ignition combustion. Figure 1.6 illustrates the ignition and flame propagation in SI engine using the conventional spark (left) and corona ignition (right). The first image corresponds to the point of ignition, and subsequent images are at a constant distance of 2.5° between frames. Corona ignition creates a significantly larger high-intensity plasma ignition source, which spreads throughout the combustion chamber early when compared to conventional spark ignition systems.

In conventional spark ignition case (Fig. 1.6), spark flashover at ignition timing ($-14.3°$ CA) and bright light from the spark is reflected. After 2.5° CA of spark, an initial hemispheric flame kernel can be detected that further moves slightly off-center in the chamber due to turbulence. At $-6.8°$ CA, i.e., after 7.5° of ignition, the large-scale wrinkling of the flame front surface increases the active flame front surface and thus enlarges and accelerates its propagation, increasing light intensity due to a magnified flame volume and higher pressure/temperature. However, in the corona ignition system, after 2.5° of ignition, five individual kernels developed around streamers, and the relatively larger area is enflamed as flame propagates [11].

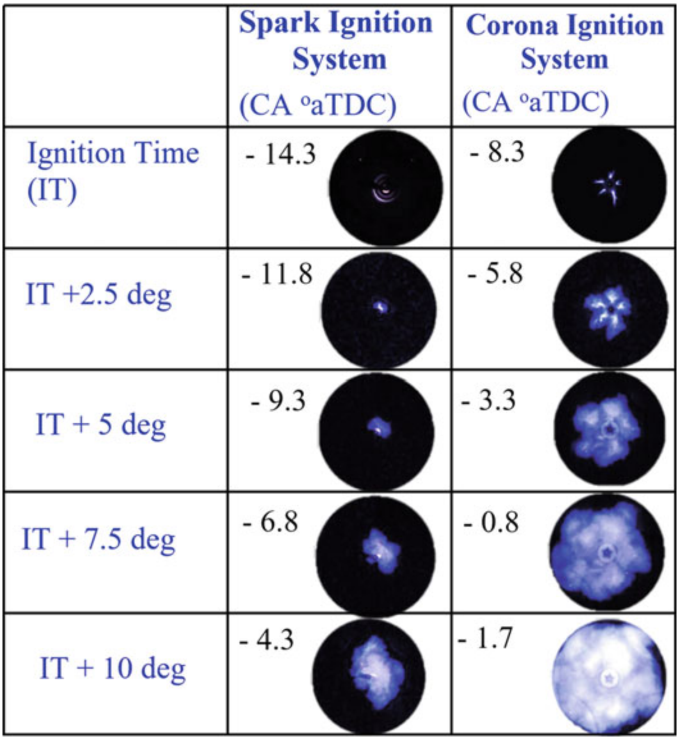


Fig. 1.6 Flame propagation in SI Engine [11]

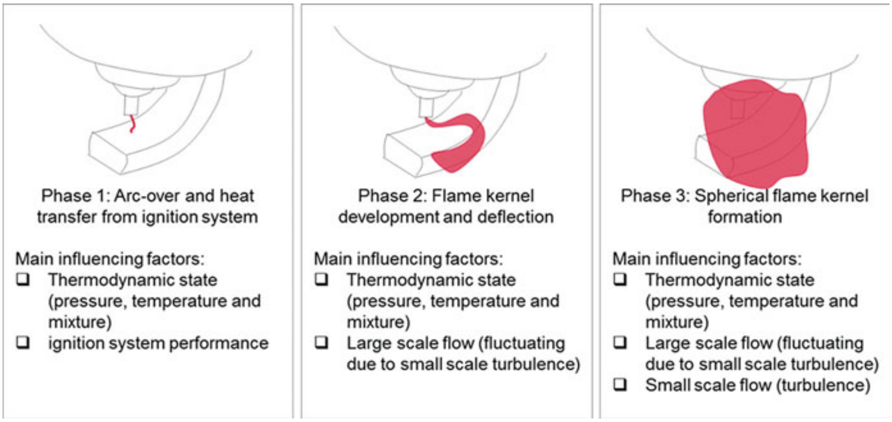


Fig. 1.7 Illustration of flame kernel development phases [12]

The charge motion around the spark plug and charge composition at the time of spark discharge is decisive for the flame development and subsequent flame propagation [5]. The process from ignition to early combustion can be divided into different phases as illustrated in Fig. 1.7. The arc-over phase is primarily affected

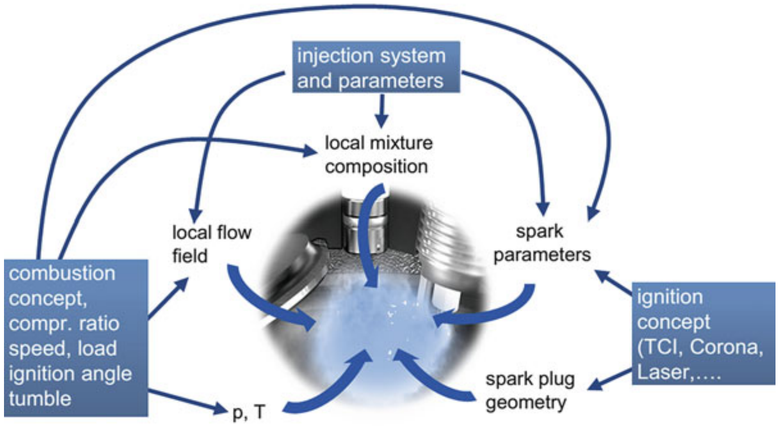


Fig. 1.8 Factors affecting the flame kernel formation in SI engine [13]

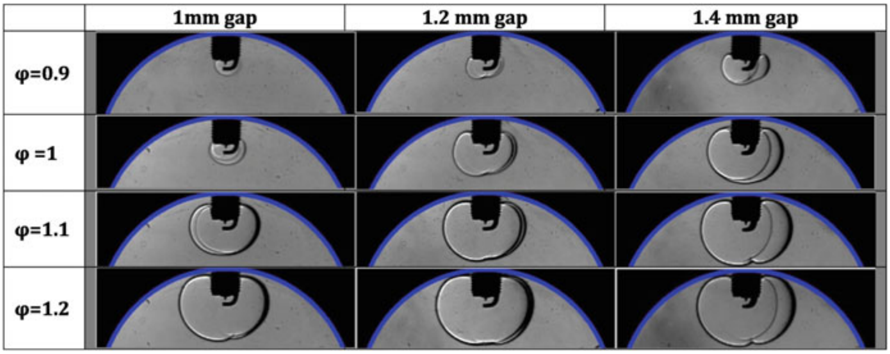


Fig. 1.9 Comparison of typical flame growth for different spark plug gaps and charge composition [14]

by the performance of the ignition system and the thermodynamic state of the charge. In this phase, chemical reactions result into an early flame kernel development. Approaching charge flow will deflect such a flame structure, but the flame kernel is too small to be influenced by small-scale turbulence. However, turbulence will affect the deflection indirectly by having an influence on large-scale flow. This deflection affects the volume activated by energy supplied from the ignition system. The early flame kernel develops to a shape which could be approximated as a sphere. The transition from a laminar to a turbulent flame occurs during this phase [12]. The flame kernel formation is affected by several factors such as local flow field and mixture composition, spark parameters, spark plug geometry, etc. which are illustrated in Fig. 1.8.

Figure 1.9 shows the effect of spark plug gap and charge composition on flame growth at 6 ms after the start of flame kernel initiation. The figures show that the difference in flame kernel growth is high between the three spark plug gaps for lean

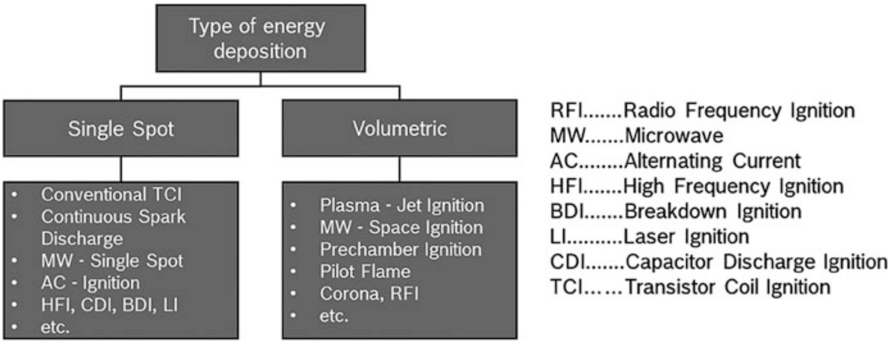


Fig. 1.10 Ignition systems in SI engines [13]

and stoichiometric conditions and flame kernel growth is not significantly affected by the spark plug gap at rich condition ($\varphi = 1.2$). Figure 1.9 also depicts that spark plug gap of 1.2 and 1.4 mm has a significantly larger flame kernel in comparison to gap 1 mm. At the beginning of the flame kernel initiation and up to 1 ms from the start of spark, the difference in the flame kernel size between different spark plug gaps is relatively small especially for larger gaps such as 1.2 and 1.4 mm. It was demonstrated that at the beginning of the flame kernel initiation and up to 1 ms from the spark timing, the difference in the flame kernel is small, but as the time after ignition progresses, the larger spark plug gaps produce a significantly larger flame kernel areas in comparison to the spark plug gap of 1 mm [14]. Thus, spark parameter and charge conditions affect the initial flame development, which leads to variations in flame propagation and heat release in SI engine.

