City Research Online

City, University of London Institutional Repository

Citation: Arcoumanis, C., Bae, C., Crookes, R. & Kinoshita, E. (2008). The potential of di-methyl ether (DME) as an alternative fuel for compression-ignition engines: A review. Fuel, 87(7), pp. 1014-1030. doi: 10.1016/j.fuel.2007.06.007

This is the accepted version of the paper.

This version of the publication may differ from the final published version.

Permanent repository link: https://openaccess.city.ac.uk/id/eprint/15549/

Link to published version: https://doi.org/10.1016/j.fuel.2007.06.007

Copyright: City Research Online aims to make research outputs of City, University of London available to a wider audience. Copyright and Moral Rights remain with the author(s) and/or copyright holders. URLs from City Research Online may be freely distributed and linked to.

Reuse: Copies of full items can be used for personal research or study, educational, or not-for-profit purposes without prior permission or charge. Provided that the authors, title and full bibliographic details are credited, a hyperlink and/or URL is given for the original metadata page and the content is not changed in any way.

City Research Online: http://openaccess.city.ac.uk/ publications@city.ac.uk

The potential of di-methyl ether (DME) as an alternative fuel for compression-ignition engines: A review

Constantine Arcoumanis [a,*], Choongsik Bae [b,1], Roy Crookes [c,2], Eiji Kinoshita [d,3]

[a] School of Engineering and Mathematical Sciences, The City University London, Northampton Square, London EC1V 0HB, United Kingdom
[b] Department of Mechanical Engineering, Korea Advanced Institute of Science and Technology, 373-1 Kusong-Dong, Yusong-Gu, Taejon 305-701, Republic of Korea
[c] Department of Engineering, Queen Mary, University of London, Mile End Road, London E1 4NS, United Kingdom
[d] Department of Mechanical Engineering, University of Kagoshima, 1-21-40 Korimoto, Kagoshima 890-0065, Japan

---

Abstract

This paper reviews the properties and application of di-methyl ether (DME) as a candidate fuel for compression-ignition engines. DME is produced by the conversion of various feedstock such as natural gas, coal, oil residues and bio-mass. To determine the technical feasibility of DME, the review compares its key properties with those of diesel fuel that are relevant to this application. DME's diesel engine-compatible properties are its high cetane number and low auto-ignition temperature. In addition, its simple chemical structure and high oxygen content result in soot-free combustion in engines. Fuel injection of DME can be achieved through both conventional mechanical and current common-rail systems but requires slight modification of the standard system to prevent corrosion and overcome low lubricity. The spray characteristics of DME enable its application to compression-ignition engines despite some differences in its properties such as easier evaporation and lower density. Overall, the low particulate matter production of DME provides adequate justification for its consideration as a candidate fuel in compression-ignition engines. Recent research and development shows comparable output performance to a diesel fuel led engine but with lower particulate emissions. NO_x emissions from DME-fuelled engines can meet future regulations with high exhaust gas recirculation in combination with a lean NO_x trap. Although more development work has focused on medium or heavy-duty engines, this paper provides a comprehensive review of the technical feasibility of DME as a candidate fuel for environmentally-friendly compression-ignition engines independent of size or application.

Abbreviations: BSFC, brake specific fuel consumption; BTX, benzene, toluene and xylene; C–C, carbon-to-carbon; CFC's, chloro-fluoro-carbons; C:H, carbon-to-hydrogen ratio; C_2H_2, acetylene; C_2H_4, ethylene; C_3H_3, proparagyl; CH_4, methane; CH_2O, formaldehyde; CI, compression-ignition; CNG, compressed natural gas; CO_2, carbon dioxide; CO, carbon monoxide; CR, compression ratio; DI, direct-injection; DMC, di-methyl carbonate; DME, di-methyl ether; EGR, exhaust gas recirculation; FIE, fuel-injection equipment; H_2, hydrogen; HC, hydrocarbon; HCCI, homogeneous charge compression-ignition; HDV, heavy-duty vehicle; IMEP, indicated mean effective pressure; LDV, light-duty vehicle; LNT, lean NO_x trap; LPG, liquefied petroleum gas; NMHC, non-methane hydrocarbon; NO_x, nitrogen oxide; P_a, ambient pressure; P_{in}, fuel- injection pressure in common-rail; PAH, polycyclic aromatic hydrocarbon; PM, particulate matter; PTFE, polytetrafluoroethylene; SI, spark-ignition; SOF, soluble organic fraction; SO_2, sulfur dioxides; SPI, sustainable process index; syngas, synthetic gas; ULEV, ultra low emission vehicle.

[*] Corresponding author. Tel.: +44 207 040 0113; fax: +44 207 040 8101.

E-mail addresses: c.arcoumanis@city.ac.uk (C. Arcoumanis), csbae@kaist.ac.kr (C. Bae), r.j.crookes@qmul.ac.uk (R. Crookes), kinoshit@mech.kagoshima-u.ac.jp (E. Kinoshita).

[1] Tel.: +82 42 869 3044; fax: +82 42 869 5044.

[2] Tel.: +44 207 882 5270; fax: +44 208 983 1007.

[3] Tel.: +81 99 285 8268; fax: +81 99 250 3181.

Contents

1. Introduction

Di-methyl ether (DME) is a liquified gas with handling characteristics similar to those of liquified petroleum gas (LPG) [1]. It can be produced from a variety of feed-stock such as natural gas, crude oil, residual oil, coal, waste products and bio-mass. Many investigations have been carried out on DME to determine its suitability for use as a fuel in diesel-cycle engines [1,2].

DME has the appearance of an excellent, efficient alternative fuel for use in a diesel engine, with almost smoke-free combustion. This is not only because of its low auto-ignition temperature and its almost instantaneous vapourization when injected into the cylinder, but also because of its high oxygen content (around 35% by mass) and the absence of C–C bonds in the molecular structure [1,2]. With a properly designed DME injection and combustion system, nitrogen oxides (NO_x) emissions can also meet ultra low emission vehicle (ULEV) limits [3]. The well-to-wheels energy efficiency of heavy- and light-duty DME-fuelled vehicles is projected to be 22.5% and 19%, respectively [4]. This is comparable to LPG and compressed natural gas (CNG) fuelled vehicles but less than the highest energy efficiency of 26% in direct-injection (DI) diesel fuelled vehicles [4]. On the other hand, the well-to-wheels carbon dioxide (CO_2) emissions of a DME-fuelled vehicle is comparable to those using DI diesel or CNG fuelled engines [4]. However, an oxidation catalyst would be necessary to meet ULEV carbon monoxide (CO) and hydrocarbon (HC) emission limits [5].

DME was also found to be an excellent gas turbine fuel with emission properties comparable to natural gas [6]. DME-fuelled turbine also allows to achieve a significant performance improvement through thermochemical recuperation with 44% higher power output and an 8% decrease of the specific CO_2 emissions compared to the present plant [7]. However, DME is not a suitable fuel for spark-ignition (SI) engines due to its high cetane number, though the burning velocity is similar to hydrocarbon fuels [8]; the easily-induced knock would limit the operation of SI engines.

Overall, the key to the development of a dedicated low-emissions, DME-fuelled engine is the performance and durability of its fuel-injection system [1,2]. In this review, the features considered to be most important in developing the potential for widespread utilization of DME to reciprocating engines are: the production and properties of DME, the fuel-injection system and the spray characteristics which contribute to the engine's performance and exhaust emissions.

2. DME properties

The key properties of DME and diesel fuel are shown in Table 1 [9,10]. In addition to the advantages above, it has a low carbon-to-hydrogen ratio (C:H) with a chemical formula of $CH_3–O–CH_3$ as shown in Fig. 1. DME in a gaseous state is invisible under standard atmospheric conditions (0.1 MPa at 298 K). When it is pressurized above 0.5 MPa, it condenses to the liquid phase. Gaseous DME is denser than air while liquid DME has a density two thirds that of water. The vapour pressure is similar to that of LPG and requires the same handling and storage precautions. It dissolves in water up to 6% by mass. However, it is not compatible with most elastomers due to its corrosiveness, so that careful selection of materials is necessary to prevent deterioration of seals after prolonged exposure to DME. The sealing of DME filled vessels, for example, can be achieved with inert material, such as polytetrafluoroethylene (PTFE).

Table 1
Properties of DME and diesel fuel

Property (unit/condition)	Unit	DME	Diesel fuel
Chemical structure		$CH_3–O–CH_3$	–
Molar mass	g/mol	46	170
Carbon content	mass%	52.2	86
Hydrogen content	mass%	13	14
Oxygen content	mass%	34.8	0
Carbon-to-hydrogen ratio		0.337	0.516
Critical temperature	K	400	708
Critical pressure	MPa	5.37	3.00[a]
Critical density	kg/m^3	259	–
Liquid density	kg/m^3	667	831
Relative gas density (air = 1)		1.59	–
Cetane number		>55	40–50
Auto-ignition temperature	K	508	523
Stoichiometric air/fuel mass ratio		9.0	14.6
Boiling point at 1 atm	K	248.1	450–643
Enthalpy of vapourization	kJ/kg	467.13	300
Lower heating value	MJ/kg	27.6	42.5
Gaseous specific heat capacity	kJ/kg K	2.99	1.7
Ignition limits	vol% in air	3.4/18.6	0.6/6.5
Modulus of elasticity	N/m^2	6.37E+08	14.86E+08
Kinematic viscosity of liquid	cSt	<.1	3
Surface tension (at 298 K)	N/m	0.012	0.027
Vapour pressure (at 298 K)	kPa	530	$\ll 10$

[a] Estimated on the basis of the equivalent chemical formula.

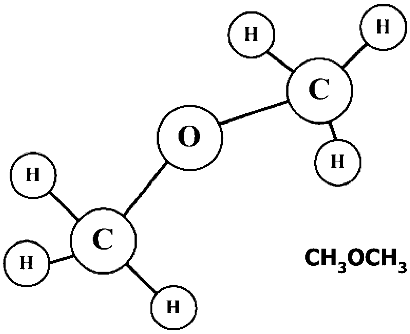

Fig. 1. Structure of di-methyl ether (DME).

DME is non-toxic and environmentally benign. At low mole fractions (a few percent by volume) the gas has hardly any odour and causes no negative health effects. Even at high mole fractions (>10% by volume) there is no effect on human health except narcotic effects after long exposure and it may be recognized by the odour. DME displays a visible blue flame when burning over a wide range of air–fuel ratios, similar to natural gas, which is an important safety characteristic. Nevertheless, the operation of DME combustion system needs the adoption of rigorous procedures for safe operation due to the wide flammability limits.

