




Review

# Prospects of Controlled Auto-Ignition Based Thermal Propulsion Units for Modern Gasoline Vehicles

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**Abstract:** Gasoline engines employing the spatially distributed auto-ignition combustion mode, known as controlled auto-ignition (CAI), are a prospective technology for significantly improving engine efficiency and reducing emissions. This review paper provides an overview of developments in various gasoline CAI technologies and discusses their attendant strengths and weaknesses. Hybrid propulsion systems powered by high-efficiency gasoline CAI engines can provide a low-carbon pathway for mobility sector decarbonisation. Therefore, this paper focuses on the challenges and opportunities of CAI implementation, especially for electrified powertrains. Different control actuators that can extend the CAI operating range are discussed, and opportunities for synergistic operation between thermal and electric components of hybridised powertrains are identified. Such synergies can remove impediments in the way of CAI system adoption and can, thus, support CAI adoption and maximise efficiency gains from its implementation. The prospects of supporting CAI combustion for different powertrain electrification levels, hybrid architectures, engine size, and energy management systems are discussed. Load levelling offered by electrified powertrains through CAI-favouring energy management strategies has the potential to substantially relax the operating point requirements for CAI-based thermal propulsion units and to remove the need for expensive actuators. The highly flexible spark-assisted partially premixed compression ignition hybrid mode (SACI-PPCI) emerges as a promising CAI strategy for conventional powertrains, and the moderately flexible spark-assisted compression ignition (SACI) configuration can be a cost-effective thermal propulsion mode for electrified powertrains.

**Keywords:** advanced compression ignition; HCCI; low temperature combustion; dedicated hybrid engines; gasoline compression ignition



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## 1. Introduction

The global transportation system accounts for about 16% of the world's greenhouse gas emissions. Around half of these originate from road passenger vehicles, which are, at present, mostly powered by internal combustion engines running on fossil fuels [1]. Moreover, harmful pollutants from combustion, particularly NO<sub>x</sub>, particulate matter, unburned hydrocarbons, and CO, deteriorate local air quality and have adverse health impacts [1]. To decarbonise the road transport sector, an estimated 3–4% reduction in CO<sub>2</sub> emissions is needed every year between now and 2030 [2], thus necessitating transformative technological innovations. Some potential transportation decarbonisation pathways include powertrain electrification [3], fuel cell-powered vehicles [4], and internal combustion engines operating on net-carbon neutral fuels, such as renewable hydrogen [5], e-fuels [6] or biofuels [7].

Decarbonisation via complete powertrain electrification, i.e., replacing internal combustion engine vehicles with battery electric vehicles (BEV), is limited by the need to decarbonise the power generation system, to expand and improve recharging infrastructure, and to improve battery technology to make it more energy-dense and economical [8,9]. Traditionally fuelled partially electrified vehicles, i.e., hybrid electric vehicles (HEV), can

help bridge this gap and make timely, cost-effective, and impactful contributions to climate change mitigation efforts, as they have relatively high technology readiness levels and benefit from an existing refuelling infrastructure [10,11].

HEVs can offer significant system-level efficiency improvements resulting from the high energy conversion efficiency of the electric components, thermal propulsion unit (TPU) operation at high efficiency points, and waste energy recuperation. These improvements, which directly correlate to CO<sub>2</sub> emission reductions, can be over 50%, on a tank-to-wheel basis, relative to contemporary spark-ignition (SI) engines [10]. Depending on the carbon intensity of the electricity grid, the relative greenhouse gas emissions of HEVs can be superior, comparable, or inferior to those of BEVs [1,10,11]. There is, however, an increasing consensus developing that, as electricity generation and battery production become less carbon-intensive, BEVs will have a significantly lower CO<sub>2</sub> footprint compared to fossil fuel-powered internal combustion engine vehicles [1]. Until then, distributing the total available battery storage capacity as smaller batteries across many mild or parallel HEVs instead of concentrating it as larger batteries in fewer BEVs or plug-in hybrids can be a more effective decarbonisation pathway [1,10].

The emission reduction potential of HEVs can be bolstered by improving the fuel conversion efficiency of their TPUs and by reducing the associated pollutant emissions. Multiple avenues for TPU efficiency improvement are being actively researched, including ultra-lean combustion [12,13], fuel chemistry improvements [10], over-expanded cycles and flexible valve trains [14–16], waste heat recovery [17], alternative cycles [18], and advanced combustion concepts [19,20]. Novel TPU designs, such as micro-gas turbines [21] and free piston linear generators [22] are also being developed.

This paper discusses advances in TPUs employing the advanced combustion concept of controlled auto-ignition (CAI) [19], with special emphasis on its potential to improve HEV system efficiencies by exploiting synergies between CAI and powertrain electrification. CAI is a thermodynamically efficient combustion mode. It is distinct from the conventional combustion modes of spark and compression ignition because of the mixture preparation and ignition strategies employed. CAI technologies have been actively researched for over two decades (initially mostly for diesel engines [23–26]) and can play an important role in transport decarbonisation. The discussion in the paper focuses on gasoline CAI TPU advancement because approximately 80% of the global light-duty vehicle fleet is powered by gasoline, and diesel vehicles have seen a decline in popularity in many automotive markets in the post ‘dieselgate’ era [1].

Broad reviews of CAI can be found in the literature [27,28], as well as reviews on gasoline direct injection (GDI) engines [29,30]. Comprehensive surveys of engine technology [31] and powertrain configuration [32] evolution for HEVs are also available [33]. The current review lies at the intersection of such compendiums and contributes the following to the automotive sector decarbonisation discourse:

1. Refreshes and re-contextualises the benefits and challenges of CAI for the current mobility landscape where powertrain electrification is becoming increasingly common.
2. Offers a consistent and up-to-date characterisation for various CAI embodiments.
3. Discusses opportunities for addressing CAI control challenges and improving system efficiency in CAI TPU-based HEVs through the co-development of thermal and electrical components.

CAI is defined in general terms, and associated advantages and challenges are discussed in Section 2, followed by a discussion of various CAI embodiments (Section 3) and engine control options (Section 4). Opportunities for complementary CAI and hybrid propulsion system (HPS) development and operation are discussed in Section 5.

## 2. Controlled Auto-Ignition

As the name suggests, CAI utilises auto-ignition (AI) as the principal energy release pathway during combustion. The other pathway commonly used in internal combustion engines is positive ignition, most commonly spark ignition (SI). In CAI, the cylinder con-

tents are relatively well-mixed, and if the post-compression pressure and temperature are sufficiently high, near-simultaneous AI is initiated at multiple points across the cylinder. This is different from traditional compression ignition (CI) ‘diesel’ combustion, where auto-ignition is spatially restricted to the vaporised fuel-air cloud around the fuel injector. Moreover, in CAI, AI is triggered not only by the compression of the trapped cylinder charge, but thermal assistance (e.g., via intake air heating or combustion product entrapment) is also needed to make the well-mixed, diluted, and lean cylinder mixture cross the self-ignition temperature threshold. CAI has thus been described as the ‘flameless’ and ‘volumetric’ energy release from the spontaneous combustion of lean and diluted, homogenous, air-fuel mixtures [34].

The onset of AI during CAI in the cylinder is governed by the chemical kinetics of the local fuel-air-residual-gas parcels and their respective pressure and time histories. This can lead to control challenges and has necessitated the development of multiple mixture preparation strategies. These strategies help tailor the AI behaviour of cylinder contents to realise efficient, clean, stable, and moderately aggressive energy release. In most practical CAI systems, a certain level of heterogeneity (in temperature, fuel, or residual gas distribution) is purposefully introduced in the trapped mixture to control AI and the ensuing heat release behaviour. However, the combusting mixtures are still reasonably well-mixed compared to those in conventional compression ignition engines, which are highly stratified. Hence, only the initial stage of conventional compression ignition combustion (the pre-mixed combustion stage) is achieved through the AI of a well-mixed fuel-air mixture. The latter stage is controlled (not kinetically) by mixing. Moreover, conventional compression ignition combustion is locally rich.

### 2.1. The Thermodynamic Case for Gasoline CAI

CAI-based energy release can offer thermal efficiency advantages through the efficient utilisation of the expansion stroke and relatively low combustion temperatures resulting from diluted and/or lean combustion. Because of the latter attribute, CAI has also been referred to as ‘low temperature combustion’ in the literature [34,35]. Both diluted (dilution via residual gases) and lean (dilution via air) combustion offer thermal efficiency benefits. The multi-site initiation of combustion in CAI can help improve engine stability and combustion efficiency by increasing the likelihood of successful and repeatable combustion. There are also associated (net favourable) effects on engine-out emissions resulting from the cylinder composition, mixedness levels, and energy release phasing.

Lean, diluted combustion presents a high thermodynamic efficiency pathway because it minimises the total in-cylinder process irreversibility and thus provides a net efficiency gain, resulting from [34]:

- A decrease in heat transfer losses because of lower temperatures of cylinder contents [+].
- An increase in specific heat ratio of the combusting mixture, which allows for more useful work to be harvested per unit expansion of the expanding gases [+].
- A decrease in the useful energy (exergy) lost with exhaust gases [+].
- An increase in combustion irreversibility because of lower combustion temperatures. For practical combustion systems, these losses range between 20 and 25% (up to 32% according to [35]) and are difficult to ‘engineer our way out of’ [–].

These thermodynamic benefits can be reduced in engines employing flame-propagation-based (deflagrative) combustion (i.e., SI engines) because the reduction in combustion temperatures slows down flame propagation. This results in the poor utilisation of the expansion stroke (thermal efficiency reduction) and makes combustion more unstable (combustion efficiency and engine stability deterioration). To counter reductions in flame propagation speed in lean SI engines, combustion chambers and air paths are designed to enhance in-cylinder turbulence levels, often at the expense of gas-exchange efficiency [36]. Simultaneous fast and low-temperature heat release is realised in CAI engines through the multi-site AI combustion of well-mixed lean and diluted mixtures, whereby combustion is slow locally because of leaning and dilution, but because energy release is distributed

across the entire cylinder, accumulated heat release is sufficiently fast. This reduces the need for high turbulence levels and permits the design of ports and valves with high flow efficiency.

When gasoline is used in CAI, another potential advantage arises from its AI chemistry. These relatively high octane number (ON) and low cetane number (CN) fuels (research ON > 60 and CN < 30 [37]) have longer chemical ignition delays compared to those of diesel, thus permitting more time for the preparation of auto-ignitable mixtures by allowing the fuel to vaporise more thoroughly and mix with air and residual gases. Gasoline fuels are also more volatile than diesel fuels (boiling range 20–160 °C vs. 160–380 °C, respectively [38]), which facilitates the preparation of a well-mixed vaporised fuel-air mixture. Hence, there is a need for very high pressure fuel injectors to promote premixed combustion, as is the case for diesel engines, that can be reduced [39].

Gasoline CAI, therefore, has the potential to improve engine efficiency by:

- i. Allowing thermodynamically efficient dilute and/or lean combustion;
- ii. Improving combustion efficiency through multi-site AI;
- iii. Reducing in-cylinder turbulence requirements to allow the design of efficient air flow passages;
- iv. Using a volatile, low-reactivity fuel to ease the formation of a vaporised fuel-air mixture.

## 2.2. Effects on Emissions

In addition to thermal efficiency improvements, CAI engines can also have net-favourable effects on pollutant emissions. The exact effects depend on the CAI strategy employed and the operating point of the engine. The effects on various pollutant emissions are listed below, with the arrows within parentheses indicating the general direction of expected change in engine-out emissions compared to conventional engines. Clearly, subsequent clean-up with aftertreatment substantially reduces vehicle-out emissions.

### 2.2.1. NO<sub>x</sub> (↓)

Thermal NO<sub>x</sub> production is likely reduced significantly in CAI engines because of lower combustion temperatures and reduced oxygen availability caused by diluted and lean operation. Moreover, because of the parallel energy release from multiple ignition sites across the cylinder, NO<sub>x</sub> is not generated from the compressive heating of initially burned gases, as happens in SI and CI engines. A reduction of 97% in NO<sub>x</sub> emissions was reported by Oakley et al. [40] for a CAI strategy relative to SI operation in the same engine.

### 2.2.2. Soot, Smoke, Particulate Matter (↓,↑)

Particulate emissions are likely to be reduced to near zero because of the high levels of pre-mixing present in most CAI engines, which results in locally lean combustion. Local fuel-air equivalence ratios less than 0.5 are considered sufficiently lean for preventing soot formation [37]. CAI strategies employing direct injection (DI) can, however, witness an increase in cylinder-out particulate matter (PM) emissions resulting from pool fires and diffusive combustion of the injected fuel [30]. The two are, respectively, caused by fuel impingement on combustion chamber walls and overlapping fuel injection and heat release events, which is indicative of insufficient fuel-air mixing. Therefore, combustion chambers have to be designed and fuel injection events controlled appropriately to keep PM emissions low. The heat capacity and dilution effects of exhaust-gas recirculation can also cause an increase in PM emissions by limiting the temperatures and amount of oxygen needed for soot oxidation, respectively. Very high EGR levels can counter these effects by lowering combustion temperatures below the soot-formation threshold [41].

### 2.2.3. Hydrocarbons (↑)

Major determinants of engine-out hydrocarbon (HC) emissions are the fuel-air equivalence ratio of the trapped mixture (rich combustion leads to higher unburned hydrocarbons), engine stability (at unstable engine operating points, HC emissions increase), fuel entrapment in crevice regions, and the oxidation of crevice-released fuel during expansion.

The overall lean nature of CAI should reduce HC emissions because of the availability of excess air for fuel oxidation, and the multi-site nature of CAI combustion should lower unburned HC emissions at traditionally unstable operating points by improving combustion efficiency. However, fuel trapped in cylinder crevices during the main combustion event can cause an increase in cylinder-out HC emissions. This is because: (i) more fuel is trapped in the cylinder crevices, and (ii) relatively less crevice-released fuel is oxidised during expansion. More fuel is trapped in the crevices because the low CN fuels used in gasoline CAI have longer ignition delays and are present in the cylinder for a longer period before combustion [37]. The likelihood of the oxidation of crevice-released fuel during the expansion stroke is lower in CAI engines because of the relatively low combustion and expansion temperatures [42].

#### 2.2.4. CO (↑)

Factors that determine engine-out CO levels are similar to those for HC, as both are products of incomplete combustion. For the same scenarios as for high HC emissions, CO emissions can increase if the fuel is partially oxidised. The oxidation of CO to CO<sub>2</sub> requires temperatures that range between 1125 and 1225 °C [19], which might not be available at all operating points during CAI operation, especially at low load-operating points. In Oakley et al. [40], CO emissions from a homogenous CAI engine increased monotonically as the load was reduced. However, at high loads, CO emissions were found to be lower than those from SI operation.

#### 2.2.5. Effects on Aftertreatment Systems

Because of their NO<sub>x</sub> and PM reduction potential, CAI engines can lower exhaust aftertreatment requirements, especially if engine operation regions that increase CO and HC (e.g., part load operation) can be avoided. Excess engine-out CO or HC emissions can be cleaned using oxidation after-treatment devices. Oxidising partially oxidised chemical species through exhaust aftertreatment is cheaper and easier to manage than simultaneously trying to reduce NO<sub>x</sub> and oxidise HC and CO using three-way catalysts (TWC), which are needed in current SI engines because all three species are exhausted out of the cylinder at unacceptably high levels [37]. However, if NO<sub>x</sub> emissions are not reduced sufficiently by in-cylinder combustion management, then TWC for stoichiometric CAI engines and selective catalytic reduction (SCR) for lean CAI engines are needed.

If engine-out PM emissions are high in DI CAI engines, they can be managed using particulate filters. The need for such filters is likely to arise only at low loads where temperatures are not high enough for soot oxidation.

One challenge for CAI combustion is that the relatively low exhaust temperatures reduce the conversion efficiency of exhaust aftertreatment devices, which typically require temperatures higher than 250 °C [1]. Although the reduction in engine-out emissions reduces the dependence on aftertreatment devices, lean ('diesel-like') aftertreatment might be required, which could increase system cost and complexity [43,44].

### 2.3. An Overview of CAI Challenges

Research on various CAI modes has been ongoing since at least 1979 [45,46] and recently CAI engines have witnessed some commercial success [47,48]. The biggest impediment to their adoption has been the difficulty in reliably controlling the onset and phasing of combustion across a sufficiently wide operating range. Failure to optimally and adequately control combustion can lead to unacceptably high pressure-rise rates (PRR) or high levels of combustion instability, in addition to having adverse effects on thermal efficiency and emissions [49]. Some of the phenomena that serve as barriers to acceptable CAI engine operation at the edges of the speed and load operating map are summarised below.

**Low Load Limit (Lean/Instability Limit):** At low loads, combustion temperatures decrease because of low peak and mean effective pressures, which causes a decrease in residual gas temperatures. The low trapped mixture temperatures retard or stop AI,

which, respectively, leads to partial-burn or misfires. Moreover, in turbocharged engines, at low loads, the exhaust gas might not have adequate energy to sufficiently boost cylinder pressure to achieve AI.

