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Effects of dimethyl ether on cyclic variations in compression ignition engines

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Abstract

Dimethyl ether (DME) can be used in compression ignition (CI) namely diesel engines as a fuel and fuel additive. This paper was compiled from the findings of the published papers on DME in diesel engines. The special procedures are employed for reduction of pollutant emissions of diesel engines. The first procedure is improvement of combustion via engine design and fuel injection modification but this is expensive and protracted. The second procedure is used an exhaust gas devices i.e. catalytic converter and particulate filter. However, these devices have adverse impact on engine performance. The last procedure is practiced the various alternative fuels and additives to reduce the emissions and also improve engine performance. The last one seems effective and economical. DME is superior for diesel engines thanks to smart fuel properties i.e. high cetane number and oxygen content. Conversely, combustion, performance and emissions in an internal combustion engine (ICE) are depended notably cyclic variations. Thus, it is vital that results of studies on DME are evaluated jointly to practice applications. Hence, this study aims to investigate the effects of DME on cyclic variations depending on current literature.

Keywords: Combustion, Compression ignition engines, Cyclic variations, Dimethyl ether

1 Introduction

Diesel (Compression Ignition–CI) engines are the most common power source for motor vehicles as they are more efficient, less pollution release, and emit less pollutant emissions (CO_2, CO, and HCs) than gasoline engines [1]. Conversely, diesel engines generate more particulate matter (PM) and nitrogen oxides (NO_x) compared to gasoline engines. Hence, a great deal of research has been done to achieve the lower emissions in diesel engines, as well as forward thinking studies on alternative fuels [2]. DME is unique among the options in support of energy security since it can be produced industrially from coal, natural gas, and a variety of biomass resources [3]. Nonetheless, the structural design and parts of diesel engines must be distorted thanks to the physical characteristics of DME, which incorporate reduced viscosity, lubricity, boiling point, and combustion enthalpy. While still at the development stage, pure DME technology is being used in CI engines, specifically in diesel engines and cars. DME has been currently used with diesel or the other alternative fuels [4]. For the purpose of place into action DME in diesel engines, it is important to appraise the outcomes of a number of studies cooperatively. Thus, the purpose of this review paper is to look into how DME affects the cyclic variations in diesel engines.

2 Fuel characteristics of dimethyl ether

As seen in Fig. 1, DME is the simple ether with the molecular formula of $CH_3–O–CH_3$ (C_2H_6O). The physical characteristics of DME and liquefied petroleum gas (LPG) are generally rather comparable. As a result, DME the storage, fuel handling, and transportation requirements of DME are comparable to LPG requirements [3]. As shown in Fig. 2, DME can be generated by direct or indirect synthetic methods. In the indirect synthetic approach, DME is produced by a dehydration reaction subsequent to the methanol synthetic reaction, whereas in the direct synthetic method, it is produced directly from syngas ($CO+H_2$) [5]. When considering energy equivalents, the production cost of DME is lower than that of gasoline or diesel fuel. When large scale plants are taken into account, the economics of producing DME are comparable to those of producing compressed natural gas (CNG) or liquefied natural gas (LNG) [6]. DME is non–toxic and gaseous at ambient temperature and atmospheric pressure. For this reason, under ambient temperature and pressure circumstances, it must be pressed to a pressure greater than 0.5 MPa in order to maintain its liquid state. To avoid vapor lock in the fuel injection system, the fuel delivery pressure should be raised to 1.7–2.0 MPa when the engine is working [7, 8].

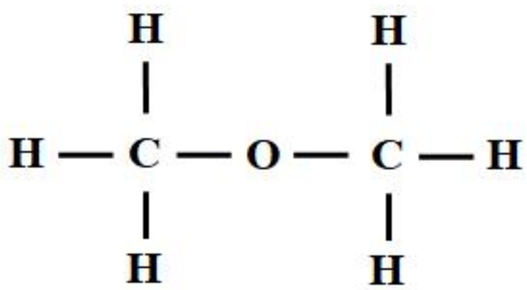

Figure 1. Chemical structure of dimethyl ether [5]

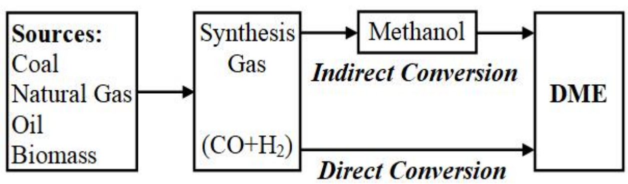

Figure 2. Production methods of dimethyl ether [9]

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Table 1. Fuel properties of diesel and DME fuels [8]

Property	Diesel	DME
Chemical formula	C_xH_y	CH_3–O–CH_3
Molecular weight, g/mol	170	46.07
Boiling point, °C	180–360	–24.9
Vapor pressure, kPa	<<10	530
Liquid density, kg/m^3	840	668
Liquid viscosity, cP	4.4–5.4	0.15
Lower heating value, kJ/kg	42500	28430
Ignition temperature, °C	250	235
Cetane number	40–55	55–60
Stoichiometric air/fuel ratio	14.6	9
Modulus of elasticity, N/m^2	1.486x10^9	6.37x10^8
Mass fraction of carbon	86	52.2
Mass fraction of hydrogen	14	13
Mass fraction of oxygen	0	34.8

Table 1 lists several benefits of DME, including excellent cetane number, adequate energy density, high oxygen content, low auto ignition temperature and high volatility. Therefore, it can assist improve engine performance, reduce emissions, and address the issue of cold starting in diesel engines when used as a pure or additive [14, 15]. Diesel fuel and DME have highly different properties, as seen in Table 1. DME is a gas fuel that has a low boiling point and a high vapor pressure at room temperature and atmospheric pressure. The heating value of DME is substantially lower than that of conventional diesel fuel. In order to accommodate DME, the engine's fuel delivery, injection, and combustion systems must be rebuilt or modified [10]. However, because DME has a higher cetane number than diesel fuel, it has a strong igniting capacity. DME can lower the mixture temperature and increase engine volumetric efficiency because it has a considerably higher latent heat of evaporation than diesel fuel. With just C–H and C–O bonds and no C–C bonds, DME contains about 34.8% oxygen. These features help explain why DME burning produces very little noise and almost no PM emissions. It can tolerate a greater EGR rate to reduce NO_x emissions more than conventional diesel fuel [6]. DME also has the advantage of not corroding metal surfaces or the structure of the fuel system [5]. The low viscosity of DME causes leaks since the fuel supply system relies on minute clearances for sealing. The moving parts of the fuel injection system experience increased surface wear as a result of its reduced lubricity properties. Consequently, while utilizing DME, it is crucial to apply the right additives to avoid leaks and surface wear. Since DME has a higher compressibility than diesel, more compression pump work is needed to move DME than diesel. DME generally erodes rubber seals because of its corrosive properties. Because of this, all rubber seals in injection systems that are currently in existence should be changed out with non–corrosive materials [1].

3 Effects of dimethyl ether on cyclic variations

The coefficient of variation (COV) is used to evaluate the stability of engine. The cycle to cycle variations are determined when cylinder pressure is measured consecutive multiple thermodynamic cycles. The cycle to cycle pressure variation is chiefly a result of variations in the combustion process from cycle to cycle [16]. The coefficient of variation of indicated mean effective pressure (COV of IMEP) is a significant indicator for cyclic variability that may be computed from recorded cylinder pressure data and it is computed as follows [17].

$$COV\ of\ X = \frac{\sqrt{\dfrac{\sum_{i=1}^{n}\left(X_i - \bar{X}\right)^2}{n-1}}}{\bar{X}} x100 \tag{1}$$

Where, Xi is a random combustion parameter such as IMEP, P_{max} etc., $\bar{X}$ is mean value of selected combustion parameter and n is the number of cycles [17, 18].

$$\overline{IMEP} = \frac{\sum_{i=1}^{n} IMEP(i)}{n} \tag{2}$$

$$\overline{P_{max}} = \frac{\sum_{i=1}^{n} P_{max}(i)}{n} \tag{3}$$

Heywood [16] declared that engine stability is negatively affected after COV values over 10%. However, other studies declared that engine stability begin to deteriorate when COV values increase beyond 5% [19, 20].

Figures 3(a) and (b) show the standard deviations and cyclic variations of two combustion stages in spark assisted or spark ignition controlled auto ignition (SI–CAI) and dimethyl ether port fuel injection (DME–PFI). It was determined that standard deviations increased significantly from CA05 to CA90 for SI–CAI while they were maintained under level of 1°CA for DME–PFI and almost no change in standard deviations during three combustion phases as seen in Figure 3(a). It was also determined that COV of (CA05–CA50) and COV of (CA50–CA90) for DME–PFI reduced by 3.5% and 28.3% compared to SI–CAI. It was declared that stability of combustion phases and each combustion stage could be attributed to more rapid and strong heat release in CA05–CA50 stage for DME–PFI which was beneficial for shortening combustion duration and stabilizing of combustion. Figure 3(b) shows the cyclic variations in SI–CAI and DME–PFI combustion modes computed from 100 consecutive cycles. It was declared that DME–PFI showed great benefit in stabilizing combustion via accelerated flame formation and propagation due to larger area of ignition source and higher ignition energy compared to SI–CAI. It was determined that cyclic variation characteristics was major problem of SI–CAI combustion and significant developments could be obtained with DME–PFI combustion and COV of CA05–CA90 reduced from 11.7% to 5.3% via DME–PFI. It was concluded that DME–PFI combustion gave great benefits in stability of combustion and shorter combustion duration compared to SI–CAI combustion [21].

