Applied Energy 406 (2026) 127133

Contents lists available at ScienceDirect

Applied Energy

journal homepage: www.elsevier.com/locate/apenergy

Variable valve actuation (VVA) for next-generation marine and off-road engines: a comprehensive review for meeting future emissions legislation

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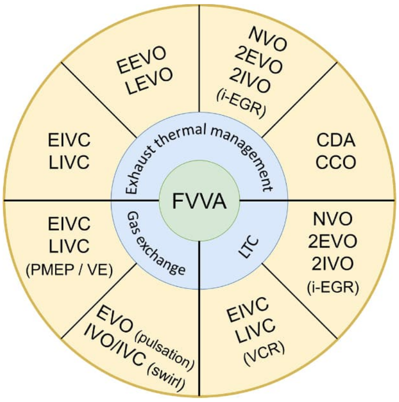
HIGHLIGHTS

- VVA's impact on marine and off-road engines is rigorously reviewed.
- 78 original research papers selected with the SLR methodology form the basis.
- CDA/EIVC/LIVC improve exhaust thermal management, even with fuel savings.
- IVC/IVO modulation enhance scavenging via optimising VE, PMEP, swirl and turbulence.
- EIVC/LIVC/i-EGR enable fast combustion phase control in advanced LTC.

GRAPHICAL ABSTRACT



ARTICLE INFO

Keywords:
VVA
Off-road engines
Marine engines
Exhaust thermal management
Gas exchange
Low temperature combustion

ABSTRACT

The marine and off-road sectors face challenges in complying with strict emission regulations. While zero-carbon fuels and advanced combustion concepts are promising, variable valve actuation (VVA) technology offers immediate benefits for reducing emissions and increasing powertrain efficiency. Contemporary VVA reviews predominantly focus on stoichiometric spark ignition (SI) engines. The possibilities that VVA offers for modern compression ignition (CI) and dual-fuel engines with advanced combustion concepts have not yet been systematically explicated and evaluated. The present study decouples VVA's impact on three critical areas in CI and dual-fuel engines: exhaust thermal management (ETM), gas exchange, and combustion. Each has been thoroughly explored and the trade-offs between emission reduction potential and efficiency have been quantified. Using a systematic literature review (SLR) methodology, findings from 78 core research papers are corroborated, focused on marine and off-road applications. There is special emphasis on VVA in heavy-duty, medium- and high-speed engines. Key conclusions highlight that cylinder deactivation and intake modulation (Miller cycle) elevate exhaust temperature at low loads without compromising fuel efficiency. This is due to adjusting in-cylinder heat capacity and heat release rate. Modulating intake valve opening (IVO) and closing (IVC) are most promising in

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https://doi.org/10.1016/j.apenergy.2025.127133
Received 9 August 2024; Received in revised form 29 September 2025; Accepted 19 November 2025

reducing pumping losses and enhancing volumetric efficiency and in-cylinder flow dynamics in high-speed engines. In medium-speed engines, a positive pressure gradient hinders the direct pumping loop effect, but the strategies can support the functioning of contemporary lambda control subsystems. Intake modulation and VVA-invoked exhaust gas recirculation allow fast combustion-phasing control by adjusting in-cylinder thermodynamic conditions. This extends the operational limits of ultra-clean, low-temperature combustion used in marine dual-fuel engines. The review underscores that VVA could be a key enabling technology for the next-generation of marine and off-road powertrains.

1. Introduction

Compression ignition (CI) engines are the primary propulsion system in both marine and non-road sectors because of their high thermal efficiency and robustness. However, conventional diesel combustion (CDC) produces hazardous air pollutants such as nitrogen oxides (NO_x) and particulate matter (PM). Gas engine technology is gaining traction in the maritime market, but even state-of-the-art, conventional dual-fuel and lean-burn spark-ignited engines face challenges in adapting to variations in fuel quality and dealing with the issue of methane slip, which exacerbates the greenhouse effect [1].

The global maritime industry currently contributes approximately 2.89 % of global greenhouse gas (GHG) emissions, with non-road mobile machinery (NRMM) accounting for about 2 % of GHG emissions in Europe [2,3]. Although these are minor contributions on a global scale, without effective countermeasures and additional policy actions, GHG emissions from global shipping is expected to increase by 0-50 % over their 2018 level by 2050. That equates to 90-130 % of the 2008 level, according to the Fourth IMO GHG Study 2020 [3]. Urgent and comprehensive measures are imperative to address this trajectory and mitigate the environmental impact of the shipping industry on a global scale.

Table 1 outlines the key emission legislation already established in the marine and NRMM sectors. For example, the International Maritime Organization (IMO) regulates sulphur oxides (SO_x) and NO_x via the MARPOL Annex V, mostly for international deep ocean-going vessels. Currently, Tier III is in effect at a global level: tighter limits apply in emission control areas (ECA). Moreover, carbon intensity is monitored and regulated by the energy efficiency design index (EEDI), the energy efficiency existing ship index (EEXI), the ship energy efficiency management plan (SEEMP) and the carbon intensity indicator (CII), via a data collection system (DCS). The initial GHG reduction strategy was adopted in June 2021, via amendments to MARPOL Annex VI. It assumed reducing the carbon intensity of international shipping in 2030 by at least 40 %, compared to 2008 levels [4]. These measures were reconsidered as unambitious in July 2023, resulting in adoption of a new IMO GHG reduction strategy [5].

Apart from IMO regulation, various government entities, including the United States (US) and the European Union (EU) have also set maritime regulations for inland waterway transport. In these regulations, the marine diesel engines are classified into off-road or non-road applications. The emission standards are implemented based on net power output, as presented in Table 1. Both the US and EU legislation regulates hydrocarbon (HC), carbon monoxide (CO), NO_x and PM. It is noteworthy that EU Stage-V also includes a particle number (PN) limit for inland waterway vessels, adding stringency to this legislation. Despite increasing use of gas engines in marine applications, a methane (CH_4) slip limit has not been considered in the US nor EU, even though

Table 1
Most relevant emission limits for marine and NRMM sector [7–10].

	Regulation	Norm or category	Net power (P)	Date	SO_x	NO_x	HC	CO	PM	PN	CO_2	N_2O	CH_4
			[kW]		[%[a]]	[g/kWh]				[1/ kWh]	[g/kWh]		
Marine	IMO	Global		2020	0.5								
		ECA		2015	0.1								
		Tier II (Global)		2011		$44\cdot n^{-0.23}$[b]							
		Tier III (ECA)		2016		$9\cdot n^{-0.2}$[c]							
	EU Stage-V	IWP/IWA-v/c-4	$P \geq 300$	2020		1.8	0.19[#d]	3.5	0.015	1×10^{12}			
	US EPA Tier 4	Marine diesel engine - Category1/2	$600 \leq P < 1400$	2017		1.8	0.19		0.04				
			$1400 \leq P < 2000$	2016		1.8	0.19		0.04				
			$2000 \leq P < 3700$	2014		1.8	0.19		0.04				
			$P \geq 3700$	2016		1.8	0.19		0.06				
NRMM	EU Stage-V	NRE-v/c-5	$56 \leq P < 130$	2020		0.4	0.19[#e]	5.0	0.015	1×10^{12}			
		NRE-v/c-6	$130 \leq P < 560$	2019		0.4	0.19[#e]	3.5	0.015	1×10^{12}			
	US EPA Tier 4		$56 \leq P < 130$	2012-2014		0.4	0.19	5.0	0.02				
			$130 \leq P < 560$	2011-2014				3.5					
	CARB Tier 5[*]		$130 \leq P \leq 560$	2029-2032		0.22	0.19[f]	3.5	0.005		724–691[##]	0.15	0.13
			$130 \leq P \leq 560$	2033~		0.04	0.08[f]						

[#] HC limits for gas engines (EU Stage-V): HC [g/kWh] = 0.19 + (1.5 × A × GER), GER: average gas energy ratio over the appropriate cycle

[##] Calculated using the reducing standard equation: CO_2 [g/kWh] = 677.5 + 2976 × (engine power [kW])$^{-0.8535}$

[a] % m/m

[b] $130 < n < 2000$, $n \geq 2000$:7.7, $n < 130$: 14.4, n: engine's rated speed [RPM]

[c] $130 < n < 2000$, $n \geq 2000$:1.96, $n < 130$: 3.4, n: engine's rated speed [RPM]

[d] A = 6 for gas engines

[e] A = 1.10 for gas engines

[f] For CARB Tier 5, non-methane hydrocarbon (NMHC), for lean-burn NG engine families remains at 0.19 [g/kWh]

[*] Proposed limits

methane has 28 times more detrimental effect for global warming compared to carbon dioxide (CO_2) on a 100-year time scale [1]. China is pioneering CH_4 emission limits for inland watercraft in its China I/II standards [6]. This is expected to influence the introduction of CH_4 regulations in other maritime standards worldwide.

In the NRMM sector, the EU has implemented Stage-V emission limits, while the corresponding US EPA Tier 4 limits apply in the US, as outlined in Table 1. Once again, both regulatory frameworks address four common emissions: HC, CO, NO_x and PM. Both have broadly equivalent limits for NO_x, HC, and CO emissions, even though the US EPA Tier 4 standard pre-dates EU Stage-V by six to nine years. However, Stage-V adopts a more stringent approach than Tier 4 to particulate emissions, with a tighter PM limit and regulation of particle number (PN).

Recently, the California Air Resources Board (CARB) proposed Tier 5 criteria, establishing new standards for pollutants and CO_2 emissions for off-road engines [7]. This proposal cuts NO_x and PM by 90 % and 75 % respectively by implementing a new NO_x idle standard and low-load cycle (LLC) standard. It also assumes a 6 % reduction in total GHG emissions, via the introduction of stricter measures to control nitrous oxide (N_2O) and methane (CH_4) as well as CO_2. The Tier 5 interim standard is expected to become effective in 2029-2033, with full adoption in 2031-2034. The EU is likely to follow suit, with its own standard equivalent to Tier 5, reflecting a shared commitment to advancing environmental sustainability in the NRMM sector.

Table 1 shows that the NRMM sector features more rigorous regulations than the marine sector, regulating a broader array of emissions with tight limit values. This disparity is primarily attributed to the fact that marine diesel engines tend to generate more excessive emissions with low quality fuels, such as heavy fuel oil (HFO). Moreover, both sectors have a considerably more lenient regulatory framework compared to on-road, heavy-duty (HD) diesel engines [11]. NRMM emissions standards are analogous to Euro/EPA standards for on-road HD vehicles, but are about two to six years behind [2,12].

This substantial regulation gap between NRMM/marine and on-road sectors enables the application of on-road emission mitigation technologies into marine and NRMM sectors, leveraging their scalability and proven robustness in terms of durability and fidelity. This not only aids achievement of strict emission targets but also reduces development time and costs for future generations. Considering the trajectory observed over several decades, it is reasonable to anticipate that the next phase of NRMM sector legislation will align closely with the on-road Euro VI (HD) standard, while the marine sector is expected to adopt EU Stage-V limits.

The stringent emission regulations have been a primary driver in both marine and NRMM sectors. However, there are also market drivers in the form of CO_2 emission reduction and the need to reduce GHG emissions such as methane slip in gas engines. These are driven by a growing emphasis on decarbonisation and the urgency of addressing rapid climate change. Importantly, the CO_2 level has significantly influenced the technology decision process.

For example, the tank-to-wheel (TTW) approach considers only downstream emissions, which encourages zero- emission solutions (battery, hydrogen or ammonia). But the well-to-wheel (WTW) approach allows wider technology selections by also considering upstream emissions. Efficiency and cost are another market co-driver, since emission mitigation strategies can carry an efficiency penalty and increased cost. However, efficiency will remain as a top priority.

There is a wide range of emission mitigation/control solutions to address the escalating stringency of emission regulations in both sectors. Table 2 outlines the latest viable options, categorising them into two types of approaches. Active solutions aim to curb the formation of emissions during the in-cylinder combustion process, whereas passive solutions focus on capturing emissions via an aftertreatment system. Typically, both strategies work together in modern diesel engines, such as exhaust gas recirculation (EGR) with aftertreatment.

Table 2
Emission mitigation technologies for marine and non-road diesel engines [11–15].

	Category	Technology	Marine	NRMM
Active	Fuels	Alternative fuels or bio-fuels / renewable fuels / e-fuels [16,17]	■	■
		Zero-carbon fuels (H_2, Ammonia) [18,19]	■ (H_2, ammonia)	■ (H_2)
	Combustion	Gas engine	△	■
		Dual-fuel (DF) combustion [20,21]	■	■
		Fuel-flexible LTC [20,22–24]	■	■
	Electrification	Hybrid & electric propulsion [25–27]	■	■
	Fuel injection system	Flexible fuel injection pressure [12,15,28,29]	○	○
		Rate of fuel injection, multiple injections [12,30]	○	○
		Fuel injection timing [12,29,31]	○	○
	Air management system	Turbocharger (e-WG, multi-stage, VGT) [32,33]	○	○
		Charge air cooling [12,15]	○	○
		EGR [34]	△	■
		VVA [35–37]	■	■
	Water addition (Wetpac)	Humidification [38]	○	–
		Water injection [39]	○	–
		Emulsion [40]	○	–
Passive	Aftertreatment	Scrubber [41]	△	–
		Diesel oxidation catalyst (DOC) [14,42]	△	○
		Diesel particulate filter (DPF) [42,43]	△	△
		Selective catalytic reduction (SCR) [42,44,45]	△	○ (single SCR) ■ (double SCR)
		Methane oxidation catalyst (MOC) [46]	■	■
		Ammonia slip catalyst (ASC) [47]	■	–

■: Promising for future regulations, △: Suitable for current regulations, ○: Legacy of previous regulations, -; not in use.

Flexibility of fuel injection pressure, timing and event is a conventional, straightforward solution in both sectors, tuning combustion to mitigate emissions. Air management technology is another effective solution, offering control over in-cylinder airflow to reduce emissions. Exhaust gas recirculation (EGR) stands out as an efficient method for NO_x abatement in the marine domain. An advanced charging system with charge-cooling further suppresses emissions by facilitating more effective EGR operation and lean burn combustion [32]. Charge-cooling has long been a standard solution in the NRMM domain, with EGR as an extra tool for NO_x control if needed. Electronic wastegate (WG) control has become common practice in both sectors. Multi-stage charging is common in marine applications; single-stage boosting or a variable geometry turbocharger (VGT) is common in NRMM.

Use of water, such as humidification, water injection and emulsion, is another means of NO_x reduction due to thermal, dilution and chemical effects, especially in marine applications [48,49]. However, it is barely used in NRMM applications due to limited space, wide ambient temperature ranges (water freezing) and durability issues. The typical trade-off between NO_x and PM in CI combustion has led to the introduction of various aftertreatment systems. Both sectors have adopted selective catalytic reduction (SCR) and diesel particulate filters (DPFs) to satisfy stringent emission limits [11,12].

DPFs are relatively new in NRMM, introduced to meet EU Stage-V's PN limit (refer to Table 1). SCR is well-established but double SCR systems are newer [50]. Exhaust gas scrubbers which reduce SO_x are common because marine fuel contains sulphur. However, IMO has gradually tightened the global fuel sulphur content limit to 0.5 %, starting from 2020, and to 0.1 % in ECA from 2015 (refer to Table 1). Low-sulphur fuel reduces use of the scrubber. Similar scrubbers are not used in NRMM because low-sulphur fuel has been mandated by the EN 590 fuel standard (max. Sulphur <0.2 %) since 1993 [51].

CDC are being gradually replaced in marine applications by gas or dual-fuel engines to comply with stringent emission standards. These include a methane oxidation catalyst (MOC) to suppress methane (CH_4) slip. Similar gas engines are promising if they are introduced into the NRMM market. With a focus on decarbonisation, there is an increased attention on various sustainable alternative fuels, including e-fuels (ethanol and methanol), renewable fuels (HVO), biofuels (biodiesel, FAME) and zero-carbon fuels (ammonia, hydrogen). Hydrogen appears to more promising in NRMM, whereas ammonia seems unappealing. Tackling the usual trade-off between NO_x and PM in CDC, advanced combustion strategies, like fuel-flexible, low-temperature combustion (LTC), look promising because they could adapt to alternative fuels and fuel blends with superior thermal efficiency.

Apart from advanced combustion and fuels, hybrid and electric propulsion systems not only reduce carbon and GHG emissions but also improve fuel consumption, vibration and noise [27,52]. Hybridisation and full electrification are both applicable and promising in NRMM, but are more limited in a marine context. The low energy density of batteries means full electric propulsion can be applied mostly in small-scale or short-voyage vessels such as passenger ferries, but not in deep sea vessels like container ships.

It is evident that an array of approaches and solutions is needed to overcome the current challenges. Integration of these innovative technologies is key for achieving highly efficient powertrains with ultra-low emissions for future NRMM and marine applications. More in-depth insights into emission reduction technologies in the marine context can be found in several review articles by Ni et al. [11] and Lion et al. [15]; and by Dallmann et al. [12] for the NRMM sector.

Variable valve actuation (VVA) has received relatively little attention in CI engine applications, yet offers significant potential, as identified in Table 2. VVA has been widely used in on-road spark ignition (SI) engines to increase breathing capacity, while improving engine performance and fuel economy over a wide operating range. This was true before turbochargers became a major trend. The majority of automotive SI engines in current production use VVA [53].

Conversely, CI engines with lean combustion always have always sufficient fresh charge due to no throttling loss and turbocharging. CI engines also have less valve clearance due to their high compression ratio (CR), so this constrains the freedom of mechanical VVA operation, for instance advancing intake valve opening and retarding exhaust valve closing to avoid collision with piston [54]. Consequently, the attractiveness of VVA in CI engines has been comparatively limited.

Although VVA is rather new to the NRMM sector, commercial marine engines by MAN, Yanmar and Wärtsilä already embrace VVA [55–57]. The most common example is Miller timing (or Miller cycle), which adjusts the intake valve closing (IVC) timing to alleviate NO_x emissions by reducing effective compression ratio (ECR). This is one of the most cost-effective NO_x reduction strategies [31]. It can be implemented simply by a conventional partial VVA set-up, such as mechanical cam-phased type or two-step system.

The majority of production VVA systems involve partial variability in timing, often using a mechanically simple hydraulic system, depicted in Fig. 1 (a). However, this approach shifts the whole valve profiles and offers limited control applications [58]. Fully variable valve actuation (FVVA) provides broader benefits but typically introduces parasitic losses, as the valves are actuated through a cam-less system. An exemplary application incorporates a brushless direct current (DC) motor and an encoder to precisely control the absolute position of the valve, as described in Fig. 1 (b). Higher actuation forces are required in large marine engines because they operate under high boost pressures with substantial inertia. Electro-hydraulic systems, exemplified by Herranen's electro-hydraulic valve actuation (EHVA) system in Fig. 1 (c), can fulfil this demand.

Both electric and electro-hydraulic systems provide fully independent control of valve parameters including timing, duration and lift for both intake and exhaust valves through sensor-based feedback control. Typically, FVVA and EHVA systems employ position sensors to track real-time valve motion, which is compared against the target profile. The controller then continuously adjusts actuation—commonly via a PID algorithm—to match the desired trajectory. Crucially, the systems are fast, allowing in-cycle control actions to be executed and in individual cylinders. This capability is a key enabler for robust combustion control in emerging LTC concepts like homogeneous charge compression ignition (HCCI) [62], or reactivity-controlled compression ignition (RCCI) [35]. Furthermore, the possibilities of FVVA and EHVA extend towards asymmetric valve opening, cylinder deactivation, dynamic skip-firing, and more. A review article by Lou et al. provides a more in-depth exploration of state-of-the-art, fully flexible cam-less VVA systems [53].

Although fully flexible variable valve actuation (VVA) systems offer several advantages, their implementation still faces challenges. First, cam-less configurations require additional hardware components such as actuators, motors, pumps, encoders, or position sensors, which increase both system and control complexity and, consequently, overall cost [53,63]. However, this cost impact depends on engine size. Since the cost of auxiliary hardware and sub-systems is largely independent of engine size, the relative cost burden decreases with increasing engine size. As a result, large-bore marine engines are less constrained by cost considerations, whereas heavy-duty off-road engines are more cost-sensitive. This necessitates an exact evaluation of the trade-off between benefits and cost implications.

The implementation cost could be reduced through the development of cost-effective sensor-less control strategies [64], for instance by employing an advanced observer with augmented extended Kalman filter [65] or nonlinear high-gain observer [66]. Furthermore, ensuring the durability and repeatability of VVA hardware and sub-systems is critical for reliable long-term operation under harsh conditions, which in turn may necessitate shorter maintenance intervals. To enable large-scale adoption of fully flexible VVA systems in mass-production engines, achieving both cost competitiveness and robust durability remains essential.

There are several well-written classical review papers on VVA. Most of them consider automotive SI engine applications, and structure the review around available VVA strategies. For instance, Hong et al. reviewed eight VVA strategies, using pressure-volume (PV) diagrams [67]. This provides valuable insights into identifying flow losses and potentials of each strategy. However, this strategy-based approach is more author-friendly than reader-friendly. It provides an academic view of how the VVA strategies can act differently, depending on the combustion concept, operating point or type of powertrain. In practice, VVA is used to achieve control of a given set of engine parameters, directly influencing emissions and performance. This can be done with concrete targets in mind.