**High Load Limit (Ringing Limit):** As more fuel is added to meet increasing torque requirements at high loads, the pressure-rise rates and peak pressure can become excessively high, leading to unacceptably loud combustion and mechanical damage. This is known as the ringing limit and serves as a high load boundary. A number of metrics have been used to index combustion intensity in CAI engines, e.g., PRR (bar/°CA [49] or bar/sec [50]), ringing intensity (MW/m<sup>2</sup> [51]), and calculated or measured combustion noise [52]. Comparisons of various combustion intensity metrics have been conducted by Ghandhi et al. [52] and Eng [50]. A limit of 3 bar/°CA was found to be an often-used PRR limit [49,53,54].

To moderate energy release at high loads, dilution levels are increased. There are, however, limits to the extent of dilution imposed by the amount of fuel that can be combusted, which is directly coupled to the engine's power output. This acts as a second high-load limit, where failure manifests as a misfire or a partial fire [55].

**Low Speed Limit:** At low speeds, combustion temperatures and, consequently, residual gas and trapped mixture temperatures decrease because of increased heat transfer. This, as is conducted at low loads, reduces the mixture's auto-ignitability and can cause combustion stability problems. For turbocharged engines, turbo-lag can also act as a limiting factor at low speeds, where sufficient exhaust flow might not be available to drive the turbine.

**High Speed Limit:** At high speeds, hot combustion gases needed for controllably triggering AI might not be retained in sufficient quantities for some valve timing-based internal EGR (iEGR) strategies. Another consequence of these iEGR strategies can be the limiting of fresh air being drawn in, as they may necessitate a reduction in maximum valve lifts (e.g., a negative valve overlap strategy). Moreover, at higher speeds, cylinder content temperatures might increase because of decreased heat transfer, which could lower ignition delays and lead to uncontrolled AI.

All of the CAI limiting effects listed above are summarised in Figure 1. The CAI operating map shown was adapted from Fu et al. [49] and was used as a baseline CAI engine throughout the paper. It represents a naturally aspirated, gasoline port fuel-injected, four-stroke, automotive engine that employs the CAI strategy of homogeneous charge compression-ignition (HCCI) and has a negative valve overlap (NVO) for trapping hot residual gases. A summary of the general discussion on the opportunities and challenges of CAI presented in the preceding sections is provided in Table 1.

**Table 1.** Comparison of CAI and conventional engine combustion technologies.

SI Inefficiencies	Can Gasoline CAI Help?	Potential Challenges
Low compression ratios <ul style="list-style-type: none"> <li>• Knock limited.</li> <li>• Infeasible (knock limited) for low-speed operation needed for heavy duty applications</li> </ul>	Yes, CAI is moderated/partially controlled knock	- High heat-release rate (HRR), PRR, and peak pressure.
Throttle <ul style="list-style-type: none"> <li>• High pumping losses.</li> </ul>	Possibly, if load is controlled via CVVL and DI, and engine runs at wide open throttle.	- Reliable load control, especially during transient operation.
Stoichiometric combustion <ul style="list-style-type: none"> <li>• TWC restriction.</li> <li>• Prevents reaping efficiency benefits of lean combustion.</li> </ul>	Yes, CAI encourages lean and diluted combustion, which lowers cylinder-out NO <sub>x</sub> emissions.	- CO and HC might increase because of un-/partially oxidised fuel in crevices and at low loads. - Need for different types of (lean) aftertreatment systems.

Table 1. Cont.

SI Inefficiencies	Can Gasoline CAI Help?	Potential Challenges
Flame propagation combustion <ul style="list-style-type: none"> <li>Slow combustion during very lean/diluted operation.</li> </ul>	Yes, through volumetric combustion.	<ul style="list-style-type: none"> <li>High HRR and peak pressures at high loads and unstable combustion at low loads.</li> <li>Controlling combustion initiation and phasing.</li> </ul>
Air-fuel mixture compression instead of just air <ul style="list-style-type: none"> <li>Low specific heat ratio.</li> </ul>	Likely not, unless engine is operated in a diesel-like CI mode.	<ul style="list-style-type: none"> <li>Diesel-like fuel injection will increase PM emissions (diffusive combustion).</li> </ul>
Carbon-intense fuel processing (21.5 vs. 16.8 gCO <sub>2,eq</sub> /MJ compared to diesel [10])	Unlikely, it would require the revision of regulations to allow low RON fuels.	<ul style="list-style-type: none"> <li>Commercialization of low RON fuels.</li> </ul>
Diesel Inefficiencies	Can Gasoline CAI Help?	Potential Challenges
'Diesel' <ul style="list-style-type: none"> <li>Post-dieselgate disfavour.</li> </ul>	Yes, with gasoline CAI.	<ul style="list-style-type: none"> <li>Gasoline has a relatively long chemical ignition delay; might necessitate lower ON fuels.</li> <li>Could be an advantage too, with more time for vaporisation.</li> </ul>
Locally rich and diffusive combustion <ul style="list-style-type: none"> <li>Soot;</li> <li>NO<sub>x</sub>.</li> </ul>	Yes, if a well-mixed, lean mixture is burned. To a lesser extent with DI CAI.	<ul style="list-style-type: none"> <li>Insufficient stratification at load limits might compromise combustion stability.</li> </ul>
Relatively slow mixing-controlled combustion phase.	Yes, through the volumetric combustion of a well-mixed mixture.	<ul style="list-style-type: none"> <li>High HRR and PRR.</li> <li>Might require lowering compression ratio.</li> </ul>
Expensive after-treatment systems.	Yes, by lowering cylinder-out NO <sub>x</sub> and PM emissions. Aftertreatment of CO and HC is easier and less costly.	<ul style="list-style-type: none"> <li>Low exhaust temperatures might require diesel-like treatment devices.</li> </ul>
Expensive high-pressure fuel injectors.	Yes, gasoline is easier to vaporise, and its AI resistance allows more time for mixing.	<ul style="list-style-type: none"> <li>Low-intensity injection-induced cylinder flows.</li> </ul>

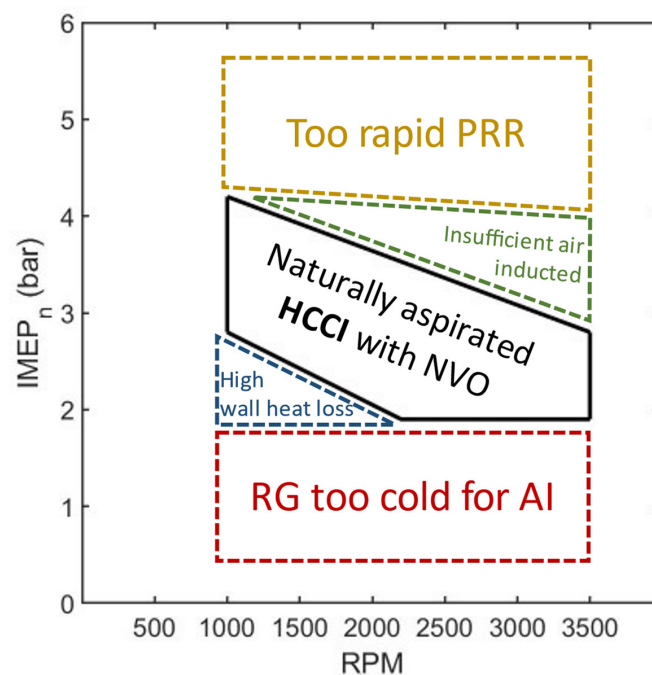


Figure 1. Limits to the HCCI operation of the baseline CAI engine. BMEP has been converted to net IMEP by assuming 90% mechanical efficiency (RG: residual gas).

### 3. CAI Characterisation and Development

The pursuit of the potential benefits of CAI whilst simultaneously avoiding the accompanying challenges has led to the development of numerous CAI realisation strategies.

*Two-stroke gasoline engine* researchers were the first to study CAI [45,46,56,57]. They were interested in it as a means to stabilise low load performance. The inherently high fraction of hot residual gases that are trapped after scavenging in two-stroke engines were exploited to initiate multi-site CAI. Thus, improving the quality of combustion and reducing HC emissions, which were exceptionally high for the port fuel-injected (PFI) or carburetted engines. Even though two-stroke CAI achieved some commercial success [58], the cost of managing exhaust HC emissions via combustion and thermal efficiency improvements was too high compared to using valved, four-stroke engines that did not suffer from the gas-exchange problems inherent to the two-stroke architecture. Two-stroke engines have witnessed a resurgence in interest in recent times because of their potential to extend the high-load CAI operating limit [18,59–62].

*Diesel engine* researchers investigated CAI as a means to reduce engine-out  $\text{NO}_x$  and soot emissions by breaking the so-called ‘ $\text{NO}_x$ -soot trade-off’ [23–26]. They hoped to avoid diffusion combustion while retaining the high efficiency of CI engines. Diesel CAI was an active research area at the turn of the century, but it has seen a downturn in the post-dieselgate era because of the general disfavour exhibited towards diesel engines by the automotive market. Some of the most promising gasoline CAI technologies have their roots in diesel CAI research.

*Gasoline four-stroke engine* researchers are interested in CAI as a means of achieving diesel-like efficiencies and lowering  $\text{NO}_x$  emissions, while retaining the low PM emissions of traditional SI engines [60,63–65]. A secondary motivation is to stabilise engine combustion at low loads to mitigate cyclic variability by replacing deflagrative combustion with more robust multi-site AI. A tertiary motivation is the desire to reduce costs, relative to diesel engines, by using cheaper, low-pressure fuel injectors and cost-effective aftertreatment devices [37].

A characterisation of various CAI strategies and a discussion on their relative strengths and weaknesses is presented next.

#### 3.1. Strategy 1: Homogeneous Charge Compression Ignition (HCCI)

This is the original ‘best of both worlds’ CAI engine concept that is a true hybrid between a conventional DI CI diesel engine and a pre-mixed SI gasoline engine. In fact, most of the discussion presented so far for CAI is directly applicable to HCCI. The use of HCCI was avoided so as not to couple CAI to just one implementation mode. Figure 2 compares various gasoline CAI modes (that are discussed below) to standard SI and CI combustion. HCCI’s placement signifies the optimum fusion of CI and SI concepts and is thus designated as a third combustion mode.

In HCCI, a diluted, lean, or stoichiometric, fully premixed (homogeneous) fuel-air-residual gas mixture is made to combust via AI by making the cylinder’s thermodynamic conditions conducive for spontaneous, self-sustaining combustion. Since the composition of the mixture is homogenous, ignition begins at multiple sites distributed uniformly across the combustion chamber. The resulting energy release is volumetric and takes place at a near-constant volume around the top dead centre (TDC). The thoroughly mixed fuel and air, which is usually in excess of stoichiometric, prevent PM production, and the low combustion temperatures thwart thermal  $\text{NO}_x$  generation.

##### 3.1.1. Mixture Preparation

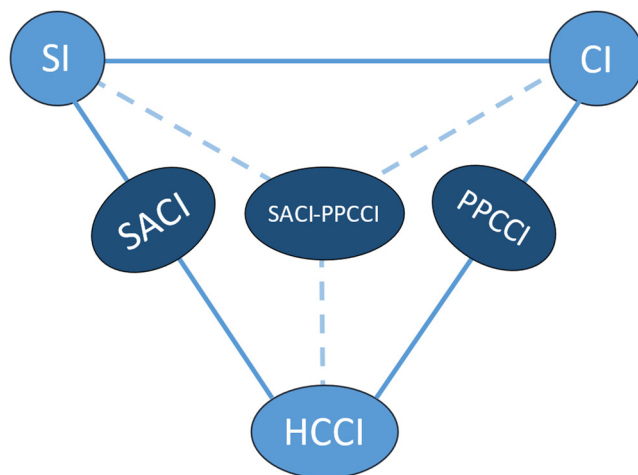
Four-stroke cycle-based gasoline HCCI engines achieve homogeneous mixing via PFI [53,64,65] or DI during the intake stroke [60,63]. Recent trends have leaned towards intake stroke DI because of improved mixing and better mixture preparation control.

Preparing a homogenous mixture is more difficult using diesel because of its poor vaporisation characteristics. Early diesel engine HCCI research explored homogeneous



fuel-air mixture preparation via PFI to allow greater time for air-fuel mixing [66,67]. Later research efforts have preferred DI during compression and expansion strokes because of improved fuel vaporisation and combustion control.

Two-stroke cycle HCCI engines have traditionally achieved homogeneity between fuel and fresh air from the direct metering of fuel into the air stream, which was then churned in the crankcase before being pushed into the combustion chamber. Newer two-stroke CAI engines use DI [59,62].



**Figure 2.** Various CAI modes and their connection to standard SI and CI combustion. Adapted from [68].

### 3.1.2. HCCI Limits

The two most-significant hurdles that have limited the adoption of HCCI engines result from its most prized feature, its combustion mode. The two hurdles are: (i) a lack of ignition and combustion phasing control, and (ii) very rapid combustion at high loads. To overcome these hurdles, multiple solutions have been proposed that try to retain as much of the efficiency and emission benefits of the best-case scenario technology. These are the CI-suffixed CAI strategies shown in Figure 2 and are discussed next. The use of the suffix CI (instead of AI) is likely an artefact of the historical efforts to deploy pre-mixed, auto-ignited combustion in CI (diesel) engines. These strategies drift from the HCCI vertex of the combustion mode triangle towards SI or CI as they attempt to address the shortcomings of HCCI by borrowing combustion management tactics from either/both of the conventional combustion modes. In one form or another, all of these strategies try to:

- i Help control the onset of combustion by providing ignition triggers to reduce the reliance on chemical kinetics and make combustion take place stably with low variability;
- ii Introduce stratification in the mixture to have the mixture be ‘pre-mixed enough’ [64] instead of being perfectly mixed to spread out energy release and moderate pressure-rise rates.

### 3.2. Strategy 2a: Premixed Charge Compression Ignition (PCCI)

Premixed charge compression ignition (PCCI) [1,23] or premixed compression ignition (PCI) [24,26], is a diesel engine strategy that came out of the challenges associated with creating homogenous fuel-air mixtures in diesel engines and the need to control and moderate HCCI combustion. In this approach, premixed rather than homogeneous cylinder gas mixtures are created by injecting fuel directly into the cylinder typically during the compression stroke at relatively advanced timings (e.g., 60°CA bTDC in [23]). This provides more time for mixing relative to diesel combustion to avoid very rich pockets, while still having relatively rich strata available to initiate AI [24,26]. The fuel-air stratification reduces the reliance on chemical kinetics for combustion initiation, and the existence of an air-fuel ratio distribution across the cylinder moderates the energy release rates. Additionally, the

premixed nature of background cylinder contents reduces soot emissions; and if appropriate levels of dilution via EGR are used,  $\text{NO}_x$  emissions can also be kept low.

In the literature, diesel engines with early compression stroke injection have sometimes [25,69] been referred to as HCCI engines, but here, they are referred to as premixed to be consistent with the mixture preparation process requirements for DI HCCI engines defined earlier, i.e., injection during the intake stroke. Moreover, as there are non-negligible heterogeneities in the combusting mixture's composition even with the relatively advanced injection, PCCI is considered a more apt characterisation.

### Fuel Injection

Most premixed CAI strategies [25,69] (including gasoline strategies discussed under strategy 2b [39,70]) employ multiple fuel injections as they allow for increased control over combustion timing and phasing. With multiple injections, the initial injections help prepare a lean, premixed background charge that is ignited by a relatively rich, combustion-triggering injection pulse close to TDC when cylinder gases are compressed and have higher temperatures. By splitting the fuel admission, energy release is spread across a wider crank angle window to avoid very rapid PRR, thus allowing more fuel to be added at higher loads without breaching PRR thresholds. The timing of the initial injection event is important. Early injection during the compression stroke when the cylinder gas density is relatively low results in higher jet penetration, which increases the likelihood of fuel adhering to combustion chamber walls [23]. This, in turn, can cause an increase in PM emissions. Therefore, designing appropriate injectors [24] and combustion chambers [26] to reduce wall wetting and/or using suitable split-injection strategies to deliver fuel in multiple, smaller jet parcels are important strategies.

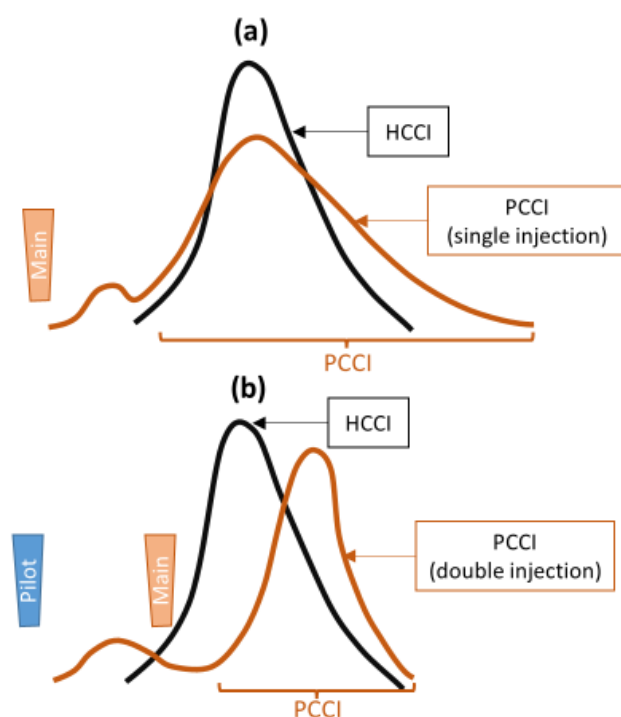
The timing and duration of the AI triggering injection pulse is also important, as it determines the gap or overlap between the fuel injection and heat release events, i.e., the ignition dwell. It is preferable to have ignition dwell, which is often taken to be an index of the extent of diffusive combustion and PM emissions [37,69], be positive. With split injection strategies, the final pulse can be shortened by metering most of the required fuel through earlier injections to maintain a positive dwell. High injection pressures can also help maintain the desired ignition dwell by injecting the required amount of fuel in a shorter period [37]. There can be accompanying fuel mixing benefits from high-pressure injection. Very high pressures can, however, cause excessive mixing, whereby the local equivalence ratios can become too lean to support combustion, and the long penetration lengths of high-pressure jets can cause an increase in PM emissions through increased wall impingement. Injection timings can have different effects on emissions depending on the specific engine combustion chamber design and injection event details. Some general trends summarised from Bobi et al. [27] are as follows: advancing injection timing causes an increase in HC CO emissions because of increased wall wetting and crevice entrapment, and it lowers  $\text{NO}_x$  and PM emissions because of reduced combustion temperatures.