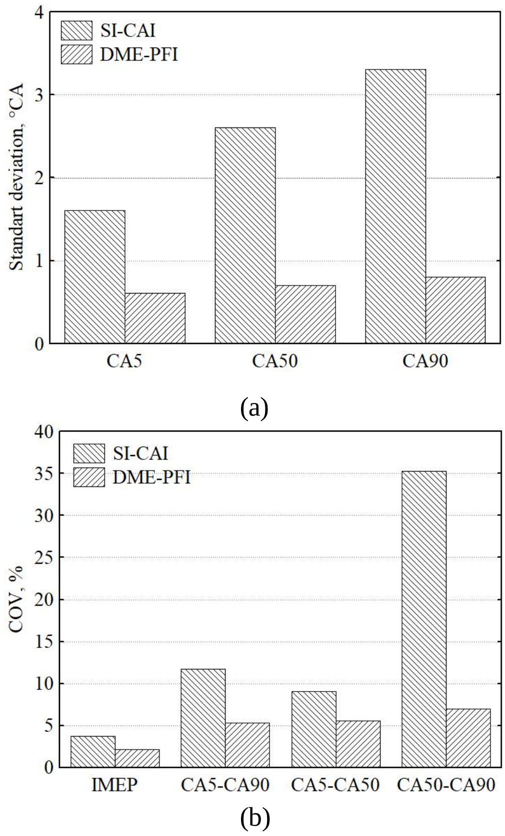


Figure 3. Variation of different cyclic variations with DME for various combustion strategies [21]

Figure 4(a) shows the variation of COV of P_{max} with homogenous charge compression ignition (HCCI) premixed ratio (r_p). It was determined that COV of P_{max} for direct injection (DI) combustion was much larger than HCCI combustion and they were 6.1 and 0.66% as seen in Figures 4(a). It was declared that cyclic variability was due to unsteady cylinder flow and injection variations in CI engines and compressibility of DME was higher in a closed system due to elasticity modulus of DME was lower hence injection variations of DME engine in DI combustion was bigger. It was also declared that pump–pipe–nozzle of fuel injection system was not adopted in HCCI combustion so DME vapor produced by vaporizer when it sent to mixer where homogeneous fuel–air mixture was formed. It was explained that ignition happened simultaneously at multiple points and better stability of HCCI combustion was achieved and thus COV of HCCI combustion was lower than DI combustion. It was determined that COV of P_{max} first reduced with rising HCCI combustion ratio and then changed little. As seen in Figure 4(a), COV of P_{max} reduced from 2.7 to 1.1% when HCCI combustion ratio increased from 33% to 47% and it increased from 0.41 to 0.51% when HCCI combustion ratio increased from 67% to 95% as seen in Figure 4(a) so it was declared that an appropriate HCCI combustion ratio should be chosen 67% for better combustion stability at 1100 rpm (revolution per minute). It was declared that similar values for COV of P_{max} were obtained at 1500 rpm and COV of P_{max} at HCCI combustion mode was much lower than DI combustion mode at 1500 rpm as seen in Figure 4(b). It was also declared that DME premixed (HCCI combustion) ratio had little negative effects on COV of P_{max} in DI–HCCI combustion modes at 1500 rpm. It was determined that COV of P_{max} was maintained at a relatively small value lower than 1% when DME ratio increased from 39% to 91% at 1500 rpm. Figure 4(c) shows the effects of brake mean effective pressure (BMEP) on COV of P_{max} in DI–HCCI combustion. It was declared that COV of P_{max} values at large DME premixed ratio (>52%) were lower than those of small DME premixed (<52%) ratios for all BMEP values as seen in Figure 4(c). It was determined that COV of P_{max} values with small DME premixed ratios reduced from 1.5 to 0.76 % when BMEP raised from 16 to 26 bar [22].

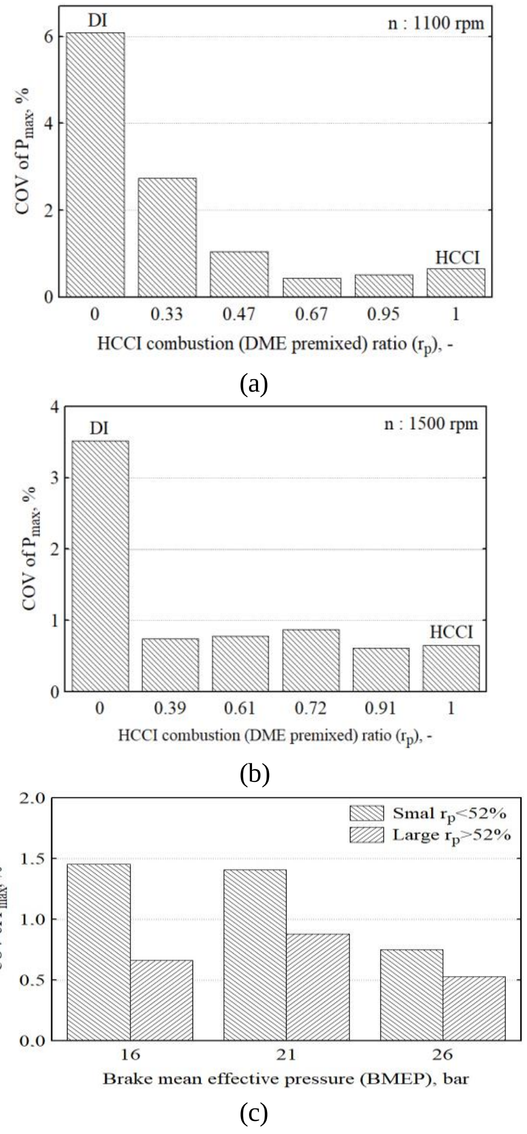
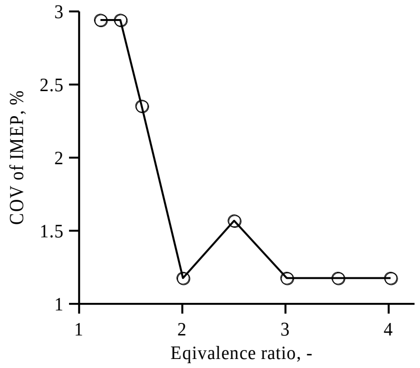


Figure 4. Variation of COV of P_{max} at a) 1100 rpm and b) 1500 rpm with HCCI ratio and c) small and large r_p ratios with BMEP [22]

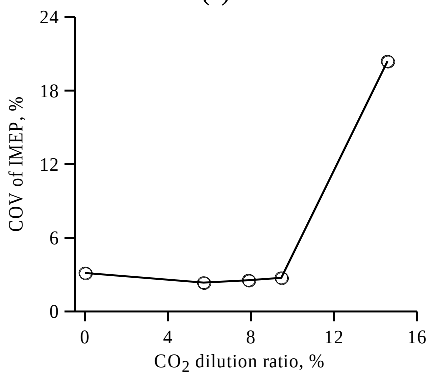
It was also determined that COV of P_{max} values had quite small values at large DME ratios for all BMEP values and they were determined as 0.6, 0.8 and 0.5%. It was declared that similar results were obtained for DME premixed ratio of

0.4 and 0.6 in DI–HCCI combustion. It was determined that COV values were 0.74, 0.93 and 0.86% for r_p of 0.4 and 0.83, 0.84 and 0.68% for r_p of 0.6 at BMEP of 11, 16 and 21 bar. It was also declared that COV was small at 1500 rpm for all BMEP values [22].

Figures 5(a) and (b) show the variation of COV of IMEP with equivalence ratio and CO_2 dilution ratios. It was declared that lean combustion of DME showed little suppression on engine stability. It was determined that rising equivalence radio improved engine stability especially after equivalence ratio of 2 as seen in Figure 5(a), although COV of IMEP deteriorated beyond CO_2 dilution ratio of 14.5 due to poor combustion in Figure 5(b). It was declared that combustion control of DME–HCCI was achieved with charge dilution and spark assistance [23].