A notable example of a well-staged, target-based VVA review is a

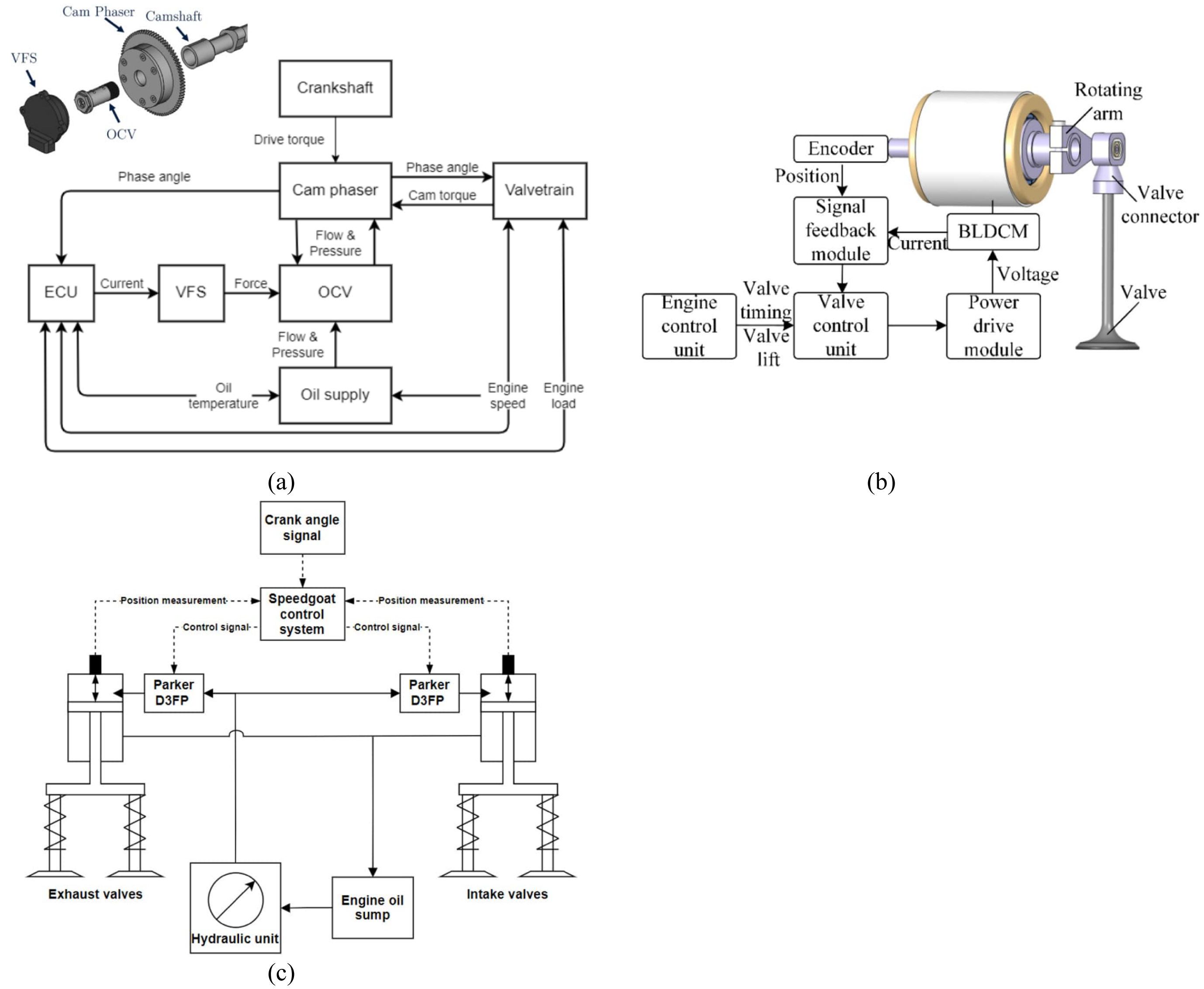


Fig. 1. Schematic diagram of various VVA system: (a) hydraulic cam-phaser type VVA, reproduced from [59]; (b) fully variable valve actuation (FVVA), reproduced from [60]; (c) electro-hydraulic valve actuation (EHVA), reproduced from [61].

1989 work by Dresner and Barkan [68]. The authors specified their VVA targets as increased fuel efficiency, increased torque, increased operational engine speed, emission reduction and residual fraction control. These targets guided the paper’s structure, providing a clear vision of how VVA can be employed to incrementally improve engine operation. However, given the study’s age, it is important to acknowledge that the review was based on stoichiometric SI engine technology, constraining the scope of the covered strategies. The review was also strictly technical, lacking the fundamental insight into the mechanism responsible for the observed trade-off.

Recently, Gao et al. [69] and Hu et al. [70] each performed a review on one specific target – exhaust thermal management (ETM). However, VVA was considered as only one of the solutions and was discussed very briefly. Both studies mostly summarized the results and trade-offs, but lack fundamental insights into the mechanism regarding exhaust gas temperature (EGT) increment, emissions and combustion behaviour. Reitz and Duraksamy [71] briefly reviewed only one VVA strategy (IVC modulation) in the context of clean and robust control RCCI. Duan et al. [72] reviewed VVA extensively and generated review tables of each VVA strategy, but only for HCCI applications. Agarwal et al. [73] reviewed EGR-related VVA strategies for all LTC strategies. In both of the last two review studies, VVA was considered as one of the solutions, and only selected VVA strategies or specific LTC concepts were covered. Furthermore, most review studies are based on small, light-duty (LD) engines, which makes investigation of VVA’s impact relatively easy.

2. Motivation, scope and methods of the review

The development roadmap carved out by the automotive industry proves challenging to navigate in the marine and off-road sectors due to energy density, packaging and robustness constraints. The evolution of the combustion engine as the prime mover is propelled along two overlapping pathways – emission reduction and the flexible integration of new fuels through advanced combustion concepts. This simultaneous pursuit demands a significant expansion of the engine calibration’s degrees of freedom. Advances in embedded control and rapid calibration methods create an opportune environment for the adoption of FVVA. This is a pivotal decision that has potential to redefine combustion-driven powertrains. But systems like EHVA entail additional parasitic losses, so the decision needs to be substantiated by rigorous evaluation of the benefits that VVA offers to the development targets that the sectors will embrace in the coming years.

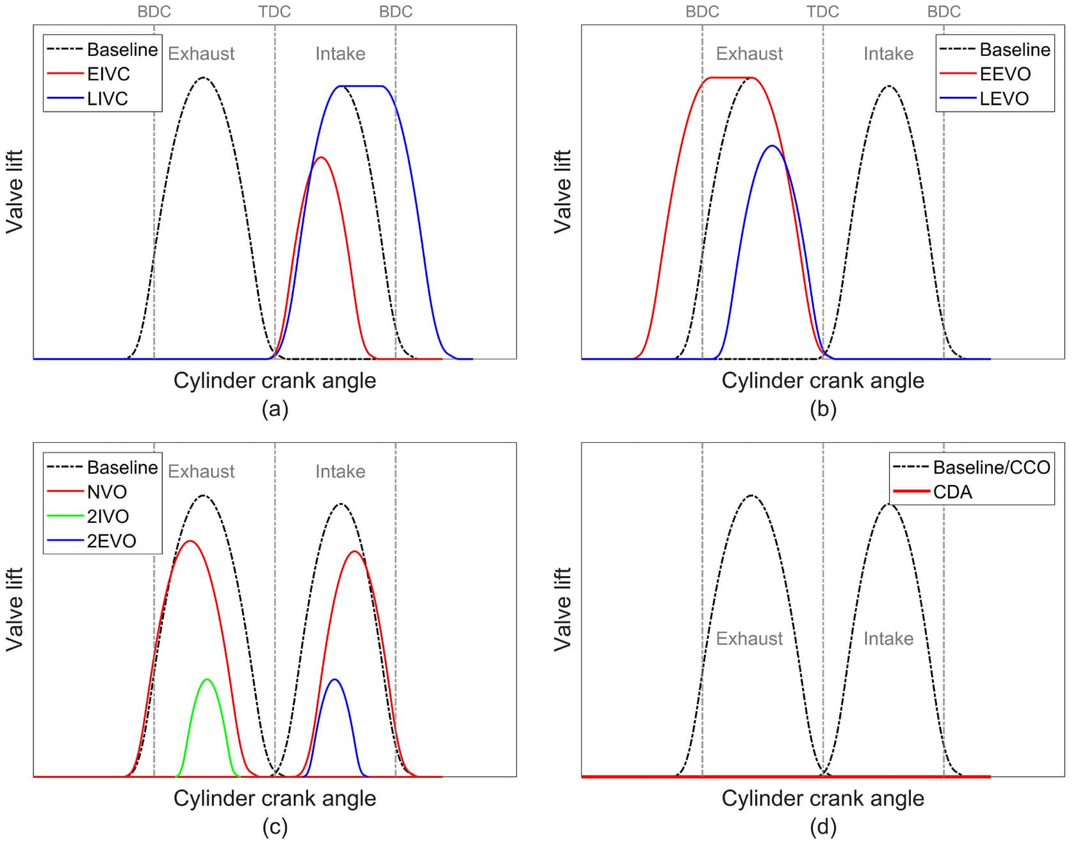


Fig. 2. VVA strategies covered in the present study for marine and off-road powertrains: (a) Intake modulation or Miller timing (EIVC, LIVC), (b) Exhaust modulation (EEVO, LEVO), (c) internal-EGR (NVO, re-breathing – 2IVO, 2EVO), (d) CDA/CCO.

VVA has received a lot of attention in recent years. Several review papers have attempted to systematise the immense complexity of the effects of different valve strategies. However, these attempts are deemed insufficient for the specific needs of the off-road and marine sectors. Those reviews predominantly focused on automotive spark ignition (SI) engine applications with a wider engine-speed range, often overlooking CI engines and emerging advanced combustion concepts in which VVA could play a critical role.

The absence of well-organised and up-to-date information regarding the role of VVA in advanced CI combustion concepts constitutes a significant knowledge gap. This study endeavours to fill this gap by offering a meticulous and industry-specific exploration of fully flexible VVA's diverse possibilities in the marine and off-road sectors. In pursuit of this, the present work identifies nine distinct VVA strategies crucial for achieving pivotal incremental development targets in marine and off-road powertrains. Fig. 2 illustrates these nine valve strategies.

The full spectrum of valve strategies can be executed through a fully flexible VVA system. Intake modulation, known as Miller timing, is achieved by adjusting IVC timing, either earlier or later. Exhaust modulation manipulates the timing of exhaust valve opening (EVO). Internal EGR (i-EGR) is realised through negative valve overlap (NVO), or alternatively, by the introduction of a second intake valve opening (2IVO) during the exhaust stroke, or by a second exhaust valve opening (2EVO) during the intake stroke. Cylinder deactivation (CDA) is accomplished by closing all valves in the deactivated cylinder, while the valves in activated cylinders continue to operate normally, introducing a fresh charge for combustion. In contrast, cylinder cut-out (CCO) maintains regular valve operation in both activated/deactivated cylinders. However, both CDA and CCO suspend fuel injection in the deactivated cylinders.

In light of the development targets for off-road and marine engines, the present study identifies three crucial areas where VVA holds significant application potential. These form the context of the critical discussion in the individual sections of this paper. *Chapter 3* delves into the application of VVA for exhaust temperature management (ETM), aiming to achieve faster light-off conditions and enhance conversion efficiency in contemporary aftertreatment systems. *Chapter 4* focuses on potential improvement in air path control to maximise controllability in air-fuel ratio, and (internal) EGR rate with minimum pumping losses. Advanced air path control is particularly important when departing from conventional diesel combustion towards more advanced combustion concepts. In this context, VVA not only supports conventional air path control but also directly influences combustion and emissions, inducing effects like valve-induced turbulence. *Chapter 5* addresses the specific dependencies of these direct effects on advanced combustion concepts. Lastly, *Chapter 6* consolidates the findings of this review to formulate a vision for the role of fully-flexible valvetrains in the development of next-generation marine and NRMM engines.

A comprehensive systematic literature review was conducted, as discussed by Thomé et al. [74]. This method minimises the risk of introducing bias to the conclusions, ensuring that the study encapsulates the majority of relevant findings. To maintain methodological rigor, the review protocol followed the Preferred Reporting Items for Systematic Reviews and Meta-Analyses (PRISMA) 2020 guidelines [75]. The selection process, covering identification, screening, eligibility, and inclusion is summarized in the PRISMA flow diagram shown in Fig. 3.

In the identification stage, Scopus and ScienceDirect databases were utilised to find the most relevant works using the following generic keywords: (i) 'variable valve actuation' OR 'timing' OR 'lift' AND 'diesel engine'; (ii) 'internal EGR' OR 'Miller' OR 'negative valve overlap' OR 'cylinder deactivation' OR 'CDA' OR 'cylinder cut-out'. The initial scientific literature search yielded a total 400 documents, encompassing various type of publications such as original research articles, conference proceedings, review articles and book chapters.

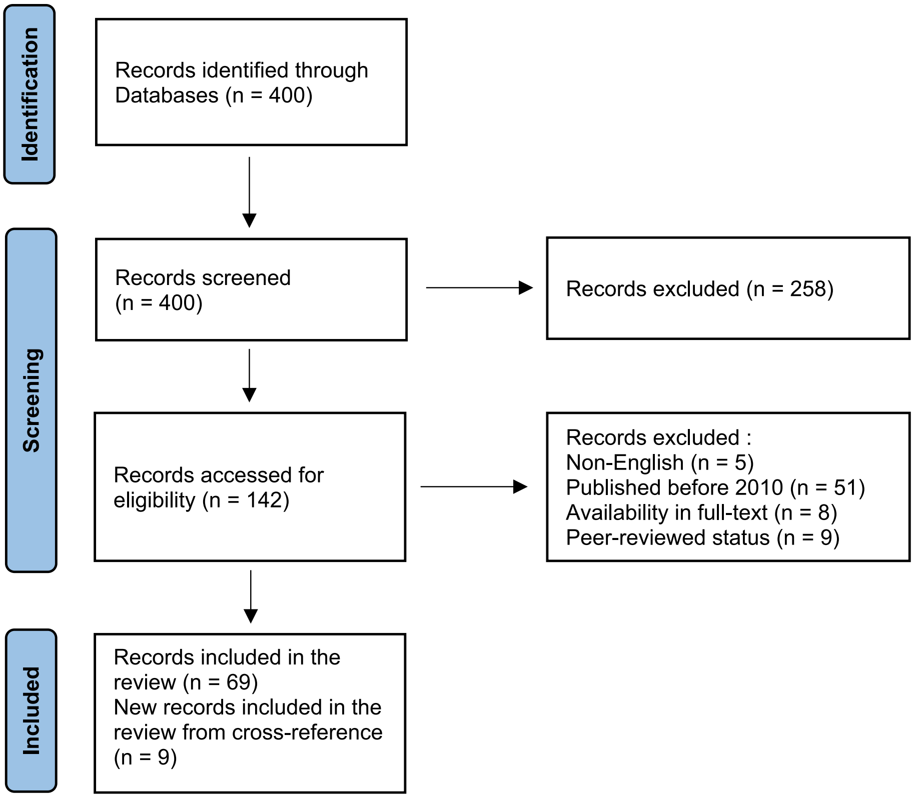


Fig. 3. PRISMA flow diagram.

In the screening stage, titles was reviewed with a focus on the most relevant keywords such as exhaust thermal management, gas exchange, swirl, turbulence, LTC, HCCI, PCCI, RCCI. Abstracts were then carefully examined to refine the selection further. A stricter criterion was also applied, emphasizing application domain. Studies mainly focused on light-duty applications (predominantly SI engine) were excluded. The review was specifically narrowed down to works addressing VVA in heavy-duty (HD) and large-bore engines, encompassing both on-road, off-road, and marine applications, and spanning four-stroke diesel and multi-fuel platforms. This step excluded 258 articles due to the lack of relevance.

In the eligibility stage, the remaining 142 articles underwent full-text review according to five inclusion domains: methodological depth, peer-reviewed status, accessibility in full text, year of publication and language in English. The articles were excluded if they were non-English, or published before 2010 or unrelated to one of nine VVA strategies defined in the present work. Additional exclusion criteria were applied to uphold the study's contemporary relevance and scholarly excellence. These included a minimum threshold for peer-reviewed publications and a requirement for Q3 tier level based on the SCImago Journal Rank (SJR) indicator [76], resulting in the exclusion of 73 additional studies that did not meet the inclusion criteria. An additional 9 papers were incorporated into the review through a cross-reference search in the final phase, further reinforcing its comprehensiveness and depth. Consequently, this process reduced the review's scope to 78 articles.

3. VVA for efficient exhaust thermal management

Modern diesel powertrains use integrated exhaust aftertreatment systems (IEATS) to comply with stringent emission limits. Generally, these systems consist of a diesel oxidation catalyst (DOC) for HC and CO reduction, a diesel particulate filter (DPF) to trap PM and selective catalytic reduction (SCR) for NO_x reduction, as shown in Fig. 4. An ammonia oxidation catalyst (AOC) could be integrated to reduce ammonia slip from the SCR. Such a system reduces harmful emissions by more than 90 % [77,78]. However, there are still some conditions that are challenging for IEATS effectiveness. These are: (i) engine cold-start & warm-up phases, (ii) idling, and (iii) low-load operation.

First, these conditions cause excessive amounts of HC, CO, NO_x, and PM due to incomplete combustion [79,80]. This is mainly attributed to low temperature of in-cylinder mixture, because engine block, coolant and lubrication temperature are all far lower than their optimum [81]. This negatively influences fuel evaporation, distribution and then combustion [82]. Second, these large amounts of harmful emissions are not mitigated in IEATS because EGT is too low to activate the catalysts [70].

Lauren et al. [83] examined two modern EU Stage-V off-road engines and observed that EGT is constantly below 250 °C whenever the engine load is below 30 %, as shown in Fig. 5 (a). Vos et al. [84] demonstrated low EGT at low engine-load (<15 % load) in a HD diesel engine (Cummins 6.7 L ISB engine), as shown in Fig. 5 (b). According to Bai

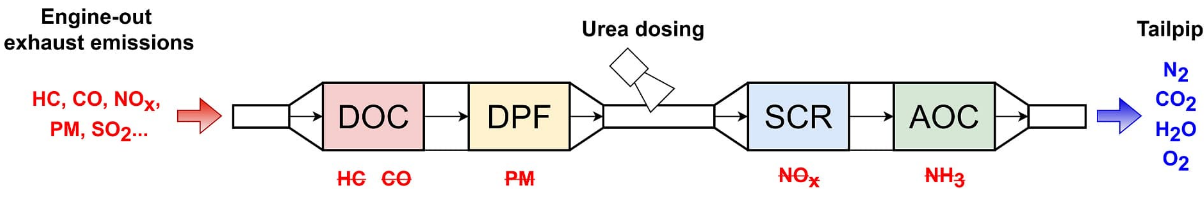
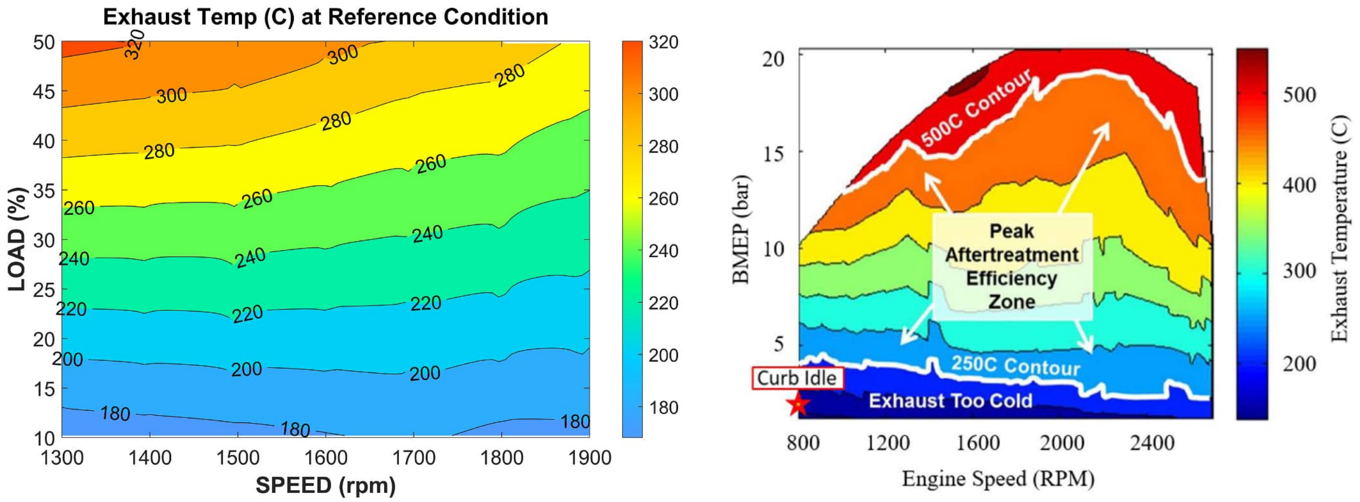


Fig. 4. Typical diesel aftertreatment system layout.



(a) (b)

Fig. 5. Exhaust gas temperature from two HD diesel engines; (a) adopted from [83]; (b) reproduced from [84].

et al. [85], EGT could be as low as 100 °C during idling. DOC and SCR systems perform very poorly (low conversion efficiency) at low EGT (< 200 °C) due to slow chemical reaction in the catalysts.

EGT should be raised to improve conversion efficiency in these difficult conditions. The light-off temperature (LOT) of a DOC is 160 °C – 220 °C [77,86]. The LOT of SCR catalysts is around 150 °C – 220 °C [87,88]. Additionally, SCR needs a working temperature of at least 200 °C in order to decompose urea completely and avoid solid deposits [89] An even higher EGT (around 250 °C – 400 °C) is beneficial for passive DPF regeneration to oxidise soot in the filter, thus trapping PM effectively without clogging over a long lifetime. [90,91]. Considering all these temperature requirements, the EGT for optimal IEATS performance ranges from 250 °C to 450 °C [92,93].

The aforementioned operating conditions such as idling and low-load points are part of emission test cycle (ISO 8178) for both applications. A new idle NO_x standard and low-load cycle test is proposed [94]. Increasingly stringent emission limits mean that ETM is crucial in modern diesel engines to guarantee sufficient emission abatement across a wide range of operating conditions. The ETM method can be classified into one of three categories: air path, fuel path and external path, as shown in Fig. 6.

Air path is mainly associated with restricting air flow into the cylinder. Less air has less heat capacity in the combustion chamber, so less heat is absorbed during combustion. This contributes to increasing the EGT. Fuel path entails adjusting injection timing and amount to raise EGT. External path involves additional devices such as a heating source to increase EGT. The air and fuel paths have been the conventional approaches, such as intake and exhaust throttling [83,95], post injection [96] or a combination of both [97]. These strategies are effective in raising EGT, but carry a substantial BSFC and BTE penalty due to increased pumping losses, high backpressure and additional fuelling [92].

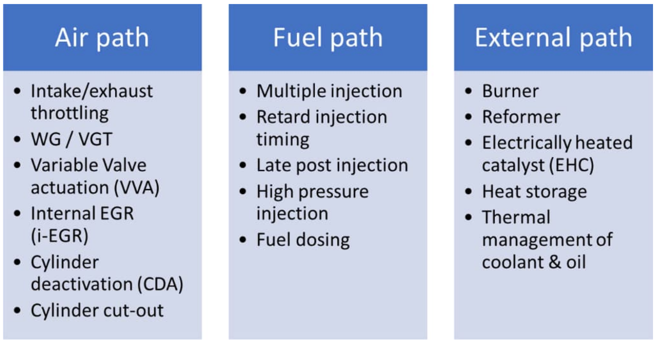


Fig. 6. Exhaust thermal management strategies for aftertreatment system in modern diesel engines, adopted from [70].