High levels of EGR have been used in PCCI engines to reduce mixture reactivity and delay the start of heat release after injection [71,72]. High EGR diesel combustion can be considered analogous to using a low reactivity fuel, such as gasoline, in diesel engines, as both EGR and gasoline reduce the mixture's auto-ignitability. This is the basis of strategy 2b.

Another way to moderate PRR is to retard combustion phasing, but in HCCI combustion, retarded combustion has been found to be very unstable, both in diesel [73] and gasoline engines [53]. A late injection event in multistage injection PCCI can help stabilise retarded combustion by providing relatively rich pockets to anchor combustion across consecutive cycles. PCCI has sometimes been realised by very late (post-TDC) injection to have long ignition dwell and to promote thorough fuel-air mixing [71,74].

Figure 3 shows qualitative heat release rate (HRR) curves comparing HCCI and PCCI combustion. The top PCCI curve is for a case when diesel is injected in a single event, and the bottom one is for a two stage (pilot-main) injection strategy. Notice the relatively mild

HRR with the PCCI strategies and the ability to control combustion phasing and distribute energy release using multiple injections. The initial crests in both the PCCI curves are caused by low temperature ( $T < 725\text{ }^{\circ}\text{C}$ ) oxidation (so-called ‘cool flame’ or low temperature heat release) reactions that occur readily in diesel-like fuels [42]. Such fuels exhibit two-stage AI chemistry, i.e., some chain branching reactions start at low temperatures, during which a part of the fuel reforms to lower hydrocarbons in advance of the main heat release event. In the PCCI cases, fuel injected during the compression stroke can undergo such reformation to release some of the fuel’s energy. Up to 10% of the total heat release can take place through such reactions [19]. Low-temperature heat release is generally not observed in gasoline engines running on high-octane-number gasoline fuels under typical intake conditions [20]. PCCI combustion can thus be considered a two-stage process, comprising a low temperature heat release stage followed by a main heat release stage. The main heat release event can be further split into two events, a premixed phase followed by a late mixing controlled combustion phase [27].



**Figure 3.** Qualitative rate of heat release plots for (a) single and (b) double injection PCCI compared to HCCI. Based on HRR curves in [25–27,69].

### 3.3. Strategy 2b: Partially Premixed Charge Compression Ignition (PPCCI)

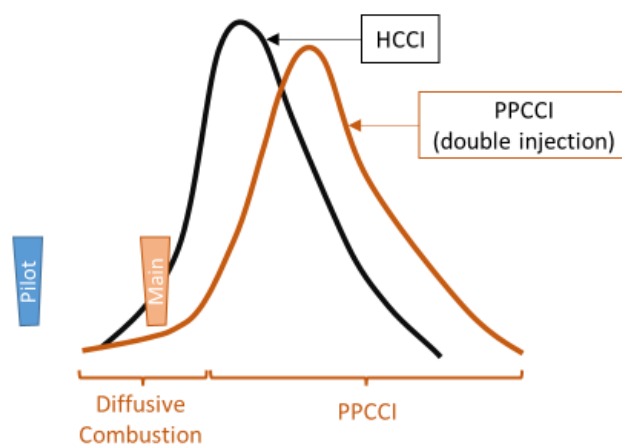
PCCI attempted to temper diesel HCCI to make it more manageable and controllable. Diesel, however, is an inherently poor fuel choice for HCCI combustion. Its high autoignition propensity (high CN) makes it susceptible to premature AI during fuel-air mixing processes at typical diesel engine compression ratios. According to Kalghatgi et al. [37], a compression ratio of 10:1 would be needed to allow injected diesel sufficient time to mix with air to form a well-mixed mixture. Conducting this so would, however, remove one of the biggest thermodynamic advantages that diesel engines possess and thus lower their efficiency. Hence, a low CN fuel, such as gasoline, can be a more-appropriate choice for moderating HCCI. This is the basis of the PCCI concept adaptation for gasoline engines, which is referred to as partially premixed auto-ignition (PPAI) [69] or partially premixed compression ignition (PPCI) [39]. In this paper, it is referred to as partially premixed charge compression ignition (PPCCI) to have consistent terminology with its diesel precursor (PCCI). Kalghatgi and Johansson described PPCCI combustion as the burning ‘of a stratified fuel cloud with a mixture of diffusion and premixed flames together with some bulk autoignition’ [37].

PPCCI can be realised in engines with compression ratios typical of diesel engines if thermal assistance is available (e.g., as intake air heating or charge heating via iEGR) to promote the AI of gasoline. Without any thermal AI assistance, gasoline fuels would require compression ratios in the range of 18–20:1 for AI. PPCCI engine compression ratios range from 16 to 18:1 [39,41,68,75]. Compared to diesel engines, PPCCI engines require relatively low fuel injection pressures because of gasoline's volatile nature, which reduces the need for atomisation to vaporise fuel, and the relatively greater time available for mixing.

In PPCCI, similar to PCCI, heterogeneity is purposefully introduced in the cylinder mixture to create a non-uniform equivalence ratio distribution to moderate and stabilise combustion. From the surveyed literature, this is always performed using multiple direct injections [41,68,69,75–78]. The first fuel injection typically takes place during the intake or early compression stroke (e.g., 150–170°CA bTDC [69,75]) but can take place later too (e.g., 26–31°CA bTDC [70]). According to Kalghatgi et al. [37], for high-ON fuels, such as gasoline, injection at 60°CA bTDC is sufficient to form homogenous mixtures for combustion. The first PPCCI injection is followed by additional injections (typically 1–3), with the last one phased around TDC to serve as an AI trigger (e.g., 16–0°CA bTDC [75]).

The intermediate injection(s), if present, create(s) a lean stratified charge, and the last injection generates a rich cloud that ignites relatively easily (shorter ignition delay). The triggering fuel injection yields a relatively rich mixture, e.g., the third injection in Liu et al. [41] prepared a fuel-air equivalence ratio 0.49 mixture, compared to 0.24 and 0.2 for the second and first injections, respectively. The proportion of fuel mass injected during the pilot and main injection events also affects engine efficiency and emissions and combustion noise [75]. Thus, injection parameters have to be selected to realise appropriate levels of fuel stratification in a globally lean mixture to optimally control ignition and combustion progression. Dempsey et al. [77] compared the relative performance of a PPCCI engine across a range of stratification levels.

Figure 4 shows a qualitative comparison of heat release traces between HCCI and two-stage-injection PPCCI. The initial rise in the PPCCI trace is because of the diffusive combustion of the ignition-triggering injection, which is followed by a rapid rise when the background lean mixture auto-ignites. The HRR, and consequently the PRR, for PPCCI is lower than that for HCCI, thus allowing for more fuel to be injected and extending the high load limit. Moreover, no observable low-temperature heat release from the compression of the early injected fuel is present [69]. The lack of early heat release allows for the injection of large quantities of gasoline during compression without concerns about premature AI. Note, however, that such reactions can occur in gasoline CAI engines under certain in-cylinder thermal and chemical conditions [79].



**Figure 4.** Qualitative HRR plots for HCCI and double injection PPCCI. Based on HRR curves in [41,69].

Recently, a new variant of PPCCI that incorporates diffusive combustion in addition to AI has been demonstrated by *Sellnau* and co-workers to extend the high load limit in a heavy duty engine to as high as 23 bar BMEP [79–81]. The strategy, referred to as ‘PPCI-diffusion’ has a late compression pilot injection (30–40° CA bTDC), which triggers an AI stage, followed by a second injection close to TDC, which initiates a diffusive combustion stage. A similar two-stage (AI + diffusion) gasoline combustion strategy was used by *Hanson et al.* [59] in a two-stroke opposed-piston PPCCI engine at high load points. High diesel-like injectors with injection pressures ranging between 1000 and 2000 bar were used in both the PPCCI-diffusion combustion engines.

#### PPCCI Limits

While PPCCI enables the extension of high load limits, there are still limits to this extension, as at very high loads, PRR becomes prohibitively high. Therefore, EGR is also used to moderate energy release rates; to counter the effects of excessive dilution, PPCCI engines are generally supercharged to further extend the high load limit [68,69]. Recent PPCCI work has also used ‘e-boost’ for operating the engine at high loads [41].

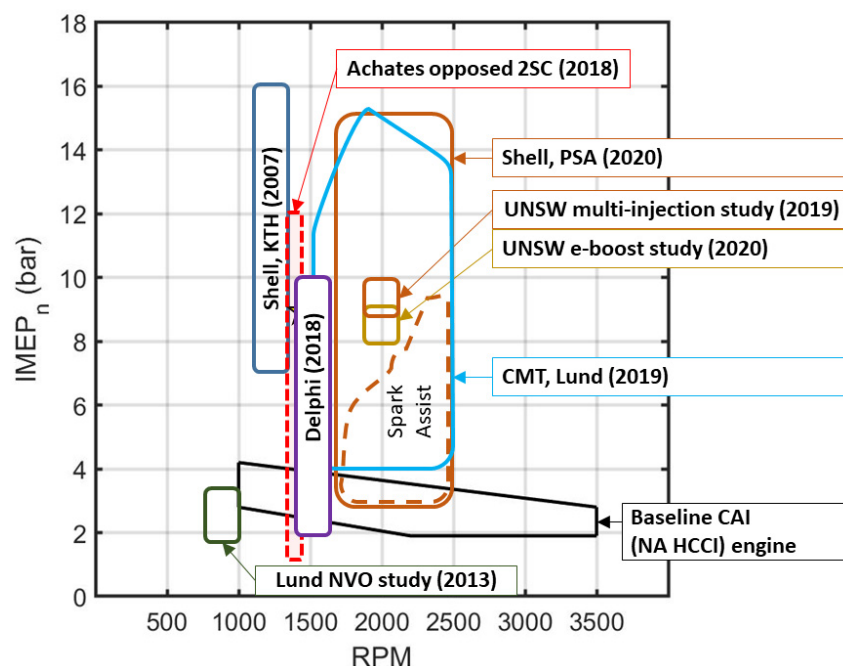
Since PPCCI engines do not have a spark plug, they cannot switch to conventional SI operation at high loads, which would be analogous to PCCI engines reverting to conventional CI at very high loads, where PRR could not be moderated using EGR, boost, and multiple injections [25,26,71,72,74]. This is one of the motivations behind the development of spark-assisted compression ignition engines (strategies 3 and 4). PPCCI engines can potentially switch to a different ‘conventional’ combustion strategies, namely diffusive (CI) combustion at the highest loads [59,79].

In PPCCI engines, low load operation can be a challenge because of gasoline’s poor AI characteristics and the relatively low temperatures at low loads. A few approaches that can enable stable low-load operation in PPCCI engines are:

- Using variable valve actuation techniques, such as exhaust rebreathing [39] or negative valve overlap [76], to raise cylinder temperatures by trapping or recycling hot combustion products (iEGR) to promote AI.
- Using a spark or a glow plug to assist with the combustion of the stratified mixture created by a late injection event [68]. This is different from spark-assisted compression ignition because ignition phasing is controlled by the late injection event and not the spark discharge.
- Avoiding low load operation by deactivating cylinders in multi-cylinder engines, e.g., in *Cracknell et al.* [68] one of the four cylinders was deactivated at low loads, and up to a 33% improvement in fuel consumption was reported.

The benefit of the latter two approaches is that they do not require expensive variable valve actuation systems and can be implemented using standard engine devices.

Figure 5 presents a speed-load map of various PPCCI studies surveyed and compares them to the baseline CAI engine from Figure 1. It can be seen that the high load limit was extended significantly with net IMEP, as high as 16 bar for a four-stroke engine [69] and 12 bar for a two-stroke engine [59], which is equivalent to 24 bar for four-stroke engines if the power enhancement from the doubling of the firing frequency is assumed to be twofold. Practically, specific power improvement realised in two-stroke engines ranges between 20 and 60% [82]. A tabulated summary of the various studies from Figure 5 is provided in the Appendix A (Table A1).



**Figure 5.** Operating regions of various PCCI engines surveyed (Table A1). BMEP has been converted to net IMEP by assuming 90% mechanical efficiency.

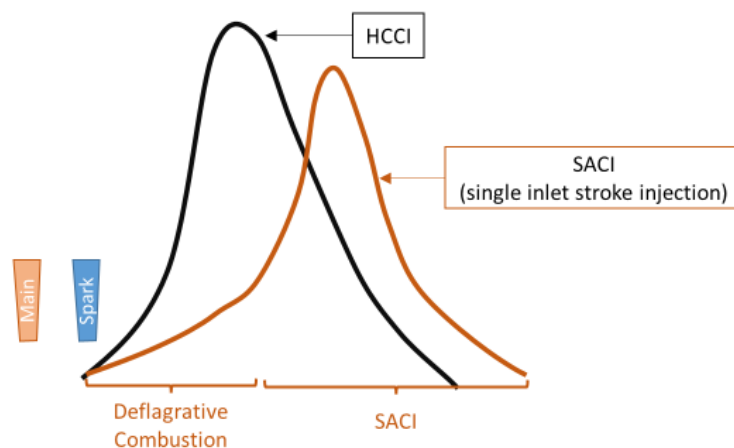
### 3.4. Strategy 3: Spark Assisted Compression Ignition (SACI)

Another CAI strategy that was developed to address the shortcomings of HCCI combustion is spark-assisted compression ignition (SACI) [63,65], which is also referred to as spark-controlled compression ignition (SPCCI) [47,48] or spark-ignited compression ignition [53]. In it, combustion is triggered in a well-mixed CAI (i.e., HCCI) engine via spark discharge, and the subsequent energy release is moderated by retarding combustion phasing and employing EGR and supercharging as needed.

The sequencing of combustion in SACI engines is as follows: a near-homogeneous mixture is prepared by early (generally during the intake stroke) direct fuel injection. A spark discharge event around firing TDC triggers deflagrative combustion. The advancing flame causes the temperature of the cylinder charge to rise through compression heating, which lowers the mixture's ignition delay and eventually triggers AI. Thus, a two-stage combustion process is realised, comprising an initial slow deflagrative stage, in which some of the fuel's energy is released, and a second AI stage that rapidly burns the remaining fuel present as a lean, homogenous fuel-air-residual gas mixture. The first stage is similar to normal combustion progression in GDI SI engines, whereas the second stage is similar to abnormal SI combustion caused by end-gas AI (i.e., knocking). Instead of being avoided, end-gas AI is purposefully initiated and managed by creating favourable conditions for its onset, e.g., higher compression ratios and warmer in-cylinder conditions. Figure 6 presents a qualitative comparison of SACI and HCCI combustion using fabricated HRR profiles. The lower HRR for SACI demonstrates its combustion-moderating influence. SACI heat release is slow at the start during deflagrative combustion and becomes faster during the main AI stage. Since both PCCI (Figure 4) and SACI are attempts at reigning in the aggression of HCCI combustion, there are similarities between their HRR profiles.

The phasing of SACI combustion is controlled by: (i) spark timing, (ii) the speed of flame-propagating combustion and the fraction of energy released through it, and (iii) the ensuing AI, the nature of which is determined by the composition and thermodynamic state of the post-deflagration mixture. Of these determinants, the principal is the spark timing. Having the correct mixture state, composition, and in-cylinder flows at the time of spark firing is thus critically important. Instabilities in initial flame propagation and early kernel growth at the dilute and lean conditions typical of HCCI combustion are major

sources of cyclic variability in SACI combustion [65,83]. This is particularly problematic at low loads where the mixture temperatures are low and cause the further slowing down of flame initiation processes [65]. Olesky et al. [63] found that, for constant combustion phasing, AI stage reactions started at around 765 °C, irrespective of the spark timing.



**Figure 6.** Qualitative HRR plots for SACI and HCCI. Based on HRR curves in [53,63].

The nature (speed and proportion) of the deflagration combustion stage can be tailored by controlling the temperature and composition of the cylinder mixture and, through that, the onset of spark-assisted AI. The flame propagating stage can be made longer at higher loads to avoid excessive PRR, e.g., by increasing external EGR (eEGR), decreasing iEGR, or decreasing intake air temperature or pressure. In Olesky et al. [63], 18 to 34% of the total energy release was made to take place via SI by adjusting the ratio of iEGR to eEGR. In Xie et al. [65], iEGR was reduced by decreasing positive valve overlap, which resulted in heat release becoming similar to that of SI combustion.