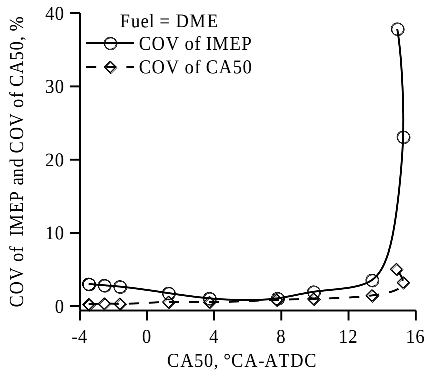
(a)



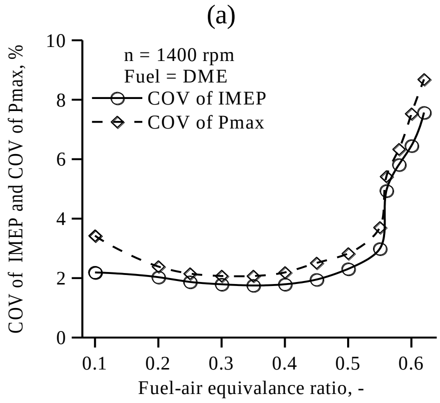
(b)

Figure 5. Variation of COV of IMEP with a) equivalence ratio and b) CO_2 dilution ratio in HCCI–DME engine [23]

Figure 6(a) shows the variation of COV of CA50 (middle of combustion phasing) and IMEP with CA50. It was declared that COV of CA50 and IMEP values for 64 consecutive cycles were relatively stable until limit of combustion phasing retarded in spite of rising volume expansion cooling as seen in Figure 6(a). Additionally, it was said that COV of IMEP increased somewhat for earlier combustion phasing due to an increase in heat transfer changes from cycle to cycle that were derived from differences in knock intensity. It was determined that COV of IMEP was only 1.21% at CA50 of 3.46°CA–ATDC (degree crank angle–after top dead center) that was middle of combustion phasing range examined. Although it was determined that delaying combustion phasing was a useful strategy for prolonging high load HCCI operation, amount of delaying that could be achieved was constrained by increasing cycle variations. It was claimed that increasing cycle variations were seen with combustion phasing retard since cycle variations in temperature during intake valve close (TIVC) were main cause of cycle variations in temperature of compressed charge. Also, even in absence of cycle to cycle variations at TIVC, heat transfer and turbulent mixing during compression stroke would result in cycle variations in temperature of compressed charge. The amount of trace species and/or unburned fuel re–circulated via EGR (exhaust gas recirculation) was also shown to vary from cycle to cycle, contributing to overall cycle variability. These predictable cycle variations were said to cause changes in auto ignition timing and burn length from cycle to cycle. These changes were said to increase when temperature rise rate decreased due to combustion phasing retard. As a result, at later combustion phasing, cycle variations of CA50 and IMEP were more significantly impacted by specific cylinder charge state variables (temperature, pressure and composition). Comparing the behaviors of cycle variations for stable and unstable operation could help identify the causes of CA50 and IMEP cycle variations [12].



(a)



(b)

Figure 6. Variation of a) COV of IMEP and CA50 with CA50 [12] and b) COV of IMEP and COV of P_{max} with fuel air–equivalence ratio [13]

Figure 6(b) shows the effect of equivalence ratio on COV of IMEP and P_{max} during 100 consecutive engine cycles for DME fueled HCCI engine. It was decided that test engines might employ a standard limit of 5% of COV value to assess engine stability. Figure 6(b) shows that the COV of IMEP and P_{max} was found to be below the 5% limit value until the equivalency ratio of 0.55, which indicated combustion stability under smooth running conditions [13].

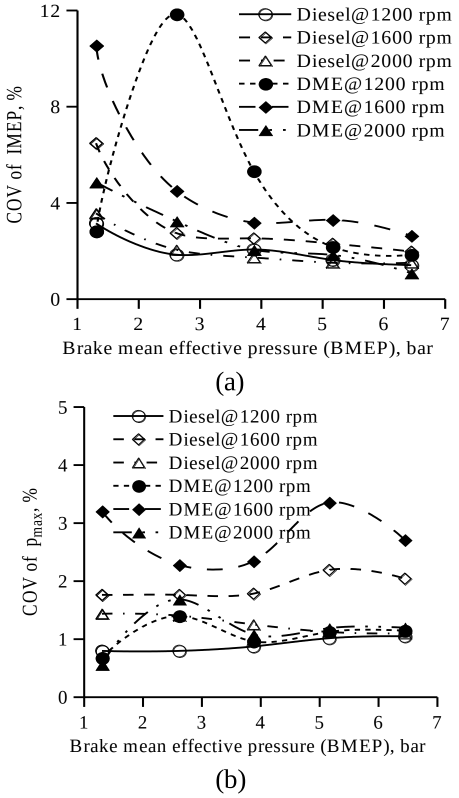


Figure 7. Variation of a) COV of IMEP and b) COV of P_{max} with BMEP [24]

Figures 7(a) and (b) show the variation of COV of P_{max} and COV of IMEP with BMEP (engine load) for diesel and DME fuels for various engine speeds. It was determined that choosing P_{max} for cyclic fluctuations was superior to choosing the maximum rate of pressure rise because it resulted in greater computation mistakes. It was found that the COV of P_{max} difference between DME and diesel was low at optimum power speed of 2000 rpm and high at maximum braking torque (MBT) speed of 1200 rpm. Additionally, for all engine speeds, the COV of P_{max} was shown to be greater at lower loads, equivalent at medium loads, and insignificant at higher loads. The higher cetane number of DME compared to diesel fuel, which results in a shorter ignition delay time during combustion, was cited as the reason why differences in P_{max} were deemed allowable. According to the statement, COV of IMEP provided information about engine combustion stability and characterized total cyclic variations by combining all fluctuations during combustion. It was shown that COV of IMEP varied more at lower loads and less at higher loads. Since there was less fuel in the cylinder between injection start and ignition initiation, it was determined that this resulted in a shorter ignition delay at greater loads. COV of IMEP for DME and diesel fuels was found to differ negligibly, indicating that DME fueling is appropriate for greater loads. It was decided that although stated COV values could appear higher than those of small bore engines, they might be suitable for use in tractor engines. Also, it was mentioned that increasing and decreasing COV of IMEP for DME was comparable to diesel [24].

Figures 8(a) and (b) show the variation in COV of IMEP with diesel–DME blends at different injection timings at 2200 rpm and full load conditions. It was determined that peak combustion pressure reduced due to shorter ignition delay (17°CA for diesel and 6°CA for 52% DME blend at injection timing of 15°BTDC (degree before top dead center) and 14°CA for diesel and 5°CA for 58% DME blend at injection timing of 12°BTDC) which supported shifting combustion phase away from top dead center (TDC) when injection timing was retarded from 15°BTDC to 12°BTDC. It was declared that stability of engine operation was analyzed by using COV of P_{max} and COV of IMEP. It was determined that these both parameters reduced when injection timing retarded from 15°BTDC to 12°BTDC as seen in Figures 8(a) and (b). It was stated that this was signed late injection was suitable for stable DME fuelled engine operation. It was also declared that COV was one of important parameters to determine limit or optimum DME ratio in DME–diesel dual fuel operation [25].

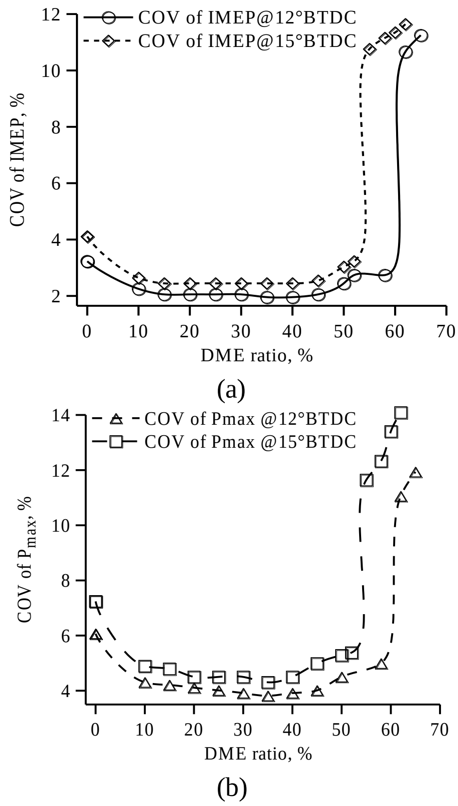


Figure 8. Variation of a) COV of IMEP and b) COV of P_{max} with DME ratio [25]

