Kaario, Ossi Tapani; Vuorinen, Ville; Kahila, Heikki; Im, Hong G.; Larmi, Martti

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Published in:
International Journal of Engine Research

DOI:
10.1177/1468087419854235

Published: 19/06/2019

Please cite the original version:
The effect of fuel on high velocity evaporating fuel sprays: Large-Eddy simulation of Spray A with various fuels

Ossi Tapani Kaario¹, Ville Vuorinen¹, Heikki Kahila¹, Hong G Im² and Martti Larmi¹

Abstract

Lagrangian particle tracking and Large-Eddy simulation were used to assess the effect of different fuels on spray characteristics. In such a two-way coupled modeling scenario, spray momentum accelerates the gaseous phase into an intense, multiphase jet near the nozzle. To assess fuel property effects on liquid spray formation, the non-reacting Engine Combustion Network Spray A baseline condition was chosen as the reference case. The validated Spray A case was modified by replacing \( n \)-dodecane with diesel, methanol, dimethyl ether, or propane assuming 150 MPa injection pressure. The model features and performance for various fuels in the under-resolved near-nozzle region are discussed. The main findings of the paper are as follows. (1) We show that, in addition to the well-known liquid penetration \((L_{\text{liq}})\), and vapor penetration \((L_{\text{vap}})\), for all the investigated fuels, the modeled multiphase jets exhibit also a third length scale \(L_{\text{core}}\), with discussed correspondence to a potential core part common to single phase jets. (2) As a characteristic feature of the present model, \(L_{\text{core}}\) is noted to correlate linearly with \(L_{\text{liq}}\) and \(L_{\text{vap}}\) for all the fuels. (3) A separate sensitivity test on density variation indicated that the liquid density had a relatively minor role on \(L_{\text{liq}}\). (4) Significant dependency between fuel oxygen content and the equivalence ratio \((\Phi)\) distribution was observed. (5) Repeated simulations indicated injection-to-injection variations below 2% for \(L_{\text{liq}}\) and 4% for \(L_{\text{vap}}\). In the absence of experimental and fully resolved numerical near-nozzle velocity data, the exact details of \(L_{\text{core}}\) remain as an open question. In contrast, fuel property effects on spray development have been consistently explained herein.

Keywords

Large-Eddy simulation, Lagrangian particle tracking, Engine Combustion Network, Spray A, fuel comparison, liquid length

Introduction

Engine Combustion Network (ECN) is an international collaborative effort to focus research and to facilitate experimental and computational data access within the engine combustion context.¹,² A focal topic within ECN is spray combustion which is approached by detailed measurements and simulations on \( n \)-dodecane, \( n \)-heptane, and gasoline. One of the best documented diesel combustion target conditions is the ECN Spray A case in which liquid \( n \)-dodecane is injected through a nozzle hole \((D = 90 \mu m)\) at a high injection pressure \((150 \text{MPa})\) in engine relevant conditions \((T = 900 \text{K}, P = 6 \text{MPa})\). Experimental, non-reacting ECN spray cases have been used as reference cases for many numerical studies²–¹² utilizing Lagrangian particle tracking (LPT), Large-Eddy simulation (LES), or Reynolds-averaged Navier–Stokes (RANS) turbulence modeling (see Table 1). Here, we use the non-reacting Spray A target condition as the baseline validation case. In addition, in the present numerical investigations, we extend the Spray A case by replacing \( n \)-dodecane with various other fuels to better understand fuel property effects on spray characteristics.

¹Department of Mechanical Engineering, School of Engineering, Aalto University, Espoo, Finland
²Clean Combustion Research Center, King Abdullah University of Science and Technology, Thuwal, Saudi Arabia

Corresponding author:
Ossi Tapani Kaario, Department of Mechanical Engineering, School of Engineering, Aalto University, 00076 Aalto, Finland.
Email: ossi.kaario@aalto.fi
(3) vapor pressure $P_f$

After the start of injection, the liquid phase reaches a maximum penetration length, commonly termed the liquid length ($L_{liq}$) that remains nearly constant during the steady period of injection. The tip of the vapor phase ($L_{vap}$) penetrates downstream with the well-established scaling $L_{vap} \sim \sqrt{t}$, where $t$ indicates time. It has been reported that $L_{liq}$ depends on the ambient and fuel injection conditions. For example, Siebers showed that $L_{liq}$ is ultimately limited by mixing, that is, the mixing rate of ambient energy and mass into the spray will determine the rate of liquid fuel evaporation. Optimally, fuel is fully evaporated before reaching the cylinder walls. However, at low ambient temperature, the liquid phase can reach the cylinder walls leading to wall wetting. Thereby, in the context of engine development process, it is essential to predict fuel property effects on $L_{liq}$.

With relevance to the present study, some key fuel properties are (1) liquid fuel density $\rho_f$, (2) latent heat $h_f$, and (3) vapor pressure $P_f$. Latent heat $h_f$ has a significant effect on the gas phase (and liquid phase) temperature during the evaporation process. Vapor pressure (or boiling point), on the contrary, has an important role in the fuel evaporation rate. In practice, the higher the $P_f$, the faster the liquid evaporates. If $P_f$ of a fuel reaches the ambient pressure, it starts to boil. Experimentally, a positive correlation has been observed between $L_{liq}$ and the fuel boiling point temperature. Thereby, a low boiling point fuel evaporates quickly and the liquid length is shorter than that of a high boiling point fuel. It is noted that in engine relevant conditions (high ambient pressure), boiling does not typically take place. A scaling law for liquid penetration shows that fuel density correlates with liquid length $L_{liq} \sim \sqrt{\rho_f/\rho_a}$, where $\rho_a$ is the ambient gas density. In the validation of the scaling law, the density ratio ranged between $13 < \rho_f/\rho_a < 220$. Here, the span of the investigated density ratios is $20 < \rho_f/\rho_a < 34$.