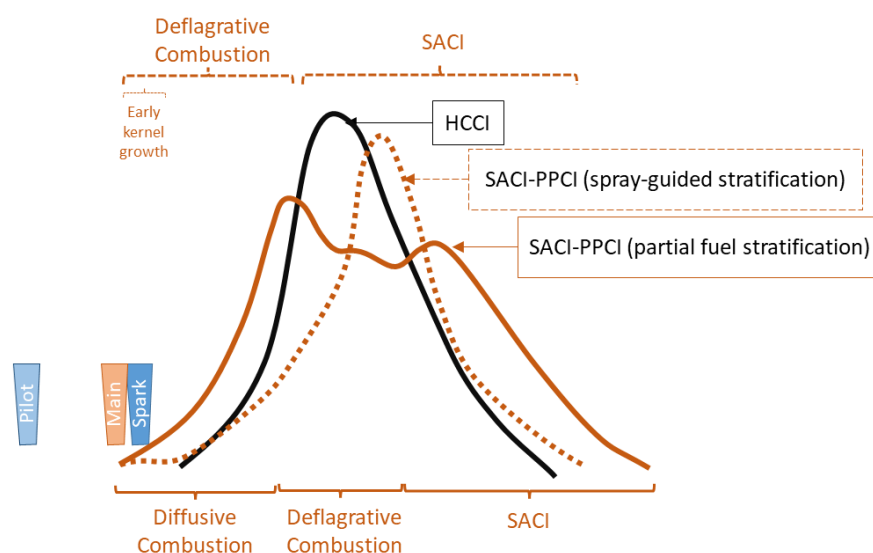
Since SACI engines are equipped with a spark plug, they can transition to conventional SI operation in regions where CAI is not feasible by making appropriate adjustments in the mixture preparation and gas exchange processes, e.g., switching to an overexpanded (Miller-type) cycle, mixture enleanment, and/or increasing EGR to avoid SI knock [84]. Operating range coverage examples from SACI engine studies are shown in Figure 8.

### 3.5. Strategy 4: SACI-PPCCI Hybrid

As the name suggests, this strategy is a combination of strategies 2b and 3 and is thus placed in between the two in Figure 2. It is categorised separately because it utilises moderating influences from both of its constituent CAI strategies, stratification via multiple DI events from PPCCI and deflagrative combustion from SACI. In the literature, however, it is more commonly referred to as SACI implementation [30,47,53]. Strategies such as SACI-PPCCI hybrid are reminders that various means of characterising CAI strategies are merely tools to facilitate structured discussions on the topic and are not constraints to CAI implementation.

In the SACI-PPCCI hybrid strategy, a late second injection event takes place shortly before (or at the same time as) spark firing in an SACI engine [83]. The additional injection introduces localised stratification around the spark plug with a homogenous background charge. This is conducted to have locally rich strata to make SI more stable and to accelerate flame propagation [53,85]. The stabilisation provided by the second injection makes operation at more retarded spark timings possible, where typically lean and diluted HCCI mixtures experience very unstable combustion [53,73]. Retarding combustion phasing by decreasing iEGR without additional support from fuel stratification reduced combustion stability in Olesky et al. [63]. SACI-PPCCI hybrid engines can have more than two injections, e.g., Hu et al. [85] employed 1–3 early injections around 300°CA bTDC to form the background well-mixed mixture.

Flame imaging experiments by Hu et al. [85] found that the pre-spark injected fuel in a ‘partially fuel stratified’ engine acted as a locally rich diffusively burning ‘super igniter’. The injector was located such that the fuel jets straddled the spark plug electrode. The resulting energy release was three orders of magnitude higher than that from a standard spark discharge. The following deflagrative stage burned a well-mixed lean mixture, and the final AI stage consumed the remaining very lean end-gas. The resulting HRR profile for such SACI-PPCCI combustion (qualitatively illustrated in Figure 7) is a combination of PPCCI and SACI profiles and has three stages: a diffusive PPCCI stage, a deflagrative SACI stage that has lower HRR than the preceding diffusive stage, and a stronger late AI stage. The nature of the first combustion stage can vary depending on the spatial and temporal distribution of the stratified fuel mixture around the spark plug. Flame imaging results from Reuss et al. [83], in which fuel injection jets were targeted just below the spark plug gap and stratification was achieved using spray-guidance (instead of wall or flow guidance [29]), did not reveal a diffusive combustion stage. Combustion progressed, as qualitatively illustrated in Figure 7, in the following stages: early kernel growth, flame propagation, and AI. Depending on the nature of fuel stratification, SACI-PPCCI HRR profiles can resemble one of the profiles shown in Figure 7 or be a combination of the two. Reuss et al. [83] also provided useful HRR and flame area-based ‘engineering metrics’ that can be used to index SACI combustion progression.



**Figure 7.** Qualitative HRR plots for SACI-PPCCI combustion. Based on HRR curves in [83,85].

SACI-PPCCI provides flexibility in tailoring energy release profiles by controlling fuel injection and spark discharge events that determine the fraction of each of the three combustion stages. CA50 was found to correlate strongly and linearly with spark timing, demonstrating that spark timing is a strong overall combustion phase controller [85].

As discussed for PCCI and PPCCI, late fuel injections can cause diffusion combustion, which can increase NO<sub>x</sub> and PM emissions. Higher than expected, yet still low, NO<sub>x</sub> emissions observed in Yoshizawa et al. [53] were attributed to the combustion of the relatively rich stratified mixture around the spark plug.

#### Operating Strategies

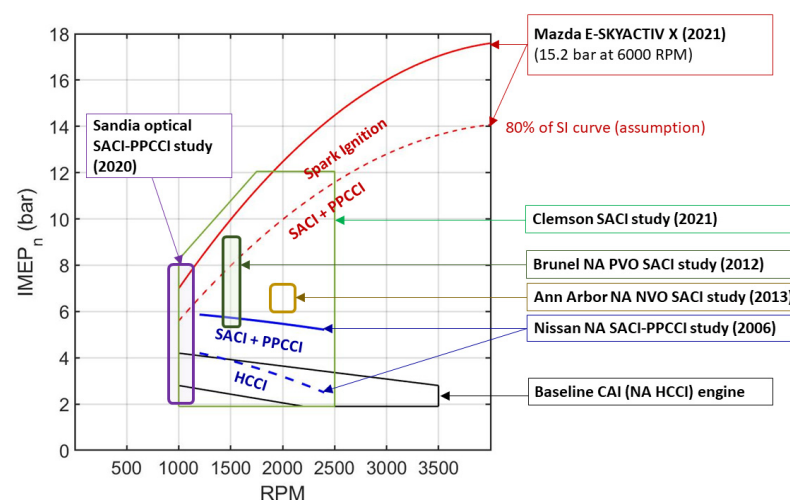
Because of the flexibility in tailoring combustion strategies, the engine operation of SACI-PPCCI can be switched between SI, SACI, SACI-PPCCI, HCCI, and PPCCI as needed to cover the desired operating region. Moderated HCCI strategies (SACI, PPCCI, or SACI-PPCCI) can be used at high loads to manage PRR, and HCCI can be used at low loads to benefit from its thermal and combustion efficiency advantages. If CAI is not possible at low loads, conventional SI operation can be used, and if PRR is excessive at very high



loads, CAI combustion can be switched to SI with accompanying adjustments in mixture preparation and gas-exchange settings. An example of a commercial SACI-PPCCI engine is presented next.

Mazda's SKYACTIV X [47] engine and the e-SKYACTIV X [48] variant for mild hybrid applications are the first class of four-stroke gasoline CAI engines to reach the market. They are GDI SI engines with high compression ratios (16:1 and 15:1, respectively) and use a split fuel-injection strategy to realise SACI-PPCCI CAI. The pilot injection takes place during the intake stroke, and the main injection occurs during the compressions stroke. The fuel is injected at the moderately high pressure of 500 bar, and the engines run very lean with an air-fuel ratio of 40:1 in CAI mode [30]. The engines also use turbulence in the form of strong swirl and tumble flows to help vaporise the injected fuel and to optimally direct it around the spark plug. A clutched supercharger is used to boost the engine as needed and is disengaged when natural aspiration is desired. A variable valve timing system creates positive valve overlap to help scavenge hot exhaust gases or operate the engine on an over-expanded cycle by early intake valve closing as needed. The SKYACTIV X engine covers most (80%) of its operating range under CAI operation and switches to conventional stoichiometric SI operation at very high loads. This is probably because the PRR moderation provided by spark assist is not sufficient at the highest load points. The engines operate in SI mode during cold start and warm up periods as well, likely due to cylinder contents being too cold to achieve stable AI. A resulting fuel economy improvement of up to 30% has been reported. The rated torque and power for Mazda e-SKYACTIV X translate to a BMEP of 13.7 bar at 6000 RPM and 15 bar at 4000 RPM [48]. The engine's operating envelope is compared to that of the baseline CAI engine and other SACI studies surveyed in Figure 8. A summary of the referenced studies is provided in Table A2 in Appendix A. A joint reading of both highlights that:

- i SACI alone cannot adequately extend the CAI operating envelope. Assistance is needed from PPCCI to provide stable, lean combustion across a wide operating range.
- ii SACI-PPCCI hybrid engines allow for using relatively high compression ratios because of the availability of two ignition control knobs (spark discharge and fuel stratification) that can provide effective PRR moderation.
- iii HCCI can serve as a stable low- to medium-load combustion mode, and operation can be switched to SI operation at very high loads and cold engine conditions.
- iv Adjusting iEGR to the eEGR ratio using various valve overlap strategies is an effective way of stabilising SACI operation and extending its load limits.
- v Greater in-cylinder flow control and turbulence, relative to non-SI CAI engines, might be needed to stabilise and accelerate combustion during the deflagrative stage.



**Figure 8.** Operating regions of various SACI and SACI-PPCCI engines surveyed (Table A2). BMEP was converted to net IMEP by assuming 90% mechanical efficiency.

### 3.6. Strategy 5: Gasoline Compression Ignition (GCI) [Redundant]

In the literature, the term gasoline compression ignition (GCI) [37,41,53,59,77] or gasoline direct injected compression ignition (GDICI) [39,68] is used as well to refer to gasoline CAI. It is more commonly used to refer to gasoline PPCCI engines [1,39,41,68,77], but it has been used for gasoline SACI engines as well [53]. A reasonable case for the use of the term for both the gasoline CAI strategies can be made, but its use in the PPCCI context seems more appropriate, as the technology is a direct offshoot of diesel engines and is often implemented directly in diesel engines. Additionally, there is no spark plug in the cylinder, and the prime initiator of AI is compression. For this reason, PPCCI engines typically have higher compression ratios than SACI engines. PPCCI-SACI hybrid engines have compression ratios between the two (around 15:1). Because of the relative vagueness of the term, the use of ‘GCI’ is avoided in this paper.

### 3.7. Strategy 6: Reactivity Controlled Compression Ignition (RCCI)

Reactivity controlled compression ignition (RCCI) combustion is a CAI strategy in which two fuels—a high-reactivity (high-CN) and a low-reactivity (low-CN) fuel—are used [86,87]. Generally, the high reactivity fuel is diesel, and the low reactivity fuel is gasoline. Appropriate amounts of each are metered to tailor the reactivity of the combusting mixture to obtain desired energy release profiles. Natural gas can also be used as a low-reactivity fuel [88]. Because the scope of the present work is focused on single gasoline fuel-based systems, RCCI is discussed very briefly below.

In RCCI, typically, a homogenous mixture is formed in the cylinder by the port fuel injection of gasoline. The mixture is then compressed, and because of its low reactivity, it does not auto-ignite during compression. AI is triggered around TDC in a conventional CI combustion-like fashion by the injection of diesel into the hot compressed mixture of gasoline and air. The diesel combustion triggers the AI of the homogenised gasoline mixture that combusts rapidly in a premixed combustion event. The amount of low- and high-CN fuels admitted, along with the high CN fuel’s injection timing, can be varied for various speed and load conditions to achieve stable combustion with low NO<sub>x</sub> and soot emissions. The challenge with this approach is that it requires two fuels, which increases the system cost and level of complexity. Octane-on-demand technologies [1] can potentially remove the need for carrying two separate fuels.

In a recently proposed RCCI variant ‘intelligent charge compression ignition’, the low-CN fuel is directly injected into the cylinder instead of being injected into the intake port. This offers better control over the low-CN–high-CN fuel stratification [89]. Another RCCI variant being developed, especially for electrified powertrains, is the ‘dual-mode dual-fuel’ concept, in which the relative proportion of the two fuels and diffusive and premixed combustion stages are adjusted to cover a wide operating map [90].

Graphical and tabulated summary comparisons of the various CAI concepts discussed here are presented in Figure 9 and Table 2, respectively. They highlight differences in fuel injection strategies and mixture formation.

**Table 2.** Comparison of various engine combustion systems.

Strategy	Fuel	Injection	Combusting Mixture	Spark Plug	Combustion Timing Control
SI (PFI)	Gasoline	PFI	Homogenous	Yes	Spark timing
SI (DI)	Gasoline	DI typically during intake	Stratified	Yes	Spark timing
CI	Diesel	DI during compression	Highly stratified	No	Injection timing
HCCI	Gasoline	PFI or DI during intake	Homogenous	No	Chemical kinetics (AI chemistry)

Table 2. Cont.

Strategy	Fuel	Injection	Combusting Mixture	Spark Plug	Combustion Timing Control
PCCI	Diesel	1–2 DI during compression	Partially mixed background, possibly with stratified injection region	No	Chemical kinetics and/or last injection timing
PPCCI	Gasoline	DI during intake/compression + second DI late into compression	Homogenous background charge, stratified injection region	No	Last injection timing
RCCI	Gasoline + Diesel	PFI (Gasoline) + DI (Diesel) around TDC		No	Diesel injection timing
SACI	Gasoline	DI during intake	Homogenous	Yes	Spark timing
SACI-PPCCI	Gasoline	DI during intake/compression + second DI late into compression	Homogenous background charge, stratified injection region	Yes	Spark and last injection timing

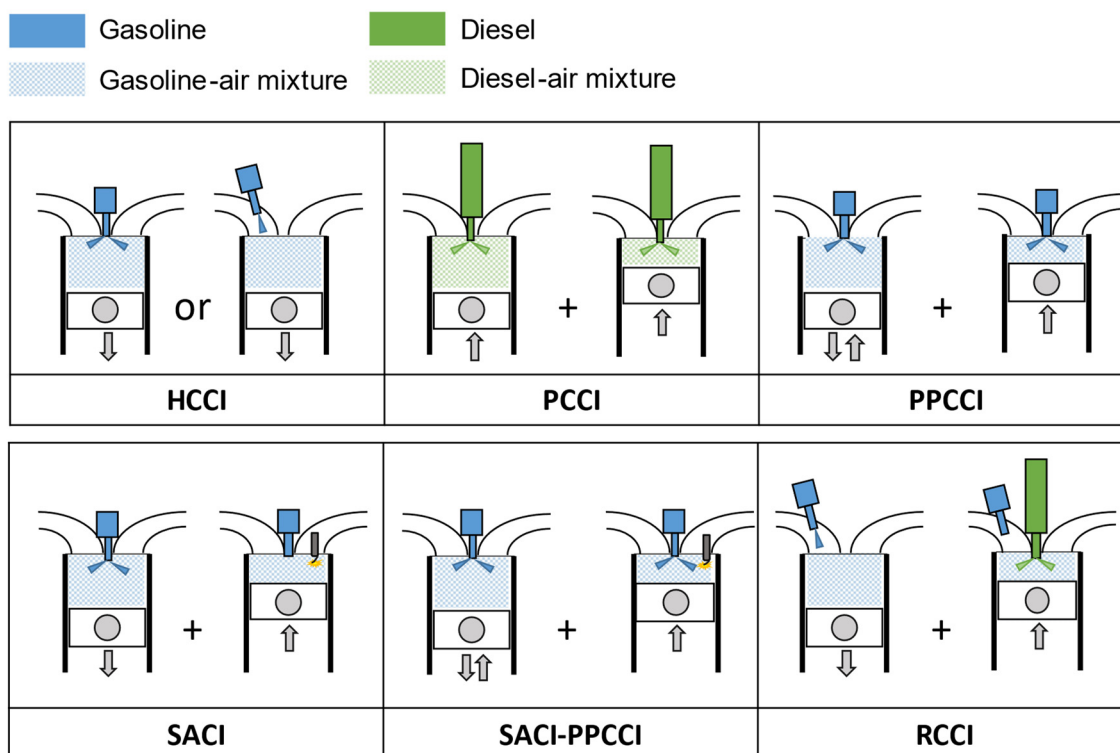


Figure 9. An overview of mixture preparation processes in various CAI technologies. Adapted from [1].