Figure 9(a) shows the variation COV of P_{max} for 100 consecutive combustion cycles. It was determined that largest port DME quantity produced highest P_{max} value, while absence of port DME premixing produced lowest P_{max} value. It was mentioned that P_{max} is a crucial mechanical limitation in engine design, making its variations crucial to examine. It was shown that P_{max} increased as DME quantity increased and that COV of P_{max} decreased as DME quantity decreased. It was determined that a high DME ratio increased knock tendency, which may lead to more intense pressure oscillations. Thus, COV of P_{max} rose as DME quantity increased because of increasing knock tendency. But as Figure 9(a) showed, COV of P_{max} values were relatively low (COV of P_{max}<3%), indicating that overall cyclic changes in P_{max} were smaller. It was determined that changes in average gas temperature cycle by cycle affected changes in heat transfer from cylinder walls, necessitating an investigation into T_{max} fluctuations. Figure 9(b) shows the variation of T_{max} for 100 consecutive combustion cycles. It was determined that T_{max} increased with rising DME quantity due to early combustion of DME similar to P_{max} and COV of T_{max} also increased with rising DME quantity. It was determined that rising knock intensity caused more heat transfer from cylinder liner and improved cylinder temperature distribution, which raised the COV of T_{max} at higher DME quantities. Statistical variation in T_{max} is rather minor in DME–diesel dual fuel partly charged compression ignition (PCCI) combustion, as COV of T_{max} values were found to be less than 3% in all tests. Tests showed that T_{max} diverges in a relatively narrow range, suggesting that lower COV of T_{max} values may indicate almost negligible flame propagation in spark ignition (SI) engines. According to declaration, rate of heat release (ROHR) measures speed at which fuel chemical energy is transformed into thermal energy during combustion. Variations in ROHR from cycle to cycle should fall within an ideal range for efficient engine performance. Figure 9(c) shows the cycle by cycle variations in $(dq/d\theta)_{max}$. It was determined that amount of HCCI combustion increased and DI combustion reduced during DME–diesel dual–fuel PCCI combustion which leading to a decrease in a value of $(dq/d\theta)_{max}$ with rising DME quantity. Additionally, it was found that higher temperatures and pressures led to a more thorough and rapid combustion of diesel, which decreased cycle variations in $(dq/d\theta)_{max}$ for direct injection compression ignition (DICI) combustion. As a result, cycle variations in $(dq/d\theta)_{max}$ decreased as amount of DME increased. It was stated that when HCCI combustion gradually took over combustion and $(dq/d\theta)_{max}$ appeared in the HCCI high temperature reaction (HTR) process, amount of HCCI combustion may surpass the amount of diesel DICI with further rise in DME port quantity. In that case, cycle by cycle variations in $(dq/d\theta)_{max}$ did not reduce but they increased with rising DME quantity. It was determined that combustion noise is directly related to cyclic variations in rate of pressure rise (ROPR), and cyclic changes in $(dP/d\theta)_{max}$ are also frequently employed for cyclic variations. Figure 9(d) shows the cycle by cycle variations in $(dP/d\theta)_{max}$ for 100 consecutive cycles. It was found that for diesel–DME dual–fuel PCCI combustion process, variations in ROPR follow the same trend as ROHR. It was determined that trend of the cycle fluctuations in $(dP/d\theta)_{max}$ and cycle variations in $(dq/d\theta)_{max}$ was comparable. Additionally, COV of $(dP/d\theta)_{max}$ was shown to decrease initially as DME quantity rose, but to increase over time. Additionally, even though ROPR was noise sensitive, it was determined that changes in $(dP/d\theta)_{max}$ could roughly correspond to changes in the maximum rate of heat release. Therefore, it may be possible to estimate the quick early calculation of a combustion phase using the crank angle position $(dP/d\theta)_{max}$. It was stated that primary benefit of employing position of $(dP/d\theta)_{max}$ was that it gave a controller more time to acquire values of other control parameters and turn on actuators to regulate the subsequent cycle. It was determined that drivability of engine is directly impacted by COV of IMEP. According to numerous reports, drivability issues in cars typically begin when the COV of IMEP surpasses 10%. Figure 9(e) shows the cycle by cycle variations in IMEP for 100 consecutive combustion cycles. It was found that IMEP displayed clear oscillations and was dispersed over a larger range as DME quantity increased. This was due to rising DME quantity with increased amount of HCCI combustion, which in turn caused cycling to cycle variations in ignition timing and burning rate, which in turn caused variations in IMEP. COV of IMEP rose as quantity of DME grew; it was determined [14].

Figures 10(a) and (b) show COV of P_{max} as an indicator of combustion performance for assessment of combustion characteristics. COV of P_{max} for single injection mode shows a larger fluctuation in maximum pressure, as seen in the figures, but variation in P_{max} for pilot injection mode was similar over a wide injection timing range. Based on these differences in maximum pressure, it can be inferred that pilot injection improved cyclic fluctuations for diesel–biodiesel–DME blend and was a successful measure for combustion stability [26].

Figures 11(a) and (b) show the variation of COV of IMEP with ethanol (ETH) energy ratio for DME–ETH blends at various intake temperatures. It was determined that combustion turned to unstable (COV of IMEP > 5%) and sudden drops in IMEP and indicated thermal efficiency at intake temperature of 20ºC as seen in Figure 11(a) when ethanol energy ratio was raised to 20%. This was determined to be result of high temperature reaction (HTR) starting too late, which caused whole combustion to take place during expansion stroke. A similar trend was also determined at intake temperature of 40ºC. It stated that rising intake temperature reduced COV of IMEP at stable energy input. As seen in Figure 11(b), COV of IMEP ≤ 5% increased with ethanol energy ratio from 26.3% and rising intake temperature to 40ºC and 60ºC. It was declared that onset timings of HTR without low temperature reaction (LTR) could be expected at 40ºC and 60ºC intake temperatures for ethanol ratios larger than certain values and so deteriorations raised in combustion due to much ethanol addition [27].

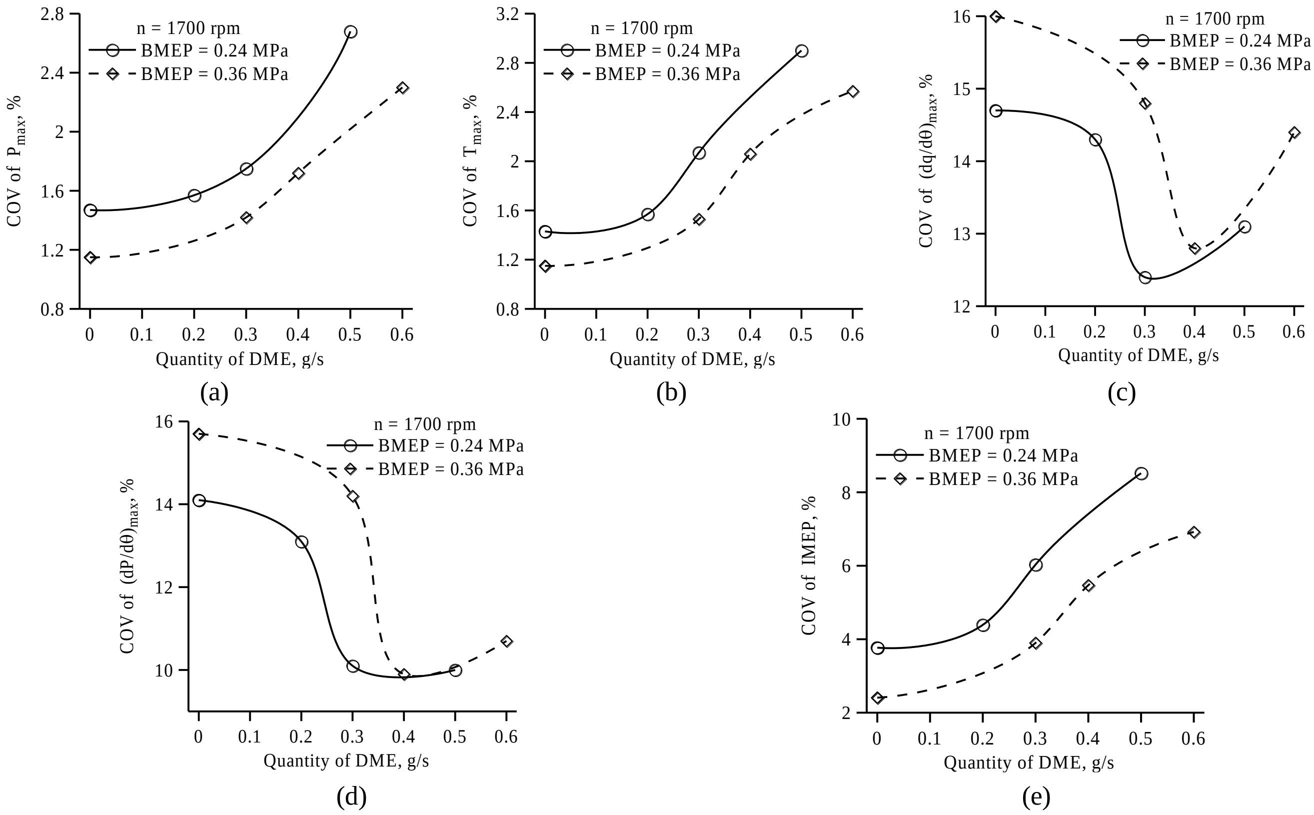


Figure 9. Variation of various kinds of cyclic variations with quantity of DME [14]