Figure 1.10 summarizes the different types of ignition systems for SI engines based on how the flame kernel volume is activated during the ignition process. The “single spot ignition” means the deposition of ignition energy within a fixed location in the cylinder, and the energy distribution is not actively controlled by the ignition system itself [13]. However, a spark volume is characterized by the spark channel and even enlarged by charge motion (e.g., deflecting the spark). The deflection of spark leads to activation of flame volume, which results in potential to ignite the dilute mixtures in gasoline engines. Volumetric ignition concepts such as corona or plasma-jet ignition systems actively provide a flame kernel volume even in low charge motion conditions. The ignition systems can also be classified based on how the primary energy for operating the circuit is made available, i.e., battery ignition systems and magneto ignition systems [7].

The overall combustion process in SI engines can be considered to consist of three phases: (1) flame development phase, (2) flame propagation phase, and (3) flame termination phase [10, 15]. On the firing of spark, plasma discharge ignites the air-fuel mixture in a small volume between and around the spark plug gap. Sustained combustion reactions result in the development of a turbulent flame which propagates outward from the spark plug. The combustion reactions depend on both temperature and pressure, the nature of the fuel and proportion of the exhaust residual gas. At first, the flame propagates very slow, and very small pressure rise in the cylinder pressure is noticed until the flame kernel grows into a fully developed flame. A fully developed flame becomes turbulent and like a spherical wave propagates at high speed across the combustion chamber. A variety of fluid motions such as swirl, tumble and squish, and turbulence is generated in the cylinder by proper design of intake ports and the combustion chamber shape to accelerate the combustion process (increase flame speed). The turbulent flame speed depends on the turbulence intensity and is several times higher than the laminar flame speed of a flame propagating through a fuel-air mixture of similar composition and thermodynamic state. The combustion rate depends on various factors such as temperature and pressure of charge, mixture strength, residual gas fraction or charge dilution by external EGR, turbulence, and fuel chemistry. With the progress of flame propagation, cylinder pressure rises, and the unburned mixture in the front of the flame gets compressed. The pressure and temperature of unburned charge rise till the peak pressure is achieved in the cylinder. The unburned charge can autoignite if it remains at sufficiently high temperatures for a significant time period. The autoignition of charge leads to knocking, which is a different (abnormal) process than the combustion by normal flame propagation. The detailed discussion on heat release rate and knocking is provided in Chaps. 7 and 9, respectively.

Well-mixed (normally called homogeneous) stoichiometric operation is the leading combustion mode for gasoline SI engines used for automotive applications. However, several factors limit the engine efficiency of stoichiometric operation (without exhaust gas recirculation, EGR) particularly at low and intermediate loads [16]. First, throttling used for load control increases the pumping loss at part-load conditions. Second, high combustion temperatures result in both high heat transfer losses and unfavorable thermodynamic properties of the combustion products (lower γ). The lower ratio of specific heats (γ) leads to reduction of the work extraction efficiency. Third, the stoichiometric combustion is not able to fully complete near TDC due to dissociation of CO_2 in the hot O_2-depleted gases. A limiting 10–90% burn duration of 30 CAD was found at the location of peak efficiency irrespective of operating conditions, which means that the 10–90% burn duration determines efficiency and that combustion deteriorates significantly beyond this 30 CAD offsetting any efficiency gains from thermodynamics effects, reduced pumping work, or reduced heat transfer losses [17].

To overcome some of the demerits of well-mixed stoichiometric SI engines, gasoline direct injection (GDI)-based engines are developed [18]. The direct injection spark ignition (DISI) engines can operate unthrottled as well as on the leaner mixture. In this engine, the fuel spray plume is injected directly into the cylinder, generating a fuel-air mixture with an ignitable composition at the spark gap at the time of ignition. The GDI engines have improved fuel economy over PFI engines due to a substantial reduction in pumping loss, reduction in heat loss, higher compression ratio, increased volumetric efficiency, and less acceleration-enrichment requirement [18].

The fuel-lean operation can improve engine efficiency. However, the challenge is to maintain stable and efficient combustion despite a reduction of flame speeds in

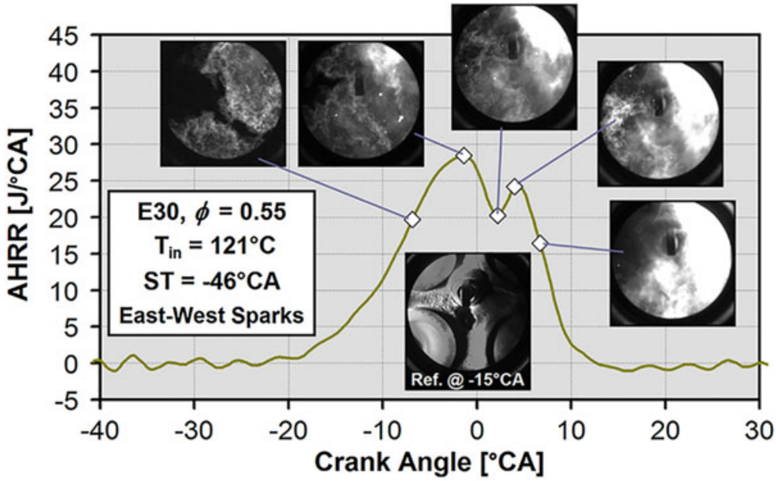


Fig. 1.11 Mixed-mode operation in DISI at lean E30 operation with two gable-mounted spark plugs without swirl [19]

fuel-lean mixtures. The reduction of flame speeds becomes a particular problem from the perspective of ignition, and effective flame spreads throughout the charge. During lean engine operation, high combustion efficiency, shorter burn duration, and faster inflammation provide higher thermal efficiency [16]. It was also found that short delay from spark to main combustion (i.e., fast inflammation) reduces cyclic variations of the deflagration-based combustion for both lean and dilute operation. To obtain short combustion duration, several possibilities are explored including multi-spark plug and advanced volumetric ignition systems such as corona, increasing the turbulence, intake heating, etc.

To maintain a sufficiently short burn duration, mixed-mode combustion is proposed for $\phi < 0.6$ in DISI engine [19]. In the mixed-mode combustion, a combination of deflagration and end-gas autoignition occurs, and the end-gas reactants reach to the point of autoignition by the compression of pressure rise due to the deflagration-based combustion [16, 19]. Figure 1.11 illustrates the mixed-mode combustion process using the flame imaging of transition from turbulent deflagration to end-gas autoignition. The mode can ensure sufficiently short burn duration for ultra-lean SI operation such that relative efficiency gain of roughly 20% can be realized [19]. The mixed-mode combustion is conceptually the same as spark-assisted compression ignition (SACI). However, in this mode, a larger fraction of the combustion is based on flame propagation, and the level of internal residuals (or external EGR) is much lower. This combustion mode introduces a noise concern, and it should be restricted to very lean ($\phi \leq 0.55$) or highly dilute ($\phi_m \leq 0.65$) operation while ensuring very repeatable deflagration-based combustion to avoid occasional knocking cycles [16].

1.1.3 Compression Ignition Engine

A four-stroke compression ignition engine also experiences the four strokes of intake, compression, expansion, and exhaust similar to four-stroke gasoline engines. However, due to differences in fuel characteristics, the formation and ignition of fuel-air mixture differ from the gasoline (spark-ignition) engines. Diesel fuel has a higher viscosity, and lower autoignition temperature than gasoline, and it is also not susceptible to vaporizing.

In a diesel engine, only air is inducted during the intake stroke and compressed in the compression stroke that increases the pressure and temperature of the air. Toward the end of compression stroke, diesel is injected into the cylinder through a fuel injector in the hot and dense air. Fuel is injected at 400–2000 bar pressure through three to eight hole injector nozzle depending on the engine design and size. In a diesel engine, the load is controlled by varying the amount of fuel, and inlet air quantity remains the same at particular engine speed in a naturally aspirated engine. Injected fuel in the cylinder atomizes into fine droplets, and air entrainment takes place in the fuel spray. The fuel droplet evaporates and mixes with air. The combustion initiates by autoignition of premixed charge. The time period between the start of injection and start of combustion (SOC) is defined as ignition delay. After SOC, the flame spreads rapidly in the combustible mixture prepared during ignition delay period. This phase of diesel combustion is typically known as premixed combustion phase. At higher engine loads, fuel injection continues even after the start of combustion. The combustion of fuel injected after SOC depends on how fast it gets evaporated and mixed with air. During this period, turbulent diffusion process governs the fuel-air mixing and combustion rate. Thus, this phase of combustion is known as mixing controlled or diffusion combustion phase [10].