#### 4. Controlling CAI Combustion

One of the biggest challenges for CAI engine development has been combustion control. In actual engines, a host of control parameters have to be simultaneously adjusted to control CAI across the desired operating range [91]. This entails precisely controlling the chemical composition and the thermodynamic state of the cylinder contents to initiate combustion, manage heat release, and ensure adequate chemical energy availability and stable combustion, all within emissions and drivability constraints while trying to maximise efficiency. A list of potential CAI control actuators that can be used, together or in isolation, to meet these combustion control requirements is presented below.

#### 4.1. Exhaust Gas Recirculation

Combustion products can be recirculated or retained to moderate or accelerate combustion. Different EGR effects can have competing influences on combustion. Zhao [19] characterised these effects as follows:

- i Charge heating—combustion products by virtue of their temperature can increase the temperature of the cylinder gases.
- ii Dilution effect—combustion products reduce the amount of oxygen in the trapped mixture by replacing air.
- iii Heat capacity effect—combustion products contain CO<sub>2</sub> and H<sub>2</sub>O, which have relatively high heat capacities compared to that of O<sub>2</sub>. Hence, the trapped mixture's heat capacity can increase, and its ratio of specific heat can decrease.
- iv Chemical effect—partially oxidised products of combustion, e.g., CO, HC, or NO can participate in chemical reactions in the trapped mixture.

For CAI combustion, charge heating, dilution, and heat capacity effects are of primary concern. Charge heating can increase the chances of AI and make deflagrative combustion faster, whereas excessive dilution can hamper combustion by starving it of oxygen. The heat capacity effect can reduce compression temperatures and pressure to thwart AI. iEGR is primarily used to exploit the charge heating effect, whereas eEGR can be tailored via an eEGR cooling system to exploit charge heating, dilution, or heat capacity effects.

Because of these competing effects, EGR on one hand is used to extend the lower operating limit by increasing charge temperature to ensure stable AI, and on the other hand, it is used to extend the high load limit by lowering charge temperature (heat capacity effect) to allow for the admission of additional fuel. An unintended consequence of the foregoing is that the dilution effect makes combustion unviable at higher loads because of insufficient oxygen availability, leading to misfires or partial fires [55]. EGR can also be needed as a charge heater to extend the high load limit if the trapped gas temperatures are not high enough for realising AI.

Depending on the net effects of combustion product retention or recirculation, the effects on emissions could be positive or negative. NO<sub>x</sub> emissions can decrease because of lower temperatures. PM emissions can increase if insufficient oxygen is available for soot oxidation unless temperatures are lowered below the soot-formation threshold.

#### iEGR Strategies via Variable Valve Actuation

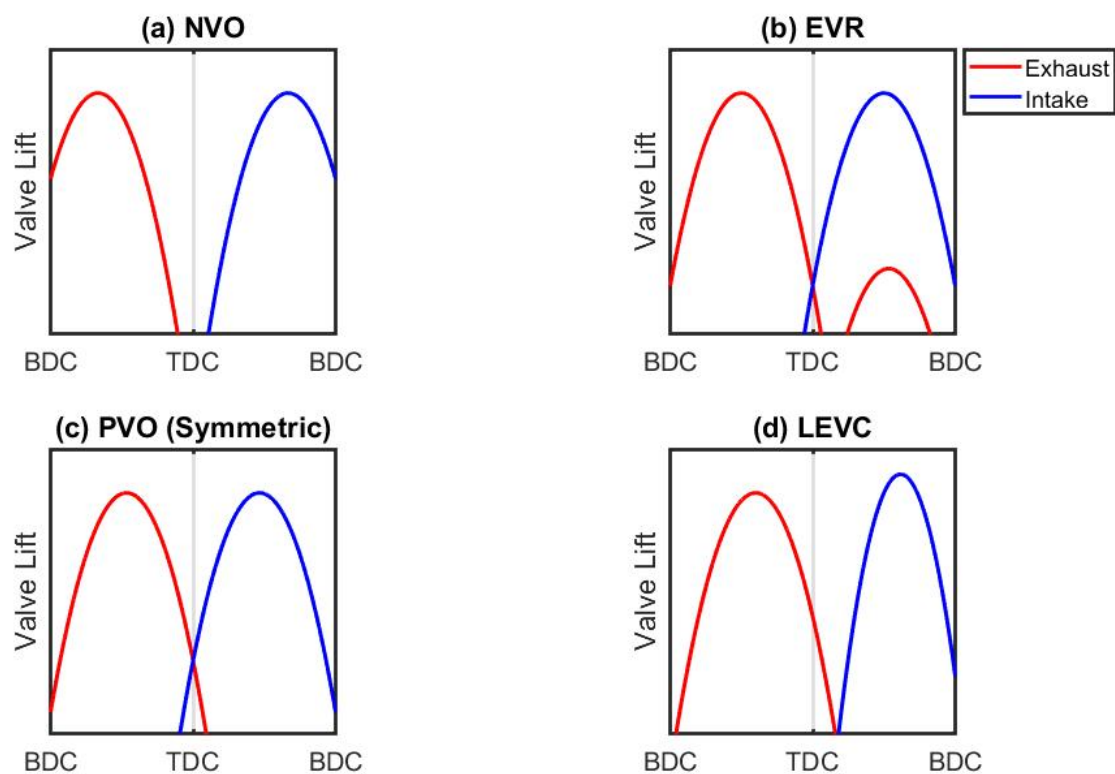
Valve timing, duration, and lift adjustments, i.e., variable valve actuation, can be used to control the amount of trapped residual gases. Variable valve actuation systems can offer different levels of flexibility, ranging from variable valve timing (VVT) to continuous variable valve lift (CVVL) systems [92]. Owing to the tremendous potential of improving engine efficiency (10–12% according to [54]) and emissions by flexibly controlling engine breathing, many major automotive companies have tried to develop such systems, e.g., BMW 'valvetronic' [93,94], Lotus 'active valve train' [95], Toyota 'valvematic' [96], Nissan 'variable valve event and lift' [97], and Fiat/Schaeffler 'Multi/UniAir' [98]. Despite some of these technologies having made their way into production engines (e.g., [98,99]) their cost and concerns over reliability and accuracy have restricted their widescale adoption [98].

Many of the surveyed CAI studies used iEGR via valve opening and closing control to support CAI (see Tables A1 and A2). The valve actuation approaches used by these studies are summarised below, and a graphical comparison is presented in Figure 10.

- i Negative Valve Overlap (NVO): Exhaust valves are closed before TDC, often accompanied by a late opening of the intake valves, to retain more combustion products than would have been retained with exhaust valve closure at/after TDC (Figure 10a). The residual gases are recompressed during the latter part of the exhaust stroke, which increases their temperature and can offer AI benefits NVO periods ranging between 60 and 220°CA were used in Borgqvist et al. [76] and between 114 and 136°CA in Xie et al. [65]. Some of the fuel's energy can be released during the NVO period

by injecting the fuel during or before it if the cylinder conditions are conducive for low-temperature heat release [76,100]. With NVO strategies, a pumping loss penalty is incurred because of recompression. These losses increase at higher loads because of higher expansion pressures [65]. Therefore, NVO might be preferable only at lower loads, where it provides hotter iEGR and greater thermal stratification [54].

- ii Exhaust Rebreathing via Exhaust Valve Reopening (EVR): To reduce pumping losses incurred because of NVO, recompression can be avoided by having a second exhaust valve opening event, typically during the intake stroke (Figure 10b). Some exhaust gases from the exhaust port would thus be readmitted into the cylinder and heat the fresh charge. Because of the recirculation of the exhaust gases, the charge heating potential of EVR iEGR is lower than that of NVO iEGR. Sellnau et al. [39] used exhaust rebreathing to extend their PPCI engine's low load performance and also to elevate exhaust temperatures to improve aftertreatment system efficiency.
- iii Exhaust Rebreathing via Positive Valve Overlap (PVO): In this approach, intake and exhaust valves are opened simultaneously beyond the overlap periods typical of automotive engines (Figure 10c). Adequate iEGR for CAI can be retained through this approach if intake and exhaust valve timings, about or on one side of TDC, are selected smartly [65]. Pumping losses are lower than for NVO with this rebreathing method as well, which makes it promising for high-load CAI operation. Xie et al. [65] realised CAI using PVO in an SACI engine. SACI engines typically require relatively less thermal assistance for AI because of the availability of a positive ignition source in the form of a spark.
- iv Exhaust Rebreathing via Late Exhaust Valve Closing (LEVC): In this exhaust rebreathing strategy, exhaust valves close after TDC, and some exhaust gases are rebreathed from the exhaust port during the expansion stroke (Figure 10d) [54]. This method produces relatively high levels of thermal stratification but not as high as NVO. It also has lower pumping losses than NVO.

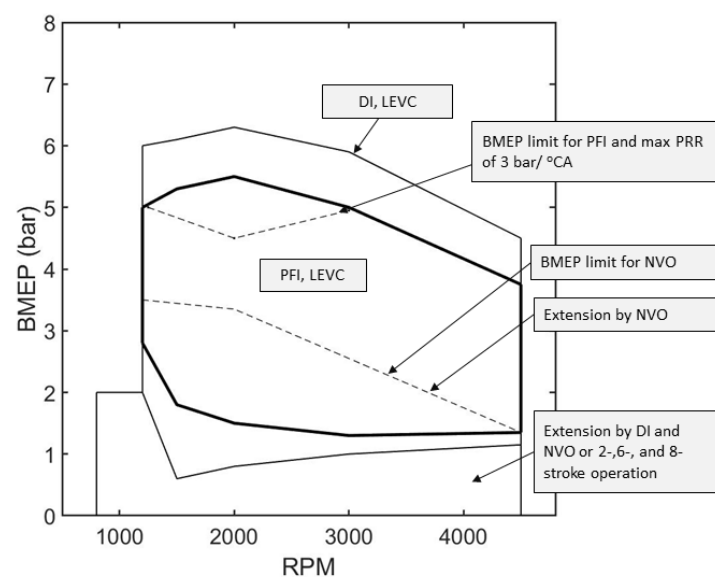


**Figure 10.** Comparison of (a) negative valve opening, (b) exhaust valve reopening, (c) positive valve overlap, and (d) late exhaust valve closing valve lift profiles.

Another strategy that has sometimes been used to adjust iEGR levels is engine exhaust throttling using a back pressure valve [39,101]. Increasing exhaust back-pressure traps more iEGR and thus supports CAI.

#### 4.2. Cycle Type

A potential way of extending the high load limit is by distributing energy release over multiple revolutions of the crankshaft instead of concentrating it all in one dedicated 'power stroke' taking place every other revolution. The most promising, and arguably the most researched, means of conducting this is by operating a two-stroke cycle, whereby the firing frequency would be doubled relative to four-stroke cycles [46], thus lowering the mean effective pressure requirements. Two-stroke engines also have the advantage of retaining high quantities of residual gases (iEGR), which can be beneficial in promoting CAI [102]. CAI operation has been utilised in two-stroke engines with various scavenging configurations. Examples include ported cross-scavenged, ported loop-scavenged, poppet-valved loop-scavenged, uniflow-scavenged with intake ports and exhaust poppet valves, and ported uniflow-scavenged designs [46,49]. To avoid significantly redesigning the valve-in-head architecture of automotive engines, the poppet-valved, loop-scavenged two-stroke cycle engine is an enticing option. Two-stroke cycle operation can be realised in a flexible four-stroke engine by changing valve timings via the CVVL module and adjusting spark and fuel injection timings accordingly. Switching from two- to four-stroke cycles can be achieved during engine operation to leverage the benefits of both cycles at appropriate operating points [54,61]. Lang et al. [54] successfully realised CAI across a wide operating map in a gasoline, dual-injection (PFI + DI) engine equipped with an electromechanical CVVL system (Figure 11).



**Figure 11.** The extension of CAI operation through iEGR management and cycle switching using a CVVL system. Adapted from [54].

Other cycle options that could be investigated are six- and eight-stroke cycles that have two consecutive firing strokes followed by gas exchange or redundant strokes [54].

#### 4.3. Low Load Avoidance

Another strategy to help control CAI can be to restrict engine operation to moderate- and high-load operating points, where AI is easier to achieve and CAI combustion is more stable. Moreover, for throttled engines, this would avoid operation at points with the worst pumping losses. Possible ways of avoiding low-load operation are:

- i Cylinder deactivation at low loads to force the active cylinders to operate at higher pressures and temperatures (i.e., at higher loads) [68]. Engine cylinders can be deactivated by closing valves [54], stopping fuel injection and spark plug firing [1]. In a CVVL-capable engine study, additional 6–10%-point fuel consumption benefits were realised through cylinder deactivation [54].
- ii Load levelling via powertrain electrification to shield the engine from low-load operation and transient changes in driving power demand by providing propulsion (entirely or partly) via the electric traction motor. This is discussed in more detail in Section 5.

#### 4.4. Boosting

An effective way to counter the EGR dilution-imposed limit on fuel addition at high loads is through boosting, whereby intake pressure is increased using a compressor. Hence, more air is forced into the cylinder, allowing for more fuel to be burned and higher torques to be generated.

Boosting can be achieved through exhaust-driven turbochargers, mechanical superchargers that withdraw some work from the engine's crankshaft, or electric superchargers that have an electric motor driving the compressor. Electric superchargers, also known as e-boost systems, can promptly respond to the engine's aspiration needs and are not limited by turbo lag at low loads and speeds [41,103]. With the availability of high-voltage circuits in electrified powertrains, e-boost systems can be adopted more widely.

#### 4.5. Fuel Injection

As discussed for the various partially premixed CAI embodiments, fuel injection strategies can play a pivotal role in moderating combustion via fuel stratification and allowing stable, retarded combustion to extend the high load limit. Locally rich mixtures can also help achieve stable combustion at low loads. Some fuel injection design considerations for CAI engines are injector location (port and/or in-cylinder, position and angle relative to spark plug in SACI variants); number of injection events and their timing, duration, and injected fuel quantity; injection pressure, injector design, and number of injectors [77]. Injection details of some of the surveyed gasoline CAI studies are provided in Appendix A.

#### 4.6. Ignition Assistance

As discussed for SACI engines, ignition aids (positive ignition devices) can be used in CAI engines to control ignition timing and combustion phasing, which can help moderate heat release and stabilise combustion to extend low- and high-load operations, respectively. The most common ignition aid is the spark plug. Some other ignition assistance devices are glow plugs [76], pre-combustion chambers (PCC), plasma ignition, and laser ignition [85]. None of the CAI studies surveyed used PCCs or any of the unconventional positive ignition devices, but because of their ability to stabilise combustion and promote efficient energy release, PCCs might warrant consideration for some SACI strategies. In some respects, PCCs and CAI are similar because both aim to promote fast combustion in lean and dilute engines by having multiple ignition sites. Depending on the CAI strategy, employing a PCC in a CAI engine might introduce redundancy. A recent commercial (non-CAI) application of PCC, albeit in a supercar, is in Maserati's Nettuno engine [104]. Mahle's dedicated hybrid engine [105] uses their PCC, the 'Mahle Jet Ignition' system [106].

#### 4.7. Charge Heating

Intake air heating can be employed to facilitate the AI of low-reactivity fuels, such as gasoline. This can be accomplished by recycling exhaust heat using an appropriate thermal management system or through electric heaters. There could be new electric heating possibilities available because of the electrification of powertrains. Direct intake air heating instead of using iEGR can help avoid associated pumping losses. Sellnau et al. [39] used a 2.3 kW electric heater upstream of the intake valves. Air heating would, however, reduce

the engine's volumetric efficiency and consequently the output power. In the surveyed CAI studies, intake air was heated to temperatures between 60 and 180 °C [85,107]. A near 1:1 negative correlation was reported between combustion phasing (indexed by CA50) and intake air temperature for HCCI engines by Kalghatgi et al. [37], i.e., every 1 °C increase in intake temperature advanced CA50 by 1°CA. For PPCCI engines, the correlation also depends on injection timing and, for SACI engines, on spark timing [85]. The CAI stage heat release increased significantly in the SACI-PPCCI engine studied by Hu et al. [85] when the intake air temperature was increased from 30 to 60 °C. Cinar et al. [108] reported the significant advancement and strengthening of CAI in an HCCI engine when air temperature was increased to 120 °C.

For operating points where AI is to be discouraged, e.g., at high loads, intake air temperature can be lowered to increase ignition delays. This can be useful in SACI engines with high compression ratios to avoid knocking under SI operation at high loads.

#### 4.8. Engine Cooling

As CAI can be controlled by managing cylinder temperatures to encourage or discourage AI, the engine cooling system can serve as a slow speed AI control actuator. If the system is adjusted to remove less heat from the cylinders, then the cylinder contents will become hotter and AI will become easier. The opposite (removing more heat) would make AI less likely. Such an approach would have similarities to 'low heat rejection' combustion concepts [109].

#### 4.9. Compression Ratio

Another CAI control actuator can be the geometric compression ratio, which traditionally has been a fixed parameter because of the complexities associated with varying it during engine operation. If, however, reliable and cost-effective means of changing the compression ratio can be developed, then the need for controlling non-compression-based AI enablers can be reduced. Hence, CAI operating limits can be extended, including for cold start conditions, and engines can be made fuel-flexible. Compression ratios can be varied at different operating points based on the trapped mixture's reactivity. Some of the means of varying compression ratios are free pistons [18], variable length connecting rods [110], multi-link crankshaft rotation mechanisms [111], and dual (split)-piston mechanisms [112]. The performance of variable compression ratio-capable CAI engines has been the subject of multiple studies [18,113–116].