The importance of the properties of DME can be identified by examining its advantages and disadvantages as a candidate fuel for replacing diesel fuel in compression-ignition (CI) engines.

2.1. Advantages

(i) *High oxygen content*: Together with the absence of any C–C bonds it is responsible for its smokeless combustion; low formation and high oxidation rates of particulates would therefore be expected under compression-ignition (CI) engine operation.

(ii) *Low boiling point*: Leads to quick evaporation when a liquid-phase DME spray is injected into the engine cylinder.

(iii) *High cetane number*: Results from the low auto-ignition temperature and almost instantaneous vapourization. Teng et al. [11] examined the high cetane number (>55) of DME from the viewpoint of its thermochemical characteristics. Its critical temperature, 400 K (127 °C) is lower than the compressed air temperature at the later stages of the compression stroke, which allows the DME injected into the cylinder to evaporate immediately. When the temperature of DME is higher than 400 K, it becomes superheated vapour and no evaporation is associated with the mixing. The chain combustion reaction, however, may be initiated by either (or both) of two competing path ways [12,13]:

(a) C–O bond fission (pyrolysis mechanism):

$$CH_3OCH_3 = CH_3O + CH_3 \tag{1}$$

(b) Hydrogen abstraction (oxidation mechanism):

$$4CH_3OCH_3 + O_2 = 4CH_3OCH_2 + 2H_2O \quad (2)$$

$$CH_3OCH_2 = CH_2O + CH_3 \quad (3)$$

The C–O bond energy is smaller than that of the C–H bond (414 kJ/mol [14]) and the distortion of the C–O bonds in the DME molecule weakens the bonding strength; thus, the C–O bond breaks easier than the C–H bond. Therefore, the pyrolysis mechanism may be more able to start the chain reaction at relatively low temperatures, which would result in the low auto-ignition temperature. The physical delay of DME is thus much shorter than that for many conventional fuels, resulting in a shorter total ignition delay and a higher cetane number.

2.2. Disadvantages

(i) *Low combustion enthalpy*: Lower than that of diesel fuel due to the oxygen content of the molecules, which necessitates a larger injected volume and longer injection period for DME in order to deliver the same amount of energy to that provided by diesel.
(ii) *Low viscosity*: Lower than that of diesel fuel, causing leakage from the fuel supply system which relies on small clearances for sealing. Its lower lubricity characteristics can cause intensified surface wear of moving parts within the fuel-injection system which requires attention.
(iii) *Low modulus of elasticity*: Lower than that of diesel fuel, with the difference being greatest at lower pressures. Thus, the compressibility of DME is usually four to six times higher than that of diesel in a closed system and the compression work of the fuel pump for DME is greater than it is for diesel fuel by up to 10% in an open system [10].

3. Fuel specification and production

3.1. Fuel specification of DME

Recommendations have been made concerning permissible levels of impurities present in DME [2]. Most impurities are recommended to be negligible such that the proportions in mass are less than 0.05% for methanol, higher alcohols, higher ethers and ketones. Water is recommended to be less than 0.01% while methyl ethyl ether should be less than 0.2%. An odorant such as ethylmercaptane can be included up to 20 ppm.

3.2. DME production

DME has been produced world wide in quantities of 100,000–150,000 tonnes per annum from 1996 or earlier [1,2], and is used as a propellant for spray cans for cosmetics instead of chloro-fluoro-carbons (CFC's). It can be produced from carbonaceous feedstock, both from fossil fuels like natural gas and renewable sources like wood. Its production method most closely resembles that of methanol.

DME production methods are of the following type [1,2,15]

(a) De-hydrogenation of methanol
(b) Direct conversion from synthesis gas (syngas)

which have essentially similar features.

Currently, most DME is produced by de-hydrogenation of methanol from syngas which is a mixture of hydrogen (H_2), carbon monoxide (CO) and carbon dioxide (CO_2), but may also contain impurities like methane (CH_4).

The direct conversion method is the simultaneous production of DME and methanol from syngas using appropriate catalysts. The first step of direct DME production is the conversion of the feedstock to syngas, most common by steam reforming of natural gas and for coal, oil residues and bio-mass by partial oxidation through gasification with pure oxygen. The second step for the direct route is via methanol synthesis using a copper-based catalyst while the third step is the de-hydrogenation of methanol to DME using alumina- or zeolite-based catalysts. With the direct conversion of syngas to DME, the second and third steps can occur simultaneously in one reactor using appropriate catalysts (different to those described above). The final step is the purification of the raw product, which may also contain some methanol and water.

Although it is possible to produce DME from both fossil and renewable sources, bio-DME is very attractive; however, producing DME from fossil sources is the only viable route at present from a cost perspective since bio-DME is very expensive relative to diesel fuel. Because natural gas at remote locations will, more likely, be the lowest-cost feedstock [4], the most cost-effective route for DME production is by direct synthesis of syngas (possibly with methanol as an intermediate step or co-product), from natural gas. From both the cost and product-yield points of view, the synthesis of DME directly from syngas may be the preferred route for large scale production. It has been claimed that DME produced from natural gas wells in Southeast Asia or the Middle East is economical enough [16] to be supplied at a lower price than diesel by up to 40% [17].

Increasing the purity requirements of the final product will increase production costs of DME. Nevertheless, it is still considered that DME, with its very low percentages of methanol and water, should not exhibit problems when used in internal combustion engines.

Flexible plants can be set-up to produce methanol and DME in different ratios, depending on the relative demand, so that a large production capacity can be realized in a limited amount of time. Converting a methanol plant to produce DME, besides building new DME facilities, is an attractive option for DME production. DME can also be produced in a coal-fired power plant and in plants close to urban or industrial areas, where waste-heat from the production process can be used for heating or generating hot water.

4. Fuel-injection system

The features of a fuel-injection system for DME and the characteristics of the flow in the injector nozzle are different from those for diesel fuel due to the different physical and chemical properties. In what follows these features and a suggested appropriate fuel-injection system for DME are summarized.

4.1. Features of the fuel-injection system

(i) *Closed pressurized fuel system*: Because of the low boiling point of DME (248 K = −25 °C), it is a gas under standard atmospheric conditions and therefore must be pressurized in a fuel system, including a storage tank, and handled like a liquefied gas. Thus, the low boiling point of DME necessitates a closed pressurized fuel system. The vapour pressure of DME, roughly the same as LPG, demands the same kind of handling and storage considerations as for LPG.

(ii) *Cavitation in the fuel-injection system*: The high vapour pressure of DME means that cavitation, which prevents stable fuel-injection operation, is unavoidable in the fuel-injection equipment [2]. Although DME is kept in the liquid phase under atmospheric temperature conditions, when pressurized to above 0.5 MPa the fuel-supply pressure (the feed pressure from the storage tank to the fuel pump) must be between 1.2 and 3 MPa to avoid cavitation in the fuel line. This is the case not only because of the higher temperatures during engine operation, when the saturated vapour pressure of DME is higher, but also due to dynamic flow effects leading to the formation of vapour zones in the fuel line [12].

(iii) *Low injection pressure*: DME gasifies immediately during injection, due to its low boiling point, even though it is injected as a liquid. Therefore, the high fuel injection pressures, such as 50–150 MPa, used in modern diesel injection systems are not required for DME and there is a consensus that 20–30 MPa is sufficient as long as the required amount of DME can be supplied during the corresponding injection period even at high engine speeds [18].

(iv) *Leakage*: Because of its low viscosity, currently available fuel-injection systems are not suitable for DME due to leakage problems. Even at atmospheric conditions the leakage of DME can be significant in regions of small clearance such as between the plunger and barrel of a rotary-type fuel-injection system. Leakage rates of DME along the plungers, of up to 40–50% of the fuel, have been reported in the literature [12]. In heavy-duty engines, leakage along the plungers is more serious than in light-duty engines where the drive mechanism is engine-oil lubricated. To prevent this, most sealing material made of conventional elastomers should be replaced with anticorrosive material such as Teflon-coated o-rings or PTFE based high-tension sealing components, if there is an expectation for DME use in fuel-injection equipment to reach mass production [19].

(v) *Low lubricity*: The lower lubricity of DME than that of diesel fuel leads to wear problems. The lubricity can be enhanced with up to 2000 ppm of a lubricity additive; examples include Lubrizol (1000 ppm) [12], Hitec 560 (100 ppm) [9] and Infineum R655 (500 ppm) [15]. Even diesel fuel, di-methyl carbonate (DMC) and fatty acids have been suggested as viscosity enhancing additives [20,21]. However, it might be possible to adapt the materials of the bearing surfaces and plunger/barrel interfaces to the low lubricity of DME [1]. It is expected that the experience gained in the past with the use of methanol in fuel-injection systems will prove useful in developing a leakage-free and durable injection system for DME operating at injection pressures not exceeding 30 MPa [1], while current common-rail systems for diesel fuel could be adapted for DME with the use of lubricity additives at injection pressures up to 55 MPa [19].

(vi) *Long injection period*: The low liquid density and low calorific value require a higher volume of DME to be injected into the cylinder, compared with that for diesel fuel. In particular, 1.8 times the volume of diesel fuel is needed (to supply the same amount of energy) which necessitates a longer injection period and advanced injection timing.

(vii) *Large compression work*: The compression work of the fuel pump for DME will always be larger than that for diesel fuel, since DME has lower density and higher compressibility than diesel [11].

(viii) *Sealing material*: DME is not compatible with most elastomers and can chemically attack some commonly used sealing materials and other plastic components, raising questions about the durability of injection systems handling DME. A careful selection of sealing materials is necessary to prevent deterioration after prolonged exposure to DME. Sealing of DME-filled storage vessels and supply lines can, for example, be achieved with PTFE (as mentioned in Section 2).

4.2. Flow phenomena in the injector nozzle

The physical properties of DME give rise to different characteristics of the needle lift of the injector, of the pressure oscillation in the fuel line, and of the flow phenomena in the injector nozzle compared to those for diesel fuel. The flow in the nozzle is of prime importance with respect to the fuel-spray development and mixture preparation in the combustion chamber [9]. In particular flow cavitation in the nozzle orifice was found to play a crucial role in the formation of the emerging from the nozzle spray.

(i) *Needle lift behaviour*: In view of DME's lower modulus of elasticity compared to diesel, the needle lift profile of the conventional injector is expected to be different. With the resulting higher compressibility (relative to diesel) the nozzle opening stage is slower in the case of DME [22].