The fuel path alone has a limited effect on EGT increment and incurs a high BSFC penalty [95]. The cost and complexity of external path systems limit their use in large volumes for commercial production [98]. Moreover, the external path requires additional energy, resulting in a high efficiency penalty. Various research works indicate that the air path, particularly VVA, has demonstrated significant potential for effective EGT elevation without compromising BSFC and BTE. VVA-related ETM strategies involve intake/exhaust modulation, i-EGR, CDA and CCO. These are comprehensively reviewed in the following subsections.

3.1. *Intake modulation (EIVC, LIVC – Miller timing)*

This method modulates intake valve closing (IVC) timings to constrain air flow into the combustion chamber. Both early intake valve closing (EIVC) and late intake valve closing (LIVC) reduce volumetric efficiency (VE) due to reduced intake opening time and back flow respectively, as shown in Fig. 7 (a). Hence, less air is retained during the intake stroke, reducing heat capacity. The air's reduced heat capacity is less able to suppress high temperature increment during combustion, since less air absorbs less heat. To be specific, the heat capacity effect influences in-cylinder charge temperature mainly during expansion period, shortly after the peak of combustion [99]. The increased in-cylinder temperature raises EGT. Therefore, air flow is a key parameter to control EGT, as presented in Fig. 7 (b).

Garcia et al. [100] investigated EIVC and LIVC on a 1.8 L, single-cylinder, HD diesel engine. IVC was modulated up to 60-80 CAD from the baseline, with constant engine-load. Both EIVC and LIVC showed increasing EGT trend when IVC was advanced or retarded gradually. The EGT increment was proportional to the reduced air mass flow through the cylinder via EIVC and LIVC. The highest EGT increment (∼155 °C) was reported at the most retarded IVC timing where most air reduction was achieved. Under the same degree of IVC modulation from the baseline, EIVC indicated slightly higher EGT increment than LIVC, due to more reduced VE or lambda. Even though the pumping mean effective pressure (PMEP) penalty is reduced with the extreme Miller timing, BSFC penalty is increased with greater IVC modulation due to unfavourable thermodynamic properties. The highest BSFC penalty (+18 g/kWh) was recorded at the most retarded IVC. The study also found that extreme IVC modulations increased the premixed combustion behaviour

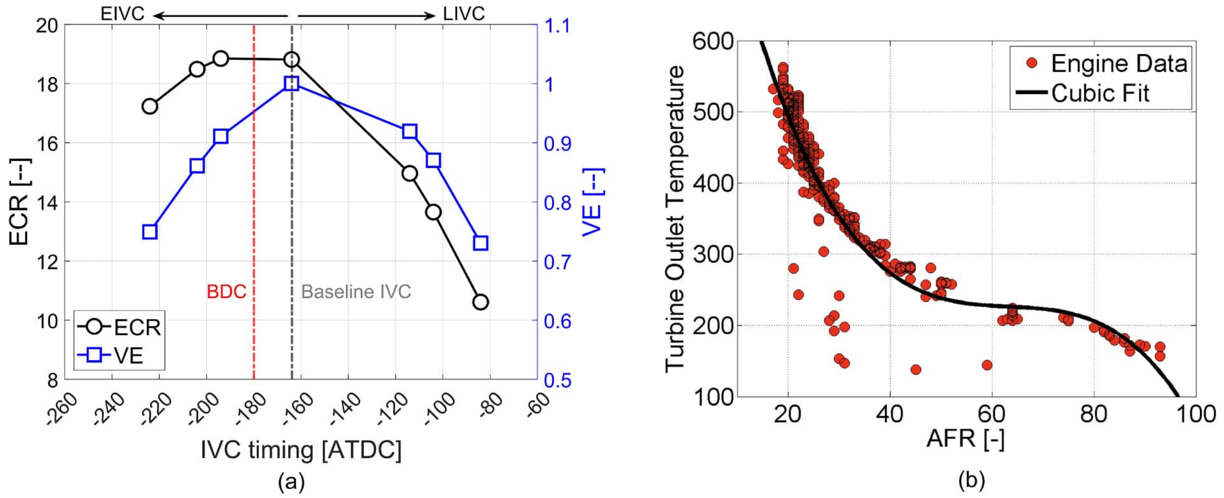


Fig. 7. (a) effect of IVC timings on VE and ECR, adopted from [100]; (b) exhaust temperature vs air-fuel ratio of HD diesel engine, reproduced from [101].

due to long ignition delay.

Gehrke et al. [102] studied the potential of EIVC and LIVC to achieve fast aftertreatment warm-up on a single-cylinder MAN D20 research engine with a cam-less valve actuation system. EIVC and LIVC increased the EGT by up to 60 °C and 120 °C respectively, with minor fuel consumption penalty, so in this case, LIVC obtained a higher EGT increment than EIVC. This was ascribed to back flow of hot in-cylinder gas contributing to the EGT increase by pre-heating the intake charge.

Garg et al. [103] also reported similar findings. Their work studied cylinder throttling via both EIVC and LIVC using a six-cylinder diesel engine equipped with an electro-hydraulic VVA system. The IVC modulations were tested under a constant engine load. Both IVC modulations could achieve effective EGT close to 250 °C (more than 50 °C increment) at turbine outlet. Both IVC modulations reduced VE and in-cylinder air mass via cylinder throttling. Interestingly, the brake thermal efficiency improved by 1.5 % due to increase in open-cycle efficiency. Furthermore, NO_x and PM emissions decreased by 40 % and 30 %, respectively because low combustion temperature and higher degree of premixed combustion (cold flame) suppress NO_x and soot formation.

Wickstrom [104] investigated various VVA strategies, including EIVC and LIVC, on a 2 L, single-cylinder, HD diesel engine. Experimental results showed an increasing EGT trend and a decreasing NO_x trend with advancing and retarding IVC. Large IVC modulations in both directions raised EGT by more than 150 °C, enabling effective warm-up at 25 % load. NO_x was reduced by as much as 35 %. However, both cases indicated a rather high BSFC penalty of up to 11 %, despite reduced pumping work. It was claimed that the BSFC penalty stems from low thermal efficiency due to the low ECR.

Guan et al. [98] performed experimental study of LIVC under low engine-load conditions on a 2 L, single-cylinder, HD diesel engine. LIVC raised EGT by 31 % (52 °C) and also lowered both NO_x and soot emissions, with a moderate BSFC penalty of 5.3 %. However, hydrocarbon (HC) and carbon dioxide (CO) emissions increased due to low combustion temperature. A similar trend has been observed in several works [100,104]. The combustion process is analysed in detail, revealing that the ignition delay is prolonged because the cylinder temperature at the end of compression is low due to low ECR. The longer ignition delay allows more time for fuel to be mixed with fresh air. This enhances premixed combustion intensity. Moreover, CA50 is delayed due to the delayed combustion start by long ignition delay.

Brückner et al. [105] observed a reversal trend whereby extreme Miller timing with single injection could increase NO_x emissions in a 4 L marine diesel engine, due to long ignition delay and greater premixed combustion. However, it was shown that a split injection strategy could lessen NO_x emission while extending the boundary of the extreme Miller timing.

Various studies have demonstrated that EIVC and LIVC are effective methods of increasing EGT over conventional fixed valve timings in modern HD diesel engines, but at the expense of increasing fuel consumption. However, many of these studies fail to describe in a systematic manner why fuel consumption increases. It is essential to know why BSFC suffers and how the penalty could be minimised. Since conventional strategies have a high BSFC penalty, powertrain manufacturers and their customers are keen to find better ETM solutions that provide a high EGT but low BSFC penalty. Actually, EIVC and LIVC's BSFC penalty is strongly associated with combustion behaviour, since pumping work is not relevant because it is reduced. The long ignition delay (or delayed CA50) shifts the whole combustion, so the main heat release occurs after the sweet spot near TDC. Thus, the work transfer is less efficient, which increases BSFC.

The combustion phase should be advanced in order to minimise the BSFC penalty while increasing EGT. De Ojeda [106] adjusted start of injection (SOI) to maintain CA50 as baseline in his study of EIVC, using a 6.4 L, V8 diesel engine. EIVC could increase EGT by 95 °C with BSFC even improved by 4.5 %. Another potential method is to use internal or external EGR to reduce ignition delay. Guan et al. [98] demonstrated the combined effect of LIVC and i-EGR. The i-EGR effectively advanced combustion phase and peak heat release by reducing ignition delay with hot residual gases. The increased cylinder temperature helped to reduce HC and CO emissions to some extent. In the end, LIVC with i-EGR raised EGT by 62 °C (+10 °C over LIVC only) with lower engine-out emissions and a lower BSFC penalty (-0.7 %) compared to just LIVC.

In applying EIVC and LIVC, high EGT increment is attained at the most extreme IVC timings, where the greatest air mass reduction is achieved. However, the extreme IVC modulation is constrained by several factors. The first is PM. The reduced air flow enriches the mixture, increasing PM emissions. Therefore, modern diesel engines have threshold AFR or lambda to avoid excessive PM emissions. However, in some research, the strong premixed combustion induced by the extreme EIVC and LIVC could suppress PM emissions [103]. Further investigation is required to establish the borderline. The second constraint is that the Miller effect increases HC and CO emissions due to low combustion temperature [98,100,104]. Thus, extreme IVC modulation would be compromised with an emissions penalty. Third, extreme Miller timing increases incomplete combustion because the excess oxygen level is decreased by large IVC modulations [98]. This corresponds to increased HC and CO emissions. However, additional boost could improve the emissions (soot, HC and CO) by reducing the locally rich mixture and promoting complete combustion, but it will decrease EGT benefit. Lastly, excessive IVC modulations delay the entire combustion

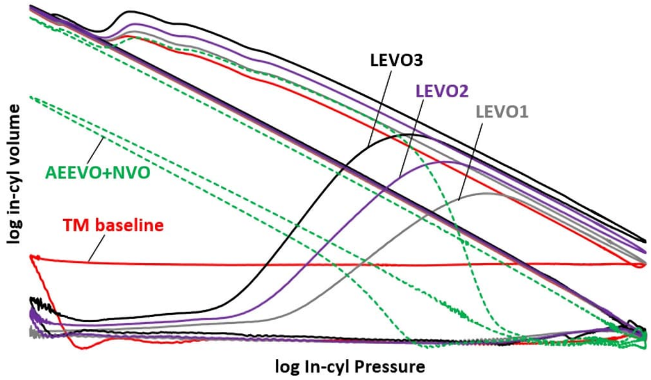


Fig. 8. P-V diagram in a log scale for different LEVO timings (LEVO3: 160 CAD delay, LEVO2: 150 CAD delay; LEVO1:135CAD delay, AEEVO – aggressive EEVO: 100 CAD advanced, TM baseline: Thermal management baseline with delayed injections and elevated exhaust pressure), reproduced from [109].

and heat release, negatively affecting BSFC and brake thermal efficiency. Therefore, these factors must be carefully considered in applying EIVC and LIVC for efficient ETM.

3.2. Exhaust modulation (EEVO, LEVO, exhaust phase)

Exhaust modulation is another common application of VVA to increase EGT. Early exhaust valve opening (EEVO) could increase EGT substantially. Before opening the exhaust valves, the combustion process occurs where high pressure and hot combustion gases expand and generate the useful work. Advancing the EVO (EEVO) halts the expansion cycle earlier, which might reduce engine performance. Since less expansion occurs at EEVO, the in-cylinder charge temperature is rather higher than the baseline, thus increasing EGT.

Roberts et al. [107] applied EEVO on a six-cylinder Cummins diesel engine with an electro-hydraulic VVA system, operating under constant load conditions. The results showed increasing EGT trend with advancing EVO. EEVO increased EGT above 250 °C at low-load operation (BMEP 1.3 bar, 2200 RPM), but with a high BSFC penalty (>18 %). The highest BSFC accompanied the highest EGT increment (extreme EEVO), due to the large loss of power. This study also found that PMEP slightly decreased with advancing EVO, although the reduction is rather minimal (<0.15 bar).

A study by Gosala et al. [108] had similar findings. They tested EEVO at loaded idle (800 RPM, BMEP 1.3 bar) on a six-cylinder diesel engine. EEVO achieved an EGT increment of around 100 °C but at the cost of 56 % higher fuel consumption and 12 times higher PM emissions compared to the baseline. The higher PM emissions stemmed from insufficient soot oxidation given by EEVO. This study also experimented with EEVO combined with i-EGR (NVO). This strategy could raise EGT by up to 150 °C, with a BSFC penalty 14 % less than just EEVO. Testing included a transient test in the Heavy-Duty, Federal Test Procedure (HD-FTP) cycle. EEVO with NVO gave a faster warm-up in the cold cycle, and 90 % SCR conversion efficiency was reached 100 s earlier than the baseline. Gehrke et al. [102] raised EGT by 65 °C but at the cost of reducing expansion work. This study also mentioned that adjusting the VGT vane position could achieve an effect comparable to EEVO.

Other works have investigated late exhaust valve opening (LEVO). This increases EGT due to recompression when EVO occurs after BDC. The recompression increases the pressure and temperature of the in-cylinder charge, thus raising EGT.

Joshi et al. [109] researched LEVO by delaying EVO by 135 CAD, 150 CAD and 160 CAD gradually against the stock EVO timing. Increasing delay produced higher EGT. EGT rose to 200 °C above the baseline, but with 86 % higher fuel consumption, and soot emission rose by a factor of five. The large EGT increment mainly is associated with the high degree of recompression by retarded EVO. However, this recompression induces high pumping work, as seen in Fig. 8. Excessive EVO delay carries a significant fuel consumption penalty. Turning to PM emissions, LEVO has a lower increment than EEVO due to longer oxidation time, but PM is still higher than the baseline due to rich fuel injection needed to obtain the same torque output.

Vos et al. [110] examined and compared both EEVO and LEVO under idling operation on a Cummins HD diesel engine, while maintaining exhaust valve closing timing. Both EVO modulations enabled high EGT increment compared to the conventional idle valve setting. LEVO outperformed EEVO by raising EGT by 85 °C, while maintaining comparable levels of exhaust flow and emissions (NO_x and PM) relative to EEVO. However, LEVO incurred a high BSFC penalty (25 %). The authors pointed out that the reduced valve lift and opening duration of exhaust valves also hinder the discharge process during gas exchange process, making the engine run harder to expel the in-cylinder charges.

Wickstrom [104] studied both exhaust and intake phase shift in a single-cylinder HD engine. Exhaust phase was advanced and intake phase was retarded. EGT rose by up to 154 °C and 88 °C at 25 % load and 0 % load respectively. This was rather higher than with EIVC and LIVC. The high EGT increment was due to the combined effect of EEVO and LIVC. NO_X emission was reduced due to the Miller effect. However, these strategies increased BSFC by +8.5 %, and soot emission increased drastically.

Bharath et al. [111] achieved an EGT increment of up to 101 °C at the DOC inlet by employing EEVO at low-load points (< 3 bar BMEP). Their comprehensive analysis, combining 1D simulation and 3D combustion modelling, was conducted on a 1.9 L light-duty diesel engine running in gasoline/diesel RCCI mode. In general, RCCI produces excessive unburned hydrocarbons (UHC) and CO emissions at low loads due to low combustion temperature. The increased EGT proved sufficient to activate the DOC, which contributed to the 98 % conversion efficiency for engine-out UHC and CO. Nevertheless, excessive EVO modulation resulted in a decline in performance (BMEP) and a substantial BSFC penalty, which was up to +9.6 % compared with the conventional diesel engine.

In summary, both EEVO and LEVO attain EGT increment, but at the expense of high fuel consumption. Since it is nearly impossible to reduce BSFC, it is important to minimise the BSFC penalty. This entails finding an optimum compromise between EGT increment and increasing BSFC. Additionally, excessive soot emission has been observed in EVO modulation due to increased fuel injection and short oxidation time. This should be carefully considered in the optimisation and calibration stage.

3.3. Internal EGR (NVO and reintroduction)

Conventionally, EGR is employed to suppress NO_x emissions. However, internal EGR (i-EGR) also can serve to increase EGT, using the mechanism similar to intake modulation. It allows more residuals in the combustion chamber during the gas exchange process. A large amount of residual gases restricts fresh charge induction, and so the mixture's reduced heat capacity causes the EGT increase. Furthermore, the hot residual gases trapped in the cylinder also contribute to elevating EGT directly. In general, the more residual gases remain in the cylinder, the higher is the EGT achieved.

There are several ways of using VVA to enable i-EGR. The first is negative valve overlap (NVO) by advancing EVC and/or retarding IVO. Early EVC shortens the exhaust period, trapping hot residual gas fraction in the cylinder. This decreases air flow in the next cycle. Another method is reopening the intake or exhaust valves, which is called 2IVO (second intake valve opening) / 2EVO (second exhaust valve opening) or IV (intake valve)/ EV (exhaust valve) bump.

2IVO occurs during the exhaust stoke, allowing exhaust gases to leak into the intake manifold. These residual gases are reintroduced to the cylinder, thus diminishing air induction in the next cycle. 2EVO occurs during the intake stroke, allowing exhaust gases to flow back to the

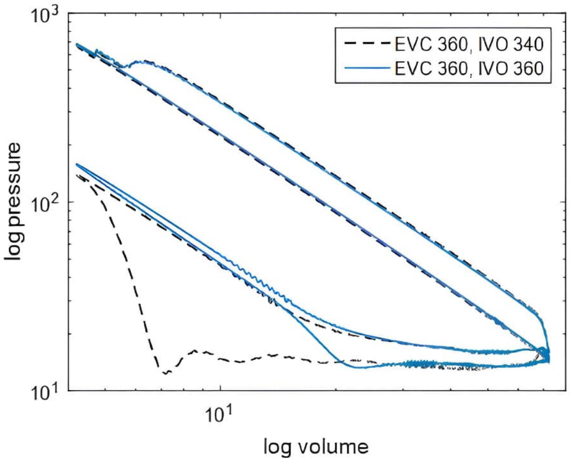


Fig. 9. PV diagram comparing the effect of IVO timing in NVO application, reproduced from [113].

combustion chamber due to the pressure gradient, because exhaust pressure is higher than intake pressure in most turbocharged diesel engines. This reduces intake flow in the same cycle. The reduced air induction caused by trapped residual gases raises EGT due to the lower heat capacity effect.

Joshi et al. [112] investigated internal EGR via exhaust reintroduction and NVO on a HD diesel engine under the idling condition (800 RPM and BMEP 1.3 bar). NVO was realised by EEVC with LIVO, and reintroduction was achieved by 2EVO. NVO increased EGT by +38 °C, with fuel consumption 12 % higher than the baseline. 2EVO raised EGT by +52 °C, with fuel consumption 3 % higher than the baseline. 2EVO's greater EGT increment is due to higher in-cylinder dilution, which corresponds to lower air induction and thus reduced in-cylinder heat-capacity. Furthermore, the higher dilution contributed to supressing more NO_x: both strategies cut NO_x by more than a half.

NVO's high fuel penalty stems from high pumping losses by recompression caused by early EVC. Conversely, 2EVO's low fuel penalty is associated with less pumping loss by lower air induction and equalisation of in-cylinder pressure with exhaust during reinduction. Other factors contributing to 2EVO's lower fuel penalty include increased specific heat ratio (γ) due to the enhanced dilution effect, and reduced in-cylinder heat loss compared with NVO. These factors improve closed-cycle efficiency.

However, both 2EVO and NVO increased soot emission by more than 2.5 times, due to high in-cylinder dilution. UHC emissions from NVO were 1.5 times higher than the baseline, whereas they were 20 % below the baseline with 2EVO. For this reason, the engine-out HC emission would be a limiting factor for high EGT in NVO application.

Gehrke et al. [102] examined NVO in a 1.6 L HD single cylinder diesel engine, modified from MAN D20 engine platform. Increasing NVO period resulted in an EGT rise of up to +69 °C, penalising fuel consumption by +7.8 %. It was pointed out that the elevated charge temperature and additional compression of EGR lead to increased fuel penalty to some extent. While engine-out NO_x was maintained by external EGR, PM increased about 2.4 times due to the reduced air-fuel ratio from both internal and external EGR.

Joshi [113] performed extensive experimental study on NVO and reintroduction (2EVO) with a heavy-duty, single- cylinder diesel engine. The NVO work investigated the effect of IVO timing on fuel efficiency and the amount of i-EGR. He found that the delayed IVO timing enables more fuel-efficient operation because the pumping work induced by recompression is reduced by expansion with delayed intake opening, as shown in Fig. 9.

Moreover, the delayed IVO traps more residual gases (high EGR effect) because the higher suction pressure created by retarded IVO prevents the residual gases from escaping into the intake manifold. This is beneficial to increasing EGT. Joshi also examined the influence of cam phasing with NVO. Greater EVC advancement traps more exhaust gases, which will increase EGT in the next cycle. The corresponding delay in IVO contributes to reducing pumping loss and improving fuel efficiency.

Joshi examined the effect of exhaust valve lift in the exhaust reintroduction study (2EVO). Higher valve lift produced more EGT elevation with a lower fuel penalty. A higher turbine outlet temperature (TOT) is achieved by the reduced flow rate due to increasing dilution effect. The high dilution also increases gamma (γ) which in turn, minimises the fuel penalty. Furthermore, the high lift reduced NO_x (-80 %) effectively, but soot emission was three times higher. Table 3 summarises the effects of varying key parameters of i-EGR in Joshi's work.

Wickström [104] observed a similar trend while varying intake and exhaust valve lift during exhaust reintroduction. Both strategies achieved an EGT increment of more than 120 °C at 25 % load at the highest lift (4 mm). The fuel penalty was below 10 %. In both cases, NO_x was reduced, but soot emission increased because less excess oxygen hampers the complete oxidation to CO_2. Importantly, the author stated that 2IVO might cause accumulation of soot particles in the intake manifold.

Zhang et al. [114] investigated 2IVO and 2EVO on a 1 L, single-cylinder, HD diesel engine. A higher degree of 2IVO and 2EVO, achieved by increasing lift and duration, elevated EGT by as much as 55 °C and 78 °C respectively, under low engine-load. 2EVO attained greater EGT increment than 2IVO because the hot exhaust gases are directly introduced into the combustion chamber. It is noted that both reintroduction strategies reduced efficiency by only 1 %, assuming that the start of the injection was adjusted to compensate the reduced ignition delay caused by high compression temperature.