The fuel distribution in the cylinder is not uniform, and local fuel-fuel ratio also varies from zero (only pure air present) to the infinity (only fuel present). Images in Fig. 1.12 illustrate the heterogeneous nature of diesel combustion and diffusion flame development and its progress. At high combustion temperatures (2000–2500 °C), carbon particles in the diffusion flame have sufficient luminosity and appear as a yellow region, and radiation from particle varies as flames cool down [15]. The combustion has just initiated in the first image (Fig. 1.12), and all the fuel sprays have developed diffusion flame in the second image. Subsequent two combustion images have the highest intensity, depending on the heat release rate. The heat release rate is also affected by the different swirl number (SN) as illustrated in Fig. 1.12 [20]. The premixed combustion phase is not observed in modern diesel engines with high fuel injection pressures (Fig. 1.12), which typically occurs in the low-pressure mechanical fuel injection (see details in Chap. 6). Based on engine load, only a very small portion of premixed heat release appears in modern diesel engines due to the accelerated mixing process and reduced ignition delay period because of high fuel injection pressures [5].

Figure 1.13 schematically illustrates a modern diesel engine showing different systems. Diesel engines used microprocessor control with direct fuel injection and wastegate turbochargers around the year 1989. Later, EGR with both high pressure

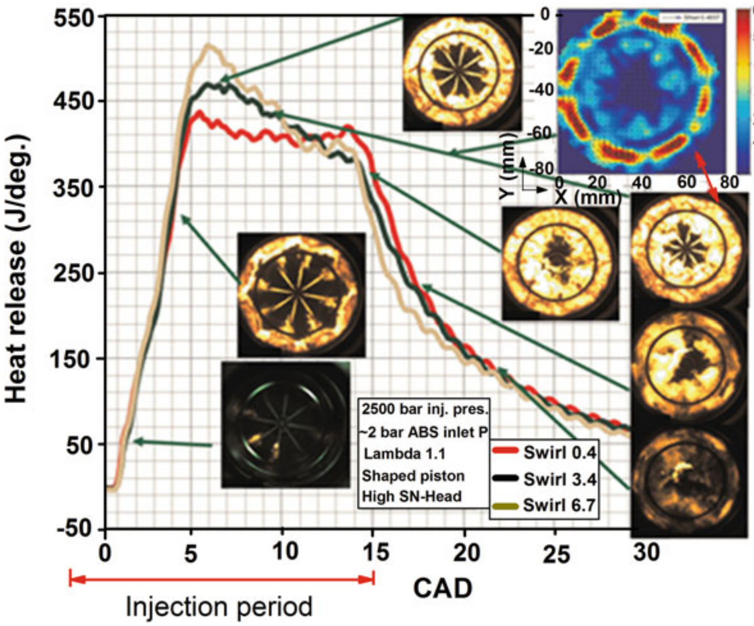


Fig. 1.12 Heat release rate and flame luminescence images of the combustion chamber in a modern diesel engine at a load of 20 bar IMEP, 2500 bar fuel injection pressure and 1000 rpm [20]

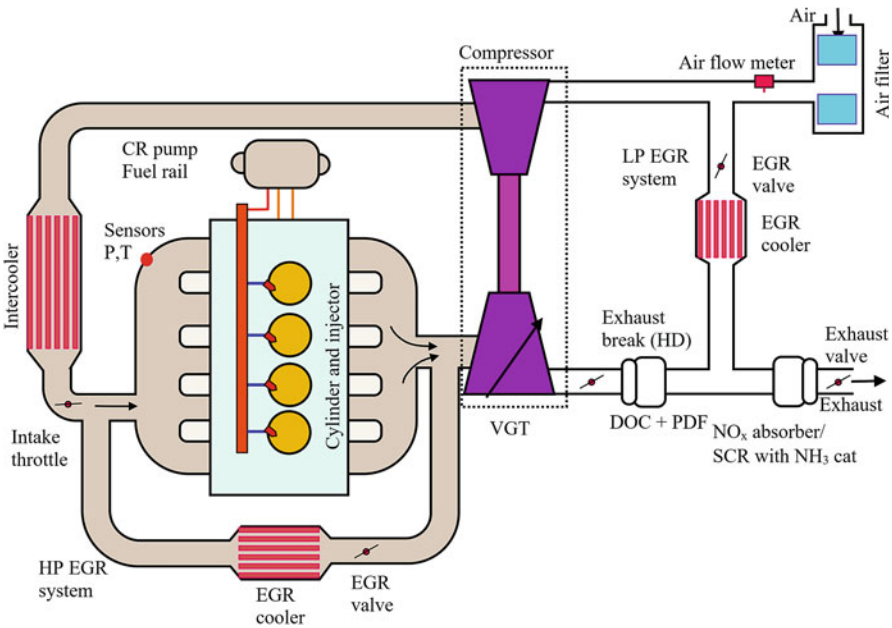


Fig. 1.13 Schematic diagram of a modern diesel engine showing different systems (adapted from [6])

(HP) and low pressure (LP), oxidation catalyst, and turbochargers with variable geometry (VGT) are developed. Presently diesel engines are equipped with common rail diesel injection (CRDI) system with very high fuel injection pressure, piezo injectors, multiple fuel injection capability, high EGR rates, twin turbochargers or VGT chargers, and recent aftertreatment technologies (DPF, SCR, DeNO$_x$-catalyst) as illustrated in Fig. 1.13 [5].

The combustion is a very complex process in the diesel engines, where fuel injection, atomization, vaporization, mixing, as well as combustion can simultaneously occur in the cylinder. Vaporization and mixing are the slowest processes and thus control the combustion rate. The speed of diesel combustion is limited by the mixing between the injected fuel and the air in the cylinder, and thus, it also limits the maximum engine operating speed. Diesel engine characteristics be can be roughly classified into six groups: (1) fuel injection characteristics, (2) fuel spray characteristics, (3) combustion characteristics, (4) engine performance characteristics, (5) ecology characteristics, and (6) economy characteristics [21]. Figure 1.14 shows the diesel engine characteristics and their interrelationship. All these characteristics are dependent on the most basic parameters such as fuel type or injection system type and on several process characteristics such as the injection process, fuel spray development, atomization, mixture fuel/air formation, ignition and combustion, etc. [21]. Figure 1.14 depicts that fuel injections' characteristics significantly affect the performance, combustion, and emissions' characteristics of the diesel engine.

The most important injection characteristics are fuel injection pressure, injection duration, injection timing, and injection rate history [21]. All these parameters affect the diesel engine characteristics (Fig. 1.14). In conventional diesel engines, fuel is injected through a mechanical fuel injection system, which is illustrated in Fig. 1.15. In this system, the pump pressurizes the fuel which reaches to the fuel injector via a

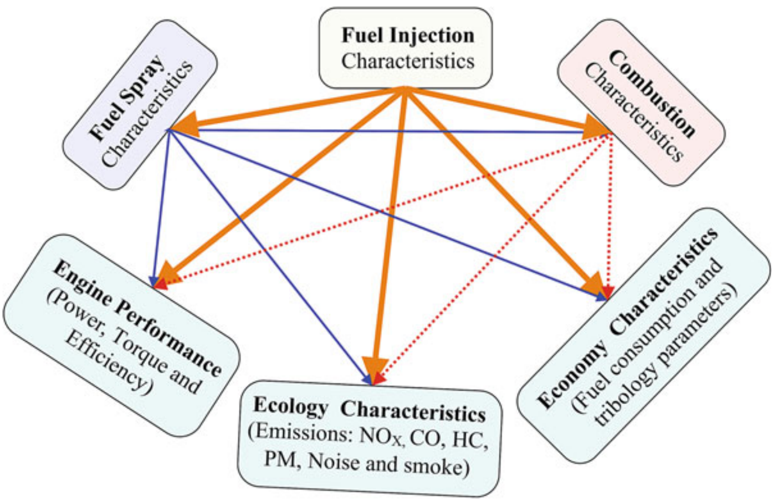


Fig. 1.14 Diesel engine characteristics and their interrelationships [21]