Kajitani et al. [23] suggested that the feed pressure is an important factor affecting the nozzle-opening time and that the nozzle opens earlier when the feed pressure is higher. With DME, the feed pressure in the fuel line is required to supply the fuel from the storage tank to the fuel pump. Thus due to the higher feed pressure of DME in the fuel line, the start of nozzle opening should be earlier for DME than for diesel fuel. Considerable residual needle bouncing was also observed with the low nozzle-opening pressure (6.86 MPa), because of the reduced spring preload on the nozzle needle. This is in contrast to the relatively well-defined injection behaviour at higher opening pressures (8.82 MPa) and, as a result, it has become necessary to optimize the injector specifications in order to prevent needle bouncing.

(ii) *Pressure drop*: In the case of DME, there is a considerable pressure drop across the valve seat due to the high fuel compressibility [22]. A loss of pressure across the valve seat reduces the pressure drop across the nozzle holes which, in turn, reduces the velocity of the injected spray.

(iii) *Pressure oscillations and residual pressure in the injection line*: After injection, DME exhibits larger pressure oscillations and a higher residual pressure in the injection line between the fuel pump and the injector, compared to diesel fuel [10]. The reason for the larger pressure oscillations with DME is believed to be its higher compressibility compared to diesel. A larger amount of energy is stored in the compressed DME than in the diesel fuel during the injection process which gives rise to larger pressure oscillations in the injection line; these slowly decay between injections giving rise to the high residual pressure.

(iv) *Flow phenomena and flow rate in the nozzle orifice*: Kapus and Ofner [9], using model orifice experiments and numerical methods, described the flow in the model orifice for DME to be in the gas–liquid two-phase regime. In contrast, the same analysis showed diesel fuel to exhibit both liquid single-phase and gas–liquid two-phase flow characteristics.

For a high pressure drop between the inlet and the outlet edge of the orifice, gas bubbles emerging from cavitating locations with high fluid velocities and low static pressures (mainly at the inlet edges of the contraction) are mixed with the liquid to form an apparently homogeneous zone of gas and liquid. Once this zone covers the whole cross-sectional area of the orifice, the flow characteristics become independent of the back pressure. This means that, for a given inlet pressure, the flow rate can not be increased further by lowering the back pressure. In this case, complex gas-dynamic processes lead to an increase in the exit angle of the discharging spray.

4.3. DME fuel-injection system options

The two methods of developing a fuel-injection system for DME involve (1) modification of conventional pump–pipe–nozzle fuel-injection systems or (2) new system with a variable displacement pump or common-rail. In light-duty vehicles, it is common to use either dual-fuel or retrofit engines by converting existing pump–pipe–nozzle systems. However, in view of the high reliability requirements, this method is difficult to use and it is uncertain whether it can be applied in practice to light-duty vehicles [1,2]. State of the art common-rail diesel systems are also promising for implementation in passenger car engines operating with DME [18,22,23]; this is because these systems may require no more than 8 MPa injection pressure to operate satisfactorily. In heavy-duty vehicles, however, new dedicated DME fuel-injection systems are under development based on the common-rail concept. For demonstration purposes it is probably more cost effective to convert conventional fuel injection equipment [1]. On the other hand, Tsuchiya and Sato [18] found that an in-line fuel-injection pump (Jerk type) with conventional spring-loaded needle injector was more suitable than a common-rail system for heavy-duty DME engines due to the maximum injection pressure characteristics.

In order to meet future emissions legislation for vehicles, it is necessary to develop a fuel-injection system that is able to control injection timing, injection duration, quantity of injection and rate-shaping in combination with EGR systems. Injection rate-shaping which reduces the rate of injection during the ignition delay period, can decrease the rate of heat release just after ignition ('premixed' combustion) as well as the peak firing pressure and temperature in the cylinder, thus leading to NO_x and noise reduction [5]. High EGR rates can also be employed due to the very low soot-producing capability of DME. It is considered that by optimization of the injection system at high EGR rates, NO_x and PM can both be reduced simultaneously. For NO_x reduction using a catalyst, post-injection after the main injection event is crucial for controlling the exhaust gas temperature and composition. This is important for new generation diesel engines equipped with storage type De-NO_x catalysts which necessitates development of flexible multi-injection systems capable of variable post-injection patterns.

4.3.1. Common-rail system

A purpose-built common-rail fuel-injection system is probably the best solution for a dedicated DME engine [1]. The independence of injection pressure on engine speed gives a significant boost to the DI engine irrespective of the fuel used and allows optimization of injection timing, injection duration, rate of injection, and rate-shaping to be achieved. AVL has developed a DME common-rail fuel-injection system with solenoid valves (which are three-way-valves) specifically for large truck and bus engines [24]. This system, which has leakage control with double seals and a purge system, has shown promising results for engine performance, exhaust emissions and safety in operation.

In order to accurately measure the rated injection of DME and to investigate the possibility of rate-shaping of the injection period, Ikeda et al. [25] connected an injection rate meter (based on the Zeuch method) to a common-rail fuel-injection system. It was shown that, if the dimensions of the measuring system are modified appropriately, the injection rate meter can be applied to the common-rail fuel injection system to characterize the DME injection pattern. The maximum injected quantity seemed to depend on the cavitation factor (number) in the case of DME. The cavitation factor was defined as the pressure drop of the nozzle normalized by the inlet pressure minus the vapour pressure [9,25]. They also suggested that multiple fuel-injection, including pre- and post-injection, was only possible by the use of a common-rail fuel-injection system. Ikeda et al. [25] calibrated and assessed the accuracy of the injection rate meter with multiple injection rates and quantities, including five-stage injection, and showed that the common-rail injection system was able to realize the required programmed injection accurately. Yu and Bae [19] also utilized commercial diesel common-rail and solenoid-driven diesel injectors for DME injection tests with the help of pneumatic pumps to compress DME up to 55 MPa. They measured DME injection rates from the solenoid injector using the so-called Bosch tube method based on the hydraulic pulse theory.

4.4. *A liquid DME fuel tank*

Although the DME tank could be designed similarly to those used for LPG, a low-pressure pump and cooling system would be needed between the DME fuel tank and the injection pump, further complicating the fuel design. To simplify the McCandless et al. [26] developed a liquid DME fuel tank by considering the thermodynamic behaviour of DME. The fuel tank, of cast aluminum with a capacity of 40 l, was essentially a thermodynamic pump containing two fluids: liquid DME and a vapour–liquid mixture of driving fluid (mainly propane) with a diaphragm separating the two fluids. The propane pressurized the DME into a sub-cooled-liquid state and functioned as a driving fluid that pumped the liquid DME from the tank to the injection pump by means of its vapour pressure.

5. Spray characteristics and combustion

5.1. *Spray characteristics*

When it is injected into the engine cylinder, DME vapourizes immediately due to its low boiling point at atmospheric pressure; as a result of its low critical temperature and critical pressure, the behaviour of liquid DME is sensitive to the in-cylinder thermo-dynamic conditions [11]. Investigations into the spray characteristics of DME have been carried out by various groups [9,10,19,23,27,28]. Wakai et al. [27] performed experiments using a Schlieren optical system on a DME spray injected by means of a diesel injector into a vessel in the low- to middle-pressure range of 0.1–1.5 MPa. Sorenson et al. [10] studied high-pressure liquid DME injection into a nitrogen environment at temperatures <413 K (40 °C) and pressures of 1.5, 2.5, 4.0 and 5.5 MPa, using a lubricity additive to the fuel. Teng et al. [11] analysed the hydrodynamic behaviour of a DME spray in a DI diesel engine cylinder by referring to the investigations carried out by Glensvig et al. [28] and Sorenson et al. [10]. Kajitani et al. [23] observed sprays of DME at atmospheric pressure while Yu and Bae [19] characterized the injection rate and spray development with a common-rail system operating at pressures up to 55 MPa. The experimental conditions of these investigations are shown in Table 2 and the spray behaviour of DME can be classified according to the ambient pressure P_a as explained and summarized below.

5.1.1. *P_a < saturated vapour pressure of DME or low ambient pressure (0.1 MPa) [19,23,27]*

When DME is injected into a lower ambient pressure than its saturated vapour pressure, the pressure of the DME spray falls below the saturated vapour pressure and flash boiling occurs. Several rugged clusters have been observed at the peripheral region of the spray, when viewed on a relatively large scale, considered to be the vapour cloud. The tip penetration of the DME spray is less than or almost the same as that of a diesel fuel spray but a wider spray angle is observed with DME compared to diesel. The

Table 2
Experimental conditions of DME spray investigations

Investigators	Wakai et al. [27]	Sorenson et al. [10], Glensvig et al. [28], Teng et al. [9]	Yu and Bae [19]
Injection system	In-line type Bosch injection pump, single-hole-nozzle	In-line pump, single-hole-nozzle and pintle-nozzle	Common-rail, solenoid-driven injector five-hole mini-sac nozzle
Nozzle diameter (mm)	0.2	0.513	0.168
Valve opening – or common-rail pressure (MPa)	9.8	13 at $P_a = 1.5$ and 2.5, 20 at $P_a = 4.0$ and 5.0	25, 40 and 55 (common-rail pressure)
Ambient pressure (MPa)	0.1 and 1.5	1.5, 2.5, 4.0 and 5.5	0.1 and 3.0
Ambient temperature (K) (°C)	293 K (20 °C)	411 K (38 °C)	<413 K (40 °C)

increased spread of the DME spray (wider spray angle and larger spray volume) is thought to be mainly due to the enhancement of fuel atomization by flash boiling and faster evaporation of DME.

5.1.2. Saturated vapour pressure of DME < P_a < critical pressure of DME [10,11,27]

In this case the behaviour of the DME spray is similar to that of diesel fuel in terms of general shape. The tip penetration is less than or almost the same as that of the diesel fuel spray and the spray angle is greater than that of diesel fuel. The evaporation of the DME spray is more rapid than that of the diesel fuel spray and enhanced evaporation rates have been observed at higher pressures. Unlike a diesel fuel spray, the boundary of the DME spray seems to be irregular and this irregularity increases with increasing cylinder pressure. At cylinder pressures near the DME critical pressure, breakup of the DME spray occurs unlike the behaviour of the diesel fuel spray.