This slight efficiency penalty is attributed to greater in-cylinder heat loss as the in-cylinder gas temperature was increased by introducing hot residuals. Furthermore, the combustion rate slowed with the reduced lambda. Both reintroduction strategies achieved low HC and CO emissions because their oxidation was promoted by hot residual gases. However, higher smoke emission was observed in 2EVO due to more significant residual concentration and temperature stratification than with 2IVO.

The 2EVO strategy also has been examined in a natural gas/diesel RCCI engine at low load [35]. Here, EGT was raised by 101 °C by retaining hot residual gases. This could make a substantial improvement to conversion efficiency of a methane catalyst. Additionally, 2EVO raised combustion efficiency by introducing i-EGR, which increased IVC temperature and combustion temperature. Consequently, engine-out methane (CH_4) emissions were reduced by about 98 %. In fact, i-EGR can be used to control combustion phase and emissions in LTC applications. This is discussed further in *Chapter 5.2*.

In short, both NVO and reintroduction show the potential to elevate EGT via the dilution effect, with moderate fuel penalty. The fuel penalty mainly relates to increased pumping work and slow combustion caused by the EGR effect. For efficient thermal management, advanced fuel injection could compensate for the slow combustion and fuel penalty in both cases. In addition, it is useful to delay IVO timing in NVO to minimise the pumping loss. Although there is clear merit in harnessing the EGR effect for NO_x reduction, extreme EGR operation is constrained by excessive soot emission due to the NO_x – PM trade-off. However, increasing boost could mitigate soot emission with penalty of EGT increment. Thus, EGT increment should be a considered compromise with the emission and BSFC penalties in internal EGR operation.

3.4. Cylinder deactivation (CDA) and cylinder cut-out (CCO)

Cylinder deactivation (CDA) is another prospective strategy for effective and efficient EGT elevation. CDA involves deactivating both fuel injection and valve operation in selected cylinders. This has the effect of reducing engine displacement. Deactivated cylinders do not produce any work, so active cylinders need to operate at higher load to

Table 3
Summary of influence of sweeping key parameters in i-EGR operation, adopted from [113].

	Negative valve overlap (NVO)		Reintroduction (2EVO)		
Valve profile varied	Delayed IVO (for given advanced EVC)	Cam phasing	Delayed 2EVC	Increased lift	Phased profile
Parameters					
i-EGR quantity	↑	↑	↑	↑	↓
Fresh air flow	↓	↓	↓	↓	↑
Fuel consumption	↓	↑	↓	↓	↑
Engine-out NO_x	↓	↓	↓	↓	↑
Engine-out soot	↑	↑	↑	↑	↓

maintain torque demand. This entails more fuel injection and higher combustion temperature. Ultimately, the increased heat release in active cylinders elevates EGT.

CDA has been studied mainly in SI engines to improve fuel consumption at low- and part-load conditions by reducing throttling losses [115]. There is limited research on diesel engines, but Zammit et al. [116] demonstrated high reduction of soot, HC and CO emissions with CDA. Growing interest in ETM has prompted some more experimental and simulation investigation into CDA in diesel engines. In particular, a research group from Purdue University in the US has performed extensive experiments on CDA and various flexible VVA strategies. For example, Magee [101] performed an experimental study of CDA using a 6.7 L, six-cylinder, HD diesel engine, deactivating half the cylinders. This halved the fresh air charge flow and raised the EGT by 100 °C at low-load operation. In addition, the brake thermal efficiency (BTE) was improved by 10 % compared to six-cylinder operation, due to reduced pumping loss from low air induction and discharge process. In addition, it revealed that the engine was working more efficiently at low load, but not at high load, with CDA.

Gosala et al. [117] investigated CDA on the same engine and observed similar results. Deactivation of three cylinders produced a TOT increment of 50 °C, with 11 % reduction in fuel consumption at loaded idle. The authors pointed out that lower total heat loss due to the reduction in active cylinders contributes to improved fuel efficiency. Simultaneously, higher-load operation triggered by CDA to maintain the same engine power helps to increase EGT by increased heat release.

The same engine was used again by Joshi et al. [118]. They observed low PM, NO_x and HC emissions when using CDA to maintain engine-out temperatures above 200 °C at idling operation. In addition, deactivation of half the cylinders accelerated the warm-up during the cold-start cycle in transient operation (HD-FTP). Compared to the standard six-cylinder operation, CDA cut the time to reach 50 % and 80 % SCR efficiency by 24 s and 73 s, respectively. CDA reduced total engine-out NO_x by around 20 % and achieved a 3 % fuel saving over the HD-FTP drive cycle.

Some studies have demonstrated that CDA is more fuel-efficient in raising EGT than other VVA strategies, such as i-EGR and intake modulation. Joshi et al. [112] compared CDA with i-EGR via 2EVO and NVO, under idling operation. CDA showed exceptional performance to increase EGT by +70 °C, even accompanied by a 4 % fuel saving.

Vos et al. [93] also evaluated CDA with EVIC and LIVC under high-speed and low-load operation (2200 RPM and 3.9 bar BMEP), using it to replace a conventional thermal management method (exhaust manifold pressure control). CDA with four firing cylinders provided a higher EGT elevation (+117 °C) than EIVC and LIVC, even with a 1.5 % fuel saving.

This significant TOT increment is enabled by increased fuelling per active cylinder to achieve the same load, and by a large reduction in air flow by cylinder deactivation.

Cylinder cut-out (CCO) is another approach to raise EGT. This is rather similar to CDA in operation and implementation. CCO disables fuel injection for the selected number of cylinders, but unlike CDA, intake and exhaust valves operate as usual. The key difference is that CCO allows non-firing cylinders to breath without fuel, so they do not influence total incoming charge flow. CCO does not require any modification to valves, so is easier to implement than CDA.

Previously, CCO was studied as a fuel-saving technique. Konrad et al. [119] used 1D simulation to demonstrate a 26 % fuel saving by CCO in a medium-speed, dual-fuel, marine engine. Yang et al. [120] presented a 1-13 % improvement in fuel efficiency with an off-road diesel engine for an excavator. In addition to fuel saving, CCO also could enhance EGT in the similar manner as CDA. The increased heat release in active cylinder raises EGT substantially.

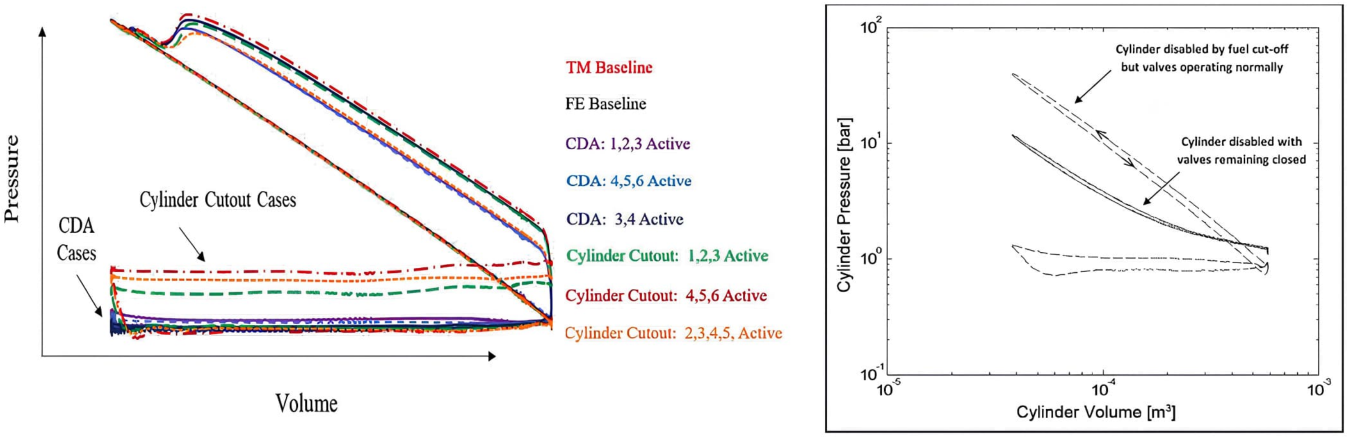
Gosala et al. [121] performed more extensive study on flexible valve modulation of non-firing cylinders using a HD diesel engine. Intake or exhaust valves, or both, were partially opened in deactivated cylinders, and only half the cylinders were active. The study demonstrated that non-firing cylinders with at least 4 mm valve opening show a similar EGT trend and fuel efficiency as CDA, due to low gas-exchange loss. Hence, partial valve opening with CCO could minimise the fuel penalty.

Vos et al. [84] examined and compared both CDA and CCO in a six-cylinder HD diesel engine at idling condition (800 RPM and 1.3 bar BMEP). Both CDA and CCO could effectively raise EGT to the same level, with 40 % and 17 % less fuelling respectively, compared to a conventional thermal management strategy using an over-closed VGT vane. CDA's superior fuel saving was due to less pumping work in both active and deactivated cylinders, as depicted in Fig. 10 (a). Zammit et al. [116] also reported high pumping loss in a non-firing cylinder with CCO, shown in Fig. 10 (b).

Liu et al. [122] found similar results in simulation work. CCO showed four times higher pumping loss than CDA due to the gas exchange process of the normal valve operation. CCO has large pumping work in a firing cylinder, arising from a combination of an over-closed VGT and high charge flow [84]. Non-firing cylinders in CCO contribute to a decreasing dilution effect by discharging non-combusted gases. This requires high backpressure to increase the amount of EGR for effective NO_x control. However, CCO suffers from the need to obtain the required backpressure without exhaust pressure control, unlike CDA. Hence, the normal valve operation in CCO has a negative impact on both firing and non-firing cylinders.

Kim et al. [99] also investigated CDA and CCO in an off-road diesel engine platform with 1D simulation. It was noted that CCO has less capability to elevate EGT compared to CDA. In CCO, normal valve operation in deactivated cylinders discharges cooler charges that mix with hot exhaust gases from activated cylinders in the exhaust manifold. This mixing process significantly reduces exhaust thermal energy, constraining the EGT increment. In some case, it was reported that EGT actually decreased instead of increasing. Liu et al. [122] also reported EGT decrement by 10 °C with CCO. However, CDA increased EGT by +50 °C.

Regarding the emissions, a high amount of HC, CO and soot was reported due to the locally rich mixture caused by high amount of main fuel injection under constant injection pressure as the baseline. However, they can be mitigated below the baseline by increasing injection pressure. A boost pressure sweep with CDA and CCO revealed that boost pressure control is strongly associated with high pumping loss. Additionally, high boost pressure helps improve fuel efficiency by promoting more premixed combustion, but it reduces the EGT increment due to the heat capacity effect. Moreover, extra boost enables better air-fuel mixing and alleviates the locally rich mixture which could suppress the



(a) (b)

Fig. 10. PV diagram to compare pumping work between CDA and CCO in (a) firing cylinder, reproduced from [84]; and (b) non-firing cylinder, reproduced from [116].

formation of HC, CO and soot to some extent.

Liu et al. [122] performed a numerical study on a medium-speed, high-power diesel engine (220 L, 16 cylinders) with a three-dimensional CFD combustion model. The objective was an accurate evaluation of the influence of CDA and CCO on emissions. The simulation was carried out at light-load condition (450 RPM and 0.57 bar BMEP), with half the cylinders deactivated. CO emission in both cases was reduced by 75 % compared to normal operation with high injection pressure where rail pressure varies according to load and engine speed. The high injection pressure enhances air-fuel mixing and combustion. The elevated combustion temperature improves oxidation rate and evaporation of fuel on cylinder surfaces, offsetting both over-penetration and wall-wetting due to the high injection pressure. Soot emission was also decreased by 62.9 % in both cases. CFD simulation revealed that soot formation was increased with rich air-fuel mixing, but most of them were oxidised during expansion cycle due to high combustion temperature. However, the high combustion temperature resulted in higher NO_x emission in both cases.

Zammit et al. [116] also studied emissions reductions in CDA operation. Soot emission was cut by 60 % due to increased oxidation with high combustion temperature. Vos et al. [84] presented a soot reduction of more than 50 % in both CDA and CCO by maintaining a relatively high AFR ratio. Engine-out NO_x increased due to higher cylinder temperatures in both cases, but EGR mitigates NO_x emission drastically, with minor fuel penalty (+0.8 %).

Implementation of CDA and CCO carries one disadvantage – greater noise, vibration and harshness (NVH). The reduced firing frequency and suspended cylinder operation could result in unbalanced rotation and vibration [112]. Zammit et al. [116] claimed that the higher rate of pressure rises in the active cylinders during CDA could increase combustion-generated noise and vibration in a light-duty diesel engine.

Hushion [123] asserted that deactivation should be used at less than 30 % load for vibration and power delivery reasons. However, Archer and McCarthy [124] observed acceptable NVH limit with CDA with a HD engine below IMEP 3 bar. Pieczko et al. [125] demonstrated the feasibility of mitigating the vibration for a HD engine with CDA at idling. NVH can be effectively mitigated by using a dual-mass flywheel or a centrifugal pendulum absorber [126]. Furthermore, an advanced CDA concept, so-called rolling CDA [127] and dynamic skip fire (DSF) [128], also improves NVH characteristics under CDA operation. Cummins and Tula demonstrated this technology on on-road HD vehicles, achieving a 20 % fuel saving, efficient thermal management, a 75 % NO_x reduction and a CO_2 reduction of 5 % [129]. These approaches enable more flexible CDA operation across a wider range of conditions and support smooth transient operation by gradually adjusting firing order and frequency in response to load demand – an important consideration given the significant reduction in air flow during CDA.

The similar approach can be applied to off-road HD engines and large bore marine engines to improve NVH and transient performance to some extents. However, the implementation of the rolling CDA and DSF on the large engines is constrained by system inertia, where even a state-of-the-art valvetrain, such as an electronic-hydraulic valve system, has a slow response [53,130]. For this reason, Liu et al. [122] applied conventional CDA to a mid-speed locomotive diesel engine.

CDA operation is also influenced by turbocharger. Unlike CCO, CDA reduces the air flow and consequently lowers exhaust energy (enthalpy), which decreases turbocharger efficiency at low load. Since turbochargers are generally optimised for high-load conditions, their performance at low load is inefficient [131]. Kim et al. [99] reported that CDA shifts the operating point closer to the surge line due to reduced mass flow and higher pressure ratio. Kakani et al. [132] emphasized the need for higher boost levels to supply sufficient air and keep PM emissions within acceptable limits. The use of a variable geometry turbine (VGT) can enhance turbocharger efficiency, extend the operating range and transient response by providing higher boost at low load under CDA [123]. However, this additional boost may reduce EGT due to the increased heat capacity.

In a nutshell, these studies found CDA and CCO showed similar performance in elevating EGT by increased heat release, with lower emissions (HC, CO, and soot). But CCO is less effective and efficient, because normal valve operation in deactivated cylinders leads to mixing of cool non-combustion gases with hot combustion gases which restricts the EGT increment. In addition, it has a negative impact on the scavenging process in both firing and non-firing cylinders. However, partial breathing could improve the gas exchange process, allowing fuel savings close to CDA level. Both strategies are rather more efficient than other VVA methods. Additionally, unlike CDA, the performance of the thermal management system in CCO is sensitive to the geometry of the split exhaust manifold, exhaust pressure control and location of the EGR loop in multi-bank engines such as V6 or V8 [84]. Therefore, it is important to determine not only the operation boundary, but also the number and location of CCO's non-firing cylinders, for safe and effective operation.

3.5. *Comparison of all VVA strategies*

Table 4 summarises the literature studies discussed above to compare each VVA strategy and its effectiveness for ETM. The summary weights the reported EGT increment against the change in BSFC and the

Table 4
Summary of literature reviews on exhaust thermal management.

Refs.	VVA strategy	VVA modulation	Type of Engine	Operating condition	Results				Notes
				Speed / Load	ΔEGT	ΔBSFC	ΔNO$_x$	ΔPM or Soot	
				[RPM] / [bar]	[°C]	[%]	[%]	[%]	
[100]	EIVC	IVC: 0 to -60°CA	HD diesel, 1.8 L, one-cylinder, HP-EGR*	1160 / 17.6 (IMEP)	+117	–	+5.2	–	NO$_x$ is maintained by EGR
	LIVC	IVC: 0 to +80 °CA			+155	–	+5.2	–	
[102]	EIVC	IVC: 0 to -70 °CA	HD diesel, 1.6 L, one-cylinder	1250 / 5.4 (BMEP)	+58	+1.2	0	+104	External EGR is used for constant NO$_x$ emission
	LIVC	IVC: 0 to +70 °CA			+116	+2.6	0	+234	
[103]	EIVC	IVC: 0 to -65 °CA	HD diesel, six-cylinder, VGT, HP-EGR	1250 / 2.5 (BMEP)	+55	BTE: +1.3 %	-40	-35	VGT and EGR remain as baseline setting
	LIVC	IVC: 0 to +100°CA			+50	BTE: +1 %	-34	-44	
[104]	EIVC	IVC: -50 to -110 °CA	HD diesel, 2.0 L, one-cylinder	1200 / 25 % load	+175	+6.5	-35	+170	
	LIVC	IVC: 0 to +80°CA			+213	+11	-16	< 0.15 FSN	
[98]	LIVC		HD diesel, 2.0 L, one-cylinder, EGR	1150 / 2.2 (BMEP)	+52	+5.3	-13.5	-81	
	LIVC+ iEGR				+62	+4.6	-2.3	-85	
[106]	EIVC	IVC: 0 to -90 °CA	HD diesel, 6.4 L, eight-cylinder** HP-EGR	2050 / 4.3 (BMEP)	+90	-4.5	0	-66	Const. CA50 NO$_x$ control with HP-EGR
[107]	EEVO	EVO: 0 to -90 °CA	HD diesel, six-cylinder, HP-EGR	2000 / 1.3 (BMEP)	+52	+18	–	–	
[108]	EEVO		HD diesel, six-cylinder, HP-EGR	800 / 1.3 (BMEP)	+76	+49	-30	+900	External EGR is used
	EEVO + iEGR				+142	+38	+26	+1550	External EGR is not used
[102]	EEVO	EVO: 0 to -70 °CA	HD diesel, one-cylinder	1200 / 5.4 (BMEP)	+65	+20.6	0	+233	External EGR is used for constant NOx emission
[109]	LEVO	EVO: -135, -150, -160 °CA	HD diesel, six-cylinder, HP-EGR	800 / 1.3 (BMEP)	+93	+53	+54	+26	
[110]	EEVO		HD diesel, six-cylinder, HP-EGR	1200 / 1.3 (BMEP)	+47	+57	-8	-12.5	
	LEVO				+89	+78	-11	-12.5	
[104]	EV + IV shift***	EV: 0 to -40 °CA IV: 0 to +40 °CA	HD diesel, 2.0 L, one-cylinder,	1200 / 25 % load	+154	+8.5	-51	+370	
[112]	NVO		HD diesel, six-cylinder, HP-EGR, VGT	800 / 1.3 (BMEP)	+38	+12	-64	+240	External EGR is not used
	2EVO				+52	+3	-80	+300	Exhaust restriction is used External EGR is not used
[102]	NVO	NVO: 0 to ±80 °CA	HD diesel, 1.6 L, one-cylinder	1200 / 5.4 (BMEP)	+69	+7.8	0	+242	External EGR is used for constant NO$_x$ emission
[104]	2EVO	Lift: 1 to 4 mm	HD diesel, 2.0 L, one-cylinder	1200 / 25 % load	+123	+6	-62	> 1.0 FSN	
	2IVO	Lift: 1 to 4 mm			+144	+9	-81	> 2.5 FSN	
[114]	2EVO		HD diesel, 1 L, one-cylinder****	1.3 (IMEP)	+78	-1.6	0	> 0.1 FSN	Injection is adjusted to maintain CA50
	2IVO	Lift: 5.8 mm			+55	-1.2	-39	< 0.04 FSN	Const. injection (10 mg/cycle)
[35]	2EVO		HD diesel engine, six-cylinder, HP-EGR	1000 / 3 (IMEP)	+101	–	–	–	Natural gas/diesel RCCI
[101]	CDA	three cylinders deactivated	HD diesel, 6.7 L, six-cylinder, HP-EGR, VGT	1200 / 13 % load	+100	-10	–	–	
[117]	CDA	three cylinders deactivated	HD diesel, 6.7 L, six-cylinder, HP-EGR, VGT	800 / 1.3 (BMEP)	+50	-11	–	–	
[118]	CDA	three cylinders deactivated	HD diesel, six-cylinder, HP-EGR, VGT	800 / 1.3 (BMEP)	+60	-4	-56	-75	
[112]	CDA	three cylinders deactivated	HD diesel, six-cylinder, HP-EGR, VGT	800 / 1.3 (BMEP)	+69	-4	-62	-65	HP-EGR is used for NO$_x$ control
[93]	CDA	two cylinders deactivated	HD diesel, six-cylinder, HP-EGR, VGT	2200 / 3.9 (BMEP)	+117	-1.5	+4	+19	
[122]	CDA	eight cylinders deactivated	Locomotive diesel, 220 L, sixteen-cylinder	450 / 0.57 (BMEP)	+50	-11	+76	-62.9	
	CCO				-10	+1.8	+143	-62.9	
[99]	CDA	two cylinders deactivated	HD diesel, four-cylinder,	1900 / 26 % load	+205	-1.4	+10	+169	Maintain const. Engine load Excessive emissions due to const. Injection pressure

(*continued on next page*)

Table 4 (*continued*)

Refs.	VVA strategy	VVA modulation	Type of Engine	Operating condition	Results				Notes
				Speed / Load	ΔEGT	ΔBSFC	ΔNO_x	ΔPM or Soot	
				[RPM] / [bar]	[°C]	[%]	[%]	[%]	
	CCO	two cylinders deactivated			+65	+14.6	-3	+387	
[84]	CCO	three cylinders deactivated	HD diesel, six-cylinder, HP-EGR, VGT	800 / 1.3 (BMEP)	+104	+34	-40	+124	HP-EGR is used for NO_x control

[*] 11 L, six-cylinder is modified to one-cylinder operation by deactivating five cylinders.

[**] Engine is modified to enable low-temperature combustion.

[***] Exhaust and intake valves are advanced and retarded, respectively. (EEVO + EEVC + LIVO + LIVC).