#### 4.10. Altering Mixture Reactivity

The scope of the current paper emphasises CAI for gasoline-fuelled engines; therefore, the potential benefits of altering charge reactivity by introducing non-standard species have not been explored in detail.

According to the literature, the ideal gasoline CAI fuel should have a research octane number (RON) between 70 and 80 [37,75]. This is considered a good compromise between the need for a high-reactivity fuel at low loads where cylinder conditions are difficult for AI and the need for a low-reactivity fuel at high loads where high temperature and pressure might initiate AI prematurely. Some strategies that have been discussed in the reviewed literature to chemically adjust the mixture's auto-ignitability are:

- i Introducing a high reactivity fuel in a low-reactivity fuel mixture to promote AI when desired. This is the RCCI approach.
- ii Using high-ON fuels, such as ethanol, to prolong ignition delay and thus allow operation at high compression ratios. Christensen et al. [107] realised stable CAI, albeit at relatively low loads, using ethanol in an engine with a very high compression ratio of 21:1.
- iii Introducing radical producing species, such as ozone, in the intake air [117] or partially reforming the injected fuel via low-temperature heat release [100] to lower AI temperature.



- iv Fuel blending, e.g., 80% gasoline with 20% diesel to form ‘dieseline’ that has lower ON than gasoline [37] or mixing n-heptane and iso-octane via separate injectors in intake ports to control fuel ON for HCCI operation [118], or blending hydrogen and methane [119].
- v Using low ON fuels, such as naphtha (RON 60–85 [30]) to achieve AI at low compression ratios and mixture temperatures. Such fuels can also offer a reduction in CO<sub>2</sub> on a well-to-wheel basis, as they need less processing [68].

If engine operation is limited to moderate and high load points via low-load avoidance, then optimal RON would be higher than 70. Thus, market gasoline fuels that typically have a RON greater than 70 might offer natural advantages for HPS CAI operation.

A summary of the control actuators listed above is presented in Table 3. The table also identifies whether the effects of adjusting a given control parameter can help extend operational limits of a naturally aspirated HCCI engine. Moreover, the response rates of various means of adjusting the control parameter have been characterised (in a slightly arbitrary and subjective manner) as either being ‘fast’, ‘intermediate’, or ‘slow’.

**Table 3.** CAI control parameters, their effects and benefits, and implementation options. LL, LS, and HL, respectively, identify if the effect can help extend low-load, low-speed, or high-load HCCI engine limits. ‘Interm.’ is short for intermediate.

Control Parameter/Actuator	Effects/Benefits	Implementation Strategies	Response Rate
iEGR	<ul style="list-style-type: none"> <li>- Charge heating (LL, HL)</li> <li>- Dilution (HL)</li> <li>- Heat capacity effect (HL)</li> <li>- Chemical effect (LL)</li> </ul>	<ul style="list-style-type: none"> <li>- Negative valve overlap</li> <li>- Exhaust rebreathing via PVO</li> <li>- Exhaust rebreathing via EVR</li> </ul>	Fast
		<ul style="list-style-type: none"> <li>- Controlling backpressure via exhaust devices</li> </ul>	Slow
eEGR	<ul style="list-style-type: none"> <li>- Charge heating (LL, HL)</li> <li>- Dilution (HL)</li> <li>- Heat capacity effect (HL)</li> <li>- Chemical effect (LL)</li> </ul>	<ul style="list-style-type: none"> <li>- EGR loop</li> <li>- Dedicated EGR</li> <li>- Second engine [69]</li> </ul>	Slow
Cycle type	<ul style="list-style-type: none"> <li>- Distribute energy release over successive revolutions to reduce dependence on release every second revolution (HL)</li> </ul>	<ul style="list-style-type: none"> <li>- Two-stroke cycles</li> <li>- Injection during NVO [76]</li> <li>- Six-stroke cycles [54]</li> </ul>	Fast
Operating load	<ul style="list-style-type: none"> <li>- Prevent operation at non-AI-conductive low temperature, low load conditions (LL)</li> </ul>	<ul style="list-style-type: none"> <li>- Powertrain electrification</li> </ul>	Interm.
		<ul style="list-style-type: none"> <li>- Cylinder deactivation [68]</li> </ul>	Interm.
Intake pressure	<ul style="list-style-type: none"> <li>- More air available for burning fuel (HL, LS)</li> <li>- Increased compression pressure for AI (LL, LS)</li> </ul>	<ul style="list-style-type: none"> <li>- Turbochargers</li> </ul>	Slow
		<ul style="list-style-type: none"> <li>- Mechanical superchargers</li> <li>- e-boost</li> </ul>	Interm.
Fuel injection	<ul style="list-style-type: none"> <li>- Heat release moderation via sequential stratification (HL)</li> <li>- Combustion stabilisation and ignition control via stratification (LL, HL)</li> <li>- Homogenous mixture preparation (LL)</li> </ul>	<ul style="list-style-type: none"> <li>- PFI</li> </ul>	Interm.
		<ul style="list-style-type: none"> <li>- Early DI</li> <li>- PFI + late DI</li> <li>- Multiple DI</li> </ul>	Fast

Table 3. Cont.

Control Parameter/Actuator	Effects/Benefits	Implementation Strategies	Response Rate
Ignition assistance	- Heat release moderation via combustion phasing control (HL)	- Spark plug and discharge parameter control	Fast
	- Combustion stabilisation and AI promotion via ignition energy provision (LL, HL)	- Pre combustion chamber	
Intake air heating	- Promote AI (LL)	- Thermal management systems using waste heat	Slow
		- Electric intake heaters	
Intake air cooling	- Discourage AI (HL) - Prevent knock for SI operation (HL)	- Refrigeration circuit	Slow
Engine cooling	- Modulate AI by increasing or decreasing cylinder gas temperature (HL, LL)	- Cooling fluid temperature and/or flow adjustment	Slow
		- Water DI	Fast
Geometric compression ratio	- Modulate AI directly by adjusting compression pressure (HL, LL)	- Free pistons directly coupled with electric machine [18] - Variable length conrod [110] - More listed in text	Interm.
Mixture reactivity	- Increase mixture reactivity to promote AI (LL)	- Using lower/higher ON fuels - Ozone dosing [117] - Low temp. heat release [100] - RCCI, ICCI [89] - Octane on demand [1]	Interm.

### 5. Opportunities for CAI-Based HPS

The discussion presented thus far has reviewed various CAI technologies without focusing on the powertrain configuration, whether electrified or conventional. The suitability of CAI engines for electrified powertrains and the accompanying challenges and opportunities are discussed in this section. Synergistic opportunities unique to CAI TPUs, i.e., system efficiency improvements in addition to the usual ones from electrification, are emphasised. An overview of hybrid propulsion systems is provided in Appendix B.

Electrification decisions are strongly influenced by the particular powertrain's design needs, as marginal efficiency benefits have to be weighed against the associated increase in system cost and complexity. The cost of CAI engines is expected to be higher than that of SI engines in general because of higher operating in-cylinder pressures and high-complexity engine control systems. Some of these costs might be offset by relaxation in aftertreatment system requirements due to lower engine-out emissions. Opportunities for electrification to complement CAI engine operation can be categorised into the following three non-mutually exclusive groups:

1. **Operating Map Expansion:** Using electronically actuated/controlled levers to increase CAI efficiency and extend the CAI engine's operating map. This could include the use of systems, such as electronic valve actuation, electric heating, and e-boost systems, to remove constraints to CAI operation at limits of CAI operability. Indirect references to this approach have been found in the CAI literature [21,39,41], but the authors have not come across any studies dedicatedly employing this approach, possibly because of the associated increase in cost and system complexity.
2. **Load Levelling:** Using EMs to provide or absorb torque to or from the engine to limit its operation to CAI-operating regions or, in the case of multi-combustion-mode engines, maximise high-efficiency CAI operation. For CAI TPUs, multi-mode

combustion strategies are to be expected, e.g., HCCI-SI [120], SACI-SI [84], RCCI-CDC [121], HCCI-SI-RCCI [122]. Variants of load-levelling approaches that maximise engine operation in high-efficiency regions, e.g., by traversing the optimal engine operation line, are common among all classes of HPS, CAI-based or otherwise. For CAI HPS, load levelling can be achieved implicitly by pairing appropriately sized engines and EMs in suitable hybrid train architectures to maximise the CAI operation of the engine, e.g., in a parallel HPS with a naturally aspirated CAI TPU that has a part load CAI operating region (similar to that of the baseline CAI engine), and frequent CAI operation would be realised during urban driving [121]. Additional explicit instructions can also be sent by the supervisory control module to prioritise engine operation in CAI regions by using electric and absorption drive in power-split or parallel architectures [120] or having an energy management strategy for a series HPS that prioritises engine operation at CAI points [51].

3. **Torque Tracking:** Engaging EMs to maintain smooth output torque during combustion mode changes to ensure smooth drivability. Combustion mode transitions require adjustments of the in-cylinder conditions, which are limited by transport delays and actuator response times and can lead to suboptimal energy release cycles and emission excursions in this period. Torque tracking is commonly used to complement load-levelling supervisory control strategies [84].

These strategies, employed in isolation or collectively, can serve as enablers for the use of CAI TPUs in HPS and yield significant efficiency and emission improvements. The improvements can be maximised by co-designing the thermal and electric propulsion units as so-called dedicated hybrid engines [105,123]. Such engines rely on the electric machine to perform functions necessary for engine operation.

### 5.1. CAI HPS Studies

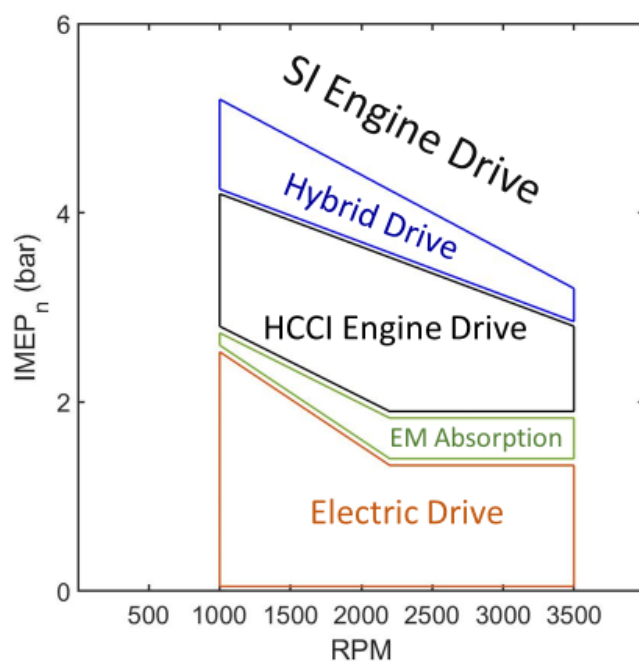
A summary of surveyed studies that used electrified powertrain components to complement CAI TPUs is presented below.

#### 5.1.1. HCCI

Delorme et al. [51] used a vehicle model to study the performance of a mid-size car powered by a dual-mode (HCCI-SI) TPU for parallel, power-split, and series HPS architectures at varying levels of electrification, ranging from micro to plug-in electric. HCCI operation was realised by managing iEGR levels through NVO adjustments. Two energy management strategies were used, an optimum-line-based strategy and a modified version that prioritised HCCI operation. Fuel consumption improvements between 2% for the high electrification level and 15% for micro HPS were realised relative to SI TPU powertrains using the base strategy. Improvements for the high-electrification level HPS increased to 10%, while those for micro HPS almost stayed unchanged when the HCCI prioritising the load-levelling strategy was used. Power-split HPS benefited more from having an HCCI TPU than a series HPS because the engine operated more frequently at the low speed and torque points for which HCCI was available. The series HPS required engine operation at higher torques to charge the battery, which fell outside the HCCI operability region and in the conventional SI regime. The study also predicted that drivability issues could arise from restricting engine operation to low torque points.

Lawler et al. [120] used a vehicle model with an integrated engine model to index the efficiency benefits from hybridising a dual-mode HCCI-SI engine in a parallel configuration for a mid-size passenger car. The engine used a symmetric NVO strategy to realise HCCI operation. A supervisory control module called 'e-HCCI' was used to maximise HCCI engine operation and reduce the number of mode changes. Two electrification levels (mild and medium with 5 and 10 kW electric motors, respectively) were studied. The e-HCCI strategy deployed electric, engine, hybrid, and absorption drive as needed to negotiate the driving demand with engine operating mode, battery SoC, and electric machine power to meet the HCCI maximisation objective. A qualitative illustration of the e-HCCI strategy

after being adapted for the baseline CAI engine is presented in Figure 12. The hybrid and EM absorption modes brought load points in the vicinity of the HCCI operating region into the HCCI fold by providing or absorbing torque. Very low- and high-load points were handled by electric drive and SI engine drive, respectively. At the highest loads, SI-hybrid drive was used. The resulting efficiency improvement with HCCI-SI operation over SI-only mild HEV was 18.4%, which was higher than the individual benefits from hybridisation and HCCI combustion, demonstrating the synergistic relationship between the two. The marginal efficiency benefits decreased for the medium hybrid case because the higher degree of electrification excluded more low-load points at which the SI engine efficiency was very low. A similar analysis of the power-split hybrid architecture determined that the efficiency benefits were relatively small because the engine operated at high loads most (about 98%) of the time where SI mode efficiency was high. An engine size sensitivity sweep was also performed, in which the displacement volume was increased from 2.5 to 3 litres. This brought 38.3% of the operating points into the HCCI fold, as the larger engine operated increasingly in the part-load regime. The fuel economy improved by 20% as a result.



**Figure 12.** Qualitative illustration of the expansion of the HCCI operating region of the baseline CAI engine via parallel hybrid drive with the e-HCCI supervisory control strategy from [120].

Lawler et al. [124] improved the e-HCCI system to further reduce engine mode transitions by establishing dead-bands at mode boundaries to mitigate efficiency penalties associated with engine on/off events and mode transitions. Additionally, the battery management system that previously maintained battery SoC between 45 and 55% was made ‘passive’ by introducing a ‘motor torque scaling factor’. This prioritised HCCI operation and reduced time in electric and hybrid drive modes, and it increased the absorption-mode duration. The model also accounted for drivability considerations in determining the gear shifting schedule. Hence, over the US UDDC driving cycle, around 90% of the operating points were packed into the HCCI region, which led to a fuel economy improvement of 48% over the conventional SI vehicle and 12% over the SI-only parallel HPS. The study also performed an engine-size displacement sweep from 1 to 3 litres, while proportionally changing vehicle weight and EM power. Increasing engine size enhanced fuel economy improvements because of a rise in the proportion of HCCI operation.

An engine data-validated vehicle simulation model was used by Solouk et al. [125] to compare a mid-size series HEV operating on a gasoline HCCI (PFI) TPU to a conven-

tional SI engine-based HEV. A single engine operating point with a power demand of 24 kW was selected based on the vehicle power requirements for highway driving and powertrain transmission losses. The CAI engine-powered HEV exhibited 12.6% higher fuel economy relative to the SI TPU-based HEV. Various energy management strategies (dynamic programming, thermostatic rule-based control, and model predictive control) were compared. The dynamic programming strategy led to greater fuel efficiency gains than the other approaches, despite having more engine starts and stops.

#### 5.1.2. SACI

Robertson et al. [84] compared the torque tracking performance of two parallel HEVs—a mild P0 and a medium P2 vehicle—that were powered by an SACI-SI TPU. The study used a rule-based ‘minimisation of mode switching’ strategy to provide and withdraw engine torque via the EM to smooth the combustion-mode transition fluctuations. A charge-sustaining mode strategy was used, whereby torque was absorbed from the engine by the generator at low battery charge levels, and electric/hybrid drive was engaged at high loads when SoC was greater than 50%. The torque fluctuations were found to be around 23 Nm during SACI-SI transitions and required 5 kW of electric support, which could be provided through mild hybridisation. Hence, the number of mode transitions were reduced by 15–40% for different drive cycles, which offered up to 5% fuel economy benefits, and the transitions became smoother. Efficiency improvement was, however, not prioritised in the study.

#### 5.1.3. PPCCI

Not many hybrid studies surveyed employed PPCCI technologies, even though such engines can benefit from low load avoidance via load levelling, as highlighted by one of its early developers as follows: ‘*electric drive [can] cover some of the deficiencies of GCI [PPCCI] such as very-low-load operation*’ [37].

García et al. [126] used vehicle models to study two-series hybrid vehicles powered by a low-ON gasoline (RON = 75)-fuelled PPCCI TPU. Both vehicle models were fed with experimental engine performance and emission data, and engine operation was restricted to regions of relatively high thermal efficiency (above 4 bar BMEP). A rule-based control strategy was used to sustain battery levels between 40 and 80%, while minimising fuel consumption and NO<sub>x</sub> emissions. A multi-objective Pareto optimisation approach was used to study the effects of splitting the engine map into multiple states (load levels). The optimum number of states depended on the driving cycle and vehicle class, which determined the engine power levels used for battery recharging. The combination of available states and load requirements that resulted in engine operation at high loads yielded significant efficiency improvements and achieved comparable fuel consumption values to those of diesel engines. Such approaches as limiting engine operation to high-efficiency windows and meeting SoC targets by traversing these regions can enable high-efficiency CAI operation without needing an extension in the CAI operating map using costly actuators, such as CVVL.