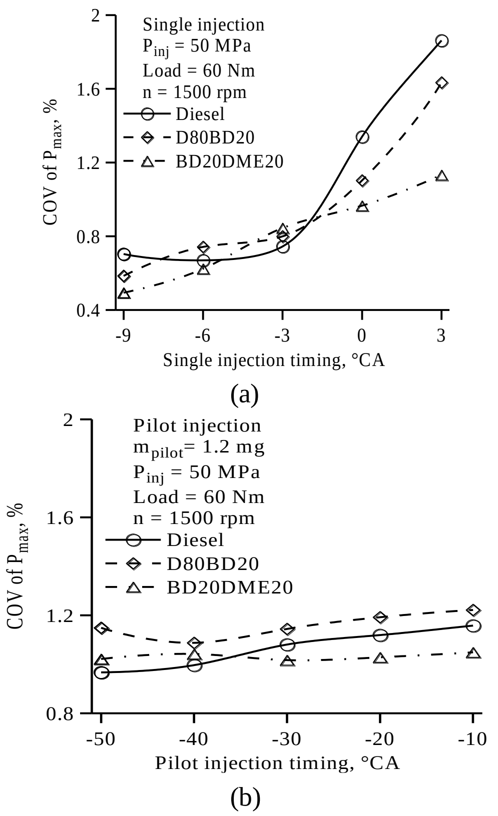


Figure 10. Variation of various kinds of cyclic variations with DME blend at various injections [26]

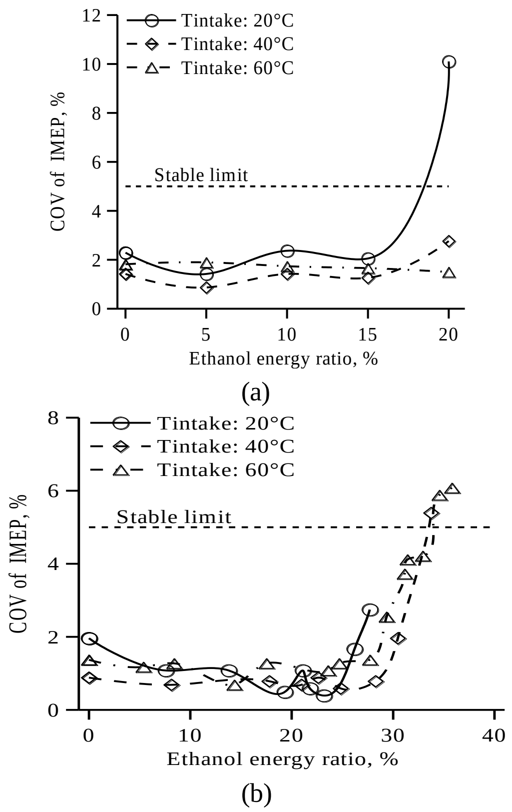


Figure 11. Variation of COV of IMEP with ethanol energy ratio for DME–ETH blends at various intake temperatures [27]

Figure 12(a) shows the variation of COV of IMEP with ethanol energy ratio for DME–ETH blends at various equivalence ratios. Unstable operation at equivalency ratio of 0.3 with 20% ethanol energy ratio was also discovered, along with quick combustion with knocking aroused at the greatest equivalence ratio (ϕ=0.38), in all experiments with lower indicated thermal efficiency. In contrast to equivalency ratios of 0.30 and 0.38, it was determined that carefully regulated onset timing with ethanol addition, particularly more than 15%, and an equivalency ratio of 0.34 had the potential to be beneficial. This demonstrated how crucial ignition timing control is and provided a way to switch between different equivalency ratios with varying amounts of ethanol injection, allowing the engine to run at desired loads without compromising indicated thermal efficiency. Figure 12(b) shows the variation of COV of IMEP with ethanol energy ratio for DME–ETH blends at various engine speeds. It was determined that maximum IMEP improved under a certain ethanol energy ratio with rising indicated thermal efficiency. It was declared that intake temperature was set at 40ºC to avoid earlier misfiring with ethanol addition at high engine speeds. It was stated that stable operation with COV≤5% was continued at almost all operating conditions and increase in maximum IMEP and indicated thermal efficiency for almost ethanol energy ratios with rising engine speed [27].

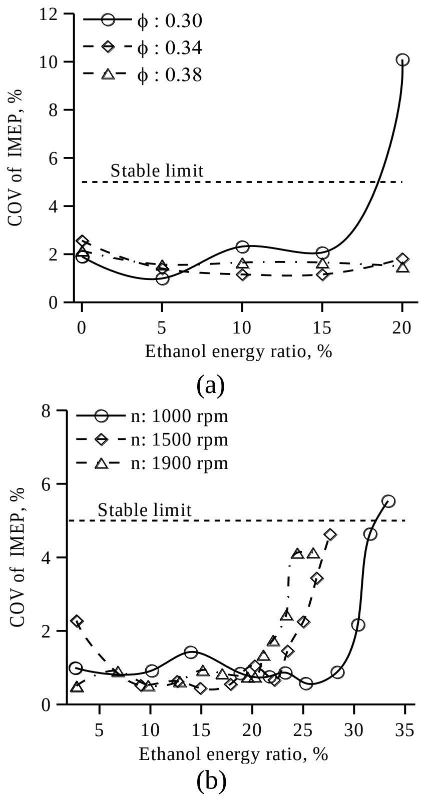


Figure 12. Variation of COV of IMEP with ethanol energy ratio for DME–ETH blends at a) various equivalence ratios and b) various engine speeds [27]

Figures 13(a) and (b) show the variations of cycle to cycle variations for ammonia (NH_3)–DME blends. It was determined that addition of ammonia increased COV of P_{max} and COV of CAD of P_{max} as seen in Figure 13(a). The COV of P_{max} and COV of CAD of P_{max} for pure DME were similarly found to be modest, at roughly 1% and 0.11%, however at high engine loads; they increased to 8% and 0.44% for 60%DME–40%NH_3 blend. This increased variance was shown to be caused by more temperature loss as a result of more ammonia being delivered at higher loads, which lengthens the ignition delay and increases variability. Additionally, it was reported that the 40%DME–60%NH_3 blend burned quite steadily when compared to the 60%DME–40%NH_3 blend; nonetheless, the COV of P_{max} and COV of CAD of P_{max} were still greater than those of pure DME, which were 5% and 0.16%, respectively. It was claimed that the results for HCCI engines were consistent and that the rise in HC and CO emissions was caused by incomplete combustion in some cycles. Additionally, it was claimed that unstable combustion resulted from the ammonia's evaporation lowering the cylinder temperature during the compression process. It was shown that rising engine load reduced the cycle variability of 40%DME–60%NH_3 blend. When the engine load grew, the cylinder temperature rose, causing the engine to reach stable combustion [15].

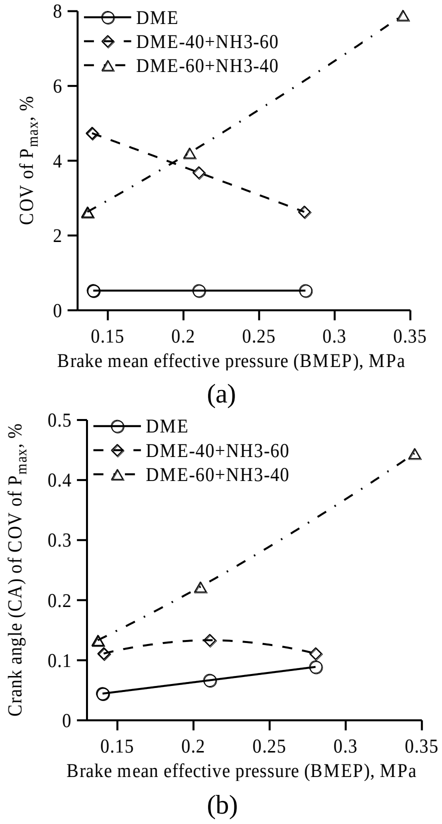


Figure **13.** Variation of various kinds of cyclic variations for diesel and DME–NH_3 blends [15]

Figure 14(a) shows the variation of COV of P_{max}, COV of IMEP, COV of $ROPR_{max}$ and COV of N for 100 consecutive cycles with load at 1400 rpm. It was declared that upper limit for each parameter in internal combustion engines is 5% of COV value. As shown in Figure 14(a), COV value was over limit value of 5%, resulting in unstable combustion with severe knocking, whereas COV value for other parameters was found to be below limit value until load reached 24%, indicating smooth engine running with stable combustion. Also, Figure 14(a) illustrates how knocking intensity (KI) varies by load at 1400 rpm. KI rose noticeably in HCCI combustion mode powered by DME with rising load. It was stated that limit value for start of knock was 5 MW/m^2. It was determined that KI was below the limit value suggesting smooth engine operation for loads of 5–24% and KI was over limit value of 6.64–13.82 MW/m^2 which resulting in a high rate of pressure rise and causes to uncontrolled combustion [28].