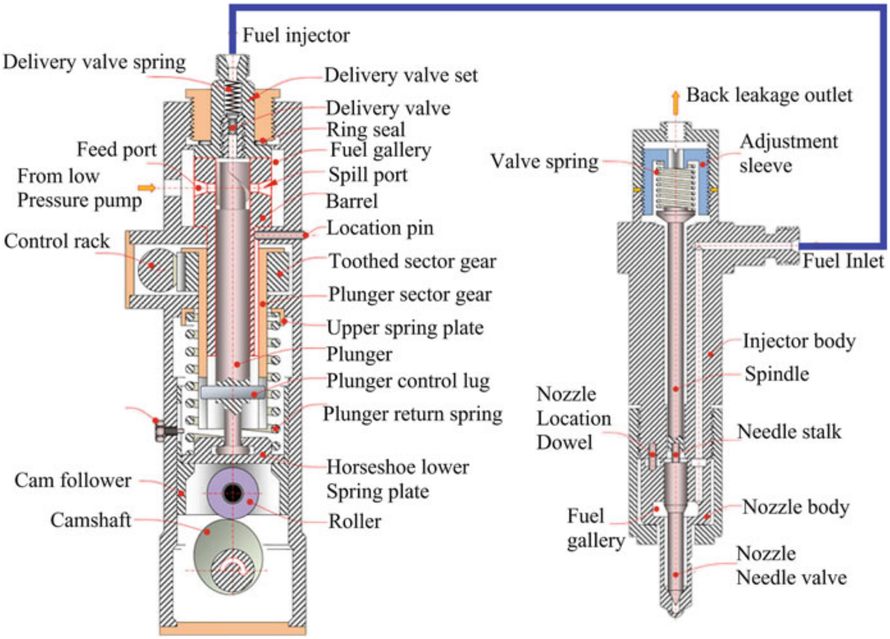


Fig. 1.15 Schematic diagram of mechanical fuel injection system in diesel engine [21]

high-pressure line. The fuel injector is a spring-loaded system; when the line pressure overcomes the spring force, the fuel injection occurs in the cylinder. In this case, the pressure is not constant throughout the engine cycles, and high pressure is generated when fuel injection is required. The timing of fuel injection is governed by the cam, which drives the plunger of the fuel pump. The quantity of fuel injected is governed by plunger lug, which rotates the plunger and the orientation of helix changes. In the mechanical fuel injection system, the fuel injection pressure is relatively lower (~400 bar or less). In a modern diesel engine, electronic fuel injection systems are used, which is often referred as common rail direct injection (CRDI) systems. The CRDI system mainly fulfills the three requirements of modern diesel engine, i.e., (1) high-pressure capability and injection pressure control, (2) flexible timing control, and (3) injection rate control. The schematic of a typical CRDI system is presented in Fig. 1.16.

In CRDI systems, the generation of high fuel pressure and fuel injection events is separated. The fuel pressure is independent of the engine speed and load unlike the mechanical in-line jerk and distributor pumps. The high-pressure fuel is fed to a rail (accumulator), which delivers the fuel to individual injectors (Fig. 1.16). The CRDI systems consist of four main components, i.e., (1) high-pressure pump, (2) high-pressure distribution rail and pipes, (3) electronic fuel injectors, and (4) electronic control unit (ECU). The common rail is connected to the fuel injectors through short pipes. The main advantage of the CRDI system over the conventional in-line jerk pumps is that the fuel injection pressure is constant and independent of the engine

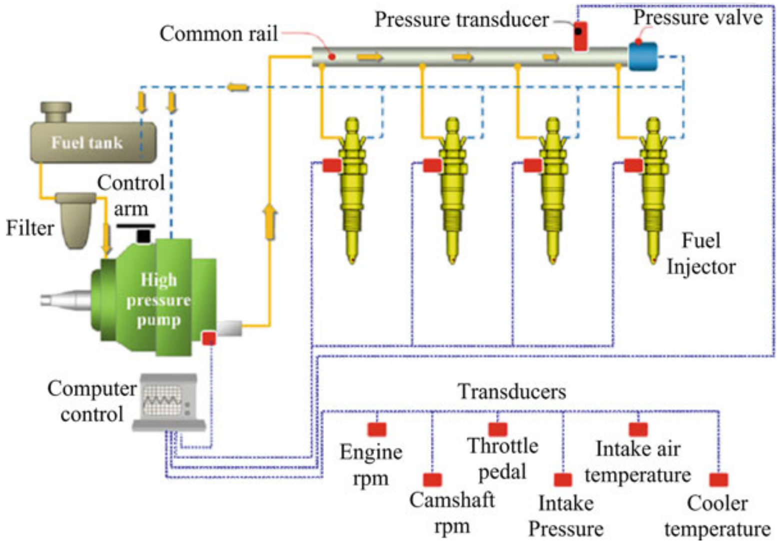


Fig. 1.16 Schematic diagram of CRDI system in modern diesel engines [21]

load and speed. Due to this independence, the maximum flow rate (and maximum torque required to drive the pump) does not have to coincide with the injection event (which is the case with the distributor pump), and thus, CRDI system requires a lower fuel pump maximum torque.

Due to electronic solenoid-based control of fuel injectors, the multiple injection events in a cycle are possible. The pilot injection is typically used to control the diesel engine noise. Figure 1.17 illustrates the components of electronic CRDI fuel injector and their associated signals. The injection quantity and injection timings are controlled by the opening and closing of the solenoid valve (Fig. 1.17). In the closed condition of the solenoid valve, the fuel pressures in the working chamber and in the needle chamber are equal to the rail pressure, and the needle is in the "closed" position. When the solenoid valve opens, the pressure in the working chamber falls, and the common rail pressure in the needle chamber lifts the needle, which initiates the fuel injection in the cylinder. The working chamber is pressurized again when solenoid valve closes [21].

The characteristics of the fuel injection system affect the fuel spray characteristics (droplet size distribution, spray angle, and spray tip penetration), which affect the combustion and emissions' characteristics of the diesel engine. The diesel engines are widely known for their higher fuel economy. The main advantages of using diesel engine are higher fuel conversion efficiency, high torque, low HC and CO emissions, durability and reliability, and typically low fuel and maintenance cost [22]. However, there are several demerits in diesel engines, which include cold start difficulty, noisy-sharp pressure rise, inherently slower combustion (low engine speed), lower power-to-weight ratio, expensive components, NO_x and particulate matter emissions, and low air utilization.

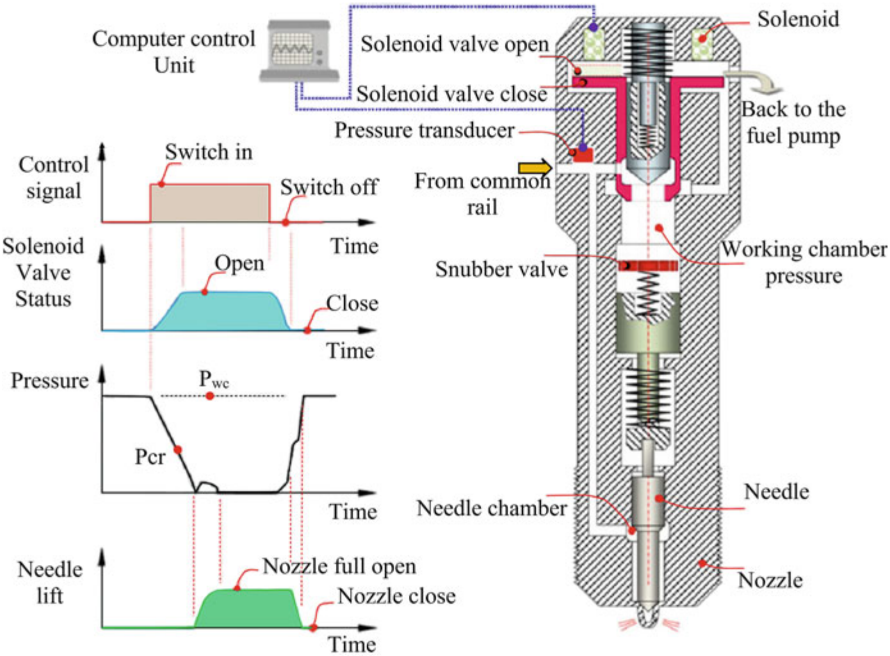


Fig. 1.17 The CRDI nozzle of a modern diesel engine [21]

1.1.4 Low-Temperature Combustion Engine

Spark ignition (SI) and compression ignition (CI) engines powered by gasoline and diesel, respectively, are the most widely used reciprocating engines. Diesel CI engines are preferred choice for medium- and heavy-duty applications due to their higher fuel conversion efficiency as it is operated at higher compression ratio, on the lean fuel-air mixture and unthrottled. However, the CI engines emit higher nitrogen oxides (NO_x) and particulate matter (PM), and there exists a trade-off in the emission of two species. The aftertreatment of NO_x and PM is difficult in diesel engines due to leaner engine operation. To meet the future emission legislation, a combination of in-cylinder strategies as well as exhaust aftertreatment devices is proposed for emission reduction. Employing currently available aftertreatment devices has limitations of higher cost, fuel economy penalties, durability issues, and larger space requirements [23]. Moreover, to compete in the market, higher efficiency at minimal cost is required. Thus, there is a need for drastic improvement in the in-cylinder combustion strategies to achieve higher fuel conversion efficiency along with a decrease in engine emissions which can reduce dependence on the exhaust aftertreatment technologies.