[****] Six-cylinder engine is modified to one-cylinder operation by deactivating five cylinders.

relative effect on engine-out emissions. Fig. 11 visualises the most relevant trade-off, between EGT and BSFC. Most of the discussed VVA strategies show potential to elevate EGT between 50 °C and 150 °C at low load (IMEP 1-5 bar), but there are large variations in their effects on BSFC. Fig. 11 shows that CDA is clearly the most fuel-efficient strategy, enabling an EGT increment of around 200 °C, while BSFC is reduced by up to 10 %. Conversely, the most inefficient strategy is exhaust modulation, which carries a 20-90 % BSFC penalty due to a large power deficit (EEVO) and a large pumping loss (LEVO). Intake modulation and internal EGR both have rather similar BSFC outcomes of -5 % to +10 %. Its BSFC penalty can be partially mitigated with CA50 control. Importantly, intake modulation and CDA can offer the highest EGT increment than others. Optimising the combustion phasing indicates further room for BSFC improvement, but not all the works included in this review considered this option. CCO deactivated several cylinders, like CDA, but it has a high BSFC penalty due to large pumping loss in both activated and deactivated cylinders. This shows that valve operation has a huge impact on gas exchange, efficiency and EGT when cylinders are deactivated.

One can conclude from Table 4 that VVA has influenced emission characteristics because it affected combustion, by changing trapped mass and mixture pressure and temperature before combustion occurs. However, it is difficult to observe a consistent trend, since each study has different sub-systems and emission-control strategies, while operating at different loads and engine speeds. It is possible to identify a trend if the same engine and hardware setup is used. The latest work by Kim et al. [99] analysed and compared VVA's impact on emissions under the same engine platform, using numerical simulation with a predictive combustion model thoroughly verified with experimental data. The study reveals that VVA has negative impacts on soot, HC and CO but high NO_x reduction. That study's EGT-BSFC trends and conclusions were similar to those depicted in Fig. 11. This emission penalty can be improved to some extents with increasing boost, but it will decrease EGT. For this reason, mild boost control rather than aggressive boost control is recommended to balance EGT increment and emission penalty.

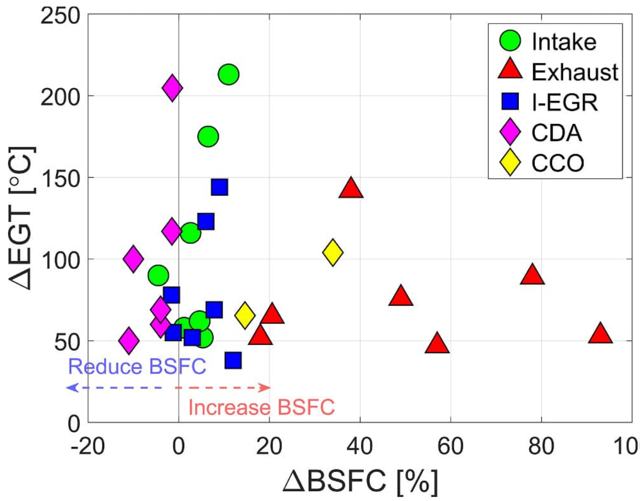


Fig. 11. Comparison of all VVA strategies, showing EGT increment and BSFC penalty, based on selected literatures.

Lastly, it should be noted that majority of existing studies are conducted under steady-state condition. However, evaluating transient performance (time-dependent or transient cycle) – specifically, how rapidly EGT can reach the target value with given aftertreatment system – is equally important for real-world applications.

4. VVA to control gas exchange process

The gas exchange process is pivotal for internal combustion engines, expelling burned gases and introducing fresh charges. The significance of this process lies in its direct impact on engine efficiency, performance and breathing capacity. Moreover, gas exchange plays a key role in shaping in-cylinder air-fuel mixture (lambda, stratification, turbulence), influencing combustion and emissions. Various strategies have been developed to optimise this process, each with its own advantages and limitations.

For instance, turbochargers are used to reduce pumping work and improve efficiency by regulating intake and exhaust pressures [133]. Intake manifold geometry optimisation [134,135] and turbocharging the intake air [136] are used to enhance volumetric efficiency (VE). Advanced piston designs and variable compression ratio (VCR) engines [137,138] have been used to adjust compression ratio to achieve the optimum effective compression ratio (ECR). Cylinder-head modifications [139] and advanced intake port and manifold designs [140] can optimise in-cylinder motion, specifically swirl and turbulence.

However, many of these methods lack the flexibility to adapt dynamically to engine operating conditions, and often require fundamental changes to the engine, making them less practical. In contrast, VVA offers real-time adjustments of valve operation, allowing dynamic control of the gas exchange process. This flexibility not only enhances efficiency and performance, but also enables use of different strategies to suit varying operating conditions, such as load and speed. Furthermore, combining VVA with other traditional methods, such as turbocharging, can lead to even better results. Therefore, VVA seems to be a more promising and adaptable approach compared to traditional methods.

This chapter reviews how different VVA strategies can be applied to achieve efficient gas exchange in HD engines. Although some studies have focused on on-road HD or super-duty (SD) diesel engines, their findings are relevant and extendable to marine and off-road applications. Different aspects of the gas exchange process, along with their relevant VVA strategies, will be explained in separate sections.

4.1. *Intake modulation for reduction of pumping work*

Pumping work is an indicator of gas exchange efficiency. Modern diesels in both on-road and off-road applications grapple with high exhaust backpressure, resulting in undesirable pumping losses. This backpressure is a consequence of emission mitigation measures such as IEATS (SCR and DPF) and external EGR (backpressure control) [12]. Elevated backpressure significantly augments pumping losses because the engine expends additional effort to overcome the increased pressure. The adverse impact of high pumping losses is evident in reduced engine efficiency, compromised performance and diminished fuel economy. However, intake modulation (EIVC/LIVC) can reduce these high pumping losses.

Wang et al. [141] studied how Miller timing via EIVC affects performance, combustion and emissions of a 7.2 L, six-cylinder HD marine diesel engine equipped with two-stage turbocharging. The experimental testing was conducted at four different loads: 25 %, 50 %, 75 % and 100 % load. PMEP was reduced with increase of Miller timing at all load conditions. PMEP reduction was highest at 100 % load. However, the benefit of PMEP reduction was rather small at low engine-loads. It was identified that this trend was mainly associated with two-stage turbocharger control, especially high-pressure turbine bypass valve operation. Also, NO_x emission was reduced due to the Miller effect's lower combustion temperature. However, the Miller timing induces reduced peak HRR and delayed combustion phase, which will offset the benefit of reduced PMEP.

Thomas [142] investigated how EIVC and LIVC affects pumping losses on a 6.7 L gas engine, equipped with cam-less VVA. The IVC timing was modulated up to ±100 CAD at part load. Pumping work was reduced with increasing and decreasing IVC timing, as illustrated in Fig. 12 (a). The lowest PMEP was observed at the most advanced IVC (IVC-100) and retarded IVC (IVC + 100) timings. PMEP at IVC-100 (EIVC) was 24.8 % lower than at the stock IVC timing. The results showed that IVC modulation was able to enhance fuel economy by 4 % by minimising pumping work. Additionally, it was reported that IVC modulation delays combustion, an observation previously discussed in *Chapter 3.1*.

Similarly, Mahendar et al. [143] explored numerically the benefits of the Miller strategy (EIVC) to enhance the efficiency of an 11.7 L, six-cylinder HD gas engine running on ethanol and methanol. The work revealed that at an excess air ratio (λ) = 1, EIVC reduces PMEP and delivers a 2-3 % gain in BTE. However, at lean conditions (λ = 1.6), a considerable pumping loss was observed when increasing the degree of IVC modulation. High pumping losses are attributed to the need for increased boost pressure to maintain the same trapped air mass as the baseline, resulting in higher backpressure caused by the turbine. Furthermore, the efficiency gain at lean operation was significantly reduced by low in-cylinder turbulence and increased burn duration. However, this can be mitigated by increasing turbocharger efficiency, especially at the earliest Miller timing. This was further verified by Kovacs and Eilts's work [144].

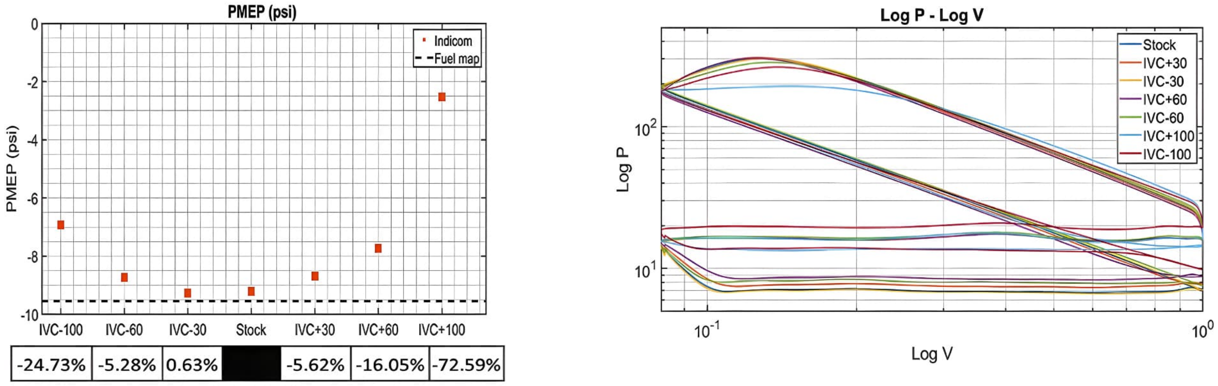
Baratta et al. [145] investigated the impact of EIVC on a 7.8 L, six-cylinder HD gas engine running on compressed natural gas (CNG). It was reported that Miller timing with EIVC reduced pumping work by more than 50 % at part load, contributing to a BSFC reduction of around 6 %. Moreover, adoption of EIVC at full load led to a 35 % increase in engine torque and 20 % more power. This was achieved by reducing in-cylinder temperature and turbine inlet temperature (TIT) with EIVC. The lower in-cylinder temperature brings more margin on knocking; the lower TIT enables further boosting.

Ickes et al. [146] also examined Miller timing via EIVC and LIVC on a 13 L, HD diesel engine operating under gasoline/diesel dual-fuel combustion. They concluded that both strategies reduce mass flow through the engine, thus reducing pumping work and increasing gas exchange efficiency. However, they noted that the reduction in trapped mass leads to a reduction in thermodynamic efficiency.

Various studies have demonstrated that Miller timing can be applied not only in diesel engines but also in gas and dual-fuel engines to minimise pumping losses. Although effective to control PMEP, it leads to reduced VE or in-cylinder trapped mass. This negatively affects combustion, emissions and engine performance, which could offset the benefit of reduced pumping work. Increasing intake pressure with two-stage turbocharging can be an effective solution to resolve this issue. Turbocharger efficiency and control strategy with Miller timing have strong impact on PMEP.

In addition to Miller timing, other innovative VVA methods have also been used for controlling pumping work. Kumar and Zhang [147] experimentally investigated the effect of EEVO, positive valve overlap (PVO) and exhaust reintroduction (2EVO) on gas exchange of a 14.9 L, six-cylinder gasoline compression ignition (GCI) engine. It was found that EEVO had higher pumping losses due to recompression of the trapped exhaust gases, while 2EVO had the lowest due to reverse flow where intake pressure is increased and the gas-exchange loop is reduced. PVO's PMEP was moderately higher than 2EVO's, attributed to the increasingly aggressive throttling of the back-pressure valve for EGR control. Overall, 2EVO had a more efficient gas-exchange process, resulting in lower BSFC.

Mahrous et al. [148] analysed the influence of atypical intake valve timing, EVC timing and IVO timing on the gas exchange performance of an HCCI engine. Atypical intake valve timing entailed actuating the pair of intake valves separately, at different phase angles. The results showed that this approach widened the operating range of the exhaust valve timing and increased the low-load range of the HCCI combustion, while minimising pumping losses. Notably, a reduction of 15 % in pumping power was observed with EVC timing at 40 CAD before TDC (BTDC) and IVO timing at 97 CAD after TDC (ATDC) with non-identical intake valve



(a) (b)

Fig. 12. The effect of IVC modulation on (a) PMEP and (b) cylinder pressure (misfire at IVC + 100), reproduced from [142].

timing.

In summary, VVA strategies, particularly Miller timing through EIVC and LIVC, have shown significant potential in reducing pumping work and improving engine efficiency in HD engine platform. These techniques effectively enhance performance at lower loads, but their application at higher loads is limited by a reduction in in-cylinder trapped mass, which can negatively impact combustion and overall engine efficiency. Moreover, the benefit of PMEP by the Miller timing can be penalized by the turbocharger (size and control). This limitation could be mitigated by turbocharger re-matching or multi-stage charging system.

4.2. *Intake modulation for optimisation of volumetric efficiency (VE) and effective compression ratio (ECR)*

Volumetric efficiency (VE) and effective compression ratio (ECR) are critical factors for gas exchange in internal combustion engines. VE measures the engine's ability to take in and use air efficiently, affecting combustion efficiency and power output. Altering the ECR influences in-cylinder compression, fuel flexibility and emissions. Fixed values for VE and ECR in advanced engine concepts like RCCI and HCCI are insufficient to achieve optimal efficiency while meeting emission standards. Therefore, it is essential to optimise VE and ECR dynamically.

Several studies have highlighted adjustment of both VE and ECR using VVA strategies, and these offer a deeper insight on their practical application. Thomas [142] conducted a comprehensive study on a Cummins 6.7 L natural gas engine, examining the effects of IVC timing on both VE and ECR. The study revealed that setting an early IVC at 30 CAD (IVC-30) gave the lowest PMEP, as depicted in Fig. 12. This outcome was attributed to the simultaneous increase in both VE and ECR, contributing to enhanced fuel efficiency and open-cycle efficiency. Notably, ECR decreased with both LIVC and EIVC, except at IVC-30. The dual impact on both VE and ECR highlighted in this study underscores the key role of precise VVA settings in optimising gas exchange.

Modiyani et al. [149] conducted an in-depth study on a 6.7 L Cummins ISB engine to assess the effects of IVC timing on both VE and ECR. Their research aimed to determine the extent of control over gas exchange and ECR achievable through precise manipulation of IVC timing. The study revealed that IVC timing emerged as the most influential factor on ECR, while VGT and EGR positions had a negligible impact. Maximum ECR was observed at 20 CAD after BDC (ABDC). Interestingly, the study also noted that adjustments in IVC timing had a significant effect on VE. Specifically, VE peaked at the baseline IVC timing. Deviations from this, either early or late IVC timing, resulted in a decline in VE. The interplay between ECR and VE illustrates complex engine dynamics, where changes in ECR affect VE, crucial for combustion and trapped air in the cylinder. The study shows that manipulating IVC timing controls ECR and VE, but careful attention is needed to find the optimal balance.

Garcia et al. [100] studied the effects of IVC modulation on both VE and ECR in a 1.8 L, single-cylinder, HD diesel engine. Their findings revealed that the engine's baseline IVC timing, set at 20 CAD ABDC, resulted in the highest VE. Additionally, they observed that EIVC was more detrimental than LIVC to VE. The study also noted that the decrease in ECR associated with the use of Miller valve profiles (EIVC and LIVC) was symmetric around BDC. This study applied two conditions of IVC modulation: one with constant intake pressure; the other with constant lambda, achieved through higher boost pressure. Under constant pressure conditions, they incurred a fuel penalty due to a weakened air-fuel mixture and slower burn rate, although excessively late and early IVC profiles resulted in lower peak cylinder pressure, higher exhaust temperature and lower required EGR. Interestingly, improvements were achieved without compromising fuel efficiency or increasing emissions when implementing the Miller cycle in conjunction with a high-efficiency turbocharger (maintaining constant lambda).

Lu [150] investigated how valvetrain modifications could enhance efficiency in a six-cylinder, direct-injection diesel engine. His study revealed that EIVC by 30 CAD gave a notable 3 % increase in VE through dynamic charging. This enhancement contributed to an improvement of approximately 1.25 % in fuel efficiency, without compromising NO_x emissions. The research also delved into the effects of IVC modulation on thermal efficiency and open-cycle efficiency (OCE). It observed that adjustments in IVC timing, varying from the nominal 565 CAD to the range of 540 CAD to 640 CAD, could mitigate piston-motion-induced compression and reduce ECR. These modifications not only enhanced OCE but also facilitated advanced combustion processes, thereby improving overall thermal efficiency. However, the study issued a cautionary note against excessive IVC modulation, emphasizing its potential to diminish volumetric efficiency and introduce backflow issues, ultimately adversely affecting engine performance.

The above studies addressed VVA's combined impact on both VE and ECR. The following studies focus exclusively on either VE or ECR. These targeted explorations into each parameter offer a deeper understanding of their distinct roles in enhancing engine performance and efficiency.

Vos et al. [151] manipulated IVC timing to enhance diesel engine efficiency under high-speed and high-load conditions. The study illustrated that this approach effectively improves VE, yielding notable fuel economy improvement. Specifically, delaying IVC by 20 CAD at 2200 RPM and 12.7 bar BMEP increased VE by 5 %. Similarly, the delayed IVC led to a 3.1 % increase in VE at 2200 RPM and 7.6 bar BMEP. These VE enhancements translated into fuel savings of 1.2 % and 0.8-1.9 % respectively at the specified operating conditions. The study highlighted the significance of optimising valve timing as a means to achieve greater efficiency without compromising emissions, especially in the context of high-speed, high-load operations.

Exploring alternative VVA strategies for controlling VE, Mahrous et al. [148] examined the effect of atypical intake valve timing (intake valves actuated at different timing) and variation in IVO and EVC on the gas exchange process of a four-valve direct injection HCCI engine. The results revealed that when the IVO timing had the same distance from the TDC as the EVC timing but in the opposite direction, it resulted in the highest VE. In contrast, an early or late IVO timing relative to this point resulted in a penalty to the VE. The study also showed that using atypical intake valve timing could mitigate up to 50 % of this penalty by reducing early and late backflow and heat transfer over the intake valves. Furthermore, it was found also that atypical intake valve strategy can lead to wider operating range of HCCI engines.

Tomoda et al. [152] focused on ECR adjustment by implementing variable valve timing and lift technology to improve thermal efficiency and reduce emissions of a 2.2 L diesel engine. The research revealed that IVC timing is an effective means of controlling ECR. Altering IVC timing relative to the base setting (30 ABDC) within the range of -30 to +20 CAD resulted in corresponding changes in ECR from 13.8 to 12.5, respectively. The study emphasized that, particularly during high-load conditions, lowering ECR through LIVC can significantly improve the balance between NO_x and fuel efficiency by reducing the temperature at the end of compression. However, late IVC timing may lead to backflow and diminished combustion efficiency. Conversely, increasing ECR can reduce HC emissions and enhance cold-start performance.

Ickes et al. [146] also explored the impact of IVC modulation on ECR, highlighting that the effects of EIVC and LIVC on ECR vary, depending on baseline valve timing and the methods used to calculate ECR. Their research underscored that reducing ECR enhances gas exchange efficiency but results in lower trapped mass, consequently increasing combustion temperatures and heat transfer losses. This study illuminates the intricate trade-offs involved in optimising ECR through VVA.

In summary, studies show that controlling ECR and VE through valve adjustments primarily involves EIVC and LIVC. Regarding ECR, EIVC reduces compression, benefitting NO_x emissions reduction and fuel economy under high-load conditions, but can decrease power output. LIVC increases ECR, enhancing power and efficiency at low to medium loads, but raises the risk of engine knocking and higher NO_x emissions.

Turning to volumetric efficiency (VE), optimal intake valve timing is crucial. EIVC generally reduces VE by shortening the intake period, while a moderate amount of LIVC can increase VE by allowing more air intake. Deviations from optimal IVC timing in either direction can lower VE and compromise combustion efficiency, but are sometimes necessary to control combustion phasing. VVA provides the flexibility to fine-tune both VE and ECR, balancing power output, fuel efficiency and emissions management across different operational conditions.

4.3. EVO modulation for improved low-end torque via pressure pulse effect

The exhaust pressure pulsation effect is a phenomenon in multi-cylinder engines whereby the opening of an exhaust valve in one-cylinder influences gas exchange in other cylinders. This occurs when high-pressure exhaust gases are expelled into the shared exhaust manifold, creating pressure waves which travel through the system and interact with neighbouring cylinders. Such pressure pulsations can disrupt the scavenging process, affect the air-fuel mixture and increase pumping work. Consequently, it is imperative to comprehend and regulate exhaust pressure pulsations. VVA can serve as an effective control measure, but scientific and technical reports aiming to verify this thesis are rather scarce.

Kim et al. [153], investigated the correlation between pressure pulsation and turbine upstream geometry in the context of a 1 L, three-cylinder, boosted gasoline engine with significant exhaust back pressure and relatively low turbocharger efficiency. Their study demonstrated that reducing the exhaust manifold volume also reduced pumping loss, improving fuel consumption by up to -2.4 %. Furthermore, this adjustment effectively preserved high-pressure pulsation energy, thereby improving turbine power. Ultimately, this contributed to 15 % more low-end torque.

Most of the knowledge on the impact of VVA on pressure pulsation in diesel engines comes from Dahodwala et al. [154] and Tomoda et al. [152]. The first study explored various VVA techniques to downsize heavy-duty diesel engines while maintaining performance. It focused on regulating pressure pulsation through valve timing adjustments. The research showed that manipulating exhaust valve timings can significantly increase exhaust pressure and turbine power. Optimised valve timings in a four-cylinder engine at 800 RPM were found to reduce pumping losses and improve volumetric efficiency by managing pressure conditions during valve overlap. This optimisation entailed retarded exhaust valve timing and advanced intake valve timing. This enhances scavenging, which is not achievable with standard timings, due to high exhaust and in-cylinder pressures.

Tomoda et al. [152] examined variable valve timing (VVT) systems to enhance low-end torque in diesel engines by leveraging pressure pulsation. Specifically, they retarded EVO timing, creating a negative peak of exhaust pressure pulsation near TDC, followed by an intake-exhaust valve overlap. This induced a scavenging effect, leading to a substantial increase in the introduced air mass. The study found that pressure pulsation control, facilitated by a shutter valve, gave a 40 % uplift in torque at 1400 RPM, and up to 24 % more without the shutter valve.