5.1.3. P_a > critical pressure of DME [10,11]

When the pressure of DME droplets enters the supercritical region, the spray turns into a miscible turbulent jet (a single-phase phenomenon). In this case, the liquid–vapour interface disappears and the droplets spread to the neighborhood in an explosive way. The spray angle increases as the ambient pressure decreases, whereas a diesel fuel spray demonstrates the opposite tendency. The transition from a spray to a miscible turbulent jet may be encountered during the spray development. If the size of the droplets is small enough, the droplets may completely vaporize before reaching the spray tip due to the temperature distribution in the spray. In this case, during the spray development, the dispersing droplets may dominate the spray hydrodynamics in the early stages, while air entrainment may considerably affect the later stages of the spray evolution.

Wakai et al. [27] suggested that, in the low-pressure regions in the nozzle sac chamber and in the holes, cavitation in DME occurs more easily than in the diesel fuel. It was considered that such cavitation formation in the form of vapour bubbles can enhance the atomization of the DME spray and influence the spray characteristics. Kapus and Ofner [9] also suggested that complex gas–liquid two-phase dynamic flow with cavitation, may occur for DME which increases the exit angle of the discharging spray. The effect of cavitation on the enhancement of atomization and on the spray global characteristics for DME is still unclear.

Teng and McCandless [29] conducted a comparative study of the characteristics of diesel fuel and DME sprays under realistic engine conditions on the basis of momentum conservation. Their analysis revealed that the DME spray in the diesel combustion system may not develop as well as that of the diesel fuel at high engine loads and speeds due to the longer injection duration and increased spray angle. Being less dense and more volatile than diesel, DME has a higher gas-to-liquid density ratio in the spray, which also tends to increase the spray angle. Because of the large spray angle, the fuel-rich boundaries (due to the rapid fuel evaporation) of the DME sprays from the different nozzle holes tend to overlap under long injection duration and create strong swirl which is unfavourable for complete combustion. The characteristics of the DME spray suggest that the DME may need a larger combustion chamber for better mixing and combustion than diesel fuel. Fig. 2 shows typical DME spray images compared to diesel spray.

5.2. Spray combustion of DME [26–29]

Wakai et al. [27] also performed experiments on the spray combustion characteristics of DME, using a constant volume vessel at 3.1 MPa and 823 K, and found that the DME spray was ignited at 2.6 ms after the start of injection. The first ignition occured around the middle portion of the spray, and the flame spread mainly in the downstream direction. The flame was non-luminous and light emission from the flame was very weak. Under lower ambient pressures (2.1 and 1.1 MPa), the ignition delay of the DME spray was shorter than that of a diesel fuel spray, whereas it was almost the same as the diesel fuel at higher ambient pressures (4.1 and 3.1 MPa). At an ambient pressure of 1.1 MPa, the DME spray could be ignited at temperatures >770 K, contrary to the diesel fuel which could not be ignited.

5.3. Numerical simulation of DME spray combustion

Numerical simulations of spray combustion for DME were carried out by Golovitchev et al. [30,31]. Their numerical model was based on the KIVA-3 code, with modified sub-models accounting for spray atomization, detailed oxidation chemistry, soot formation and the effects of chemistry-turbulence interaction. The leading role of the methyl formate reaction path was recognized in the ignition process and the detailed chemical mechanism (43 species and 222 reactions), including soot formation, was used. The predictive capability of the model was evaluated through comparison of numerical and experimental data on liquid and vapour penetration and on ignition in a constant-volume chamber under diesel-like conditions for both n-heptane and DME. The predictions of the ignition point and delay time were found to be in agreement with the experimental data. The calculations showed ignition to start after evaporation of a major portion of the fuel which allowed an auto-ignition wave to propagate in a partially premixed mixture. The flame propagation process exhibited oscillatory characteristics where the auto-ignition kernels moved forwards and backwards, due to pressure waves and the cooling effect of the evaporating fuel. A high-temperature zone became visible in the central part of the spray. Simulation of an engine fuelled with neat DME was also carried out using the same CFD model. The DME fuel spray was rapidly atomised and vapourised. In the initial stage of

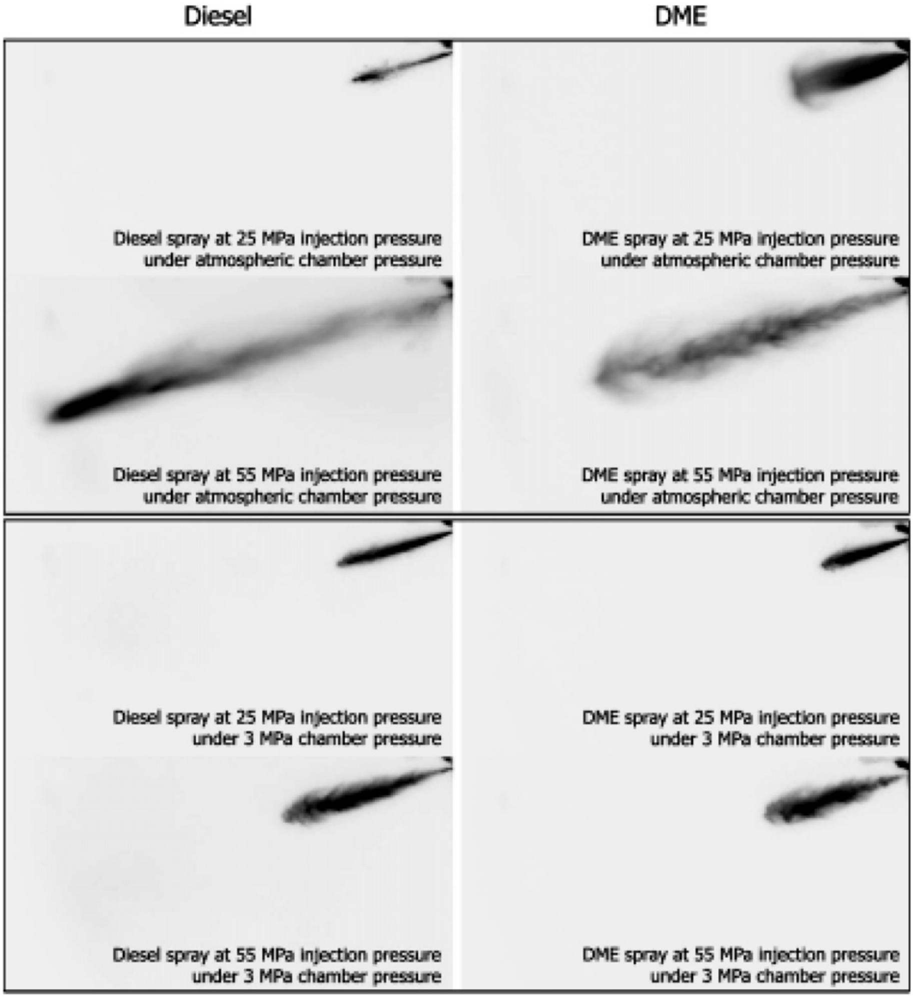

Fig. 2. DME sprays at various conditions compared to diesel sprays [18]; atmospheric (10.1 MPa) and high pressure (3 MPa) ambient conditions with fuel-injection pressures at 25 and 55 MPa.

ignition the heat release zone appeared to be associated with a diffusion flame but later on the flame propagated in a partially premixed mixture. Under these conditions, the ignition mechanism showed the well-known two-step characteristics, with the maximum heat release taking place after TDC. Better performance and peak combustion pressures were demonstrated at lower injection pressures and reduced injection velocities (<170 m/s). The validation data for the CFD model was given by Zhao et. al [32]. Following this it was argue that the application of neat DME would require the development of low-pressure fuel-injection equipment, properly matched to the combustion system.

6. Engine performance and emissions

6.1. Exhaust emissions and fuel consumption

In previous investigations [3,5,9], it has been demonstrated that emissions of particulate matter (PM), NO_x, and combustion noise from compression-ignition engines are significantly lower when DME is used instead of diesel fuel, while DME fuel consumption can be kept close to the DI diesel value on an energy basis. Fig. 3 illustrates some of the most recent data on brake specific fuel consumption (BSFC), NO_x and CO_2 specific emissions for DME compared with diesel fuel from a six-cylinder 7 l turbocharged/intercooled heavy-duty diesel engine operating in the Japanese JE-05 test mode [18]. The data demonstrates the reduction in NO_x and CO_2 emissions that can be achieved with DME at a similar fuel economy. Smoke (PM) and THC together with NO_x are shown in Fig. 4 from experiments obtained in a small single-cylinder engine of 638 cm^3 [33]. The data in Figs. 3 and 4 show the reduction in NO_x and particulate emissions that can be achieved with DME at equal power and torque and equal fuel economy. It can be argued that any particulate emissions come from the lubricant oil and not from DME. Because ignition delay is shorter with DME, NO_x emissions should, in principle, be lower than those for diesel fuel.

6.1.1. Particulate matter (PM)

It is well known that soot is formed in fuel-rich regions under high temperature conditions. The precursors of soot

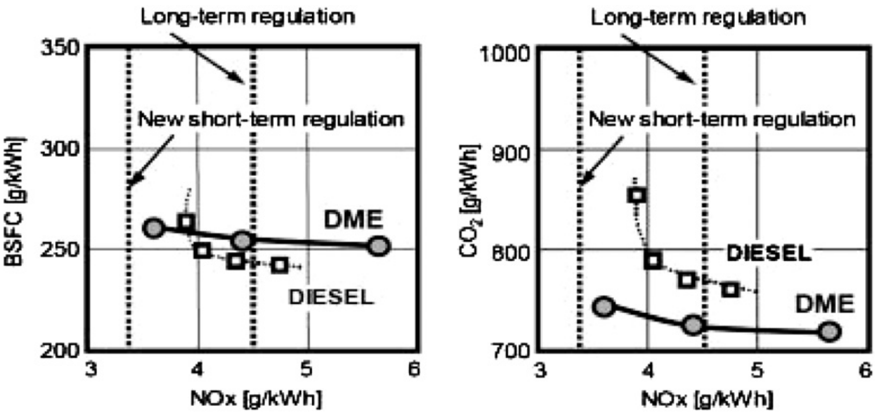

Fig. 3. Fuel consumption (BSFC), NO_x and CO_2 emissions for DME compared to diesel from a six-cylinder 7 l turbocharged/intercooled heavy-duty diesel engine operating in Japanese D13 mode [18].