The NO_x formation in the engine cylinder is mainly dependent on the combustion temperature and air-fuel ratio (oxygen availability) [15]. The NO_x formation increases exponentially after the combustion temperature reaches around 2000 K.

In conventional engines (SI and CI), the maximum combustion temperature can occur in the range 2500–3000 K, which leads to the higher NO_x formation in the cylinder. Due to the higher combustion temperature occurrence, the conventional SI and CI combustions are referred as high-temperature combustion (HTC). Soot formation tendency increases as the local fuel-air mixture becomes richer. Ideally, to avoid the soot and NO_x formation in the combustion chamber, the engine should be operated on the premixed fuel-air mixture and at lower combustion temperature. The homogeneous charge compression ignition (HCCI) strategy employed in reciprocating engines is precisely working on the same concept by burning well-mixed fuel-air mixture and leaner combustion (lower temperature) [5]. Due to the limitations of the power density of HCCI engines, several partially stratified charge concepts are developed, and the common name for all the premixed combustion technologies is low-temperature combustion (LTC). All the premixed LTC modes have the common characteristic of relatively higher fuel conversion efficiency and simultaneous drastic reduction in soot and NO_x emissions. Figure 1.18 illustrates the LTC technology on local equivalence ratio-temperature (ϕ-T) map.

Figure 1.18 depicts the dependency of the formation of NO_x, soot, HC, and CO on the local combustion temperature and local equivalence ratio in the combustion chamber. In the φ-T map, the shape and size of the soot formation area depend on the fuel characteristics [25]. However, the region of NO_x formation is dependent on φ-T and independent of fuel. The conventional SI engine typically operates at stoichiometric mixture and operates at significantly higher temperatures, which lead to mainly higher NO_x formation (Fig. 1.18). However, diesel combustion is heterogeneous, and there is a large variation in the local equivalence ratio. Thus, there will be

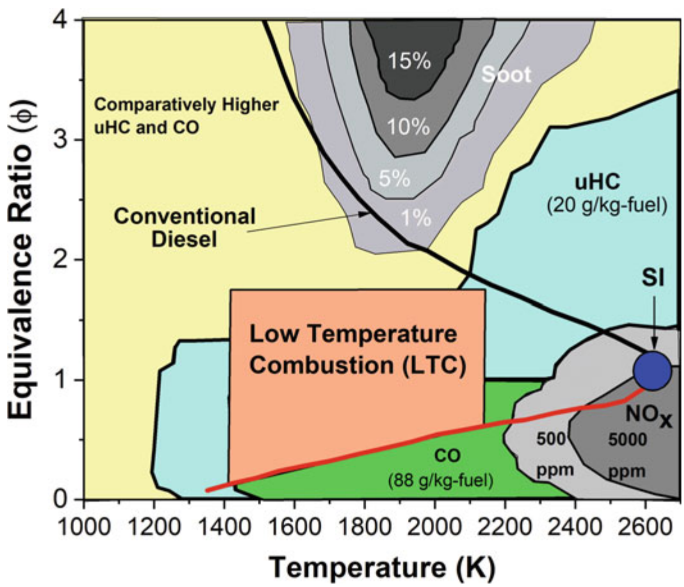


Fig. 1.18 Operating region of low-temperature combustion mode on the ϕ-T map (adapted from [5, 24–27])

higher soot formation along with the NO_x emission in conventional diesel engines (Fig. 1.18). When combustion temperature is decreased to reduce the NO_x formation, the soot formation kinetics is also slowed down [28, 29]. However, soot oxidation rate decreases more than the soot formation rate at relatively lower combustion temperatures [30], and therefore, soot increases in the engine exhaust. At lower combustion temperature, the HC and CO emissions increase, which is another major challenge. The HC and CO formation is also dependent on the combustion temperature and local equivalence ratio (Fig. 1.18). Near-complete oxidation of HC and CO occurs at higher combustion temperatures and leaner mixtures. In the φ-T map, there is a region which has relatively lower HC and CO emissions, and NO_x and soot emission can be avoided. This region is defined as low-temperature combustion (LTC) region (Fig. 1.18), and engine operation in this region leads to ultra-low NO_x and soot emissions [5]. The basic combustion mode in this region is HCCI combustion.

Figure 1.19 illustrates the working process of the HCCI engine and its comparison with conventional SI and CI engines. In HCCI combustion, the well-mixed

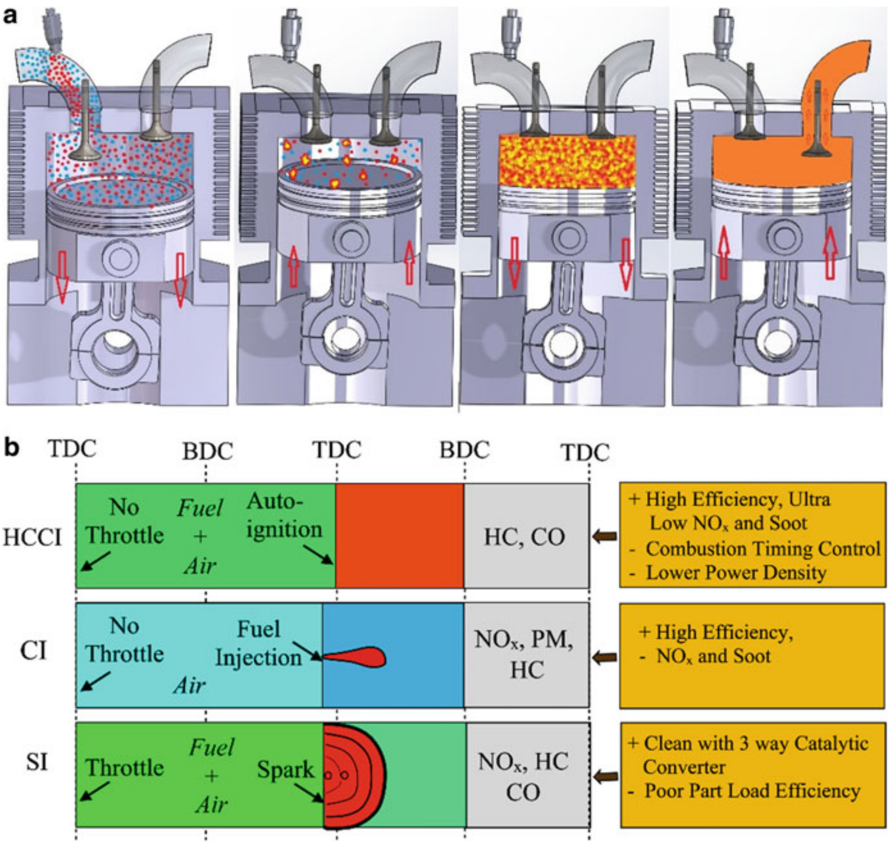


Fig. 1.19 (**a**) HCCI combustion process illustration, and (**b**) comparison of HCCI combustion process with conventional SI and CI combustion in the four-stroke cycle [5]

charge is prepared typically by PFI system. The fuel is inducted in the inlet manifold, and it gets mixed with air during intake and compression stroke. Toward the end of the compression stroke, when charge temperature reaches its autoignition temperature, the combustion starts in the entire combustion chamber (Fig. 1.19a).

In HCCI combustion, the autoignition of entire well-mixed charge occurs, and thus, the ignition timing is controlled by pressure and temperature in the combustion chamber. Therefore, there is no direct control of combustion timing in HCCI engine as spark timing in SI engine and fuel injection timing in the CI engines. There is no throttling and flame propagation in the HCCI combustion, which is there in conventional SI engines. The advantages and challenges of HCCI combustion engine are shown in Fig. 1.20. The solutions proposed to overcome the challenges are also presented in Fig. 1.20. The main advantages of the HCCI engine are the higher thermal efficiency and ultra-low NO_x and PM emissions. However, the major challenges are limited operating range and control of combustion phasing. Since combustion starts with autoignition of fuel, thus, there will be a cold start problem with high-octane fuels. There are several solutions proposed to overcome the challenges in HCCI combustion engines (Fig. 1.20). The detailed combustion process, autoignition chemistry, effect of different engine parameters on HCCI