Encouraged by these promising results in heavy-duty, high-speed diesel engines, the authors subsequently investigated the impact of exhaust pressure pulsations in a six-cylinder, medium-speed marine diesel engine, focusing on the pulsations' influence on gas exchange and pumping losses. Fig. 13 visualises exhaust pressure pulsation at exhaust runner at cylinder 1, together with an exhaust valve profile and cumulative exhaust mass flow rate via exhaust valves. The pressure pulsation by other cylinders is obvious after closure of exhaust valves (EVs). It is noteworthy that the first three key pulses directly impact gas exchange during the opening of EVs: one from the opening of exhaust valve in the same cylinder, a second pulse which seems a reflection of the first pulse, and a third from the subsequent cylinder's valve opening. Since there is reverse flow at the peak of the second pulse, some modifications were applied to the exhaust pipe's geometry. These impacted the middle pulse but did not significantly remove the reverse flow and did not enhance overall engine performance.

The authors also explored delaying exhaust valve opening. This altered the pulses slightly but did not improve engine performance. They concluded that EVO-modulated pressure pulsation strategies prove ineffective for state-of-the art, four-stroke marine engines, due to positive pressure gradient ($P_{\text{intake}} > P_{\text{exhaust}}$) evoked by highly efficient turbocharging. However, the mechanism can be deployed to reduce pumping work and improve low-end torque in high-speed engines with positive backpressure across the engine, as used in the off-road sector.

4.4. IVO and IVC modulation to control in-cylinder flow motion – Swirl and turbulence

Swirl is the rotational motion of air within the combustion chamber when the fresh charge is drawn. It plays a significant role in gas exchange, particularly in CI engines. First, swirl helps the even distribution of air and fuel, thus creating a more homogenous mixture. A well-distributed, homogenous mixture promotes more efficient, complete and stable combustion. This helps reduce harmful emissions like UHC

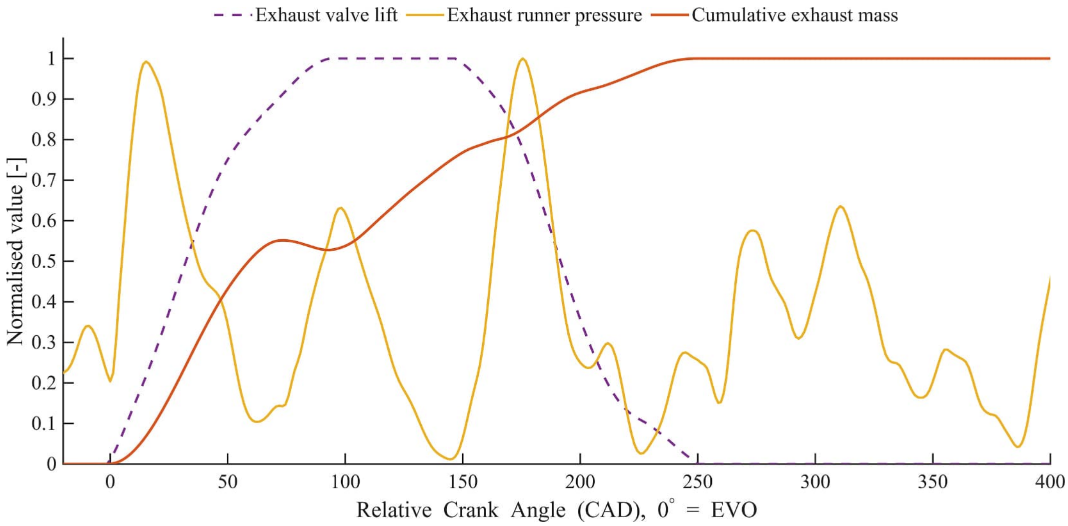


Fig. 13. Visualisation of pressure pulsation and their effect on cumulative exhaust flow rate.

and PM, which are formed by fuel-rich pockets. Second, swirl induces in-cylinder turbulence which enhances interaction between fuel droplets and air and then promotes better evaporation. Swirl also influences residual gas fraction (RGF) and in-cylinder oxygen concentration. Consequently, most CI engines have a swirl-oriented cylinder head or swirl-induced intake port or a swirl flap in the intake port. These geometries are not flexible, but swirl motion can be further controlled by VVA.

Tomoda et al. [152] studied the effects of variable valve timing and lift on a light-duty diesel engine's performance. They explored two approaches to controlling the swirl ratio at low- and medium-load conditions, as shown in Fig. 15. Their first approach involved adjusting IVC timing of one valve relative to the other to control swirl strength at low loads, which reduces smoke emissions and improves fuel consumption. Their second approach involved advancing the intake valve opening (IVO) timing before TDC to enhance swirl strength and increase introduced air mass at medium loads. This aims at a better trade-off between fuel consumption and emissions. They found that the early intake valve opening (EIVO) is more suitable for high load, because it improves VE by increasing the flow coefficient. However, EIVC (for one valve) is more suitable for low-load levels because it strengthens the swirl ratio, which is more important than the flow coefficient at low loads.

Peng et al. [155] used 3D-CFD simulation to investigate the effect of NVO on gas exchange of a small diesel engine model operating diesel HCCI. This revealed that extending NVO duration gave increased swirl motion and a more pronounced vortex formation beneath the intake valve. This phenomenon was attributed to the intensified annular jet flow through the valve curtain area during the intake stroke. Specifically, a pronounced tumble motion was generated during the early induction stage when NVO was 60 CAD. Nevertheless, there was no significant variance in the values of swirl ratio and turbulence intensity at the end of the compression stroke across different NVO settings.

Cengiz [156] conducted an in-depth investigation focused solely on the correlation between EIVC and swirl. This entailed a 3D-CFD simulation of a 2.1 L, HD diesel engine, running in premixed charge compression ignition (PCCI) combustion mode. Fig. 14 illustrates EIVC's substantial impact on swirl ratio. Specifically, EIVC gave a 50 % reduction in both charged mass and swirl ratio compared with the baseline configuration. This diminished swirl adversely affected combustion by impeding the penetration of the main injection into the zone of fresh air amid the combustion products from the pilot injection. This led to incomplete combustion, contributing to high unburned carbon emissions. However, the study suggests that a well-calibrated, early split-injection and substantial EGR could potentially alleviate the negative effects of EIVC, such as the reduction in charge mass and swirl.

Hong et al. [67] stated that late IVO (LIVO) induces turbulence for incoming air-fuel mixture, which promotes homogenous combustion. LIVO creates suction pressure below intake manifold pressure: a large pressure difference results in high velocity upon opening of intake valves, which creates turbulence. However, a deep vacuum pressure results in high pumping losses. Mikulski et al. [35] used multizone combustion modelling to implement LIVO on a HD diesel engine operating under natural gas-diesel RCCI mode at low load (3 bar IMEP). This showed that LIVO induces high intake velocity due to low pressure in the cylinder. This promotes high turbulence during intake stroke, but it completely quenches before combustion starts. Increased mixing energy associated with LIVO was found to have negligible influence on simulation results.

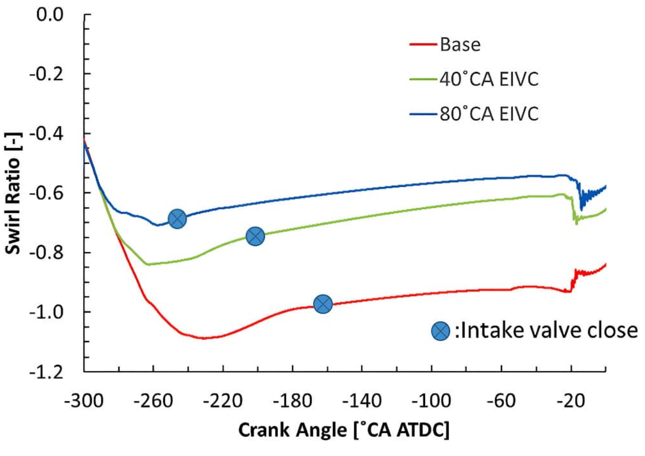


Fig. 14. In-cylinder swirl generation time histories of baseline and EIVC cases, reproduced from [156].

Conversely, studies have shown that Miller timing (EIVC) could reduce in-cylinder turbulence, especially in SI engines. Reduced turbulence slows flame speed, which extends end gas residence time and increases knock propensity [157,158]. Mahendar et al. [143] reported that EIVC reduces turbulence kinetic energy (TKE) and increases combustion duration with retarded CA50. This offset the potential thermodynamic efficiency gain of EIVC at lean operation ($\lambda = 1.6$) of a HD SI engine operating with ethanol and methanol and retrofitted with diesel cylinder head and swirl intake ports. Pertaining to Miller timing, several studies indicate that LIVC has less of a turbulence penalty than EIVC [159,160]. Thus, LIVC performed better at a knock-limited operating condition [161].

In summary, IVC and IVO modulation have emerged as crucial factors influencing the motion of the in-cylinder charge, specifically swirl and turbulence. Increased in-cylinder dynamic motion enhances charge introduction, fostering fast and uniform combustion. Conversely, reduced swirl and turbulence has shown detrimental effect on combustion and emissions. The negative effect can be alleviated through adjustment of injection timing and EGR. It is important to note that implementing this strategy requires careful consideration to balance benefits against potential drawbacks, such as increased pumping losses or knock propensity.

4.5. *Comparison of all VVA strategies enabling efficient gas-exchange*

Tomoda et al. [152] conducted a comprehensive examination of VVA's ability to enhance gas exchange over the entire operating map of a diesel engine. Fig. 15 compares the different VVA approaches and their effectiveness on the gas exchange process. Although this research was conducted on an automotive engine, the findings illustrate VVA's adaptability and its ability to enable the engine to respond efficiently to varying operating conditions. For instance, early IVC and late IVC modulations are shown to control the effective compression ratio and improve scavenging, respectively, at low-medium loads. Early IVO

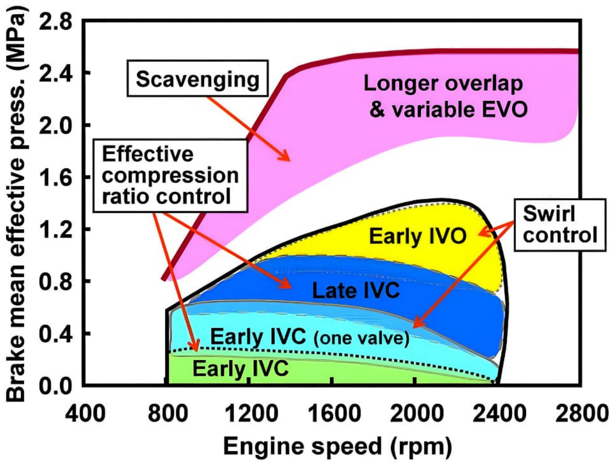


Fig. 15. Implementation of various VVA strategies for better gas exchange over the entire engine map, reproduced from [152].

enhances VE and swirl control at medium loads.

This chapter reviewed various studies of VVA's role in regulating gas exchange. Table 5 summarises all these studies, listing the VVA approach, the target parameters and the outcomes. This review shows that among the various VVA approaches, IVC modulation seems to have the most significant effect on gas exchange. Focusing on VE and ECR, there is an optimal IVC timing for each that leads to the highest VE and ECR values, while both late and early IVC timings reduce these values. EIVC leads to lower pumping losses and an increase in power, but excessive EIVC could suppress in-cylinder turbulence, potentially impacting combustion negatively. Similarly, moderate LIVC can reduce the pumping work but may lead to increased knock propensity. Both strategies have trade-offs that must be carefully managed to optimise engine performance.

Other than IVC, IVO modulation is also used widely. LIVO leads to increased turbulence during the intake stroke but can result in higher pumping losses. EIVO improves VE by enhancing the flow coefficient, making it more suitable for high-load conditions. Turning to exhaust valve modulation, strategies such as LEVO are applied to control pressure pulsation and enhance low-end torque through scavenging effects. 2EVO shows low pumping losses, whereas EEVO shows high pumping losses. Other than normal VVA approaches, some innovative methods like atypical intake valve timing and NVO have been shown to affect gas exchange, such as reducing pumping losses, enhancing volumetric efficiency and enhancing swirl motion.

Fig. 16 sets out the VVA control mechanisms for the various elements of gas exchange. The findings in this chapter indicate that VVA has significant potential to provide the flexibility required by modern engines to deliver optimum performance across a wide range of operating conditions. Even though most reviewed studies have been conducted under steady-state conditions, it is important to assess transient performance. Unlike large-bore marine engines, non-road HD engines are frequently subjected to substantial transient load during both official emission testing and real-world operation. While VVA strategies such as intake modulation reduce intake air mass flow rate, effective boost control using VGT and wastegate can help meeting the air flow demands during transient events.

5. VVA to control low-temperature combustion

Conventional diesel engines have their inherent issues of high combustion temperatures and mixing-controlled combustion within fuel-rich pockets which result in a large amount of NO_x and soot emissions. A diesel engine's well-known NO_x-soot trade-off means that it is challenging to reduce both emissions through in-cylinder emission control strategies. Driven by the need to comply with stringent emission regulations, IEATS such as SCR and DPF have demonstrated effective reduction, but their wider application is hindered by factors such as high cost, durability, fuel consumption penalty and substantial space requirement [71,162,163]. For this reason, low-temperature combustion (LTC) strategies have been emerging as a viable solution to mitigate them simultaneously.

LTC strategies such as homogeneous charge compression ignition (HCCI) achieve both ultra-low NO_x and soot emissions by locally low combustion temperature and enhanced air-fuel mixing with premixed lean combustion [164]. Despite these significant advantages, real-world implementation faces several challenges related to narrow operating load range, combustion stability, difficult transients and poor ETM [72,73]. Low load operation is limited by misfire and unstable combustion with high HC and CO emission. In contrast, high load operation is constrained by rapid pressure rise rate excessive combustion noise and knocking.

These challenges mainly arise from combustion control [72]. HCCI's lack of an immediate combustion trigger, in the form of spark discharge or self-igniting diesel spray, means combustion onset is determined by in-cylinder charge temperature at IVC and cylinder thermal stratification. Reactivity-controlled compression ignition (RCCI), a dual-fuel realisation of LTC, partially resolves the controllability problem by adjusting the reactivity with blend ratio or pilot injection timing across the operating map. Nevertheless, RCCI retains many of the same operational challenges.

To ensure safe and smooth LTC operation, various experimental studies impose explicit boundaries on key combustion parameters. A maximum pressure rise rate(MPRR) limit below 10 bar/CAD [165–168] and a ringing intensity limit below 5 MW/m^2 [169–171] are typically used to supress knocking, combustion noise, and excessive mechanical stress. Stable combustion is further ensured by constraining the coefficient of variation (COV) of IMEP to 4-5 % [165,168,172].

Because LTC lacks direct combustion control mechanism, ignition timing and combustion phase are strongly influenced by multiple factors such as temperature and pressure of intake air, fuel properties, blend ratio, injection strategy & timing, thermal & reactivity stratification, concentration of reacting species, residuafl gas fraction (EGR), equivalence ratio, engine geometry and operating parameters [72]. These sensitivities of combustion mean VVA's effect is much more pronounced in LTC than in conventional engines, and thus is widely considered as an enabler for stable and ultra-efficient RCCI. This chapter reviews three control concepts supported by VVA: variable compression ratio, internal EGR and in-cylinder fuel reforming.

5.1. Intake modulation for variable compression ratio

Variable compression ratio (VCR) can be used to extend high-load range, to mitigate HC and CO emission and to control combustion phasing in LTC because CR influences thermodynamic status of the in-cylinder charge. In general, high CR is ideal to maximise thermal efficiency in ICEs. Even though it is limited by a high potential for knocking in SI engines, high CR promotes auto-ignition and chemical reactions in LTC. Aceves et al. [173] performed a numerical simulation of CR's effect in an HCCI engine. The results indicated that the optimal operation region with high thermal efficiency varied for each CR. This suggests that the CR can be tailored to suit the operational conditions and to extend an HCCI engine's operational limit. VVA can achieve that by varying the effective/apparent compression, instead of using complex and slow variable-geometry CR systems.

A numerical study by Kulkarni et al. [174] showed long ignition delay while reducing ECR by EIVC. The simulation was based on a 6.7 L, HD diesel engine operating under PCCI mode at medium-load condition. Lower in-cylinder pressure and temperature by EIVC (low ECR) induces long ignition delay, which promotes premixed combustion. However, CA50 is retarded due to delayed combustion initiation. As a result, pressure rise rate (PRR), peak cylinder pressure and peak burned zone temperature are lower with decreased ECR. This is beneficial for NO_x reduction. However, excessive EIVC risks misfire where the fuel-air equivalence ratio is above the stoichiometric value, since in-cylinder oxygen is insufficient to burn all the fuel. This limits the reduction of ECR via EIVC. The same behaviour was observed in excessive LIVC timing [175].

To resolve this issue, Nevin et al. [175] increased boost pressure, while retarding IVC to maintain the same air-fuel ratio. The engine testing was conducted on a 2.4 L, single-cylinder HD engine. Combustion was delayed with constant AFR, due to low combustion temperature invoked by low ECR. NO_x decreased by nearly 90 % at excessive LIVC by the Miller effect. Low soot was maintained up to moderate LIVC. However, excessive LIVC increased soot rapidly because low flame temperature hampered soot oxidation. CO and HC increased due to the low combustion temperature caused by LIVC.

Cengiz et al. [156] performed a 3D-CFD simulation of EIVC at part load (3 bar BMEP) to control start of ignition (SOI) and CA50 under PCCI mode on a 2.1 L, single-cylinder HD diesel engine. EIVC delayed SOC and CA50 by 5CAD and 4CAD respectively, due to the low cylinder pressure and temperature conditions created by EIVC. Even though EIVC

Table 5
Summary of literature reviews on gas exchange process.

Refs.	Type of engine	VVA strategy	VVA modulation	Gas exchange aspect	Main effect	Penalty
[141]	HD marine diesel engine 7.2 L, six-cylinder two-stage TC	EIVC	IVC: 0 to -60 CAD	PMEP	Reduced PMEP, NOx,	Reduced peak HRR, delayed combustion
[142]	HD gas engine 6.7 L, six-cylinder HP-EGR, TC	EIVC, LIVC	IVC: -100 to + 100 CAD		Reduced PMEP	Reduced VE
[143]	HD diesel engine (ethanol/methanol) 11.7 L, six-cylinder TC	EIVC	IVC: 0 to -90 CAD		Reduced PMEP ($\lambda = 1$) Increased PMEP ($\lambda = 1.6$)	
[144]	HD diesel engine 9.5 L, six-cylinder two-stage TC	EIVC	IVC: 0 to -60 CAD		Reduced PMEP by increasing TC efficiency	
[145]	HD gas engine 7.8 L, six-cylinder TC	EIVC			Part load: reduce PMEP and BSFC Full load: increase torque and power	
[146]	HD dual-fuel engine (gasoline/diesel) 13 L, six-cylinder HP-EGR, two-stage TC	EIVC, LIVC			Reduced PMEP and increased gas exchange efficiency	Reduced thermodynamic efficiency
[147]	HD GCI* engine 14.9 L, six-cylinder HP-EGR, VGT	EEVO, PVO, 2EVO	EVO: 0 to -40 CAD PVO: EVC -30 CAD and IVO +15 CAD		Pumping work comparison: EEVO > PVO > 2EVO	
[148]	HCCI engine 0.5 L, single-cylinder	IVO, EVC, atypical intake valve timing	IVO: 0 to 110 CAD EVC: 30 to 100 CAD		Atypical intake timing: Reduced PMEP and low-load extension	Atypical intake timing + EVO + IVO: power loss
[142]	HD gas engine 6.7 L, six-cylinder HP-EGR, TC	EIVC, LIVC	IVC: -100 to + 100 CAD	VE / ECR	Increased VE and ECR: lower PMEP	
[149]	HD diesel engine 6.7 L, six-cylinder HP-EGR, VGT	EIVC, LIVC	IVC: -60 to 80 CAD	VE / ECR	IVC: crucial factor to determine ECR and VE	Trade-off: ECR and VE
[100]	HD diesel engine, 1.8 L, one-cylinder, HP-EGR	EIVC, LIVC	IVC:0 to -60 CAD IVC:0 to +80 CAD	VE / ECR	EIVC is more detrimental than LIVC to VE	Fuel penalty by slow combustion
[150]	HD diesel engine Six-cylinder, VGT	LIVC	IVC: 0 to +30 CAD	VE / ECR	LIVC30: +3 % VE	Excessive IVC modulation: backflow, lower VE/performance
[151]	HD diesel engine Six-cylinder, HP-EGR, VGT	EIVC, LIVC	IVC: -15 to +40 CAD	VE	IVC + 20: +3–5 % VE, up to -2 % fuel saving	
[148]	GDI engine (HCCI) 0.56 L, one-cylinder	IVO, EVC, atypical intake valve timing	IVO: 0 to +110 CAD EVC: 30 to +100 CAD		Atypical intake valve timing: mitigate VE drop by excessive IVO modulation	Too early or late IVO: VE
[152]	LD diesel engine 2.2 L, four-cylinder	EIVC, LIVC	IVC: -30 to +20 CAD	ECR	Low ECR (LIVC): improve NO_x and BSFC High ECR: improve HC and cold-start performance	Backflow and combustion efficiency at high loads
[146]	HD dual-fuel engine (gasoline/diesel) 13 L, six-cylinder two-stage TC (HP VGT)	EIVC, LIVC	EIVC: 240-225 BTDC LIVC: 130-110 BTDC	ECR	Low ECR improves gas exchange efficiency	Low trapped mass, increased combustion temperature, high heat losses
[154]	HD diesel engine 5 L, four-cylinder TC	Optimised valve timing	EV**: retarded IV***: advanced	Pressure pulsation	Optimised VVA: reduce PMEP and VE drop by improving pressure pulsation	
[152]	LD diesel engine 2.2 L, four-cylinder	LEVO			LEVO: reduce peak of pressure pulse ➔ improve scavenging, VE, low-end torque	
[152]	LD diesel engine 2.2 L, four-cylinder	EIVC (only one valve)	IVC: 0 to -73 CAD	Swirl ratio	Increased swirl at low load	
[155]	LD diesel engine 0.5 L/cylinder	NVO	NVO: 0 to 180 CAD		Large NVO increased swirl	
[148]	HD diesel engine (PCCI), EGR 2.1 L, one-cylinder	EIVC	IVC: 0 TO -80 CAD		EIVC: reduce swirl and fuel penetration ➔ incomplete combustion and high HC	
[67]	SI engine one-cylinder	LIVO	IVO: +50 CAD	Turbulence	LIVO: high swirl due to large pressure gradient	
[35]	HD diesel engine, Natural-gas/diesel (RCCI) six-cylinder	LIVO	IVO: 0 to +160 CAD		LIVO: improve mixing energy (fast intake velocity and high turbulence)	

(*continued on next page*)

Table 5 (*continued*)

Refs.	Type of engine	VVA strategy	VVA modulation	Gas exchange aspect	Main effect	Penalty
[143]	HD diesel engine (ethanol/methanol) 11.7 L, six-cylinder, TC	EIVC	IVC: 0 to -60 CAD		EIVC: reduce turbulence and delay combustion	

GCI*: Gasoline compression ignition, EV**: Exhaust valves, IV***: Intake valves.