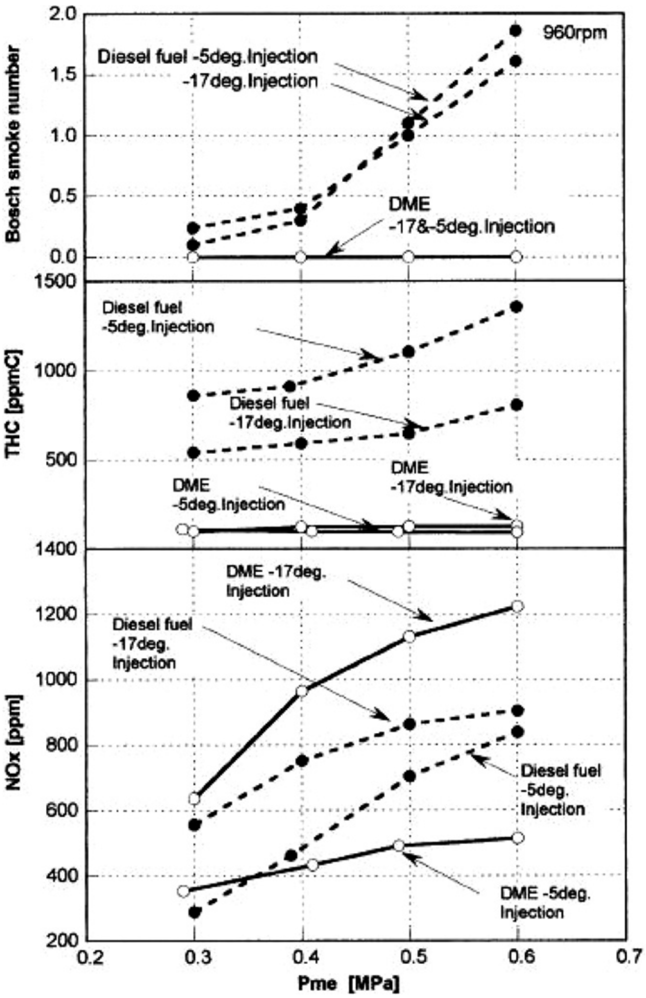

Fig. 4. Smoke, THC and NO_x emission for DME and diesel from a single cylinder of 638 cm^3 DI compression-ignition engine [33].

are unsaturated hydrocarbons such as acetylene (C_2H_2), ethylene (C_2H_4), and propargyl (C_3H_3), found in diesel combustion products [34]. The proportion of fuel carbon forming soot precursors has been found to decrease with increased oxygen content in the fuel [34] and with decreased number of C–C bonds [35].

The effect of oxygen content in the fuel on the PM from a diesel engine was investigated experimentally by Miyamoto et al. [36]. The PM emissions reduced from the value for conventional diesel fuel with no oxygen content to less than 2% of that value when the oxygen content of the fuel was 25–30% by mass. Curran et al. [37] showed the influence of the oxygen content of fuels on the production of soot precursors using a detailed chemical kinetic model; the calculated values of the soot precursors decreased with increased oxygen content in the fuel. Therefore, it can be concluded that the soot formed in DME combustion should be almost zero at an oxygen content of 35% and no C–C bonds. This implies that DME-fuelled engines would not need a particulate filter in the aftertreatment system. Sidhu et al. found in [38] that DME yields much lower particulates (0.026%) than other fuels under compression-ignition conditions such as diesel (0.51%), biodiesel (0.52%) and CNG (0.30%). A DME-fuelled engine was also found to have in the exhaust gas higher (71%) soluble organic fraction (SOF) than in a diesel (20%) of total carbon.

6.1.2. *NO_x*

Comparative values of NO_x emissions from DME CI engines and those from diesel fuel seem to vary depending on the engine conditions and the fuel supply system, according to various sources in the literature. NO_x was found to be lower with DME than with diesel fuel [3,9], which was attributed to the shorter ignition delay for DME than for diesel, the smaller amount of fuel injected during the ignition delay period and the smaller amount of fuel burned during the pre-mixed burning phase. The initial enthalpy change was thus lower, resulting in a reduction of the peak combustion temperature. However, it is possible that higher NO_x can be produced from DME than from diesel fuel for an early start of injection since the duration of the peak combustion temperature would be longer in the initial combustion period due to the shorter ignition delay of DME [39]. When injection retardation is optimized for each fuel, NO_x from DME is lower than from diesel fuel [39,40]. Teng et al. [11] suggested that, although DME is an oxygenated fuel, the release of free oxygen is very limited as revealed by chemical kinetics calculations [12] which makes the effect of the oxygen content in DME on NO_x formation insignificant. Egnell [22] studied NO_x formation for DME using a reaction enthalpy analysis, by maintaining similar injection conditions for DME and diesel fuel. It was shown that the rate of reaction is not fundamentally different for DME and diesel fuel under these conditions. One explanation for the lower NO_x emissions with DME, when compared to diesel fuel injected at considerably higher pressures, was that the local equivalence ratio was higher for DME, giving a lower local oxygen concentration and thus NO_x levels. It was suggested that because the gaseous specific heat capacity of DME is higher than that of diesel fuel, the adiabatic flame temperature for DME is slightly lower than for diesel fuel (represented in this analysis by dodecane). This resulted in the proposition that lower NO_x emissions for DME should be expected if all other conditions are identical; this could again be explained by the lower rate of energy release [41]. A zero-dimensional multizone model has also been implemented to qualitatively simulate model the combustion characteristics of DME within a compression-ignition engine with particular reference to NO_x formation; the predictions indicated that DME forms less NO_x than diesel fuel [42,43].

Exhaust gas recirculation (EGR) reduces both burned gas temperature and NO_x formation inside the burned gas region. DME engines allow a higher EGR rate for NO_x reduction without being restricted by the NO_x-soot tradeoff, because DME has a high cetane number and fuel droplets in a DME spray within the engine cylinder evaporate – about three times faster than diesel fuel [44]; further NO_x reduction can be achieved with higher EGR rate [45]. Teng and McCandless [46] found that “engine-out” NO_x emissions were less than the 2007 regulated level during tests in a heavy-duty truck engine. The longer injection duration for DME with current fuel-system technology was, however, unfavourable for NO_x reduction; to prevent this a multiple-injection strategy was proposed [47].

6.1.3. *HC and CO*

Table 3 shows HC and CO emissions in the literature from CI engines fuelled with DME, compared to those with diesel fuel. It can be seen that HC emissions from DME are usually lower than or equal to those from the combustion of diesel fuel; HC emissions consist of partially or completely unburned fuel, produced in locations where combustion takes place under fuel-rich conditions, due to incomplete air-fuel mixing. Teng et al. [11] explained that, because DME has a short ignition delay period, the over-rich and over-lean mixture regions formed during the ignition delay period might be smaller, resulting in significantly reduced HC emissions. It should be added that DME is an oxygenated fuel containing 35% by mass of oxygen, has good mixing characteristics and is a superheated vapour after entering the engine cylinder. The volume of fuel-rich regions existing during the combustion period could thus be less, resulting in reduced HC emissions.

The data of CO emissions show some contradictions depending on the engine system and operating conditions. The higher CO emissions sometimes associated with DME could result from the longer injection duration, coupled with lower injection pressures and larger spray holes. Since there is production of CHO and CH_2O involved in the combustion of DME, depending on the reaction process, a larger amount of CO may be produced compared with diesel fuel [24]. Impingement of the DME fuel spray on the cooled combustion chamber wall could raise HC and CO emissions by quenching the DME reaction process [48]. CO may also be produced in locations of over-lean conditions. Due to the faster evaporation and better mixing with air, despite the shorter ignition delay for DME, more over-leaning can occur and, if the local equivalence ratio becomes too low to support combustion, an increase of CO emissions may result. Higher emissions of CO and HC have been reported along with lower NO_x in many investigations [49–53].

On the other hand, DME has good mixing characteristics, so that the locations of the fuel-rich regions in the combustion period could be reduced, resulting in lower CO emissions [11]. Egnell [22] argued that the lower emissions of HC and CO are linked to the structure of the DME molecule, similarly to soot formation. The low C/H ratio, the lack of C–C bonds and the high oxygen content of

Table 3
Comparison of HC and CO emissions for DME with those for diesel fuel

Year		Test condition	HC	CO
1995	McCarthy et al. [54]	ECE R49 13-mode cycle – estimation	Almost equal	Higher
1995	Fleisch and Meurer [3]	ECE R49 13-mode cycle – estimation	Almost equal	
1995	Kapus and Cartellieri [5]	US-FTP 75 cycle – estimation	Almost equal Lower (with oxid. cat.)	Higher Higher (with oxid. cat.)
1997	Kajitani et al. [39]	Steady-state test	Lower	
1998	Kajitani et al. [23]	Steady-state test	Lower	Lower
1998	Ofner et al. [24]	ECE R49 13-mode cycle test	Lower	Higher
2001	Egnell [22]	Steady-state test	Lower	Lower
2005	Goto et al. [21]	Japanese D13-mode	Lower	Lower

the fuel should give faster and more effective oxidation of intermediate species. The interaction between fuel-injection system behaviour and the combustion chamber design should be optimized to minimize HC and CO emissions. HC and CO could nevertheless be treated relatively easily with an oxidation catalyst in the aftertreatment system.

6.1.4. Combustion noise [3,15]

Since ignition delay is shorter for DME than for diesel, the pressure rise for DME during the premixed burning stage is slower than with diesel fuel which results in reduced combustion noise.

6.1.5. Non-regulated exhaust emissions

It is likely that some formaldehyde (CH_2O) is formed during combustion of DME in a CI engine, but significant emissions of sulfur dioxide (SO_2), polycyclic aromatic hydrocarbons (PAH) and benzene, toluene and xylene (BTX) are expected. The formaldehyde emissions could be 1–10% of the non-methane hydrocarbon emissions [1,35]. Aldehydes may also be high because associated CO emissions are high, although formaldehydes can be easily reduced by oxidation catalysts to a negligible level [55].

6.2. Effect of fuel-injection equipment (FIE) and combustion system parameters

The effect of the fuel-injection equipment and combustion system parameters on fuel consumption, NO_x emissions and combustion noise has been investigated for various DME-fuelled engines [5,9,18,36,56]. The parameters examined included injection pressure, start of injection, rate of injection, compression ratio, piston-bowl shape and EGR level; each of these parameters is examined later on. Fig. 5 shows an example of exhaust emissions affected by injection pressure [18].