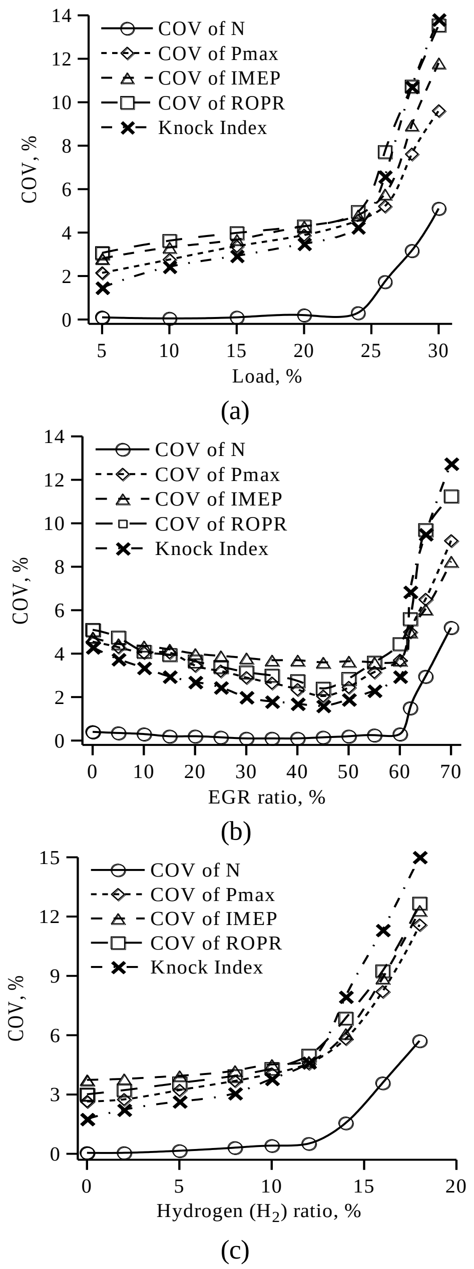


Figure 14. Variation of various kinds of cyclic variations for DME–hydrogen blends at various operating conditions [28]

Figure 14(b) reveals the effect of EGR rate on COV of P_{max}, COV of IMEP, COV of $ROPR_{max}$, and COV of N and KI at maximum load limit of 24% and 1400 rpm for DME fuelled HCCI engine. It was found that until the EGR rate reached 60%, COV values of all parameters and KI were below the acceptable limit value, indicating stable engine running with acceptable combustion noise. It was found that over limit values for COV and KI above 60% EGR rate caused unstable combustion with strong knock, multi peak heat release, and significant pressure fluctuations during combustion. DME fueled HCCI engine running at 1400 rpm was found to have a maximum EGR rate of 60%. Figure 14(c) shows the effect of hydrogen energy ratio (HER) on COV of P_{max}, COV of IMEP, COV of $ROPR_{max}$ and COV of N and KI for optimized EGR of 35% for DME fueled HCCI engine at 1400 rpm. It was determined that COV and KI values was below limit value until 12% HER which indicating stability of engine with smooth running conditions. COV and KI values above 12% HER were over limit value of COV>5% and KI>5 MW/m^2 which ensuing in unstable combustion with intensive knock. It was concluded that maximum HER was 12% at 1400 rpm [28].

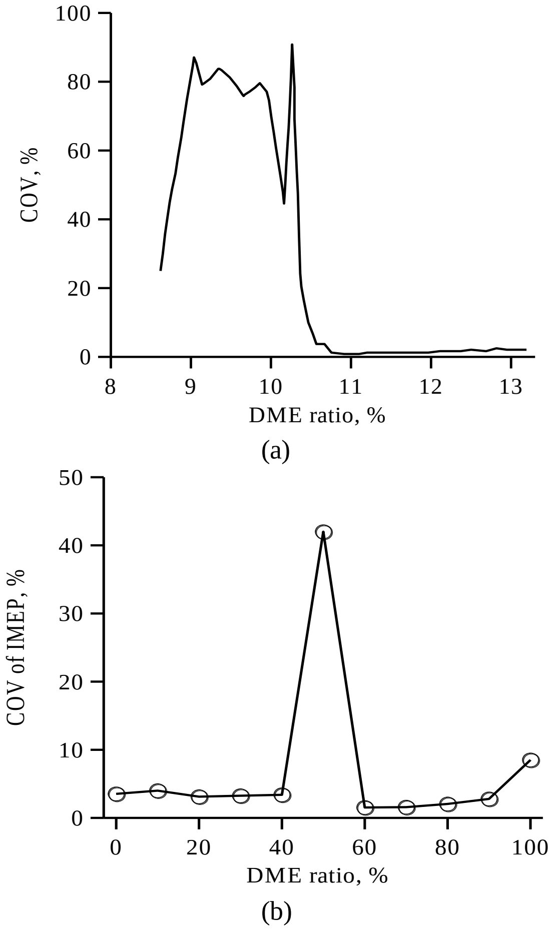


Figure 15. Variation of a) COV for methane–DME [11] and b) COV of IMEP for DME–LPG blends [29]

Figure 15(a) shows the COV for methane flows of 8.17 LPM (liter per minute) and DME flows from 0.75 to 1.21 LPM with 0.01 LPM increments in HCCI engine. It was determined that rising DME flow changed DME equivalence ratio from ϕ_{DME}= 2.02 at lowest to 2.20 at highest DME ratio and equivalence ratio of methane was varied from ϕ_{CH4}=1.77 to 1.80. Ignition does not occur and results in negative IMEP at lower DME ratios. It was declared that more DME addition leaded to irregularly firing cycles with high COV and COV reduced to 45% near 10% DME ratio but it increased to 90% at DME ratio of 10.3%. Additionally, it was stated that greater DME ratios resulted in cycles of fire and misfire, which alternatively caused early ignition and high rates of pressure rise. However, the charge in the subsequent cycle was diluted by the reacted residual gas, which caused misfire. Further DME ratios were found to induce stable engine running, with a maximum IMEP of 10.5% DME ratio. It was stated that higher DME ratios drawn CA50 earlier and P_{max} as heat was released at lower cylinder volume. Thus, IMEP reduced due to rising relatively higher heat losses and P_{max} reached to 10 bar/°CA limits at 11.3% DME ratio. It was continued rising DME ratio for showing trends beyond this point, but in general such high pressure rise rate was considered potentially damaging and was avoided [11]. Figure 15(b) shows COV of IMEP for various DME ratios for DME–LPG blends. As seen in the figure COV of IMEP reached its peak of 42% at DME ratio of 50% while it's normal value should be within 8% [29].

Figure 16(a) shows the variation of COV of IMEP with injection timing for diesel and DME–n–Butane (BUT) blends. It was stated that injection pressure and fuel amounts were set at 8 mg and 250 bar for blends and 12 mg and 1500 bar for diesel. For diesel, the most stable combustion was found at 9°CA–BTDC, and for blends, at 12°CA–BTDC. The IMEP COV was less than 5% under these operating conditions, which was regarded as the engine stability cutoff threshold. Thus, it was stated that fuel injection timing was fixed at 9°CA–BTDC for diesel and 12°CA–BTDC for blends for all engine tests. It was found that a sudden drop in IMEP and a sharp rise in COV of IMEP were induced by delayed injection time after 12°CA–BTDC. Because poor auto ignition property of n–Butane caused a delayed start of combustion, it was determined that the engine's operational zone was limited. Because engine combustion took place during the compression stroke, it was also mentioned that engine output decreased at advanced timing prior to 20°CA. Figure 16(b) show the variation of COV of IMEP with engine load (IMEP) for diesel and DME–BUT blends. Under all operating conditions, with the exception of 40% n–Butane content at 1 bar IMEP, less than 5% COV of IMEP was produced. It was said that 5% COV of IMEP was the cutoff that determines combustion stability. The primary cause of unstable combustion was attributed to poor auto ignition caused by a low cetane number of 40% n–Butane. Due to their quick vaporization, mixes were found to have better combustion stability than diesel [30]. Figure 16(c) shows the variation of COV of IMEP with injection timing for DME–LPG (Iso–butane) blends. It was determined that COV of IMEP was 1% for DME80LPG20, and DME90LPG10 blends and diesel and DME70LPG30 blend gave deteriorated combustion stability when injection timing was fixed after TDC as seen in Figure 16(a). It declared that irregular combustion occurred because late injection timing which concluded with longer ignition delay [31].