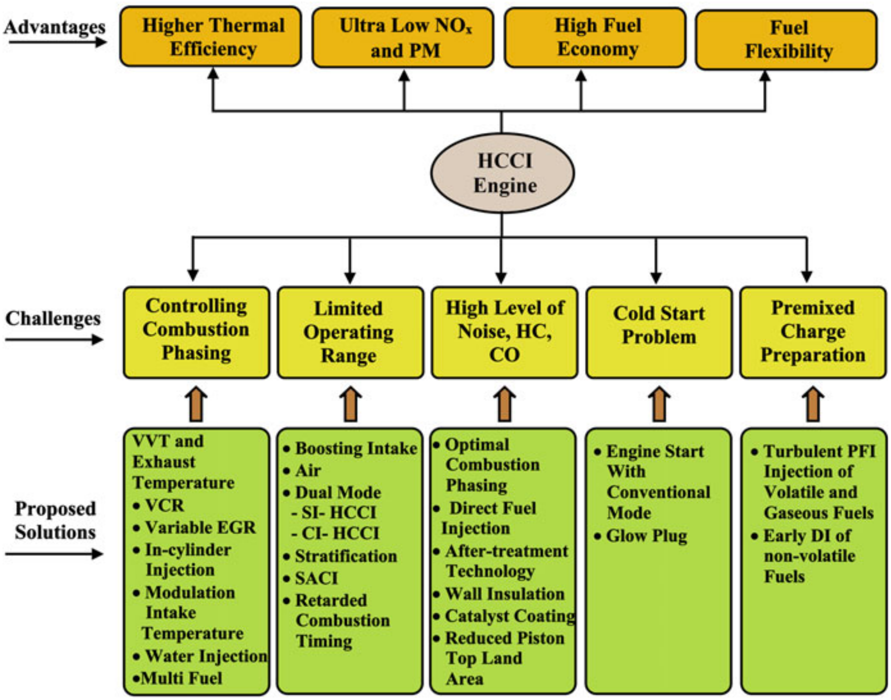


Fig. 1.20 The HCCI engine advantages, major challenges, and their proposed solutions [5]

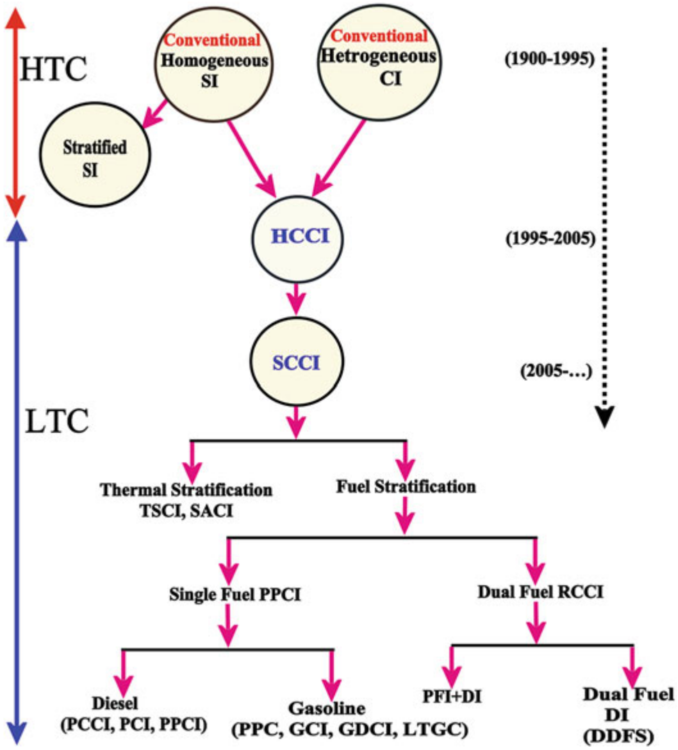


Fig. 1.21 Evolution of different combustion concepts for reciprocating internal combustion engines [5]

combustion, and control strategies of HCCI engines can be found in author's book [5].

To improve the power density and combustion control of the HCCI engine, various stratified charge compression ignition (SCCI) concepts are investigated by various researchers with different levels of stratification in fuel as well as temperature. Figure 1.21 illustrates the evolution of different combustion concepts in LTC and HTC regime. In high-temperature combustion (HTC) regime, the conventional SI and CI combustion occurs. Several LTC strategies have been investigated with various acronyms and names (Fig. 1.21), with different levels of inhomogeneity of the charge, but all of them are compression ignition. All the LTC strategies can be classified into two groups as HCCI and SCCI combustion. In the SCCI regime, the combustion depends on two possible stratifications of the charge by temperature inhomogeneity or fuel inhomogeneity (Fig. 1.21). The natural thermal stratification occurs in all the strategies. In thermal stratification group, the thermally stratified compression ignition (TSCI) and spark-assisted compression ignition (SACI) concepts are present. Fuel stratification can be created in the combustion chamber by single or dual fuel with different injection strategies. Using different direct injection strategies of single, fuel composition stratification is created in the combustion chamber; while using dual fuel, reactivity stratification is also created along with the composition stratification. In the fuel stratification regime, gasoline partially premixed combustion (PPC) and dual-fuel reactivity controlled compression ignition (RCCI) are the two major combustion strategies that are extensively investigated because of their potential of better combustion phasing control along with the advantages of HCCI engines. Combustion concepts in the PPC range are also termed as gasoline compression ignition (GCI) or gasoline direct injection compression ignition (GDCI) with a minor difference in injection strategies. In the dual-fuel stratification strategy, two fuel injections strategies are possible: (1) PFI injection of low reactivity fuel and direct injection (DI) of high reactivity fuel and (2) direct injection of both the fuels, which is termed as dual direct injection fuel stratification (DDFS). The detailed discussion of all these LTC concepts can be found in reference [5].

1.2 Engine Testing and Combustion Diagnostics

In general, engine testing is conducted to improve the design and configuration, to integrate the new technology, and to find out performance characteristics prior to the production and employing into a vehicle. Earlier, the engine tests were conducted to determine the power and fuel consumptions along with an additional test to find out the effectiveness of cooling, vibration, and noise, lubrication, controllability, etc. Present stringent emission legislation to limit the engine emissions leads to the more and more sophisticated engine testing. Test beds for internal combustion engines are generally differentiated by the following areas and types of application (objective of use): (1) research (single-cylinder engine test bed and flow test bed), (2) development (performance test bed, function test bed, endurance test bed, calibration test bed, emission certification test bed), and (3) production (end-of-line break-in test bed, quality assurance test bed) [31]. Single-cylinder engine test beds are mainly used for the purpose of research, which typically helps to assess the combustion process optically as well as thermodynamically. The flow test beds are utilized to investigate the charge motion, which has a significant role in the engine combustion process. The performance test bed is used for the determination of engine power output in the whole operating range in the predefined test conditions. The performance, combustion, and emission characteristics are analyzed using this type of test beds. Function test beds serve to optimize, verify and secure engine related overall system features. Dedicated calibration test beds are typically used in series development to determine a specific and optimal engine behavior on engine control units. Final homologation tests are conducted on the exhaust emission certification engine test bed. Endurance test beds are used to investigate and ensure the durability and long term stability of the engine and its components.

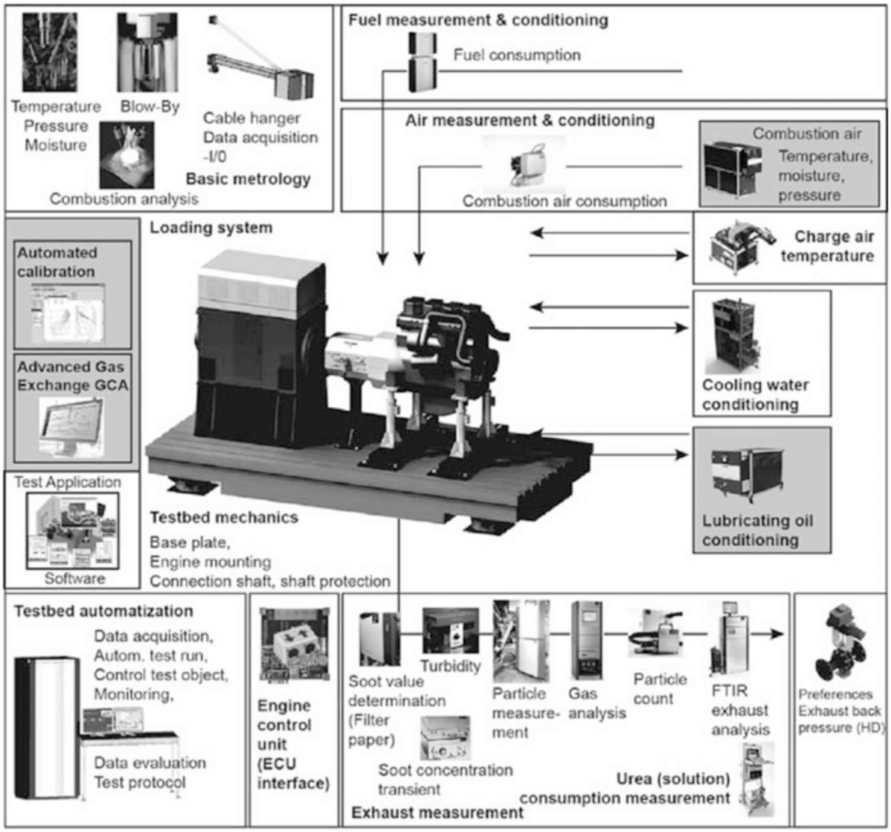


Fig. 1.22 Scope typically covered by an internal combustion engine development test bed [31]

To fulfill the broad range of objective, typical test beds for the development of reciprocating combustion engines contain several components. Figure 1.22 shows the typical test bed for internal combustion engines depicting the main components. The main components of a typical test bed are dynamometer (for torque measurement and control), engine media conditioning, fuel, air consumption measurement systems, instrumentation for combustion diagnostics, temperature and pressure measurement at the engine periphery, emission measurement devices, communication interface to the ECU, test bed automation (control/simulation), and calibration tools [31]. Test beds are classified as steady-state test beds and non-steady-state test beds based on the used test bed technologies, particularly engine dynamometer and test bed automation system.