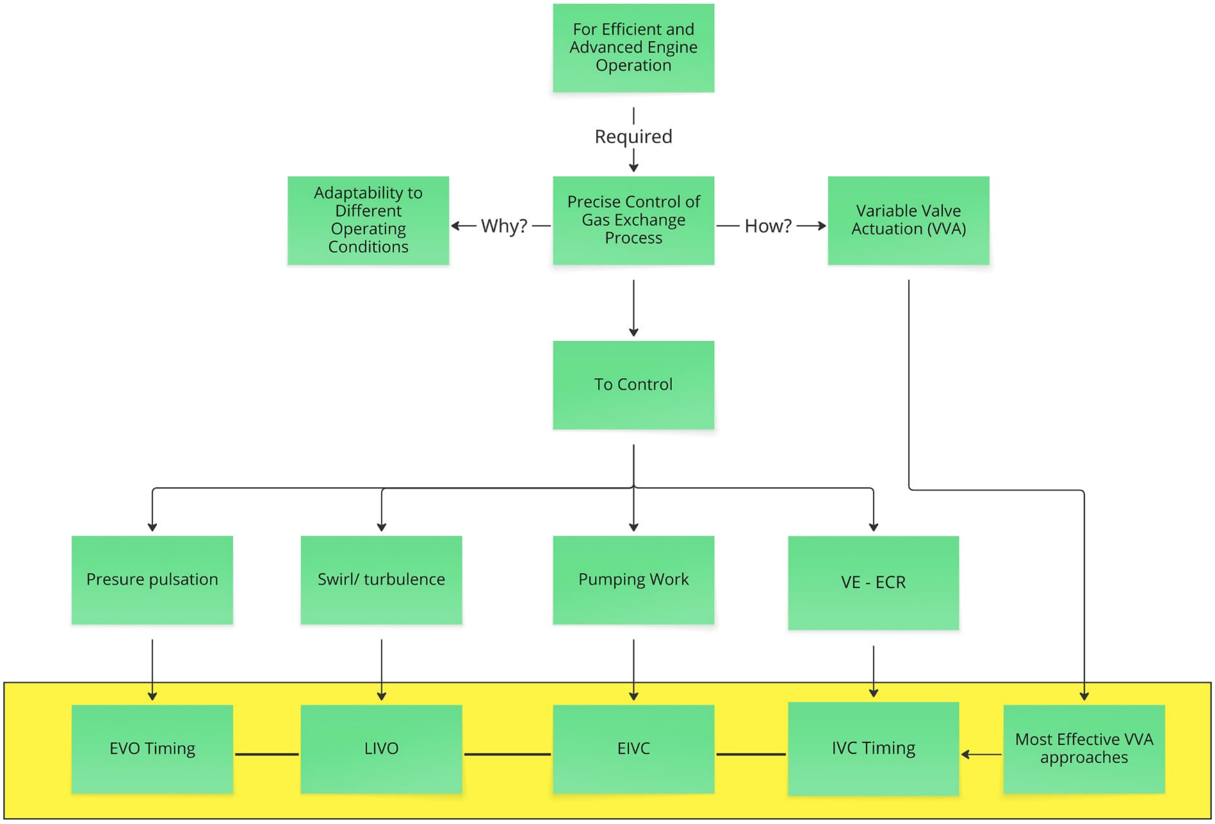


Fig. 16. Control mechanism of efficient gas exchange with VVA.

reduced peak cylinder pressure, peak HRR was increased by enhanced premixed combustion with extended ignition delay. Furthermore, EIVC led to slow evaporation and reduced the swirl ratio by up to 50 %, causing poor air-fuel mixing. The lower oxygen content due to EIVC and the less entrainment by the reduced swirl meant combustion was 4 % less complete. This increased CO emission. In addition, soot increased due to the low-pressure condition leading to poor atomisation, creating a fuel-rich zone. Nevertheless, simultaneous reduction of NO_x and soot is feasible [175–177].

Xu et al. [177] numerically investigated start of injection (SOI), EGR rate, pressure and temperature with EIVC and LIVC. The 3D-CFD simulation was performed on a six-cylinder HD diesel engine operating under PCCI mode at low- load condition (5 bar IMEP). The numerical optimisation showed the optimal solutions varied with different IVC strategies. EIVC is more suitable with high intake pressure and high EGR rate of up to 70 %, while LIVC is beneficial with low intake pressure and moderate EGR rate (~40 %). This indicates that the optimal IVC timing varies and is influenced by other parameters.

Molina et al. [178] studied LIVC and injection strategy (injection timing and multiple injection) in both 3D-CFD simulation and experiment using a single-cylinder HD diesel engine under gasoline/diesel RCCI mode. At high load, LIVC contributed to achievement of 100 % load (≈24 bar BMEP) with a single diesel injection and reducing premixed gasoline ratio. At mid load, LIVC led to a long ignition delay of around 10 CAD by reducing ECR from 14:1 to 11:1, which shows potential to control combustion phase. García-Valladolid et al. [179] observed similar trend in a medium- speed marine diesel engine operating under natural-gas/diesel RCCI mode.

Benajes et al. [172] also experimentally compared two ECRs (14:1 and 11:1) implemented by EIVC on a single-cylinder HD diesel engine under gasoline/diesel RCCI mode. Low ECR (11:1) achieved significant load extension (≈22 bar BMEP) and also suppressed excessive pressure rise rate (PRR) with increasing load. Xu et al. [180] optimised gasoline/diesel RCCI combustion with EIVC and LIVC across a wide load range. The study used 1D simulation (gas exchange) coupled with 3D simulation (combustion) of an HD diesel engine. EIVC with high premixed ratio (PR) showed better thermal efficiency and lower soot. LIVC with lower PR indicated better ringing intensity (RI).

Pedrozo et al. [181] performed experimental research with LIVC on a 2 L, single-cylinder HD engine operating in natural-gas/diesel RCCI mode. Methane slip and NO_x emissions were up to 80 % lower with RCCI combustion and LIVC than in the conventional dual-fuel mode. Thermal efficiency was also high. It was noted out that the high reduction of methane slip was achieved by the high conversion efficiency in the IEATS, because LIVC increased EGT up to +75 °C.

Mikulski et al. [35] investigated EIVC for high-load extension. This was a numerical study of a natural-gas/diesel RCCI engine. Boost pressure was increased with advancing IVC to provide sufficient air for high-load operation. Fig. 17 illustrates the comprehensive simulation results, including combustion, performance and emission parameters. With fixed CR of 14:1 (black circles), 21 bar IMEP was not attainable due to excessive peak cylinder pressure (P_{max}). However, EIVC (cases 4-6, red crosses), gave a lower charge temperature (T_{IVC}). This induced a long ignition delay, leading to slower combustion and lower PPR. This enabled the engine to be pushed to 23 bar IMEP, while maintaining the P_{max} limit of 190 bar. However, combustion efficiency fell by four percentage points, due to the degraded combustion process. This led to high methane emissions, but this penalty can be mitigated with increasing boost demand.

Xu et al. [182] compared and evaluated two different VCR strategies: ECR control via VVT, and control of geometric compression ratio (GCR). High CR was employed at low-load operation for both cases, to accelerate the combustion for high thermal efficiency. In the low-load optimisation cases, GCR control uses a high EGR rate and low intake pressure, whereas VVT control applies relatively lower EGR rate and high intake pressure. At high-load operation, both require low CR to control PPRR and peak cylinder pressure. GCR control is more beneficial than ECR control via VVT (LIVC) for high-load operation because VVT led to decreased volumetric efficiency and misfire.

5.2. Internal EGR (NVO and reintroduction) to improve combustion efficiency

EGR is an established method of mitigating NO_x emissions. However, EGR also can be used to control combustion in LTC applications. LTC is driven by combustion kinetics, and its lack of combustion control is one of the biggest challenges. Spontaneous ignition and combustion are strongly influenced by the thermodynamic status (pressure and temperature) of the in-cylinder mixture and mixture composition. EGR can change the in-cylinder charge condition by trapping hot exhaust gases. Increasing in-cylinder temperature by raising the EGR rate advances ignition delay and start of combustion (SOC) [73].

Additionally, EGR has a dilution effect which reduces oxygen concentration. EGR also affects heat capacity due to CO_2 and water vapour, which influence combustion temperature, HRR, PRR and NO_x [73,183,184]. Lastly, EGR exerts a chemical impact whereby unburnt combustion by-products, including HC, CO, CO_2, NO, H_2O, etc., actively participate in chemical reactions that moderately influence the rate of the main combustion reaction [73]. Therefore, introducing hot recirculated exhaust gas is an effective way to advance combustion onset and improve combustion efficiency in LTC, especially at low load. The present study discusses only internal EGR (i-EGR) which can be realised by VVA strategies such as NVO and exhaust reintroduction.

5.2.1. NVO

Chapter 3.3 already discussed how NVO can be used for EGR. Shi et al. [185] experimentally evaluated internal and external EGR on a 2.1 L, naturally aspirated, single-cylinder diesel engine, running diesel HCCI. I-EGR was realised by NVO. Increasing NVO advanced SOC, with short combustion duration and higher peak HRR. Hot residual gases increased intake charge temperature, which accelerated the combustion rate of HCCI. So, NVO is beneficial to achieve auto-ignition temperature of high-octane fuel like gasoline, but with low CR (typically spark-ignited engines) due to low self-ignition propensity [73,186]. In addition, large NVO gave an increasing internal EGR effect and accelerated vaporisation due to the high in-cylinder temperature. This phenomenon was demonstrated in a 3D-CFD simulation [187]. It enables formation of a homogeneous mixture, which reduces smoke emission. Large NVO increased NO_x, but it was reported that it is more sensitive to engine load [185]. External EGR delayed SOC because cooled EGR reduces the charge temperature. This is beneficial for high-cetane fuel like diesel, with a high self-ignition propensity.

Borgqvist et al. [188] examined NVO on a small diesel engine operating under gasoline partially premixed combustion (PPC) mode. The main aim was to elevate in-cylinder temperature by trapping hot EGR to promote auto-ignition of high-octane fuel (gasoline) for low-load extension. Increasing NVO successfully raised temperature at IVC and advanced combustion phase with short ignition delay. Advanced combustion increased peak HRR, improving combustion efficiency. Ultimately, large NVO extended the low-load limit from 3.2 bar (IMEP) to 2.2 bar (IMEP).

However, the opposite trend was observed at the most extreme NVO case, giving long ignition delay, prolonged combustion duration and low peak HRR. Excessive dilution in the mixture is likely to reduce reactivity, thereby decelerating the auto-ignition chemistry. Moreover, application of NVO is constrained by admission of large amounts of residual exhaust gases which diminish air [62]. For this reason, NVO has been demonstrated mostly at low engine-loads. Despite potential alleviation via use of a boosting system, such as turbocharging or supercharging, NVO at high loads is still limited by an excessive PRR [189].

Even though NVO is effective to increase in-cylinder charge temperature and control combustion phase, some studies have shown that the recompression during NVO generates additional thermal losses and so reduces overall engine efficiency [190]. In contrast, Rodriguez and Cheng [191] reported that optimised NVO can increase fuel efficiency by 7 % at part loads compared to positive valve overlap operation in a gasoline engine. Despite these conflicting results, this issue can be resolved by re-opening intake or exhaust valves (reintroduction or re-breathing) [192]. This agrees with the findings from other literature studies in *Chapter 3.3.*

5.2.2. Reintroduction or re-breathing (2IVO and 2EVO)

Another way to achieve i-EGR is reintroduction or re-breathing by 2EVO, or a second intake valve opening (2IVO), as discussed in *Chapter 3.3.* Mikulski et al. [35] modelled 2EVO at low engine-load (3 bar IMEP). The numerical simulation was carried out on an HD diesel engine operating under natural-gas/diesel RCCI mode. I-EGR or residual gas fraction (RGF) was controlled by an intake opening and closing valve

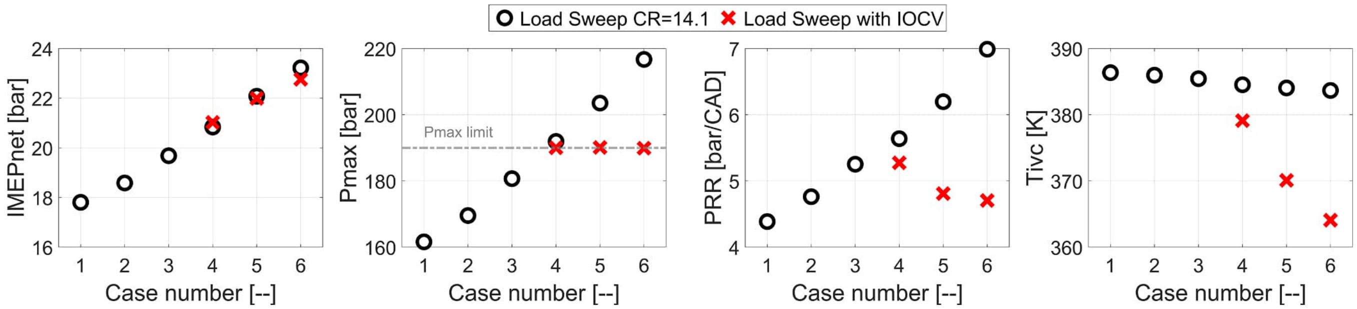


Fig. 17. Simulation results of load sweep with fixed CR = 14:1 (black circles); with reduced CR via LIVC (red crosses), adopted from [35]. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

(IOCV) at each cylinder's intake port. Late IOCV opening blocked fresh air during the intake stroke, allowing the introduction of more burned gases, thus increasing the i-EGR rate. A large amount of hot i-EGR elevated the in-cylinder charge temperature (T_{IVC}) and then advanced ignition and combustion (CA50), as shown in Fig. 18. Furthermore, the improved combustion mitigated engine-out CH_4 emissions by more than 98 %. Simultaneously, EGT was elevated by 100 °C which allowed efficient operation of a methane catalyst at low load.

Pedrozo et al. [193] conducted experimental and numerical research on re-breathing via 2IVO for low-load extension (3.2 bar IMEP). The engine testing was conducted on a 2 L, single-cylinder HD diesel engine operating in ethanol/diesel RCCI combustion mode. RGF (i-EGR) was controlled by increasing back pressure with constant re-breathing valve profile. This mechanism induced a high pumping loss with high RGF, which could penalise engine efficiency. High i-EGR induced large RGF and a high global equivalence ratio. The increased mean cylinder temperature accelerated evaporation and combustion. This reduced cycle-to-cycle variation at low load, thus enabling stable combustion and low-load extension.

Additionally, the higher cylinder temperature improved oxidation of unburned HC and CO, leading to a remarkable reduction of over 65 %. This enhanced combustion efficiency from 87.7 % to about 96 %. However, further reduction of HC and CO was constrained by the fact that the increased RGF lowered the oxygen concentration. Diesel injection timing was advanced in order to avoid too much advanced combustion phase. The revised timing resulted in long ignition delay and maintained SOC and CA10. Soot was reduced by the long ignition delay and better mixture mixing, and NO_x was cut by dilution and the high heat-capacity effect.

Borgqvist et al. [194] compared the effect of 2EVO with NVO on a gasoline PPC engine at low load. Both had a rather similar effect on combustion, but 2EVO increased combustion stability. Moreover, 2EVO showed better gas exchange efficiency since NVO suffered from high heat loss during the recompression stage. This gave 2EVO higher indicated efficiency than NVO. High unburned hydrocarbon was observed at near-idling operation, regardless of valve timings. This was caused by the long ignition delay due to rather low charge temperature, even with NVO, at lightly loaded points. The delayed combustion reduces combustion temperature and time for oxidation, thus necessitating a three-way catalytic (TWC) converter. But the low EGT ($<$200 °C) is challenging for catalyst operation, so ETM is required for effective reduction.

Nakamura et al. [195] studied exhaust re-breathing via 2EVO, combined with external EGR, to mitigate PRR by retarding combustion phase for high-load extension. Testing was performed with a small HCCI engine with dimethyl ether (DME). Cooled, external EGR was the main method of delaying the combustion phase, while hot EGR by 2EVO amplified the benefit of thermal stratification at IVC. The results indicated a significant retardation in the combustion phase, which prolonged combustion duration and lowered the maximum PRR. This mitigated the knocking tendency, benefitting high-load operation. However, too much delayed combustion threatened unstable combustion, such as misfire or partial burn.

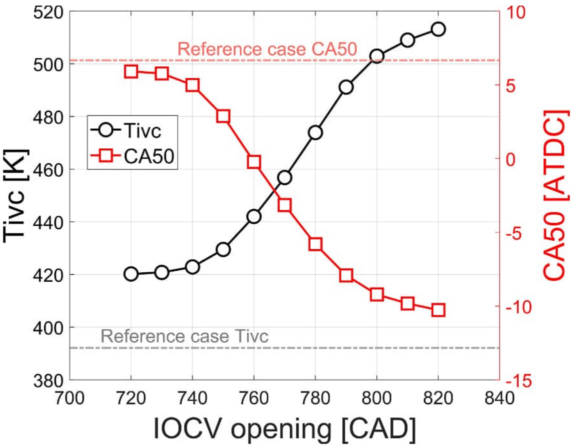


Fig. 18. Simulation results of 2EVO in an RCCI engine: in-cylinder temperature at IVC and CA50, adopted from [35].

These re-breathing studies clearly indicate that high charge temperature at IVC with 2EVO, due to increased RGF and reduced lambda, shortens ignition delay and advances combustion (CA50). This improves combustion efficiency and combustion stability at low load. However, the shift in combustion phase can be effectively counteracted, if required, by adjusting parameters such as diesel injection timing and/or blend ratio (BR). Furthermore, 2EVO mitigates methane emission by up to 98 % due to improved combustion. In contrast to NVO, substantial pumping loss is not anticipated, because the charge flow through the intake valves is seamlessly replaced by the exhaust valves. Lastly, 2EVO could raise EGT, enabling rapid warm-up for catalysts. Unlike this form of i-EGR, cooled external EGR has the opposite effect, which delays combustion onset and lowers PPRR because the introduction of cooled exhaust gas reduces intake charge temperature.

5.3. NVO for in-cylinder fuel reforming

Instead of being used for i-EGR, NVO can be employed for in-cylinder fuel reforming, through direct fuel injection into the hot residual gases during the NVO period, as depicted in Fig. 19. The primary function of NVO is to induce a high thermal state through recompression, raising the in-cylinder temperature to 700-1000 K [196]. This initiates exothermic chemical reactions. Processes such as fuel vaporisation, reforming and potential oxidation during the NVO period exert a significant influence on both in-cylinder temperature and the quantity of intake air [186]. Oxidation of unburned hydrocarbons and CO during NVO also contributes to the in-cylinder temperature dynamics [197]. Consequently, the thermodynamics of NVO dictate the compression temperature histories during the main event and play a crucial role in controlling autoignition timing [198]. However, injection timing during NVO and NVO fuel fraction affect the main combustion event to a large extent [196].

The impact of NVO fuel injection on the intake process and subsequent main combustion event can be characterised by three prominent effects: (i) the energy consumption associated with fuel vaporisation, (ii) the heat release during partial fuel oxidation in the NVO phase, and (iii) the chemical conversion of fuel resulting from the reforming process [196]. All the studies reviewed in this section highlight an interconnected relationship between chemical and thermal effects, serving as the facilitator for enhanced LTC control.

Urushihara et al. [199] conducted an empirical investigation into fuel reformation, employing NVO on a 0.5 L, single-cylinder engine operating in gasoline HCCI mode. The study found that direct injection

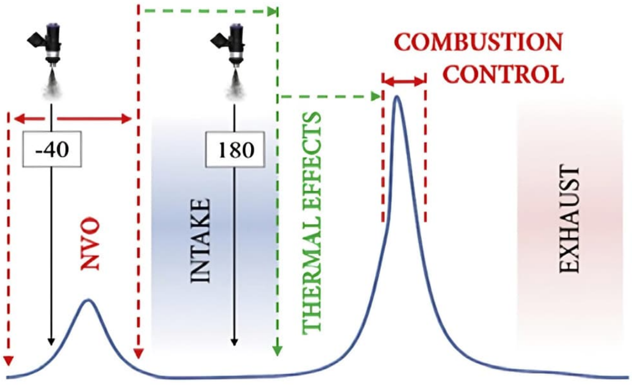


Fig. 19. In-cylinder fuel reformation with NVO and direct fuel injection in HCCI, adopted from [196].

of the entire fuel dose during NVO advances combustion, due to fuel reformation. In addition, it extends the lean-operation limit of HCCI without increasing NO_x. However, excessive fuel reforming had an adverse impact on fuel efficiency, attributable to premature ignition. A partial fuel reformation approach was introduced to mitigate this challenge, using split injection. This yielded improvements in both fuel efficiency and the lean-operation limit. Notably, the research underscored the need to reduce fuel injection quantity during NVO with increasing engine load to attain optimal fuel efficiency.

Hunicz et al. [62] experimentally investigated NVO to extend high-load limit with fuel reforming, using a single-cylinder SI engine operating in gasoline HCCI mode. Two injections were used: one during NVO, another during the early stage of IVO. The NVO was realised by retarding the whole exhaust valve profile by hydraulic VVA system. At mid load, retarding EVC while maintaining overall fuel dilution led to higher oxygen content because there was less i-EGR. This promoted more fuel reformation, which consequently increased HRR. At high load, the HRR was more sensitive to EVC timing, since less oxygen is trapped in the residuals. In addition, i-EGR tended to be hotter at low loads, reducing the need for fuel reforming to ensure proper combustion onset. The study showed that at high load, NVO with less i-EGR and reduced NVO fuel injection (for fuel reforming) is an effective way to mitigate PRR, which is beneficial for high-load extension.