Abdul-Manan et al. [10] used a vehicle simulation model to compare the greenhouse gas reduction potential of various mild (P0 and P2) and full hybrid powertrain configurations (parallel, power-split, and series) powered by a ‘GCI’ TPU. A low-octane-rating gasoline fuel with a RON of 80 was used for the study to highlight the life cycle benefits of such a fuel. The series hybrid configuration reduced emissions most significantly, 43% relative to a non-hybridised CAI-based engine on a tank-to-wheel basis, followed by around 20 and 15% reductions, respectively, for power-split and parallel full HEVs. Mild hybrid yielded only 6% improvements. A rule-based supervisory control strategy that considered motor and engine load demand, vehicle speed, and battery SoC as control parameters was used. The detailed engine-level control priorities are not discussed in the study, but from the system level priority of maximising efficiency and minimising emissions, the control strategy used is considered to be of the load-levelling type. The study also estimated that,

on a well-to-wheel basis, CAI-based HPS will still be preferable to BEVs in 2040 in many regions of the world where the electric grid has high carbon intensity, including China, India, the Middle East, Africa, and Russia. BEVs, on the other hand, will be preferable in most of Europe, while the carbon footprint of BEVs and the HPS will be comparable in North America.

The Mazda e-SKYACTIV X engine, which is of the SACI-PPCCI type, was used in a mild hybrid configuration with a 24 V battery in the new Mazda 3 mid-size vehicle [127]. The engine was also equipped with cylinder deactivation to switch from four- to two-cylinder operation.

#### 5.1.4. RCCI

Although RCCI-based HPS TPUs are burdened by relatively high system complexity and cost, they have been highly researched because of the control flexibility they offer (e.g., [118,128–130]), especially by allowing multi-mode operation ranging from conventional diesel [43] to conventional port-fuel injected gasoline SI and HCCI combustion [131].

Solouk et al. [122] studied a series REEV powered by a dual-fuel (gasoline PFI and diesel DI) engine. Single-mode (SI, HCCI, and RCCI) engine operation was compared to multi-mode (HCCI-SI-RCCI) operation over three driving cycles. The HCCI strategy restricted engine operation to low powers (<14 kW), RCCI restricted it to moderate levels (<25 kW), and SI allowed operation up to 50 kW. HCCI-only operation produced the highest efficiency (12.1% improvements over SI-based HPS), followed by RCCI-only operation (9.1% improvements). Modest improvements of around 1.4% beyond single-mode CAI operation were realised from multi-CAI-mode operation.

Benajes et al. [43] used vehicle simulations to compare the fuel efficiency and NO<sub>x</sub> emissions of a mid-size car operated by three powertrains—a conventional diesel combustion (CDC) powertrain, a conventional RCCI-CDC powertrain with CDC at very low and high loads and RCCI at moderate loads (2.5–7 bar IMEP), and a series fully hybridised train in which the engine operated only in the RCCI region. Sensitivity analyses using a rule-based control energy management system were performed to find the optimum SoC level and observe the effects of having one to three engine operating points available for battery charging. Fuel penalties associated with mode transitions and switching the engine on/off were also accounted for. Variations in SoC requirements between 24 and 84% made minor impacts on the overall fuel consumption, but levels greater than 44% were recommended for on-road driving. Increasing the number of zones provided fuel-efficiency benefits. The RCCI-hybrid produced 13 and 19% fuel consumption improvements relative to RCCI-CDC and CDC powertrains, respectively.

In the following study, Benajes et al. [121] compared CDC and dual-mode RCCI-CDC performance for two parallel configurations (mild P0 and full P2). The design of the experiment and optimisation techniques were used to optimally size electric components for both the HPS configurations to maximise fuel consumption and NO<sub>x</sub> emission improvements. Full hybridisation with dual-mode TPU provided significant efficiency improvements over conventional and mild configurations, while reducing engine-out NO<sub>x</sub> emissions to below CDC levels. Exhaust after-treatment via SCR was still needed to meet Euro-6 limits.

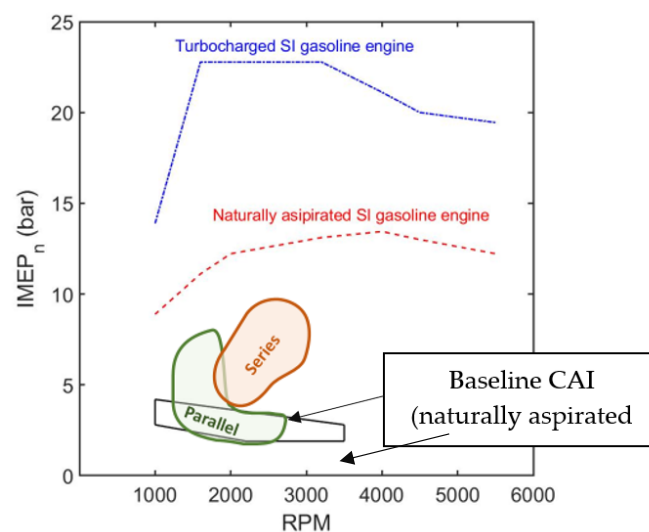
Hanson et al. [128] experimentally studied an RCCI engine-powered series HPS in a modified compact sports utility vehicle (SUV) on a chassis dynamometer to demonstrate the suitability of RCCI engine operation in a hybridised powertrain. The energy management strategy restricted engine operation to three points (15, 18, and 32 kW brake power) in the RCCI map, where the efficiencies were high, and exhaust emissions and temperatures were acceptable. From the reviewed studies, this is the only study that experimentally investigated a CAI-based HPS.

A summary of the surveyed CAI HPS studies is presented in Table 4. Figure 13 illustrates the typical CAI TPU operating regions across common light-duty vehicle driving cycles for parallel and serially configured HEVs. The regions were approximated from studies referenced in the caption, all of which used CAI-favouring load-levelling strategies.

Note that, for the parallel HEVs, at some high-speed and load operating points, the TPU operated in SI mode as well. However, as mentioned during the discussions of the referenced studies, a majority of the operation was in the CAI regions. The CAI bubbles represent a significant relaxation in the high- and low-load limit extension targets compared to those for conventional powertrains.

**Table 4.** Summary of CAI TPU-based HPS studies. \* over conventional TPU-based HPS.

Study	CAI Strategy	Hybrid Architecture	Hybridisation Level	Synergy Strategy	Max Efficiency Improvement * (%)
2010 [51]	HCCI-SI	Parallel, series, power-split	Micro/mild (parallel), full, plug-in	Load levelling	15
2011 [120]	HCCI-SI	Parallel, power-split	Mild, medium	Load levelling	20
2013 [124]	HCCI-SI	Parallel	Mild	Load levelling	12
2014 [125]	HCCI	Series (REV)	Full	Load levelling (single level)	12
2021 [84]	SACI-SI	Parallel	Mild (P0), medium (P2)	Torque tracking + load levelling	5
2019 [126]	PPCCI	Series	Full	Load levelling (multiple levels)	To diesel levels
2020 [10]	PPCCI	Series, parallel, power-split	Mild (P0, P2), full	Load levelling	-
2015 [128]	RCCI	Series	Full	Load levelling (multiple levels)	-
2017 [122]	HCCI-SI-RCCI	Series	Full	Load levelling (multiple levels)	11
2018 [43]	RCCI	Series	Full	Load levelling (multiple levels)	-
2019 [121]	RCCI-CDC	Parallel	Mild (P0), full (P2)	Load levelling	To diesel levels



**Figure 13.** Operating regions for CAI TPUs in series [43,126,128] and parallel [84,120,124] HPS for mid-size vehicles. (SI engine high load limits adapted from Fu et al. [49] by assuming a 90% mechanical efficiency).

## 5.2. CAI HPS Design Considerations

### 5.2.1. CAI Mode

The choice of CAI mode is influenced by the HPS architecture, electrification level, engine size, road load requirements, and the supervisory energy management strategy used, which collectively determine the requisite TPU operating region. If engine operation is limited to a narrow, moderate-load operating band, HCCI, which is the cleanest and most-efficient CAI strategy, should be selected. If the load requirements are more demanding, appropriate CAI range extension or multi-combustion-mode (e.g., HCCI-SI) options can be chosen that meet the operational needs while staying within system cost and complexity constraints.

As TPU operational requirements will likely be significantly relaxed from electrification, CAI strategies that offer relatively high combustion control and moderation, such as PPCCI, SACI-PPCCI, and RCCI, might not be needed. A simpler strategy, such as SACI, which requires moderately high compression ratios, mild thermal assistance for AI, low-pressure fuel injection, limited valve train flexibility (i.e., VVT), and that can transition to conventional SI operation might be well suited. For some designs where the engine operates consistently at very high load levels where conventional SI engines have high efficiencies, the use of CAI-based TPUs might not be justified.

### 5.2.2. HPS Architecture

Important TPU efficiency related considerations for HPS architecture choice are the level of shielding from transient changes in driving demand provided to the TPU and the extent of avoidance of low- and very high-load operating points needed.

Series HPS provide the highest level of shielding, followed by power-split and parallel architectures, and have the potential of offering the highest efficiency benefits relative to other full HPS [10]. The relatively large batteries in series HPS also provide more opportunities for CAI assistance from electrification. Bassett et al. [132] recommended, based on the analysis of European and American driving cycles, that TPUs used in REEVs should have a rated power of 30 kW or higher. Series HPS typically operate in charge-sustaining mode, and the target battery SoC is the key determinant of the engine load point. Normally, the energy management system limits engine operation to one or a few high load-operating points to rapidly recharge the battery. This might limit the utility of CAI TPUs because: (i) the CAI operating map might not extend to the required high load points, and (ii) the marginal efficiency improvements over conventional (SI) TPUs might not be significant. However, with a CAI-based energy management strategy, engine operation can be limited to the CAI map if an over-sized TPU is used to maintain battery SoC through part-load operation instead of high-load ‘thermostatic’ TPU operation [126]. Decisions about the latter depend on the efficiency penalty per engine restart and the part-load CAI efficiency advantage [125].

If CAI operation is limited to moderate loads with high load operation provided by a conventional combustion mode, parallel and, to a lower extent, power-split HPS might be preferable over series configuration, especially during urban driving where a majority of engine operation takes place at part loads [51,121]. In an HCCI-SI multi-mode TPU, a parallel configuration might be preferable over a power-split one because increased part-load driving will increase the proportion of CAI operation [120]. Parallel systems also have a cost advantage over series and power-split HPS because of lower electrification needs. P0, which couples EM to the engine directly via the accessory belt, is the cheapest HPS configuration but it offers limited electric torque assistance [84]. Of the parallel architectures that are capable of electric drive, P2, in which the EM is connected to the transmission input shaft, is considered the cheapest.

### 5.2.3. Electrification Level

While higher levels of electrification allow greater opportunities for enhanced CAI operation by employing one of the three CAI-enabling approaches listed above, the marginal efficiency improvement opportunities from transitioning to a CAI TPU diminish as electrification levels increase. This is because, with the increasing electrification, the engine runs for shorter durations. The highest levels of efficiency improvements from CAI adoption are thus realised for micro and mild HPS [51,120]. The study by Abdul-Manan et al. for a GCI TPU discussed above concluded that battery size (electrification level) has a stronger influence on system efficiency than the powertrain architecture [10]. From a drivability perspective, mild hybridisation has been found to be sufficient for dampening the mode shifting-induced torque output fluctuations in multi-mode TPUs [84].



#### 5.2.4. Engine Size

With increasing electrification levels, engine capacity and power rating typically decrease. For CAI TPU-based HPS, however, oversized engines that meet the power demand at moderate levels where CAI operation is possible might present efficiency advantages over high load-operating conventional combustion TPUs. Such derated engines, which would increase system cost and weight, could be used for large applications, such as SUVs and trucks [120,124].

#### 5.2.5. Energy Management

Appropriately designed energy management systems are critical for the adoption of CAI-based TPUs, as energy management decisions dictate the frequency of CAI operation and, through that, the resulting efficiency improvements. Highlights from the reviewed HPS studies regarding electrification assistance for CAI are:

- Extending the CAI engine operating range by using electrically actuated controllers will increase TPU cost and complexity, which might not be justified by the efficiency and emission improvements and might not even be needed, as discussed above.
- Load-levelling techniques can significantly increase the proportion of CAI operation (e.g., from 30% to 90% during urban driving [124]) and decrease combustion mode transitions (e.g., by 40% in [84]) in multi-mode engines.
- Electrification can assist in ensuring smooth combustion-mode transitions through torque tracking [84]. Additionally, energy management strategies can be devised to ensure compliance with drivability and NVH constraints, e.g., limiting engine [122] and electric drive [121] speeds, and momentarily (for 2–3 cycles) enriching cylinder charge during combustion-mode shifting to smooth transitions and avoid misfires [122].

### 6. Summary and Recommendations

This paper discussed the potential benefits and accompanying challenges of using controlled auto-ignition as a combustion mode in internal combustion engines, for use in both conventional and hybridised powertrains. CAI was defined as the flameless and volumetric energy release from the spontaneous combustion of lean and diluted, homogeneous, air-fuel mixtures and was identified as a third combustion regime distinct from deflagrative and diffusive combustion modes of traditional spark and compression ignition engines, respectively. Multiple CAI strategies were discussed, and a list of control actuators used in various CAI implementations was extracted. The potential for the relaxation of CAI engine operational requirements by powertrain electrification was examined, and the complementarity of the two innovations (CAI and electrification) was highlighted based on the survey of CAI TPU-powered HEV studies. Highlights of these discussion points and recommendations based on them are summarised below.

#### 6.1. CAI Advantages

The following major benefits of employing CAI in gasoline engines were identified:

- CAI can improve thermal efficiency by allowing thermodynamically efficient diluted and/or lean combustion.
- CAI can improve combustion efficiency through multi-site auto-ignition.
- Combustion stabilisation and acceleration realised through CAI can reduce the need for in-cylinder turbulence for lean and diluted engine operation. Hence, air flow passages and valves that improve volumetric efficiency can be designed, and combustion chamber designs can be simplified.
- Gasoline, because of its low reactivity (high ON) and high volatility, has inherent advantages as a CAI fuel. Its AI and vaporisation attributes facilitate the preparation of a well-mixed vaporised fuel-air mixture and reduce the likelihood of premature AI. This means that high-compression-ratio engines can be designed to achieve diesel-like efficiency, while using relatively low injection pressures.

- CAI can potentially reduce cylinder-out  $\text{NO}_x$  and PM emissions to levels where exhaust aftertreatment devices might not be needed. Moreover, if incomplete oxidation pathways, e.g., low load points, can be avoided, CO and HC emissions can be kept low as well.

### 6.2. CAI Challenges

The following challenges to CAI implementation in conventional powertrains were identified:

- The volumetric nature of CAI combustion can yield unacceptably high pressure rise rates, which can act as a high load limit. Various CAI variants try to moderate energy release by introducing chemical and thermal heterogeneities, often with accompanying efficiency and emission penalties.
- Very diluted and lean combustion can lower exhaust gas temperature and enthalpy, which can reduce the system's level efficiency and present emission after-treatment challenges.
- For a single fuel (gasoline), mixture reactivity can be altered to a limited extent (via leaning and dilution), which in the absence of expensive control actuators, such as CVVL, high pressure direct injectors, e-boost, and variable compression ratio systems limit CAI operation to moderate load levels.
- Other challenges to gasoline CAI reported in the literature are cold start and idle combustion, low load stability, acceptable transient operation, hardware (combustion chamber and injection system) optimisation, and boosting at high EGR levels [37].
- The challenge of reliably controlling the onset and phasing of combustion has been largely addressed by using chemical (rich late injection) or thermal (spark discharge) ignition triggers.

### 6.3. CAI Characterisation

Homogeneous charge compression ignition (HCCI) was characterised as the original 'best of both worlds' CAI concept that formed a true hybrid between conventional CI diesel and SI gasoline engines. The various CAI implementation strategies that have arisen out of the need to address implementation challenges associated with HCCI engines while keeping the accompanying efficiency and emission penalties to a minimum were characterised under the following headings, and the operation, advantages, and disadvantages of each were discussed:

- Premixed charge compression ignition (PCCI);
- Partially premixed charge compression ignition (PPCCI);
- Spark-assisted compression ignition (SACI);
- SACI-PPCCI hybrid;
- Reactivity controlled compression ignition (RCCI).

The SACI-PPCCI hybrid strategy that uses multiple injections to tailor the spatial and temporal distribution of mixture reactivity and uses a spark to control ignition offers the most flexibility in controlling CAI and is a promising technology for extending CAI operation across a sufficiently wide operating map needed for conventional powertrains.

### 6.4. CAI TPUs for Hybrid Propulsion Systems

In hybrid propulsion systems, the availability of an electric propulsion route changes the engine power requirements, which together with CAI-favouring energy management strategies can remove many traditional impediments to CAI implementation. CAI TPU-based HEVs have the potential of significantly (10–20%) improving system level efficiencies relative to comparable HEVs powered by SI TPUs and can thus provide a cost-effective clean transition pathway for mobility sector decarbonisation. Powertrain studies have identified synergistic relationships between CAI TPUs and powertrain electrification, whereby the efficiency improvements from the combined application of the two technologies are greater than the sum of individual improvements.