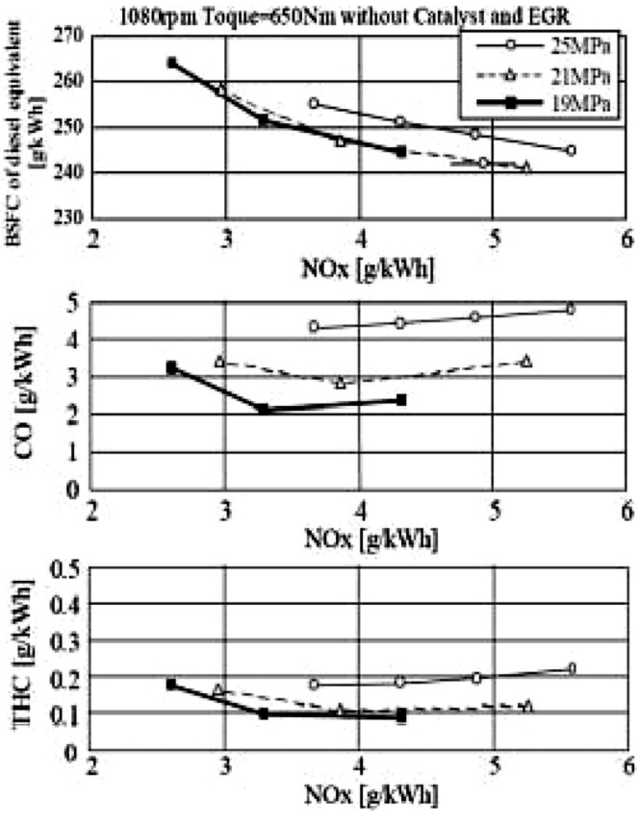

Fig. 5. Effect of fuel-injection pressure in the low speed and high load operating conditions for DME with a single cylinder of 638 ml DI compression-ignition engine [18].

Optimization of these parameters for a DME-fuelled engine (single cylinder, 2 l) equipped with a pintle nozzle injector and EGR [9] achieved combustion noise of about 76 dB(A) at both part load (IMEP = 0.2 MPa) and full load (0.75 MPa). On the other hand, for diesel fuel without EGR the noise levels were 80 dB(A) at part load and 86 dB(A) at full load. NO_x emissions were below 2 g/kWh, i.e. over 50% lower than for diesel, when the fuel consumption for DME was almost equal to that of diesel fuel.

For a DME-fuelled engine (four cylinder, 2 l) equipped with a five-hole injector nozzle and no EGR [5], although the fuel consumption was slightly higher at higher load than with diesel, the NO_x emissions were over 50% lower. At full load, the combustion noise reached levels of about 80–82 dB(A), about 10% lower than diesel fuel.

(i) *Injection pressure*: It was found that higher injection pressures, between 19 and 25 MPa, degraded fuel economy and increased CO and THC emissions in a heavy-duty engine (1.15 l/cylinder) equipped with a jerk pump type FIE, as shown in Fig. 5 [18]. However, another investigation with a common-rail system in a smaller engine (0.5 l/cylinder) showed an opposite trend [56] i.e. that higher injection pressure in the range 25–55 MPa, improved fuel economy with comparable or even reduced CO and THC emissions. Kook et al. [56] showed that the implications of better mixing between DME and air through higher injection pressure are higher power output but higher NO_x; the higher the injection pressure of DME (between 20 and 55 MPa of common-rail pressure), the higher was the IMEP, due to the improvement of the atomization process and mixing over a wide range of injection timings.
On the basis of the above it is hard to ascertain the effect of injection pressure and further investigation is needed for different fuel-injection equipment of parameters such as the injector geometry and the interaction between the emerging sprays, the in-cylinder flow and the combustion chamber geometry.

(ii) *Start of injection*: Retardation of the start of injection led to a reduction of NO_x emissions and combustion noise, but deterioration of the fuel consumption and increased CO emissions [9].

(iii) *Nozzle type*: The change of nozzle configuration, from needle-hole nozzle to pintle nozzle, not only changed the spray formation but also the injection process or the rate-shaping of injection. It was shown that the nozzle type should be matched to the engine design for better fuel distribution and mixing, which would in turn increase power output and reduce exhaust emissions.

(iv) *Injection process*: Rate shaping of injection [5,9] was carried out using multiple injections. With pilot-injection both NO_x emissions and combustion noise decreased while fuel consumption increased. As the amount of fuel injected during the initial part of the injection period decreased, combustion noise also decreased. The increase of injection pressure later on further decreased combustion noise over the whole load region.

(v) *Number of nozzle holes for a multi-hole nozzle*: Three- and seven-hole nozzles were used as alternative configurations to the standard five-hole nozzle in multi-hole nozzles [5]. With the three-hole nozzle, NO_x emissions were slightly lower than with both the five- and seven-hole nozzles but fuel consumption deteriorated considerably at higher loads. With the seven-hole nozzle, fuel consumption was improved at higher loads due to the better fuel distribution in the combustion chamber but combustion noise increased considerably. It was concluded that the five-hole nozzle was the optimum configuration for DME.

(vi) *Intake swirl level*: Kapus and Ofner [9] found that swirl flow in the combustion chamber was not necessary with DME because of its fast rate of evaporation and good mixing with air. When the swirl level decreased from 1.8 to 0.1 (AVL swirl number) the wall heat-transfer rate was also reduced due to the weaker air motion in the combustion chamber, resulting in lower fuel consumption. Combustion noise was also reduced with the reduced charge motion.

(vii) *Compression ratio (CR)*: With higher CR, ignition delay was reduced and the amount of fuel burnt during the initial energy release decreased, causing a less-rapid pressure rise in the cylinder. Both NO_x and combustion noise were correspondingly reduced.

(viii) *EGR*: NO_x emissions from DME combustion, similarly to other engine fuels, decreased with increased EGR rate, although CO and HC emissions became higher and fuel consumption increased with higher EGR. EGR is an ideal mechanism, especially at low loads, for decreasing NO_x emissions in DME engines, due to the smoke-free combustion. At high loads, CO emissions limit the use of EGR for normally aspirated engines as the air-excess ratio, which is already at a low level, is further reduced by EGR.

(ix) *Piston-bowl shape*: It is generally desirable in engines with a piston-bowl configuration that spray-wall wetting (or spray-wall impingement) should be minimized in order to allow low exhaust emissions, combustion noise and fuel consumption. Kapus and Ofner [9], found that, with respect to combustion noise and CO emissions, a normal piston-bowl (without a central cone) was better than a re-entrant bowl for the case of a pintle nozzle injector. Kapus and Cartellieri [5] also found that, with respect to fuel consumption, a wide shallow bowl offered more advantages than a narrow deep bowl when a multi-hole nozzle was used.

7. Energy efficiency and well-to-wheels emissions

7.1. Well-to-wheels analysis

The well-to-wheels energy efficiency is estimated as the product of the energy efficiency of each individual utilization step. The well-to-filling station step includes the recovery and transport of the feedstock, fuel production and fuel distribution efficiency. For the production of DME and methanol, it is assumed for financial reasons that remotely located gas sites are used. Since waste heating from the fuel production cannot be recovered for district heating, the efficiencies of the fuel production are relatively low. Consequently, the well-to-filling station energy efficiency for DME and methanol are among the lowest. Energy efficiency for the vehicle includes engine efficiency, transmission efficiency and a weight assessed correction factor. Furthermore, the well-to-wheels CO_2 emissions are determined by the CO_2 emissions during the production and distribution of the fuel, the CO_2 produced during the combustion of the fuel, and the overall vehicle efficiency. The CO_2 emissions during fuel production are primarily determined by the energy supplied for the production of the fuel but CO_2 can also be emitted due to molecular changes in the process of making fuel from the relevant feedstock. The emissions of CO_2 during combustion are determined from the carbon content of the fuel and its lower heating-value. Detailed analyses can be found in [1,4].

Table 4 shows the well-to-wheels energy efficiency and CO_2 emissions of both light-duty vehicles (LDV) and heavy-duty vehicles (HDV). The well-to-wheels energy efficiency of DME-fuelled vehicles is generally comparable to that of the LPG and CNG fuelled vehicles, while the energy efficiency is the highest for DI vehicles operating with diesel fuel. The well-to-wheels CO_2 emissions of DME-fuelled vehicles are comparable to those with DI diesel engines and CNG fuelled engines. Amongst the fossil fuel derivatives, diesel and DME from natural gas have the lowest well-to-wheels CO_2 emissions. As expected the fuels produced from renewable feedstock have very low well-to-wheel CO_2 emission levels.

7.2. Assessment of emissions by the sustainable process index

The overall ecological impact of a fuel can be judged by considering other emissions such as NO_x. Ofner et al. [24] carried out an ecological assessment of DME as a fuel, using the sustainable process index (SPI) proposed by Krotscheck and Narodoslawsky which takes into account

Table 4
Well-to-wheels energy efficiency and CO_2 emissions of light-duty vehicles (LDV) and heavy-duty vehicles (HDV) [1,4]

Fuel	Engine type	Well-to-wheel efficiency (%)		Relative well-to-wheel CO_2 emissions (Diesel = 1)	
		LDV	HDV	LDV	HDV
Diesel	CI, DI	25.7	30.3	1.00	1.00
DME	CI, DI	19.0	22.5	1.01	1.02
DME renewable	CI, DI	–	–	0.17	0.17
Gasoline	SI, $\lambda = 1$	16.9	20.1	1.47	1.46
LPG	SI, $\lambda = 1$	18.2	21.6	1.28	1.28
CNG	SI, $\lambda = 1$	16.8	20.7	1.14	1.09

both CO_2 and NO_x emissions. The concept of the SPI is based on a comprehensive knowledge of the interactions of human-induced mass and energy flows with those of the environment although this comprehensive knowledge does not really exist. With respect to fuels, the SPI is calculated by comparing the natural carbon flow of the life-cycle of the fuel to the human-induced carbon fuel flow. The natural carbon flow for fossil fuels is taken as the rate of carbon sedimentation in the oceans. The carbon flow in the natural cycle can be expressed in terms of the area required for circulation of a kg of carbon per year; for bio-fuels, this area amounts to approximately 7 m^2. For fossil fuels, sedimentation to accumulate 1 kg of carbon requires approximately 500 m^2. Thus, the natural carbon flow for fossil fuels is approximately 70 times lower than for bio-fuels. This method transforms the carbon flow estimate into 'area-year' in terms of square meters annual (m^2a). If an amount of carbon, say in wood, is used for providing energy of 1 kWh at the end use, the carbon flow during production, combustion and utilization of the fuel and the corresponding assimilation by trees, can be expressed in terms of 'area-year' per kWh.