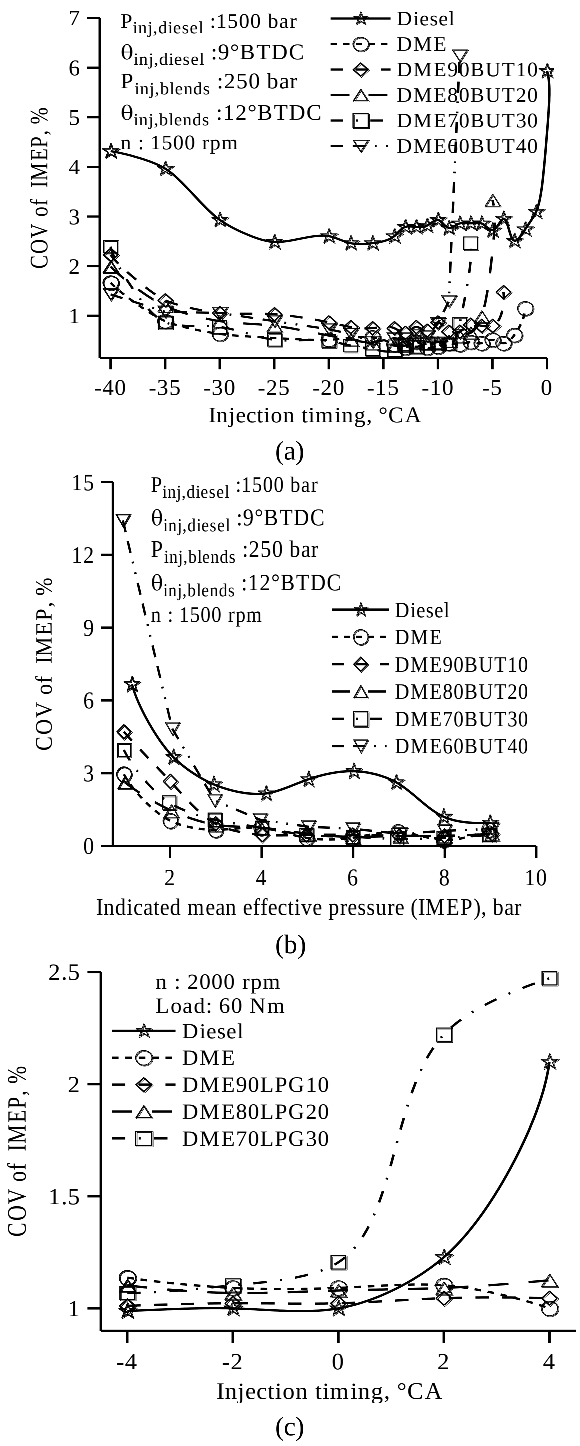


Figure 16. Variation of COV of IMEP with a) injection timing, b) IMEP for DME–BUT blends [30] and c) injection timing for DME–LPG blends [31]

Table 2. Variations in COV values with using DME

Fuel or blend	DME ratio	Engine type	Variation (%)	Ref
40% iso–octane 60% n–heptane	1.5mg/cycle~10%	PFI engine	↓1.5% for COV of IMEP and ↓3.5% for COV of (CA05–CA50) ↓28.3% for COV of (CA50–CA90) ↓6.4% for COV of (CA05–CA90)	[21]
DME	100%	HCCI engine	↓3.39% for COV of P_{max} at r_p of 0.33 and 1100 rpm ↓5.04% for COV of P_{max} at r_p of 0.47 and 1100 rpm ↓5.65% for COV of P_{max} at r_p of 0.67 and 1100 rpm ↓5.57% for COV of P_{max} at r_p of 0.95 and 1100 rpm ↓5.43% for COV of P_{max} at r_p of 1 and 1100 rpm ↓2.78% for COV of P_{max} at r_p of 0.39 and 1500 rpm ↓2.74% for COV of P_{max} at r_p of 0.61 and 1500 rpm ↓2.65% for COV of P_{max} at r_p of 0.72 and 1500 rpm ↓2.91% for COV of P_{max} at r_p of 0.91 and 1500 rpm ↓2.87% for COV of P_{max} at r_p of 1 and 1500 rpm	[22]
Diesel and DME	100%	DICI engine	↓0.32%–↑10% for COV of IMEP at 1200 rpm ↑0.66–4.1% for COV of IMEP at 1600 rpm ↓0.43%–↑1.29% for COV of IMEP at 2000 rpm ↓0.12%–↑0.6% for COV of P_{max} at 1200 rpm ↑0.52–1.44% for COV of P_{max} at 1600 rpm ↓0.87%–↑0.28% for COV of P_{max} at 2000 rpm	[24]
Diesel–DME blends	Various	CRDI engine	↓0.98% for COV of IMEP at 10% DME and injection timing of 12°CA ↓1.17% for COV of IMEP at 15–30% DME and injection timing of 12°CA ↓1.27% for COV of IMEP at 35–40% DME and injection timing of 12°CA ↓0.78% for COV of IMEP at 50% DME and injection timing of 12°CA ↓0.49% for COV of IMEP at 52–58% DME and injection timing of 12°CA ↑7.43% for COV of IMEP at 62% DME and injection timing of 12°CA ↑8.02% for COV of IMEP at 65% DME and injection timing of 12°CA	[25]
Diesel–DME blends	Various	CRDI engine	↓1.47% for COV of IMEP at 10% DME and injection timing of 15°CA ↓1.66% for COV of IMEP at 15–40% DME and injection timing of 15°CA ↓1.56% for COV of IMEP at 45% DME and injection timing of 15°CA ↓1.08% for COV of IMEP at 50% DME and injection timing of 15°CA ↓0.88% for COV of IMEP at 52% DME and injection timing of 15°CA ↑6.65% for COV of IMEP at 55% DME and injection timing of 15°CA ↑7.04% for COV of IMEP at 58% DME and injection timing of 15°CA ↑7.24% for COV of IMEP at 60% DME and injection timing of 15°CA ↑7.53% for COV of IMEP at 62% DME and injection timing of 15°CA	[25]
Diesel–DME blends	Various	CRDI engine	↓1.76% for COV of P_{max} at 10% DME and injection timing of 12°CA ↓1.86% for COV of P_{max} at 15% DME and injection timing of 12°CA ↓1.96% for COV of P_{max} at 20% DME and injection timing of 12°CA ↓2.05% for COV of P_{max} at 25 and 45% DME and injection timing of 12°CA ↓2.15% for COV of P_{max} at 30 and 40% DME and injection timing of 12°CA ↓1.56% for COV of P_{max} at 50% DME and injection timing of 12°CA ↓1.08% for COV of P_{max} at 58% DME and injection timing of 12°CA ↑4.99% for COV of P_{max} at 62% DME and injection timing of 12°CA ↑5.87% for COV of P_{max} at 65% DME and injection timing of 12°CA	[25]
Diesel–DME blends	Various	CRDI engine	↓2.35% for COV of P_{max} at 10% DME and injection timing of 15°CA ↓2.45% for COV of P_{max} at 15% DME and injection timing of 15°CA ↓2.74% for COV of P_{max} at 20–30% DME and injection timing of 15°CA ↓2.93% for COV of P_{max} at 35% DME and injection timing of 15°CA ↓2.25% for COV of P_{max} at 45% DME and injection timing of 15°CA ↓1.96% for COV of P_{max} at 50% DME and injection timing of 15°CA ↓1.86% for COV of P_{max} at 52% DME and injection timing of 15°CA ↑4.4% for COV of P_{max} at 55% DME and injection timing of 15°CA ↑5.09% for COV of P_{max} at 58% DME and injection timing of 15°CA ↑6.16% for COV of P_{max} at 60% DME and injection timing of 15°CA ↑6.85% for COV of P_{max} at 62% DME and injection timing of 15°CA	[25]
Diesel–DME blends	Various	DICI engine	↑0.62–4.75% for COV of IMEP at 0.2–0.5 g/s of DME and BMEP of 0.24 MPa ↑1.49–4.15% for COV of IMEP at 0.3–0.6 g/s of DME and BMEP of 0.36 MPa	[14]
Diesel–DME blends	Various	DICI engine	↑0.1–1.21% for COV of P_{max} at 0.2–0.5 g/s of DME and BMEP of 0.24 MPa ↑0.27–1.15% for COV of P_{max} at 0.3–0.6 g/s of DME and BMEP of 0.36 MPa	[14]
Diesel–DME blends	Various	Diesel–DME DICI engine	↑0.14–1.47% for COV of T_{max} at 0.2–0.5 g/s of DME and BMEP of 0.24 MPa ↑0.38–1.42% for COV of T_{max} at 0.3–0.6 g/s of DME and BMEP of 0.36 MPa	[14]
Diesel–DME blends	Various	DICI engine	↓0.4–2.3% for COV of $(dq/d\theta)_{max}$ at 0.2–0.5 g/s of DME and BMEP of 0.24 MPa ↓1.2–3.2% for COV of $(dq/d\theta)_{max}$ at 0.3–0.6 g/s of DME and BMEP of 0.36 MPa	[14]