Electrification can significantly relax the high and low load operation requirements of TPU using different 'load levelling' strategies to maximise operation in CAI regions. Hence,

the need for expensive control actuators to extend CAI operation can be removed, and HCCI-type TPUs can provide the required power. For mild energy release moderation and occasional high load operation, SACI engines that are capable of multi-mode operation (SI-SACI-HCCI) can be used. The use of more-advanced CAI strategies, such as PPCCI, SACI-PPCCI, and RCCI, might not be needed, nor might the costs be justified for systems already burdened by the cost of electrification. For high levels of electrification and for HPS where the share of high load operation is substantial, the efficiency benefits from CAI TPUs might not be sufficient to justify replacing SI engines.

Vehicle simulation models are expected to continue to play an important role in the development of optimised HPS components and control strategies. High-quality test-bench data of new CAI TPU designs across required load maps will be needed to effectively use such models.

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## Nomenclature

AI	auto-ignition
BEV	battery electric vehicle
BMEP	brake mean effective pressure
bTDC	before top dead centre
CA	crank angle
CAI	controlled auto-ignition
CA50	50% fuel mass fraction burned CA location
CDC	conventional diesel combustion
CI	compression ignition
CN	cetane number
CVVL	continuous variable valve lift
DI	direct injection
eEGR	external exhaust gas recirculation
EGR	exhaust gas recirculation
EM	electric machine
EVR	exhaust valve reopening
GCI	gasoline compression ignition
GDI	gasoline direct injection
HC	hydrocarbon
HCCI	homogeneous charge compression ignition
HEV	hybrid electric vehicle
HL	high load (limit)
HPS	hybrid propulsion system
HS	high speed (limit)
HRR	heat release rate
iEGR	internal exhaust gas recirculation
IMEP	indicated mean effective pressure
IMEP <sub>n</sub>	net IMEP

BMEP	brake mean effective pressure
LL	low load (limit)
LS	low speed (limit)
NVH	noise, vibration, and harshness
NVO	negative valve overlap
ON	octane number
P0-P4	parallel HPS configurations
PCC	pre combustion chamber
PCCI	premixed charge compression ignition
PFI	port fuel injection
PM	particulate matter
PPCCI	partially premixed charge compression ignition
PRR	pressure rise rate
PVO	positive valve overlap
RCCI	reactivity controlled compression ignition
REEV	range extended electric vehicle
RON	research octane number
RPM	revolutions per minute
SACI	spark-assisted compression ignition
SCR	selective catalytic reduction
SI	spark ignition
SoC	state of charge
SUV	sports utility vehicle
TDC	top dead centre
TWC	three-way catalyst
VVT	variable valve timing

## Appendix A. Summary of Surveyed GCI Articles

**Table A1.** Summary of PPCCI articles.

Ref.	Technology	$P_{in}$ (bar)	$T_{in}$ ( $^{\circ}C$ )	$P_{inj}$ (bar)	Positive Ignition	iEGR Strategy	Number of Injections	eEGR	Net Indicated Efficiency (%)
2007 [69]	PPCCI [Shell, KTH] CR: 14:1	2	40	1100			2: compression + ~ TDC	Yes	45–53.5 <sup>1</sup>
2013 [76]	PPCCI [Lund] CR: 16.5:1	Amb	Amb	500	Glow plug	NVO, Exh RB	2: injection during NVO + main		32–40
2018 [39]	PPCCI (GDCI) [Delphi] CR: 16:1	TC + small clutched super charger	40–80	350–500		Exh RB low loads CVVL-poor boost	2–3: intake + compression	Yes	38–46 <sup>2</sup>
2018 [59]	PPCCI (2SC) [Achates] CR: 18.5:1	1.12–2.3	45	326–1303		Scavenging gas exchange	2: compression (~45 and 8 $^{\circ}$ CA bTDC)	Yes	45–52
2019 [133]	PPCCI [UNSW] CR: 17.7	1	80	500			2: 170, 16–0 $^{\circ}$ CA bTDC		46–50
2020 [68]	PPCCI + SACI (GCI) [Shell, PSA] CR: 16:1	Amb	Amb	398–567	Low load SI assist		2–4: compression + expansion	Yes	28–45 <sup>2</sup>
2020 [41]	PPCCI (GCI) [UNSW] CR: 17.7:1	1–1.3 (e-boost)	80	500			3: compression (170 to 3 $^{\circ}$ CA bTDC)	Yes	48–52

<sup>1</sup> Converted from indicated specific fuel consumption by assuming: LHV = 42.5 MJ/kg. <sup>2</sup> Converted from brake specific fuel consumption by assuming: LHV = 42.5 MJ/kg and mechanical efficiency = 90%.

**Table A2.** Summary of SACI articles.

Ref.	Technology	$P_{in}$ (bar)	$T_{in}$ ( $^{\circ}C$ )	$P_{inj}$ (bar)	Positive Ignition	iEGR Strategy	Number of Injections	eEGR	Net Indicated Efficiency (%)
2006 [53]	SACI + PPCCI [Nissan] CR: 15.01:1	1–1.5	27–130	30–50	Spark Plug		2: intake + around TDC		-
2013 [63]	SACI [Ann Arbor] CR: 12.5:1	Amb	45	100	Spark Plug	NVO	1: 330 $^{\circ}C$ A bTDC	Yes	42
2013 [65]	SACI [Brunel] CR: 10.66	Amb	25	3 (PFI)	Spark Plug	Exh RB via PVO	PFI	Yes	29.5–31.7 <sup>1</sup>
2021 [84]	SACI [Clemson] CR: 18:1				Spark Plug			Yes	
2020 [47,48]	SACI + PPCCI [Mazda] CR: 16:1 15:1 (e-SKX)	Small clutched super charger	Amb	500 [30]	Spark Plug	VVT—uses PVO to scavenge hot residuals + strong swirl to enhance vaporization	2: intake + compression (before spark)	Yes	-
2020 [85]	SACI + PPCCI [Sandia] CR: 12:1	Amb	30–100	170	Spark Plug	Low RGF 4–6%	2–4: 1–3 during intake + 1 3 $^{\circ}C$ A before spark		25–38

<sup>1</sup> Converted from indicated specific fuel consumption by assuming: LHV = 42.5 MJ/kg. LHV = 42.5 MJ/kg and mechanical efficiency = 90%. Amb: ambient, CR: compression ratio, RB: rebreathing.

## Appendix B. Overview of Hybrid Propulsion System Design

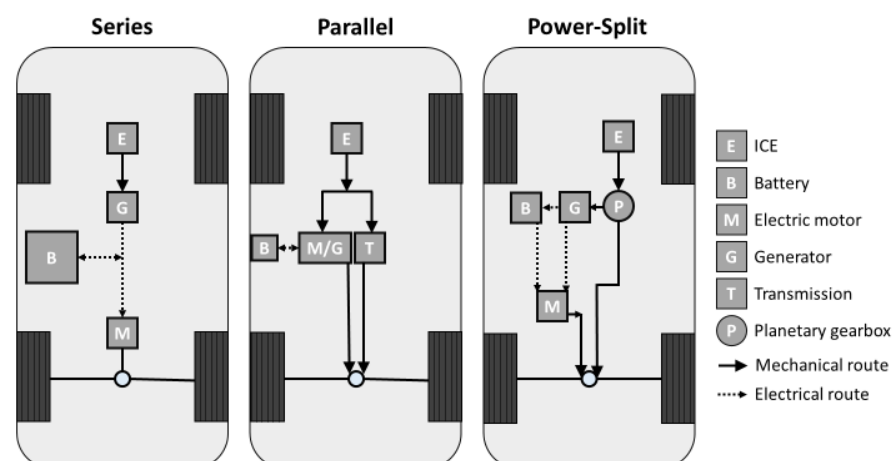
The major components of an HPS are the internal combustion engine, increasingly referred to as the thermal propulsion unit in the HPS context, and the electric machine(s) (EM) that can function as an electric generator or a motor. In the following discussion, EMs are termed generators or motors based on their primary function in the powertrain, and the following HPS operating modes are considered [120,134]:

- Engine drive: only the engine provides propulsive force;
- Electric drive: only electric motor provides propulsive force;
- Hybrid drive: both engine and electric motor provide propulsive force (also known as torque assist);
- Power split mode: engine provides propulsive force while also charging the battery;
- Absorption mode: electric generator absorbs engine power and charging the battery.

Electrified powertrains can be categorised based on their electrification levels or the relative arrangement of the powertrain components, i.e., their architecture.

### Appendix B.1. HPS Architecture

Common HPS architectures are schematically illustrated in Figure A1 and are briefly discussed below.



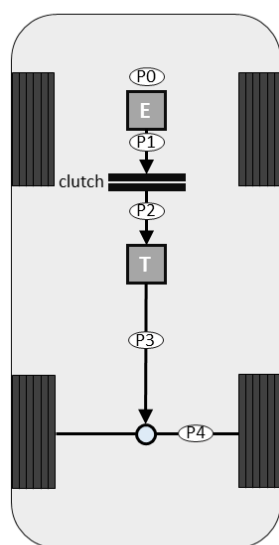
**Figure A1.** Schematics of various HPS architectures. Adapted from [135].

### Appendix B.1.1. Series Hybrid

In series HPS, the engine is connected directly to the electric generator and is decoupled from the wheels. The generated electricity is used to run the motor, which is the sole propulsion device, and excess electricity is stored in the battery (absorption mode). No mechanical transmission is needed, which reduces system cost and complexity. Since the engine is shielded from the driving demands, it operates as a generator at one or a few speed and load points where it has high efficiency levels. This leads to efficiency improvement during urban travel, which is typified by frequent starts and stops. However, at high loads (highway-type driving) where engine efficiency is relatively high, energy losses because of conversions across the engine-generator-battery-motor interfaces adversely affect vehicle fuel consumption. Series HPS require relatively large batteries (10–20 kWh [10]) to ensure adequate on-board energy availability for the EM to propel the vehicle. Concomitantly, TPUs in such configurations are generally smaller and specialised for limited load operation, e.g., 2 cylinder, 0.9 litre [132], 2 cylinder, 0.65 litre [136], and are sometimes also referred to as range extenders. Series HEVs are thus also referred to as range-extended electric vehicles (REEV). They often have plug-in functionality as well. Examples of REEVs are BMW i3 and Opel Ampera [137].

### Appendix B.1.2. Parallel Hybrid

In parallel HPS, two propulsion sources are available: the engine and the EM. The engine is connected to the wheels via a mechanical transmission. The EM, which is electrically connected to the battery and mechanically coupled to the engine and the wheels, can act as an electric generator and a traction motor. It, however, cannot generate electricity and provide traction support at the same time. Depending on the location of the EM, parallel HPS have five sub-configurations, P0 to P4 [32,138] (Figure A2). Hybrid drive is possible to varying degrees in all configurations, while configurations P2–P4 are capable of electric drive as well. Parallel HPS, especially P0 and P1, have smaller batteries compared to hybrid series because the traction requirements can be met by two propulsion devices. Engine drive, which transmits power mechanically, is preferred at high driving speeds where engine efficiency is high. This can avoid electrical transmission losses associated with electric drive. For urban driving, electric drive is more efficient. However, if the battery is depleted, engine drive at part load conditions might be needed, which can adversely affect the overall efficiency. Parallel HPS also have the advantage of not requiring a separate generator, which reduces system complexity and cost [121]. Examples of parallel HPS vehicles are Hyundai Ioniq and Kia Niro [10].



**Figure A2.** Location of EM in different parallel hybrid sub-configurations. Adapted from [138].

### Appendix B.1.3. Power-Split Hybrid

The power-split configuration attempts to realise the benefits of both series (no transmission, steady-speed high-load engine operation) and parallel HPS (engine drive at high vehicle speeds, smaller battery) by adding a tractive force transmission path from the engine in the series configuration. This is achieved using a torque-split device, typically a planetary gear system, that shunts some of the engine's power to the wheels and the remaining to the generator. A second EM that acts mostly as a motor is available to provide propulsion support (electric or hybrid drive) as needed. The torque split device also controls the engine speed and removes the need for a separate transmission by acting as a continuous variable transmission system. This allows the engine to operate at high efficiency points even at low vehicle speeds. Battery sizes for such configuration HPS range between 10 and 20 kWh [10]. The most popular example of this architecture is the Toyota Prius [139].

Fuel economy comparison between similarly sized power-split (46 miles/gal) and serially configured HPS (42 miles/gal) by Hanson et al. [128] over the Highway Fuel Economy Test provides an example of efficiency advantages of power-split architectures at high vehicle speeds. The series HPS vehicle had a low drive train efficiency of 73%, resulting from 92% and 80% generator and motor efficiency, respectively.

### Appendix B.2. Electrification Level

Vehicle electrification levels can range from micro hybrids to full BEVs that do not have a TPU and are propelled entirely by the electric traction motor. Electrification level-based electrified vehicle characterisation is summarised below.

#### Appendix B.2.1. Micro Hybrid

Micro hybrids have electric start-stop functionality and can recuperate energy from braking. However, they rely exclusively on the TPU for propulsion (no electric drive) and have low-energy electric circuits (12 V [32]). Examples include BMW 1 and 3 series and the Ford Focus [140].

#### Appendix B.2.2. Mild Hybrid

Mild hybrid vehicles can provide some traction support during transient operation to assist in acceleration and start-up using energy recuperated via regenerative braking. They have larger batteries than micro-hybrids (around 0.48 kWh [10]), which equip them with 41–160 V electrified circuits [32]. Mild hybrids typically have P0 or P1 parallel architectures as they are economical options for limited hybridisation because of relatively simple packaging and coupling requirements. Examples include Hyundai Tucson and Kia Sportage [141].

#### Appendix B.2.3. Full Hybrid

Full-hybrid vehicles have larger batteries and higher voltage electric circuits (5–18 kWh [10], 200–300 V [32]). In addition to having the capabilities of mild hybrids, full hybrids can charge batteries while driving by drawing some/all power (depending on vehicle architecture) from the TPU. They are also capable of electric drive over short distances (5–10 km [32]). These vehicles typically have parallel or power-split architectures. Examples include the Toyota Prius Prime and the Ford Fusion. Mild hybrids can improve fuel conversion efficiency up to 20%, whereas full hybrids can offer 25–35% improvement [1]. The category of medium hybrid vehicles is also sometimes used in the literature to refer to vehicles that fall in between the mild and full categories, e.g., Lawler et al. [120] studied a medium HEV that had a 1.0 kWh battery.

#### Appendix B.2.4. Plug-In Hybrid

The highest level of electrification short of the completely electrified BEV is the plug-in hybrid, which has all the functionalities of a full hybrid with the additional capability of

charging from external agents ('the grid'). They have larger batteries and longer electric drive range (>10 km [32]). Series architectures are well suited for plug-in HEVs with relatively long (>20 miles [51]) all electric range, as they have larger battery packs. Power-split hybrids can be preferable for plug-in hybrids with relatively short electric drive range.

### Appendix B.3. Energy Management

Different energy management strategies are used in HPS to direct the use of energy from on-board sources and to choose efficient energy transmission pathways while meeting emission and drivability constraints. These strategies, which are effected by the supervisory vehicle control module (via sub-modules such as engine control unit, transmission control unit, and battery management system [121]), negotiate trade-offs between multiple parameters at different road load points and ambient conditions. Some such parameters include:

- Efficiencies of energy transformation and transmission, e.g., engine drive efficiency, electric drive efficiency, engine idling/shutdown/restart penalty.
- Battery state of charge (SoC) and efficiency of charging/discharging processes at various battery levels and under different conditions. For efficient energy transfers, 40–60% [126] and 20–80% [43] SoC windows are recommended.
- Emission levels and exhaust aftertreatment temperature, e.g., avoid running the engine on a low temperature combustion mode if exhaust temperature is below the required temperature limit as needed by the aftertreatment system [128].
- Noise, vibration, and harshness (NVH) and drivability considerations, especially at low speed and during combustion mode transitions.

Energy management strategies strongly influence the fuel efficiency benefits in HPS, as they determine when, where, and for how long on the operating map will the engine be operated and how the stored energy will be utilised to assist engine operation. Popular energy management strategies can be grouped into rule-, optimisation-, and intelligence-based strategies [142]. Rule-based approaches are heuristic in nature, are relatively simple, have high levels of interpretability, and do not require the prediction of future driving behaviour. They are thus commonly used in commercial vehicles [121]. Of the optimisation-based strategies, model predictive control, equivalent consumption minimisation, and dynamic programming are popular [125], with the last one considered a benchmark for less computationally expensive strategies, as it can realise optimum solutions [84]. In Solouk et al. [125], a dynamic programming-based strategy provided a 15.3% efficiency advantage over a rule-based control strategy in a CAI-powered HPS. A frequently employed control tool to improve HPS efficiency is the optimal engine operation line, which is a locus of points at each engine power level at the speeds that yield the best fuel consumption [118,120,142]. The optimal engine line is traversed based on the HPS power requirements for the control strategy employed.

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