Fig. 6 shows the SPI in terms of m^2a/kWh-wheel for different fuels [24]. It is evident that the CO_2 emissions of fossil fuels contribute significantly to the SPI. DME produced from natural gas has a 10% lower overall impact compared to diesel EUROIII because of the reduced NO_x emissions.

The main advantage of DME is that it can be produced from both fossil and renewable resources. DME produced from renewable feedstock can be added to the DME produced from natural gas without any change of the fuel properties. As a result the overall environmental impact shifts between the values of the fossil and bio-genic primary energy source (e.g. wood) in proportion to the relative mixture ratio. From Fig. 6, it can be deduced that 17% of DME must be produced from wood in order to lower the SPI to the CNG level [24].

8. New DME concept engine

With the objective of improving both engine thermal efficiency and exhaust emissions, several new engine-design concepts such as low compression ratio DI diesel engines and homogeneous charge compression ignition (HCCI) engines have been investigated using DME as an alternative fuel.

Kajitani et al. [33] investigated the concept of using DME as fuel in a direct-injection compression-ignition engine with a low compression ratio, in an effort to identify a combustion regime with the highest possible thermal efficiency. It was shown that:

(i) The lowest compression ratio for easy start and stable operation of a small passenger-car engine was around 12:1.
(ii) The DME-fuelled engine was superior to the conventional diesel engine in terms of thermal efficiency, exhaust emissions and engine noise.
(iii) The ignition delay became longer with a reduction in compression ratio from ~18 to 12.
(iv) The brake thermal efficiency, THC and CO emissions as well as engine noise for DME remained almost constant over this compression ratio range.

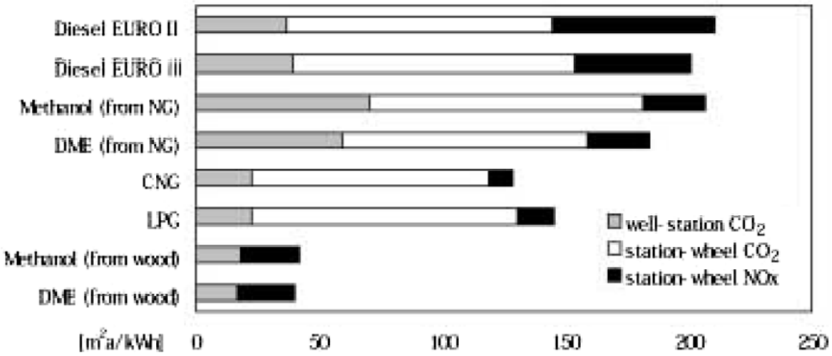

Fig. 6. Sustainable process index (SPI) for various fuels [24].

(v) NO_x emissions became lower with a reduction in compression ratio but remained at unacceptably high levels.

Lowering the compression ratio allowed reduction of the peak firing pressure in the cylinder. Consequently, engines could be designed with a lighter structure than current diesel engines and lower cost. By combining low compression ratio with rate-shaped injection, NO_x emissions, peak firing pressure and noise could all be further reduced.

There is consensus that the HCCI engine has great potential to simultaneously provide high thermal efficiency, low combustion noise, lower NO_x levels and soot-free combustion. On the other hand, it appears to be limited to lighter load operation, exhibit difficulty of controlling ignition timing and produce more unburned-HC emissions. The HCCI engine is yet in the laboratory research phase owing to the difficulty of making it appropriate for mass production in the transportation sector. To address the problems of the HCCI engine and to overcome future emission legislation limits for vehicle operation, investigations into fuel property and quality (including DME) are being conducted. Since DME injected into the cylinder as a liquid fuel quickly evaporates, it is likely that a homogeneous air–fuel mixture can be more easily formed. Rapidly evaporating DME showed near zero PM and NO_x emissions without power loss, due to the advantage of no wall-wetting [56]. DME was found to achieve HCCI operation with a diesel FIE system without engine modification. Combustion phasing could be controlled with either EGR or by the addition of high-octane/low-cetane fuels such as alcohols, methane and hydrogen, through suppression of the OH radical production in the premixed charge [57,58].

Takatsuto et al. [59] experimentally investigated auto-ignition and combustion characteristics in homogeneous lean mixtures. Auto-ignition was realized for DME over a wide equivalence ratio range compared to *n*-butane. NO_x emissions with DME were much lower than those with *n*-butane and THC concentrations were also low. For DME, auto-ignition and combustion were always found to include high and low temperature reactions, at all equivalence ratios, which were analysed through various simulation and diagnostics tools [60–63]. Since DME has a high cetane number and fuel drops in the DME spray within an engine cylinder evaporate about three times faster than diesel fuel, low-temperature combustion may be more appropriate for DME engines. This could be achieved by using a multiple-injection strategy with timing for the primary injection determined by the cylinder temperature [46]. DME as an additive or an ignition enhancer even enables the use of various other fuels for conventional diesel combustion, dual-fuel operation and HCCI operation. Propane or LPG, hydrogen and natural gas have all been tested as alternatives to diesel with the assistance of DME [22,49,64–68].

It is also expected that by employing the Miller cycle at low loads and a multiple injection strategy at high loads, combined with a high EGR rate, may allow NO_x emissions from DME engines to meet the US 2010 standard [46].

9. Summary

Numerous investigations of DME-fuelled engines have indicated that it offers excellent promise as an alternative fuel for compression-ignition operation in the automotive sector.

DME is a liquefied gas that can be produced from a variety of feed stock, is non-toxic and environmentally benign. DME is as easy to handle as LPG since it is condensed by pressurizing above 0.5 MPa. DME's main feature as an efficient alternative fuel for use in compression-ignition engines is its high cetane number. The fast evaporation of DME can lead to better mixing with air in the engine cylinder and its high oxygen content can achieve smokeless combustion through low formation and high oxidation rates of particulates. However, DME requires a higher injected volume to supply the same amount of energy as the diesel fuel, due to its lower density and combustion enthalpy. DME-fuelled systems also need lubricity-enhancing additives and anti-corrosive sealing materials to secure leakage-free operation.

The DME spray characteristics and combustion process have been investigated for various fuel-injection equipment, ranging from mechanical pump systems to advanced common-rail systems, in a variety of prototype engines with occasional slight modifications. These engine tests with DME proved its potential as a clean alternative fuel achieving smoke-free high power operation provided an optimized fuel injection equipment and the engine configuration are closely matched to reduce fuel consumption and exhaust emissions. Optimisation of the fuel injection equipment, overcoming the problem of low density, low lubricity and corrosiveness, may allow mass production of the DME-fuelled compression-ignition engines when adequate DME supplies become available in the market.

It can be argued however that the well-to-wheels energy efficiency and CO_2 emissions for DME produced from natural gas are not better than for most other fuels, as a result of the high fuel production energy and relevant CO_2 emissions, and that diesel fuel has the highest well-to-wheels energy efficiency. Nevertheless, DME has the advantage that it can be produced from both fossil and renewable resources. DME produced from renewable feedstock may be added to that produced from natural gas, without any change in fuel properties, for use as an alternative automotive fuel. However, extensive studies of the environmental impact of DME are needed to evaluate its full potential, using assessments based on different model assumptions and focusing on local conditions in urban areas or on biological processes.

Recent studies on DME-fuelled engines showed the potential of exploiting new combustion concepts such as low-compression DI diesel engines and homogeneous charge compression-ignition engines operating in a low-temperature combustion regime.

Overall, DME has been found to be a very promising alternative fuel for compression-ignition engines, capable of providing high thermal efficiency, low combustion noise, low NO_x levels and soot-free combustion; it thus merits further research and development before a final decision is taken on its potential as a mass production fuel for the automotive market.

References

[1] Verbeek RP, Van Doom A, van Walwijk M. Global assessment of di-methyl ether as an automotive fuel. second ed., 96.OR.VM.029. 1/RV, TNO Road-Vehicles Research Institute; 1996.

[2] Arcoumanis C. The second European Auto-Oil programme (AOLII). European Commission, vol. 2. Alternative Fuels for Transportation; 2000.

[3] Fleisch TH, Meurer C. DME, The diesel fuel for the 21st Century? In: AVL conference engine and environment; 1995.

[4] Verbeek R, Van der Weide J. Global assessment of di-methyl ether: comparison with other fuels. SAE Paper 971607; 1997.

[5] Kapus PE, Cartellieri WP. ULEV potential of a DI/TCI diesel passenger car engine operated on di-methyl ether. SAE Paper 952754; 1995.

[6] Basu A, Wainwright JM. DME as a power generation fuel: performance in gas turbines. In: PETROTECH-2001 conference; 2001.

[7] Cocco D, Tola V, Cau G. Performance evaluation of chemically recuperated gas turbine (CRGT) power plants fuelled by di-methyl-ether (DME). Energy 2005;31:1446–58.

[8] DjebaïLi N, Paillard C. Burning velocities of dimethyl ether and air. Combus Flame 2001;125:1329–40.

[9] Kapus P, Ofner H. Development of fuel injection equipment and combustion system for DI diesels operated on di-methyl ether. SAE Paper 950062, SAE Trans J Fuel Lubr 1995;104(4):54–9.

[10] Sorenson SC, Glensvig M, Abata D. Di-metyl ether in diesel fuel injection systems. SAE Paper 981159, SAE Trans J Fuel Lubr 1998;107(4):438–49.

[11] Teng H, McCandless JC, Schneyer JB. Thermo-chemical characteristics of di-methyl ether – an alternative fuel for compression-ignition engines. SAE Paper 2001-01-0154, SAE Trans J Fuel Lubr 2001;110(4):96–106.

[12] Edgar B, Dibble RW, Naegeli DW. Autoignition of di-methyl ether and di-methoxy methane sprays at high pressures. SAE Paper 971677, SAE Trans J Fuel Lubr 1997;106(4):625–639.

[13] Francisco JS. On the competition between hydrogen abstraction versus C–O bond fission in initiating di-methyl ether combustion. Combus Flame 1999;177:312–6.

[14] Masterton WL, Slowinski EJ. Chemical principles. W.B. Saunders Company; 1997.

[15] Alex Mills. Status and future opportunities for conversion of synthesis gas to liquid fuels. Fuel 1994;73:1243–79.

[16] Semelsberger TA, Borup RL, Greene HL. Dimethyl ether (DME) as an alternative fuel. J Power Sources 2005;156(2):497–511.

[17] Mii T, Uchida M. Fuel DME plant in East Asia. In: Proceedings of 15th Saudi–Japan joint symposium; 2005.

[18] Tsuchiya T, Sato Y. Development of DME engine for heavy-duty truck. SAE Paper 2006-01-0052; 2006.