Borgqvist et al. [194] examined NVO in gasoline PPC mode with a Volvo D5 diesel engine. A sufficiently high NVO with NVO injection during low-load operation extended the low-load limit by up to 1.5 bar IMEP, with stable combustion. However, excessive NVO gave excessive soot emission. The study also reported that the NVO fuel injection timing played an important role in HRR and main combustion. Only optimum NVO fuel injection timing achieves maximum thermal efficiency and stable combustion [196].

Mikulski et al. [37] numerically investigated NVO and in-cylinder fuel reforming (NVO injection) to improve low-load operation on an HD, six-cylinder diesel engine operating natural gas-diesel RCCI combustion mode. It was found that symmetrical NVO, where intake valves (IV) profiles are retarded and exhaust valves (EV) profiles are advanced, could not maintain stable combustion because the shorter intake valve opening duration leads to low thermodynamic compression pressure. Asymmetrical NVO, by only advancing EV profiles (~80 CAD), demonstrated high thermal condition during recompression. However, symmetrical NVO is more efficient since the work done for recompression can be recovered by re-expansion, as discussed in *Chapter 3.3*.

In-cylinder temperature during asymmetrical NVO reached above 850 K with NVO duration of 40 CAD and more, which is sufficient for diesel reforming/oxidation. Lean RCCI made better use of NVO fuel reformation, due to sufficient oxygen in the recompressed exhaust gases. It was reported that the amount of reformed fuel is strongly associated with NVO injection timing, and then this results in different amount of reactivity-enhancing sub-products, as shown in Fig. 20. In particular, the transfer of a large amount of heptyl radicals to the main combustion event enabled superior efficiency (99 % of combustion efficiency, 40.5 % net indicated efficiency) and ultra-low CH_4 emissions where NVO injection (20 % of pilot diesel injection) occurs at TDC. Substantial generation of auto-ignition-promoting species (alkyne, alkene, alkanes) occurs at early NVO injection, but their effect on combustion efficiency is negligible due to reduced consumption of high-reactivity diesel during NVO.

Nevertheless, it was reported that increasing NVO period by advancing EV profile induced a large loss of performance. This was attributed to reduction of expansion work by EEVO and high pumping loss by recompression during NVO. It was concluded that a fully-flexible VVA system is required to minimise the negative effects.

Ahmad et al. [200] investigated NVO without NVO injection, and demonstrated enhanced reactivity. The experiment was performed on a 1.4 L, single-cylinder HD engine, operating in methanol-diesel RCCI combustion mode. NVO was realised by altering EVC with late IVO, using electronic-hydraulic valve actuation (EHVA). An extended NVO period gave more hot residual gases, facilitating thermal cracking or slow partial oxidation of the fresh methanol. This may have led to production of the reactive species in the pre-flame region, influencing the reactivity/auto-ignition propensity. Consequently, increasing NVO advanced combustion phasing (CA10, CA50 and CA90) and increased CHR, IMEP and engine/combustion efficiency. In addition, NVO enabled maintenance of high combustion stability, despite a high PRR at high loads and ultra-lean condition at low loads. The NVO mode attained high combustion efficiency (>90 %) and high indicated efficiency (~50 %), with lower emissions of HC, CO and NO_x compared with the baseline valve profile.

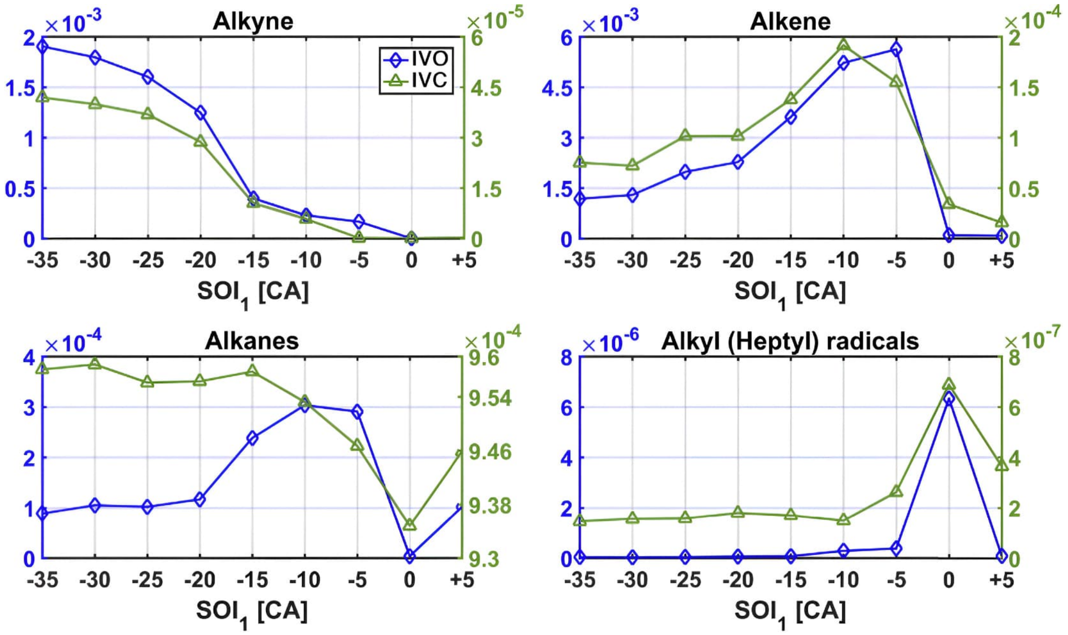


Fig. 20. In-cylinder mass fractions of alkynes (C2H2), alkenes (C2H4), alkanes (C2H6, C3H8) and alkyl (heptyl) radicals at IVO and IVC, at different NVO injection timings; lean RCCI strategy (Case B), MSR = 20 %, reproduced from [37].

5.4. *Summary of LTC-VVA effect*

This chapter has showcased VVA's potential for effective control of combustion phase in various LTC strategies. This enhanced combustion control facilitates stable combustion, extends the operational ranges at both low and high loads and also enables fast thermal management. Table 6 tabulates all the discussed studies, highlighting the findings of each. The following is a succinct summary of each VVA strategy in

Table 6
Summary of literature reviews on VVA in LTC.

Refs.	Strategy	VVA strategy	Type of engine	Fuel / LTC mode	Operating condition Speed / Load [RPM] / [bar]	Effect or Summary or Highlights
[174]	VCR	EIVC	HD diesel engine, 6.7 L, six-cylinder, HP-EGR, VGT	Diesel PCCI	2700 / 350 Nm (torque)	Low ECR by EIVC: Long ignition delay promotes PCCI combustion which advances CA50. PRR, P_{max}, peak combustion temperature are reduced.
[175]	VCR	LIVC	HD diesel engine, 2.4 L, single-cylinder EGR	Diesel PCCI	1737 / -	Low ECR by LIVC (constant AFR): Combustion phase and HRR are delayed. Low NO_x and increased CO and HC. Low soot is maintained (moderate LIVC)
[156]	VCR	EIVC	HD diesel engine, 2.1 L, single-cylinder EGR	Diesel PCCI	1100 / 3 (BMEP)	Low ECR by EIVC: Combustion (SOC, CA50) is delayed. P_{max} and combustion efficiency is reduced. Low NOx, increased CO and soot. EIVC leads to poor air-fuel mixing by reduced air, swirl ratio and slow evaporation.
[177]	VCR	EIVC LIVC	HD diesel engine, 11.6 L, six-cylinder. EGR, two-stage TC	Diesel PCCI	1600 / 5 (IMEP)	Optimal condition: EIVC: high intake pressure and high EGR rate (~70 %) LIVC: low intake pressure and moderate EGR rate (~40 %)
[178]	VCR	LIVC	HD diesel engine, 1.8 L, single-cylinder, EGR	Gasoline/ diesel RCCI	1200 / -	Low ECR by LIVC: Mid load: long ignition delay ➔ combustion control High load: load extension (~24 bar BMEP)
[179]	VCR	EIVC	Marine diesel engine, 8.8 L, single-cylinder	Natural-gas/ diesel RCCI	1000 / 10 (IMEP)	Low ECR by EIVC: Delayed combustion with long ignition delay, poor ignition process, reduced mid-load operation region
[172]	VCR	EIVC	HD diesel engine, 1.8 L, single-cylinder, EGR	Gasoline/ diesel RCCI	900–1800 / 2–22 (BMEP)	Low ECR by EIVC: High load: load extension (~22 bar BMEP), lower PRR
[180]	VCR	EIVC LIVC	HD diesel engine, 2.4 L, single-cylinder EGR	Gasoline/ diesel RCCI	1737 / 5, 9, 18 (IMEP)	EIVC + high PR***: better thermal efficiency, low soot LIVC+ low PR***: better RI
[181]	VCR	LIVC	HD diesel engine, 2.0 L, single-cylinder, EGR	Natural-gas/ diesel RCCI	1200 / 6, 12, 18 (IMEP)	Optimisation study obtains low NO_x and methane slip + high thermal/ combustion efficiency with LIVC High EGT improves the efficiency of IEATS and contributes to reducing emissions
[35]	VCR	EIVC	HD diesel engine, six-cylinder	Natural-gas/ diesel RCCI	1500 / 18–23 (IMEP)	EIVC: reduce P_{max}, high-load extension, decrease combustion efficiency
[182]	VCR	EIVC LIVC	HD diesel engine, 2.4 L, single-cylinder EGR	Gasoline/ diesel RCCI	1737 / 5, 9, 18 (IMEP)	Low load: high CR to accelerate combustion / high BTE High load: low CR to control PPRR and P_{max}
[185]	I-EGR	NVO	NA*HD diesel engine, 2.1 L, single-cylinder EGR	Diesel HCCI	1300, 1500 / 2.5–5 (IMEP)	NVO: advance combustion phase and increase combustion rate, more homogeneous mixture, low soot
[187]	I-EGR	NVO	Diesel engine	Diesel PCCI		NVO: improve fuel mixing/ evaporation/ charge homogeneity
[188]	I-EGR	NVO	Diesel engine, 0.5 L, single-cylinder	Gasoline PCCI	800 / 1.9–3.4 (IMEP)	NVO: short ignition delay, advance combustion phase, increase peak HRR, improve combustion, low-load extension Excessive NVO: long ignition delay, long combustion duration, degrade combustion
[35]	I-EGR	2EVO	HD diesel engine, six-cylinder	Natural-gas/ diesel RCCI	- / -	2EVO: short ignition delay, advance CA50, improve combustion, low methane emission, better thermal management at low load
[193]	I-EGR	2IVO	HD diesel engine, 2 L, single-cylinder EGR	Ethanol/ diesel RCCI	1200 / 3.2 (IMEP)	2IVO: enhance evaporation, accelerate combustion, stable combustion, improve combustion efficiency, low engine-out emissions
[194]	I-EGR	NVO 2EVO	Diesel engine, 0.5 L, single-cylinder	Gasoline PCCI	800 / 1.9–3.4 (IMEP)	2EVO: similar effect on combustion as NVO, better combustion stability and higher efficiency over NVO
[195]	I-EGR	2EVO	NA* Gasoline engine, 0.3 L, single-cylinder	DME HCCI	1500 /	2EVO: enhance thermal stratification at IVC
[62]	Fuel reforming	NVO	Gasoline engine, 0.5 L, single-cylinder	Gasoline HCCI	1500 / 6.1, 7.5 (IMEP)	EVC delay: enable more fuel reformation and increase HRR. NVO fuel fraction reduction: PRR mitigation and load extension.
[194]	Fuel reforming	NVO	Diesel engine, 0.5 L, single-cylinder	Gasoline PCCI	800 / 1.9–3.4 (IMEP)	NVO injection: improve combustion stability at low load and extend low load limit
[37]	Fuel reforming	NVO	HD diesel engine, six-cylinder	Natural-gas/ diesel RCCI	1000 / 3 (IMEP)	Symmetrical NVO by EV + IV shift: unstable RCCI combustion due to low compression pressure by IV shift Asymmetrical NVO by EV shift: achieve stable RCCI combustion
[200]	Fuel reforming	NVO	HD diesel engine, 1.4 L, single-cylinder,	Methanol/ diesel RCCI	1500 / 6.4–7.7 (IMEP)	NVO without NVO fuel injection: improve RCCI combustion by enhancing reactivity

NA*: Naturally aspirated, HCNG**: Hydrogen-enriched compressed natural gas, PR***: Premixed ratio relation to LTC.

VCR through intake modulation (EIVC or LIVC) has demonstrated its ability for combustion phase control, which results in long ignition delay and delayed combustion. Implementing EIVC/LIVC can achieve low effective CR. It brings down cylinder pressure and temperature near TDC, with low activation energy for combustion kinetics. This delays combustion and suppresses peak pressure rise rate (PPRR) and peak cylinder pressure, as well as knock tendency. Hence, it contributes to extending high-load operation. Additionally, it has potential to suppress NO_x by the Miller effect.

However, VCR through intake modulation has a side effect of poor air-fuel mixing, leading to high hydrocarbon emissions (CO and HC) and low combustion efficiency. This needs to be resolved further. Nevertheless, high EGT (refer to *Chapter 3.1*) enhances the efficiency of IEATS, helping to reduce tailpipe emissions. Optimising the system can effectively mitigate adverse effects, but achieving the optimal IVC timing or CR is influenced by diverse parameters in LTC, such as premixed ratio (PR), EGR rate, fuel injection strategy, engine load, plus intake pressure and temperature. Excessive EIVC/LIVC leads to misfire and unstable combustion due to insufficient in-cylinder oxygen availability.

EGR via NVO or exhaust reintroduction has shown feasibility to control LTC parameters such as SOC, HRR, PRR and CA50. Both strategies increase intake charge temperature at IVC and then advance ignition and combustion onset by introducing hot i-EGR. This improves combustion efficiency and stability, as well as engine-out emissions. Notably, exhaust reintroduction exhibits superior gas-exchange efficiency than NVO. It is worth mentioning that a majority of studies have implemented i-EGR at low-load operation. The engine load has a bearing on the outcomes. First, excessive i-EGR could constraint in-cylinder oxygen level, which leads to misfire and unstable combustion. Second, low load has more sufficient air compared with high load. Last, the advanced combustion phasing is vulnerable under high-load conditions, leading to excessive PRR and knocking. Even though i-EGR advances combustion phasing, there is a countermeasure to delay combustion phasing by adjusting injection timing and BR, especially in the context of dual-fuel operation. Therefore, i-EGR is an effective measure to control combustion at low loads. In addition, it has potential to increase EGT for ETM at low load, as discussed in *Chapter 3.3*.

Fuel reforming with NVO has demonstrated a great potential to improve combustion efficiency/stability, load limit and engine-out emissions. NVO is rather limited in its ability to control combustion, but it has been used to create high in-cylinder thermal conditions for fuel reforming. Specifically, injection timing during NVO and the fuel fraction within NVO play crucial roles in influencing both the main combustion event and the propensity for auto-ignition [72,194,196]. Higher i-EGR (large NVO) is beneficial as it provides more of the reforming substrates [37]. However, less i-EGR is required for stable combustion at high load, since high in-cylinder temperature and surface temperature trigger auto-ignition and high PRR [199]. Even though larger NVO is effective to raise in-cylinder temperature at the end of the exhaust stroke for favourable fuel reforming, excessive or large NVO significantly reduces the intake fresh charge, which causes misfire and unstable combustion, plus reduced efficiency due to heat- and pumping-losses.

Lastly, LTC is associated with poor transient performance due to high sensitivity to cycle-to-cycle variations and in-cylinder mixture conditions [73]. The limitation arises not from VVA, but from the inherent characteristics of LTC itself. To address this issue, it is recommended to shift to conventional diesel or gas combustion mode during transient events, enabling smooth transient response and extending the operational range [73,201]. Consequently, LTC is particularly well suited to marine applications where long steady-state operation and minimal transient demands dominate.

6. Conclusions

This review primarily focuses on variable valve actuation (VVA) in heavy-duty engines designed for off-road and large-bore marine engines, which hitherto have received limited attention in such research. The primary objective is to assess VVA's potential for enhancing ETM, optimising the gas exchange process and facilitating low-temperature combustion (LTC), all to ensure compliance with forthcoming emission standards. The review confines its scope to high- and medium-speed engines for marine applications and to high-speed diesel/gas engines for off-road applications. A systematic literature review (SLR) identified 78 relevant studies and these have been thoroughly discussed in individual chapters. The reviewed VVA strategies are summarized in Fig. 21, where EIVC and LIVC (Miller timing) emerge as the most commonly applied approaches.

The key conclusions derived from this comprehensive review are as follows:

In terms of exhaust aftertreatment thermal management:

- Variable valve actuation can be used to elevate exhaust gas temperature at low engine-loads by an increment of +50 °C to more than +200 °C. This is accompanied by a change in fuel consumption that varies from -10 % to +90 %, due to VVA's effect on gas exchange and combustion.
- Cylinder deactivation is the most fuel-efficient strategy, potentially improving fuel efficiency by up to 10 %, due to reduced pumping losses.
- Exhaust modulation exhibits the largest fuel penalty of up to +90 %, due to reduction of expansion cycle (EEVO) and additional recompression work (LEVO).
- Intake modulation (EIVC/LIVC) and internal EGR have a moderate effect on fuel consumption of between -5 % to +10 %, and also achieve substantial NO_x reduction. Intake modulation and CDA have the greater potential to increase EGT, by more than +200 °C.
- Even though variable valve actuation can negatively impact fuel efficiency, further combustion optimisation or use of other countermeasures could minimise this efficiency penalty.

In terms of the gas exchange process:

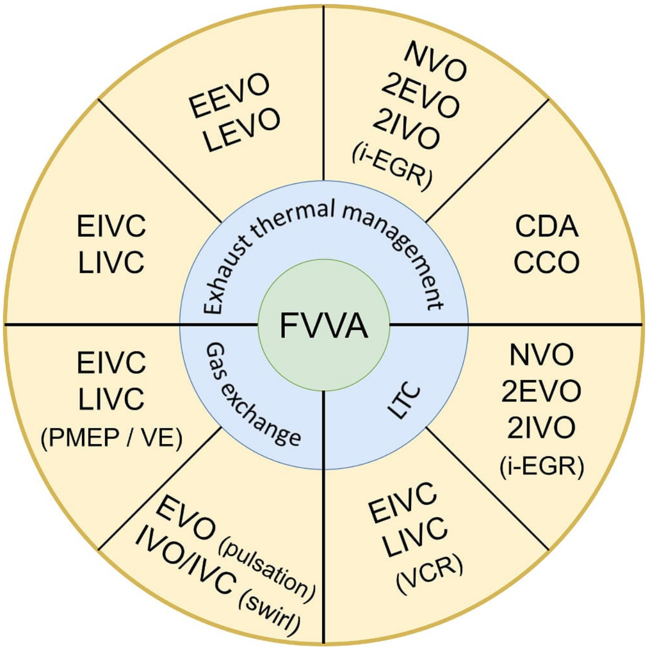


Fig. 21. VVA strategies examined in the present review.

- Variable valve actuation enables adjustment of in-cylinder flow dynamics, breathing capacity and pumping loop, facilitating efficient gas exchange in off-road high-speed engines.
- Regardless of combustion type, intake modulation could reduce pumping losses more than 25 %, which improves thermal efficiency by up to 3 % and fuel efficiency by up to 6 %.
- Intake valve modulation (EIVC/LIVC) increases volumetric efficiency by 3-5 % and achieves fuel savings of 1-2 % by adjusting effective compression ratio. This enables to balance power, efficiency, and emissions over a wide operating range.
- EVO modulation regulates exhaust pressure pulse and improves low-end torque up to 40 % with enhanced turbocharger operation.
- Intake valve opening (IVO) modulation enhances swirl and turbulence strength and thus improves combustion efficiency. Conversely, intake valve closing (IVC) modulation reduces both of them and degrades combustion, but the negative impact can be alleviated to some extent by adjusting injection timing and EGR.
- In a marine context, a positive pressure gradient ($P_{intake} > P_{exhaust}$) hinders the direct gas exchange effect, but the strategies can support lambda control.

In terms of low-temperature combustion:

- Variable valve actuation enables fast combustion phasing control by regulating in-cylinder thermodynamic conditions in marine duel-fuel engines, extending the operational limits of ultra-clean low-temperature combustion.
- Intake modulation (EIVC/LIVC) reduces compression temperature, which retards ignition and combustion onset and mitigates pressure rise rate and maximum peak pressure level, extending the high-load limit towards 24 bar BMEP.
- VVA-invoked exhaust gas recirculation enhances in-cylinder temperature by retaining hot gases, thus accelerating chemical reactions and combustion. High combustion efficiency (~99 %) and combustion stability can be achieved at low load, extending the low-load limit close to 1.9 bar IMEP.
- Negative valve overlap produces high in-cylinder thermal status (700 K ~ 1000 K) via recompression, triggering fuel oxidation and reformation of fuel injected during NVO. This could control combustion and achieve high combustion efficiency (~99 %) and combustion stability, extending both high- and low-load limits.

As a recommendation for future works:

- A fully-flexible VVA system is recommended, as it offers enhanced flexibility of valve modulation and, consequently, broader benefits for off-road and marine applications.
- More experimental-based VVA studies are required to complement the findings from a considerable body of simulation-based studies.
- A systematic comparison of different cam-less systems should be conducted, considering response time, actuation energy, system cost, carbon/ GHG footprint, and control strategies in order to identify the most suitable solution for each application.
- A systematic optimisation framework for VVA should be established to minimise adverse effects while maximise benefits.
- Practical investigations into the durability and reliability of the fully-flexible VVA system are necessary to ensure robust operation and long service life.
- Further research should address NVH issues and focus on improving transient performance to enable wider application.
- Fully-flexible VVA is identified as a key enabler for ultra-efficient, multi-fuel, low temperature combustion concept. Further studies involving with different fuel combinations, including zero-carbon fuels such as hydrogen and ammonia, are recommended to evaluate its role in next-generation clean propulsion systems.

CRediT authorship contribution statement

Jeyoung Kim: Writing – review & editing, Writing – original draft, Visualization, Project administration, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Amir Soleimani:** Writing – original draft, Visualization, Investigation, Formal analysis. **Pekka Nousiainen:** Writing – review & editing. **Martin Axelsson:** Writing – review & editing. **Maciej Mikulski:** Writing – review & editing, Writing – original draft, Supervision, Funding acquisition, Conceptualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgements

This research was conducted in the Clean Propulsion Technologies project, funded by Business Finland (ref. 38485/31/2020). The authors would like to thank David Wilcox for proofreading the manuscript.

Data availability

No data was used for the research described in the article.

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