Reciprocating automotive engine tests for product development and calibration can be classified as (1) steady-state test, (2) transient test, and (3) cold start tests. Another way of classifying the engine tests for research is based on whether test is conducted in-cylinder or out of the engine cylinder. Typical in-cylinder tests include in-cylinder pressure measurement, ion-current measurement, optical diagnostics, fast gas sampling, and flame ionization detector. Typical test conducted out to the engine cylinder includes the normal performance measurement (fuel consumption, engine efficiency, etc.), test for transient performance improvement, valvetrain testing, turbocharger performance test, friction (Morse test), and test related to aftertreatment technologies.

Understanding combustion processes would not be possible without diagnostics. Direct investigation of combustion systems reveals important information about the processes or properties including fuel spray, evaporation, mixing, ignition, flame speed, reactivity, and formation of various emission species. Combustion diagnostics provide the access to parameters such as species concentration, pressure, temperature, and flow velocity as well as to their spatial distribution and development with respect to time [32]. Combustion diagnostics can be used for serving various objectives including gaining fundamental insight of the combustion process, investigating and validating theoretical models, and real-time process optimization and control.

Combustion diagnostics is always used in engine development when unexploited potential compared with thermodynamically possible targets is ascertained during the measurement of consumption, output, and emissions [33]. Given the high target set for modern reciprocating engines, thermodynamic combustion analysis using the measurement of in-cylinder pressure is always a fixed element in the development sequence even along with the optical methods. For the fundamental studies, in-cylinder visualization of flow and combustion is typically performed using different optical methods. Optical methods can be used for various development tasks in spark ignition and compression ignition engines. In SI engines, optical methods can be used for determination of knock location, mixture formation study in the intake manifold as well as a direct injection in the cylinder, flame kernel formation and flame propagation, irregular combustion study, and noncontact temperature measurements. In diesel engines, optical methods can be used for flame image analysis, soot formation, and spray characteristics. Combustion chamber endoscopy and optical engines (optical window in piston and liner) are typically used for various optical investigations. Discussion on optical methods is out of the scope of the present book.

Figure 1.23 illustrates the measurement chain for generating various signals for combustion diagnostics other than the optical methods. Ion-current and cylinder pressure signals are the most direct methods to access the combustion process for the diagnostics. During the combustion process, ions are produce by oxidation reactions, and pressure increases due to the temperature increase by exothermic reactions. The major limitations of ion-current-based diagnostics are (1) measurement in a small portion of the combustion chamber and (2) measurement depends on the location of the sensor (mostly spark plug). The cylinder pressure is the most widely used method for combustion diagnostics in reciprocating engines. However, due to the higher cost of cylinder pressure-based measurement, other signals such as torque or instantaneous angular speed are also used for determination of combustion parameters. Instantaneous angular speed can be measured using crank angle encoder. However, some of the combustion information is attenuated in the torque and angular speed-

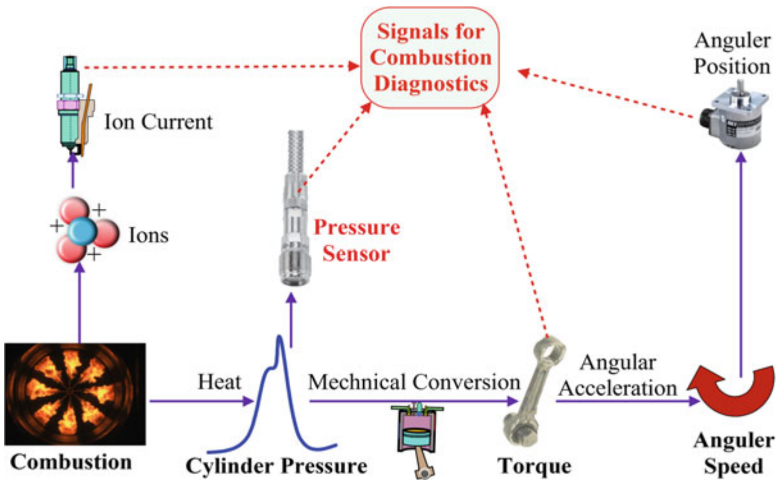


Fig. 1.23 Various signals for combustion diagnostics

based measurements. Thus, cylinder pressure is the best available signal containing most of the information of the combustion process, and presently, it can be measured easily using high-speed data acquisition and modern piezoelectric pressure sensors. Cylinder pressure-based combustion diagnostics is the main focus of the present book, and it is briefly introduced in the next section.

1.3 In-Cylinder Pressure-Based Combustion Diagnostics

In-cylinder pressure measurement and analysis is a fundamental practice used globally for research and development of reciprocating engines. Since the invention of internal combustion (IC) engines, the cylinder pressure measurement has been the principal diagnostic tool for experimenters. In-cylinder pressure measurement of the reciprocating engine has great potential for calibration, monitoring, diagnosis, validation of numerical modeling, and closed-loop control purposes. Calibration engineers use combustion analysis to improve emissions, fuel consumption, and performance calibrations of an engine. The application of methodologies based on the in-cylinder pressure measurement finds widespread applications as it provides direct combustion information with a high dynamical potentiality, which is fundamentally required for the diagnosis and control of the engine combustion process. Additionally, the in-cylinder pressure-based engine diagnostics may also lead to the reduction in a number of existing onboard sensors, which lower the equipment costs and the engine wiring complexity [34].

With the stringent legislation requirements on engine emissions and performance, the pressure-based combustion control systems have been regarded as the potential gain for the future automotive engines, which relies mostly on cylinder pressure based extracted information such as IMEP, heat release rate, combustion duration, compression condition, etc. In order to ensure the precision of the extracted parameters, high accuracy of results is required. The accuracy of the results depends on the accuracy of each component of the measuring chain, as well as the correct processing of the pressure signal.

Crank angle-based in-cylinder pressure measurement and analysis can be used to determine the variables such as piston and crankshaft loads, torque produced from combustion (equals the IMEP), gas exchange torque (PMEP), spark timing relative to MBT, time required for the combustion flame to develop and propagate, presence and magnitude of knock, cycle-to-cycle and cylinder-to-cylinder variability, etc. [35]. Additionally, the cylinder pressure-based combustion analysis can be used to achieve several development objectives such as evaluation of inlet and/or exhaust port and manifold geometries, quantifying compression ratio trade-offs, optimization of the combustion chamber geometry, selection of valve timing overlap and duration, optimization of fuel injector timing and on-time (opening duration), power/pressure rise/NO_x, automated mapping (MBT, knock, preignition control), inquiry of transient response, measurement of mechanical friction, and calibration optimization.

The various performance, combustion, and engine operating parameters that can be derived using cylinder pressure measurement and analysis are summarized in Table 1.1. Table 1.1 also briefly provides the signal processing methods along with the engine information generated. Detailed discussion on the derived parameters and signal processing methods is provided in different chapters of the present book.

1.4 Organization of the Book

Measuring and analyzing the in-cylinder pressure is an essential part of the combustion system development and calibration of an internal combustion engine. Cylinder pressure-based heat release analysis has been widely used in the field of engine research as an essential tool for understanding combustion behavior, as well as providing key data for engine and combustion modeling. Considering the advantages of in-cylinder pressure-based combustion diagnostics and a large amount of valuable information derived, this book presents a detailed discussion on different aspects of in-cylinder pressure measurement and its analysis.