Table 2. (Continued) Variations in COV values with using DME

Fuel or blend	DME ratio	Engine type	Variation (%)	Ref
Diesel–DME blends	Various	DICI engine	↓1–4.1% for COV of $(dP/d\theta)_{max}$ at 0.2–0.5 g/s of DME and BMEP of 0.24 MPa ↓1.5–5.8% for COV of $(dP/d\theta)_{max}$ at 0.3–0.6 g/s of DME and BMEP of 0.36 MPa	[14]
80% Diesel–20% Biodiesel blend	80%	CRDI engine	↓0.5%–↑0.05% for COV of P_{max} with single injection and various injection timings	[26]
80% Diesel–20% Biodiesel blend	80%	CRDI engine	↓0.05–0.17% for COV of P_{max} with pilot (multi) injection and various injection timings	[26]
DME–Ethanol blends	Various	CRDI engine	↓0.86%–↑7.81% for COV of IMEP at 5–20% ETH and T_{int} of 20°C ↓0.55%–↑1.34% for COV of IMEP at 5–20% ETH and T_{int} of 40°C ↓0.31%–↑0.08% for COV of IMEP at 5–20% ETH and T_{int} of 60°C	[27]
DME–Ethanol blends	Various	CRDI engine	↓0.91%–↑8.19% for COV of IMEP at 5–20% ETH and ϕ of 0.3 ↓0.74%–1.4% for COV of IMEP at 5–20% ETH and ϕ of 0.34 ↓0.49%–↑0.66% for COV of IMEP at 5–20% ETH and ϕ of 0.38	[27]
DME–Ethanol blends	Various	CRDI engine	↓0.42%–↑4.54% for COV of IMEP at 2.6–33.3% ETH and N of 1000 rpm ↓1.83%–↑2.35% for COV of IMEP at 2.7–27.6% ETH and N of 1500 rpm ↑0.02%–3.63% for COV of IMEP at 2.7–25.9% ETH and N of 1900 rpm	[27]
DME-NH_3 blends	Various	DICI engine	↑2.1%–4.2% for COV of P_{max} at 40% NH_3 and various BMEP ↑2.1%–7.4% for COV of P_{max} at 60% NH_3 and various BMEP	[15]
DME-NH_3 blends	Various	DICI engine	↑0.02%–0.06% for CA of COV of P_{max} at 40% NH_3 and various BMEP ↑0.08%–0.35% for CA of COV of P_{max} at 60% NH_3 and various BMEP	[15]
DME–H_2 blends	Various	CRDI engine	↑0.1%–5.6% for COV of N at 2–16% H_2 ratios ↑0.1%–8.9% for COV of P_{max} at 2–16% H_2 ratios ↑0.05%–8.6% for COV of IMEP at 2–16% H_2 ratios ↑0.2%–9.7% for COV of ROPR at 2–16% H_2 ratios	[28]
Methane-DME blends	Various	PFI engine	↓24.17%–↑65.83% for COV of IMEP at 8.6–13.2% DME ratios	[11]
DME–LPG blends	Various	DICI engine	↓2%–↑38.46% for COV of IMEP at 10–100% DME ratios	[29]
n–BUT–DME blends	Various	DICI engine	↓0.3%–3.7% for COV of IMEP at 100% DME and various IMEP ↓0.46%–2.3% for COV of IMEP at 90% DME and various IMEP ↓0.46%–4.04% for COV of IMEP at 80% DME and various IMEP ↓0.5%–2.71% for COV of IMEP at 70% DME and various IMEP ↓1.96%–↑6.75% for COV of IMEP at 60% DME and various IMEP	[30]
Iso–BUT–DME blends	Various	DICI engine	↓1.1%–↑0.15% for COV of IMEP at 100% DME and various injection timings ↓1.05%–↑0.02% for COV of IMEP at 90% DME and various injection timings ↓0.97%–↑0.11% for COV of IMEP at 80% DME and various injection timings ↑0.08%–0.99% for COV of IMEP at 70% DME and various injection timings	[31]

4 Conclusions

The effects of dimethyl ether on cyclic variations in diesel and HCCI engines are explored in this review. The following conclusions can be summarized from findings.

- It was declared that cyclic variations were the main problem of SI–CAI combustion and significant developments was occurred by DME–PFI combustion. It was determined that COV of (CA05–CA50), COV of (CA50–CA90) and COV of (CA05–CA90) for DME–PFI reduced by 3.5%, 28.3% and 6.4% compared to SI–CAI.
- It was determined that COV of P_{max} for direct injection (DI) combustion was much larger than HCCI combustion and they were 6.1 and 0.66%. It was determined that COV of P_{max} reduced with rising HCCI combustion ratio and then changed little by rising HCCI ratio and engine speed.
- It was determined that rising equivalence radio improved the engine stability especially after equivalence ratio of 2 while COV of IMEP deteriorated beyond CO_2 dilution ratio of 14.5 due to poor combustion. It was declared that lean combustion of DME showed little effect on engine stability and combustion control of DME–HCCI was achieved with charge dilution and spark assistance.
- It was declared that variations in cylinder charge state such as temperature, pressure and composition had a larger impact on COV of CA50 and COV of IMEP during later combustion phasing. It was determined that COV of IMEP and P_{max} were below 5% limit value up to 0.55 equivalence ratio which reflected combustion stability with smooth engine operation in DME fueled HCCI engine.
- It was determined that pure DME generally caused higher cyclic variations than diesel fuel but late injection of DME reduced cyclic variation. However, cyclic variations was usually higher at low engine loads than high engine loads and lean mixtures raised cyclic variations while rich mixtures reduced cyclic variability in DME fueled engine.
- It was determined that COV of $(dq/d\theta)_{max}$ and COV of $(dP/d\theta)_{max}$ reduced but COV of IMEP, COV of P_{max} and COV of T_{max} increased by rising DME premixed ratio during diesel–DME dual fuel operation in HCCI engine. Pilot injection was an effective method for combustion

stability and reduction in cyclic variations for diesel–biodiesel–DME operation.

- It is determined that cycle to cycle variations increased by rising DME in ammonia (NH_3)–DME dual fuel operation compared to pure DME. Similarly, rising ethanol ratio beyond 20% increased cycle to cycle variations in ethanol–DME dual fuel operation.
- It was determined that after than 15% hydrogen addition besides rising load and EGR ratio increased cyclic variations in hydrogen–DME dual fuel operation. It was determined that cyclic variations was reduced to 45% nearly 10% DME ratio but they were increased to 90% at 10.3% DME ratio and they reduced again extremely after than 10.3% DME ratio in methane–DME dual fuel operation.
- It was determined that butane–DME dual fuel operation provided lower cyclic variations than diesel while it caused higher cyclic variations compared to pure DME, but 40% butane ratio was determined as limit in butane–DME dual fuel operation since further butane addition increased extremely cyclic variations. It was also determined that 50% DME ratio increased extremely cyclic variations in DME–LPG dual fuel operation.
- A single cylinder test engine was employed in the most studies on using of DME. Hence, it will be helpful the using multi cylinder engines to generalize the findings on DME for future studies. It is clear that DME operation especially at high ratios raises frequently cyclic variations. Hence, new methods will be required for reduction of cyclic variations for future studies.

Similarity rate:18%

Conflict of interest

The authors declare that there is no conflict of interest.

Nomenclature

Symbols

- ° : Degree
- ϕ : Equivalence ratio
- $(dP/d\theta)_{max}$: maximum pressure rise rate
- $(dq/d\theta)_{max}$: maximum heat release rate
- N : Engine speed
- P_{max} : Maximum cylinder pressure
- r_p : Premixed ratio
- T_{max} : Maximum cylinder temperature
- Abbreviations
- ATDC : After top dead center
- BMEP : Brake mean effective pressure
- BTDC : Before top dead center
- BUT : Butane
- C : Celsius or Centigrade
- CA : Crank angle
- CA05 : Crank angle of 5% burned mass fraction
- CA50 : Crank angle of 50% burned mass fraction
- CA90 : Crank angle of 90% burned mass fraction
- CAI : Controlled auto ignition
- CH_4 : Methane
- CI : Compression ignition
- CO : Carbon mono oxide
- CO_2 : Carbon dioxide
- COV : Coefficient of variation
- CNG : Compressed natural gas
- DI : Direct injection
- DME : Dimethyl ether
- EGR : Exhaust gas recirculation
- ETH : Ethanol
- HCCI : Homogenous charge compression ignition
- HCs : Hydrocarbons
- HTR : High temperature reaction
- ICE : Internal combustion engine
- IMEP : Indicated mean effective pressure
- KI : Knocking intensity
- LNG : Liquefied natural gas
- LPG : Liquefied petroleum gas
- LPM : Liter per minute
- LTR : Low temperature reaction
- MBT : Maximum braking torque
- NH_3 : Ammonia
- NO_x : Nitrogen oxides
- PFI : Port fuel injection
- ROPR : Rate of pressure rise
- SI : Spark ignition
- TDC : Top dead center
- TIVC : Temperature at intake valve close

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