[19] Yu J, Bae C. Dimethyl ether (DME) spray characteristics compared to diesel in a common-rail fuel injection system. J Autom Eng Proc IMechE Pt D 2003;217:1135–44.

[20] Zhao X, Ren Meifeng, Liu Z. Critical solubility of dimethyl ether (DME) + diesel fuel and dimethyl carbonate (DMC) + diesel fuel. Fuel 2005;84:2380–3.

[21] Goto S, Oguma M, Suzuki S. Research and development of a medium duty DME truck. SAE Paper 2005-01-2194;2005.

[22] Egnell R. Comparison of heat release and NO_x formation in a DI diesel engine running on DME and diesel fuel. SAE Paper 2001-01-0651, SAE Trans J Fuel Lubr 2001;110(4):492–506.

[23] Kajitani S, Chen CL, Oguma M, Alam M, Rhee KT. Direct injection diesel engine operated with propane – DME blended fuel. SAE Paper 982536; 1998.

[24] Ofner H, Gill DW, Krotscheck C. Di-methyl ether as fuel for CI engines – a new technology and its environmental potential. SAE Paper 981158, SAE Trans J Fuel Lubr 1998;107(4):425–37.

[25] Ikeda T, Ohmori Y, Takamura A, Sato Y, Jun L, Kamimoto T. Measurement of the rate of multiple fuel injection with diesel fuel and DME. SAE Paper 2001-01-0527, SAE Trans J Engine 2001;110(3): 372–80.

[26] McCandless JC, Teng H, Schneyer JB. Development of a liquid-DME fuel tank – a two-fluid thermodynamic pump. SAE Paper 2001-01-0652, SAE Trans J Fuel Lubr 2001;110(4):507–13.

[27] Wakai K, Nishida K, Yoshizaki T, Hiroyasu H. Spray and ignition characteristics of di-methyl ether injected by a DI diesel injector. In: Proceedings of the fourth international symposium COMODIA'98; 1998. p. 537–42.

[28] Glensvig M. Sorenson SC, Abata DL. An investigation of the injection characteristics of di-methyl ether. ICE-vol. 29-3, ASME; 1997.

[29] Teng H, McCandless JC. Comparative study of characteristics of diesel-fuel and dimethyl-ether sprays in the engine. SAE Paper 2005-01-1723, SAE Trans J Engine 2005;114(3):1202–13.

[30] Golovitchev VI, Nordin N, Chomiak J. Neat di-methyl ether: is it really diesel fuel of promise? SAE Paper 982537;1998.

[31] Golovitchev VI, Nordin N, Chomiak J, Nishida N, Wakai K. Evaluation of ignition quality of DME at diesel engine conditions. In: Proceedings of the fourth international conference of internal combustion engines 99 (ICE99): experiments and modeling; 1999. p. 299–306.

[32] Zhao Z, Kazakov A, Dryer FL. Measurements of dimethyl ether/air mixture burning velocities by using particle image velocimetry. Combus Flame 2004;139:52–60.

[33] Kajitani S, Chen Z, Oguma M, Konno M. A study of low-compression-ratio di-methyl ether diesel engines. Int J Engine Res 2002;2:1–11.

[34] Westbrook CK. Chemical kinetic modeling of oxygenated diesel fuels in advanced petroleum-based and alternative fuels. DOE Report; 1999.

[35] Ogawa H, Miyamoto N, Yagi M. Chemical-kinetic analysis on PAH formation mechanisms of oxygenated fuels. SAE Paper 2003-01-3190, SAE Trans J Fuel Lubr 2003;112(4):2413–21.

[36] Miyamoto H, Ogawa H, Arima T, Miyakawa K. Improvement of diesel combustion and emissions with various oxygenated fuel additives. SAE Paper 962115;1996.

[37] Curran EM, et al. Detailed chemical kinetic modeling of diesel combustion with oxygenated fuels. SAE Paper 2001-01-0653, SAE Trans J Fuel Lubr 2001;110(4):514–21.

[38] Sidhu Sukh, Graham J, Striebich. Semi-volatile and particulate emissions from the combustion of alternative diesel fuel. Chemosphere 2001;42:681–90.

[39] Kajitani S, Chen CL, Konno M. Engine performance and exhaust characteristics of direct-injection diesel engine operated with DME. SAE Paper 972973, SAE Trans J Fuel Lubr 1997;106(4):1568–77.

[40] Longbao Z, Hewu W, Deming J, Zuohua H. Study of performance and combustion characteristics of a DME-fueled light-duty direct-injection diesel engine. SAE Paper 1999-01-3669;1999.

[41] Cipolat D. Combustion aspects of a compression ignition engine fueled on DME. In: Proceedings of the seventh international conference on energy for a clean environment; 2003. p. 156.

[42] Konno M, Kajitani S, Oguma M, Iwasa T, Shima K. NO emission characteristics of a CI engine fueled with neat dimethyl ether. SAE Paper 1999-01-1116, SAE Trans J Fuel Lubr 1999;108(4):460–7.

[43] Saeed A, Saeed K, Ahmed A, Malik KA. Multizones modeling of the combustion characteristics of oxygenated fuels in CI engines. SAE Paper 2006-01-0051;2006.

[44] Teng H, McCandless JC, Schneyer JB. Compression ignition delay (physical + chemical) of di-methyl ether – an alternative fuel for compression-ignition engines. SAE Paper 2003-01-0759, SAE Trans J Fuel Lubr 2003;112(4):377–89.

[45] Kinoshita K, Oguma M, Goto S, Sugiyama K, Mori M, Watanabe T. Effects of fuel injection conditions on driving performance of a DME diesel vehicle. SAE Paper 2003-01-3193;2003.

[46] Teng H, McCandless JC. Can heavy-duty diesel engines fueled with DME meet US 2007/2010 emissions standard with a simplified aftertreatment system? SAE Paper 2006-01-0053;2006.

[47] Teng H, Regner G. Fuel injection strategy for reducing NO_x emissions from heavy-duty diesel engine fueled with DME. SAE Paper 2006-01-3324;2006.

[48] Oguma M, Goto S, Hatsuzawa H, Konno M, Chen Z, Watanabe T. Chemiluminescence analysis from in-cylinder combustion of a DME-fueled DI diesel engine. SAE Paper 2003-01-3192;2003.

[49] Crookes RJ, Bob-Manuel KDH. Di-methyl ether or rapeseed methyl ester: a preferred alternative fuel option for future diesel engine operation. In: Proceedings of the fifth international colloquium on fuels; 2005. p. 181–7.

[50] Mingfa Y, Zunquing Z, Sidu X, Maoling F. Experimental study on the combustion process of dimethyle ether (DME). SAE Paper 2003-01-3194, SAE Trans J Fuel Lubr 2003;112(4):2422–9.

[51] Song J, Huang Z, Qiao X, Wang W. Performance of a controllable premixed combustion engine fueled with dimethyl ether. J. Energy Convers Manage 2004;45:2223–32.

[52] Sorenson SC, Mikkelsen S. Performance and emissions of a 0.273 l direct injection diesel engine fuelled with neat dimethyl ether. SAE Paper 950064, SAE Trans J Fuel Lubr 1995;104(4):80–90.

[53] Pana C, Negurescu N, Popa MG, Boboc G, Cernat Al. Performance of a DME fueled diesel engine. In: Proceedings of the fifth international colloquium on fuels; 2005. p. 227–32.

[54] Fleisch T, McCarthy C, Basu A, Udovich C, Charbonneau P, Slodowske W, Mikkelsen SE, McCandless JC. A new clean diesel technology: demonstration of ULEV emissions on a navistar diesel engine using a new alternative fuel. SAE Paper 950061, SAE Trans J Fuel Lubr 1995;104(4):42–53.

[55] Oguma M, Shiotani H, Goto S, Suzuki S. Measurement of trace levels of harmful substances emitted from a DME DI diesel engine. SAE Paper 2005-01-2202;2005.

[56] Kook S, Kong J, Min K, Bae C. Homogeneous charge compression ignition engine with two-stage diesel fuel injection. In: Proceedings of the THIESEL 2004 (thermo- and fluid dynamic processes in diesel engines); 2004. p. 497–510.

[57] Ogawa H, Miyamoto N, Kaneko N, Ando H. Combustion control and operating range expansion with direct injection of reaction suppressors in a premixed DME HCCI engine. SAE Paper 2003-01-0746;2003.

[58] Shudo T, Kitahara S, Ogawa H. Influence of carbon dioxide on combustion in an HCCI engine with the ignition-control by hydrogen. SAE Paper 2006-01-3248;2006.

[59] Takatsuto R, Igarashi T, Iida N. Autoignition and combustion of DME and *n*-butane–air mixtures in homogeneous charge compression ignition engine. In: Proceedings of the fourth international symposium COMODIA'98; 1998. p. 185–90.

[60] Sato S, Iida N. Analysis of DME homogeneous charge compression ignition combustion. SAE Paper 2003-01-1825;2003.

[61] Kim H, Cho S, Min K. Reduced chemical kinetic model of DME for HCCI combustion. SAE Paper 2003-01-182522, SAE Trans J Fuel Lubr 2003;112(4):1180–7.

[62] Kawahara N, Tomita E, Kagajyo H. Homogeneous charge compression ignition combustion with dimethyl ether – spectrum analysis of chemiluminescence. SAE Paper 2003-01-1828, SAE Trans J Fuel Lubr 2003;112(4):1214–21.

[63] Yamada H, Goto Y. Analysis of reaction mechanisms controlling cool and thermal flame with DME fueled HCCI engines. SAE Paper 2006-01-3299;2006.

[64] Chen Z, Konno M, Oguma M, Yanai T. Experimental study of CI natural-gas/DME homogeneous charge engine. SAE Paper 2000-01-0329, SAE Trans J Engine 2000;109(3):442–51.

[65] Narioka Y, Yokoyama T, Lio S, Takagi Y. HCCI Combustion characteristics of hydrogen and hydrogen-rich natural gas reformate supported by DME supplement. SAE Paper 2006-01-0628;2006.

[66] Crookes RJ. Biofuels performance in internal combustion engines. In: Proceedings of the fourth international colloquium on fuels; 2003. p. 201–7.

[67] Yeom K, Jang J, Bae C. Homogeneous charge compression ignition of LPG and gasoline using variable valve timing in an engine. Fuel 2007;86(4):494–503.

[68] Yeom K, Bae C. Gasoline – di-methyl ether homogeneous charge compression ignition engine, energy and fuels, in press.