The schematic of book organization is presented in Fig. 1.24. Typically, the workflow of a measurement chain consists of sensors (signal generation), data acquisition, signal processing, and data analysis. The present book provides a detailed discussion on all these aspects of measurement. The details of cylinder pressure sensors along with peripheral sensors for combustion diagnostics are provided in Chaps. 2 and 3. The acquisition of the cylinder pressure signal and their processing (phasing with crank angle position, absolute pressure referencing,

Table 1.1 Summary of performance and combustion parameters that can be estimated by processing of the measured in-cylinder pressure data [36]

Data derived from in-cylinder pressure signal	Engine information	Signal processing methods/equations
Cycle-averaged in-cylinder pressure	Firing cycle crank angle (CA)-based combustion event and quality of combustion	Averaging of cylinder pressure of different combustion cycle on crank angle (CA) basis
Motoring cylinder pressure, peak compression pressure, and its CA position	Used for tuning of heat release parameter, TDC correction, engine cylinder condition, and blowby	Measuring signal in non-firing engine cycle and averaging to a number of cycles. Calculation of TDC position using thermodynamic method using the measured pressure signal
Peak pressure	Mechanical load on the cylinder	Maximum value computation of in-cylinder pressure signal
Rate of pressure rise	Knock limit of the engine	Derivative of measured pressure signal
Crank angle at peak pressure, rate of heat release curve, and energy conversion points	Overall efficiency, combustion efficiency, qualitative exhaust values, quality of ignition system	Using first law of thermodynamics, rate of heat release (ROHR) calculation using the equation $\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta} + \frac{\partial Q_w}{d\theta}$ Energy conversion points are calculated by integrating ROHR curve
IMEP, FMEP, PMEP	Cylinder work output, combustion stability (cyclical fluctuations), friction losses, gas exchange losses	Computation of work using the cylinder pressure signal and volume curve generated from engine geometry $W_{\text{net}} = \frac{2\pi}{360} \int_{-360}^{360} \left(P(\theta) \frac{dV}{d\theta} \right) d\theta$ Calculation of work in high-pressure component and low-pressure component to calculate IMEP, PMEP, and FMEP
PV and log (*PV*) diagram	Work output and pumping losses, determination of TDC position, the polytropic coefficient of the mixture during compression and expansion	Computation of area under the curve, calculation of logarithmic value of pressure and volume curve, the slope of the curve in compression and expansion stroke to find the polytropic coefficient
Gas temperature and wall heat transfer	Qualitative exhaust values, heat transfer	Using empirical equations developed using in-cylinder pressure and mean gas temperature

High-frequency component of vibration	Knocking, combustion noise, ringing intensity	Filtering using a bandpass filter, wavelet analysis, power spectrum
Ignition delay	Formation of air/fuel mixture	Calculated from ignition (SI) or injection point (CI) and start of combustion calculated from heat release curve
Mass flow rate	Air mass flow estimation, residual exhaust gas in cylinder, backflow	Application of the Δp-method for estimating the air mass, frequency analysis of the pressure trace
Compression ratio estimation	Actual compression ratio	Computation using polytropic compression model and optimization algorithm, temperature-entropy-based method
Air-fuel ratio	Cylinder mixture strength	Cylinder pressure moment-based approach, net heat release-based estimation, g-ratio method
Torque	Engine torque	Indicated torque is estimated from the peak pressure and its location, and the load torque is observed by the estimated indicated torque
COV_{IMEP}, CA position for different heat release and mass burn fraction	Cycle-to-cycle dispersion, misfire	Statistical analysis, symbol sequence analysis, return maps
Emission	Engine emissions, Quality of combustion	Neural network-based algorithm, regression-based approach
Control parameters calculations	EGR control, noise control, emission control, online combustion failure detection, start of injection control	Online signal processing (filtering) and computation of control parameter

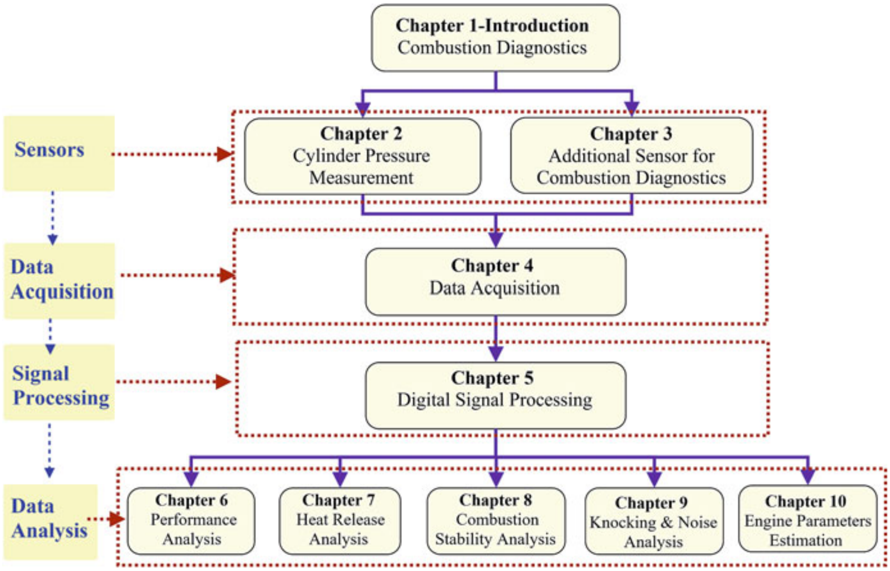


Fig. 1.24 Schematic of the book organization

smoothing/filtering, and cycle averaging) is presented in Chaps. 4 and 5, respectively. The detailed data analysis of the measured in-cylinder pressure signal is presented in Chaps. 6–10. Detailed discussion is provided on performance analysis (torque, power, and work), heat release analysis (mass fraction burned, combustion phasing, and combustion duration), combustion stability analysis (cycle-to-cycle variations) with statistical and nonlinear dynamics methods, and knocking and combustion noise analysis. The last chapter presents the discussion on estimation methods of different engine operating parameters (TDC, compression ratio, air-fuel ratio, wall temperature, trapped mass, and residual gas fraction) using the measured cylinder pressure data. Information provided in the proposed book can be effectively used for further development, optimization, and calibration of modern reciprocating engines.

Discussion/Investigation Questions

1. Describe the difference between an internal combustion engine (ICE) and an external combustion engine (ECE). Write the examples of ICE and ECE. Discuss the advantages and disadvantages of the ICE over the ECE.
2. Discuss the major environmental concerns arises due to extensive use of reciprocating ICEs powered by fossil fuels.
3. Discuss the major trends in vehicle developments to improve the engine efficiency and comply with the emissions legislation.

4. Describe the merits and demerits of ICE based vehicles over the battery electric vehicles (BEVs). Discuss the factors contributing to higher fuel economy in hybrid electric vehicles (HEV) over ICE vehicles. Justify the statement, "hybrid vehicles are mainly relevant to SI engines."
5. Discuss the reasons why the work output is typically limited by displacement over its entire speed range in SI engines while in diesel engine displacement limited only at maximum torque condition.
6. Describe why displacement volume of SI engine is lower than the diesel engine for similar power ratings.
7. Discuss the factors which govern the preference of multicylinder engine when total displacement volume is approximately above 500 cm^3 in high-speed engines.
8. Discuss the advantages and disadvantages of conventional diesel engines over the gasoline engines.
9. Describe the reasons why gasoline SI engines typically operate at stoichiometric air-fuel ratio condition.
10. Discuss the advantages of gasoline direct injection (GDI) engines over the port fuel injection (PFI) spark ignition engines.
11. Discuss the reasons why engine speed of a diesel engine is relatively lower than gasoline engines. Modern diesel engine has a relatively higher engine speed than conventional diesel engines. Investigate the development of technologies responsible for the increase in engine speed.
12. Discuss the factors governing the higher fuel conversion efficiency of diesel engines.
13. Discuss the advantages of common rail direct injection system over the mechanical fuel injection (in-line jerk type) system.
14. Write the advantages of HCCI engine over SI and CI engines. Discuss the factors contributing to higher indicated efficiency even though higher HC emissions and lower combustion efficiency found in HCCI engines.
15. Considering the ϕ-T map shown in Fig. 1.18, determine the typical local equivalence ratio and combustion temperature for LTC engines. Discuss the reasons, why NO_x and soot formation is very low in this temperature and equivalence ratio range.
16. Describe how the fuel stratification and thermal stratification help in increasing the power density of HCCI engines. Discuss the difference between fuel composition stratification and reactivity stratification.
17. Explain the importance of experiments in general? Describe the type of engine experiments performed for research and development purposed. Discuss the general instrumentation of engine for conducting various tests related to performance, combustion, and emissions?
18. Discuss the need of combustion diagnostics particularly in reciprocating combustion engines. Explain the different methods of combustion diagnostics in internal combustion engines.
19. Describe the different possible signals (other than optical signals) that can be used for combustion diagnostics in reciprocating engines. Discuss the

advantages and limitation of each signal with respect to combustion information revealed. Explain the factors contributing in the favor of cylinder pressure signal for the most widespread applications in reciprocating engine combustion diagnostics.

20. Describe the in-cylinder pressure measurement-based combustion diagnostic method. Write the typical information that can be produced by processing of measured in-cylinder pressure signal. Discuss the significance of the cylinder pressure-based diagnostics in engine research and development process.

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