There are several computational studies on single fuel LES/LPT (see Table 1). However, we are aware of only a few papers addressing the fuel property effects on spray characteristics. For example, several studies are available concerning detailed single fuel inner-nozzle simulations. Som et al. used RANS modeling for both the inner-nozzle and the ambient part of the injector comparing biodiesel and standard diesel fuels. They concluded that biodiesel has both higher liquid length and vapor penetration than standard diesel. The observation was explained by the high boiling point temperature and heat of vaporization of biodiesel. While the discussed studies were carried out in non-reacting conditions, there are a number of detailed numerical investigations carried out in reacting conditions as well. One of the few studies which included fuel property comparison in reacting conditions was carried out by Som and Longman. The study was an extension to the earlier non-reacting study. They compared biodiesel to petrodiesel and pointed out the need to develop better surrogates for the considered fuels.

![Figure 1](image_url)

**Figure 1.** Typical LES/LPT spray centerline average velocity and temperature profiles at distances $z/D < 200$ from the nozzle exit.}

According to the previous literature, the schematic picture in Figure 1 summarizes typical LES/LPT spray centerline average velocity and temperature profiles at distances $z/D < 200$ from the nozzle exit. We emphasize that this graph is solely based on simulations while respective experimental data are presently not available. Next, we discuss some of the features in the figure. The domain in Figure 1 consists of three parts. (1) Droplets lose their momentum accelerating the gaseous phase, denoted as $L_I$ in Figure 1. At the border between $L_I$ and $L_H$, the maximum axial gas phase velocity is reached, and the droplets have lost most of their momentum. Since droplet–gas phase interaction is governed by the Stokes number, where $\tau_f = \rho_dD/18\mu_g$, $U$ is the characteristic velocity, $d$ is the droplet diameter, and $\mu_g$ is the molecular gas viscosity, it is expected that within $L_I$, $St$ number dictates the acceleration of gas phase and the respective deceleration of droplets. Increasing $St$ number implies reduced
rate of momentum transfer from droplets to gas phase and, hence, increased $L_I$. In addition, droplet evaporation is initialized which has a cooling effect on the gas phase. (2) At the beginning of $L_{II}$, the slip velocity is low. While the core of the jet is still unaffected by the shear layer, the potential core or spray core ($L_{core}$) is established at the centerline of the spray analogous with the findings on single phase jets. The length of $L_{II}$ is related to the diameter of the gas jet at the end of $L_I$. (3) After the grown instabilities have reached the spray axis at the end of $L_{II}$, the mixing governed velocity decay starts. Based on momentum and mass conservation, the axial velocity decays in single phase jets as $U_0/1/z$. In addition, due to mixing, the spray centerline temperature starts to increase towards the ambient gas phase temperature. In the absence of experimental and fully resolved numerical near-nozzle velocity data, Figure 1 only indicates typical LES/LPT modeling outcome. For future development of such models, we propose that the connection between $L_{core}$, $L_I$, and $L_{II}$ needs to be better understood for various fuels.

Thereby, the following hypothesis are formulated for the present numerical work: (1) liquid length between the studied fuels is proportional to $L_{core}$, that is, $L_{liq} \sim L_{core}$, since the mixing of hot gases into the spray core starts only after $L_{core}$; (2) liquid length is also affected by other properties, such as liquid density, heat of vaporization, or vapor pressure, that is, $L_{liq}(\rho, h_f, P_f, \ldots)$; and (3) vapor penetration is proportional to $L_{core}$, that is, $L_{vap} \sim L_{core}$, after the beginning of the mixing zone (beginning of $L_{III}$), the vapor jet penetrations are similar.

As discussed above and summarized in Table 1, there are not many detailed numerical studies assessing fuel property effects on spray characteristics. With relevance to engine R&D process, understanding LES/LPT model performance for various fuels is important. Therefore, our aim is to bridge the observed research gap by computational LES/LPT modeling of the mixing and evaporation characteristics of various fuels. In order to answer the above stated research hypotheses, the below objectives have been formulated for the present numerical study:

1. Validate the LES model in the non-reacting ECN Spray A case for n-dodecane;
2. Compare diesel, dimethyl ether (DME), methanol, and propane to n-dodecane in the Spray A conditions;
3. Assess injection-to-injection variation between the fuels;
4. Understand the effect of liquid density on liquid length;
5. Analyze the local equivalence ratio differences between the fuels.

In larger context, the model features and performance for various fuels in the under-resolved near-nozzle region are discussed. In order to answer the objectives, the study is limited to non-reacting sprays. However, we note that the near-field metrics of non-reacting and reacting Spray A are essentially the same.

**Governing equations**

**Fluid motion**

The governing equations for the gaseous phase describe the conservation of mass, momentum, energy, and species mass fractions, and they are written as follows

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u_i}{\partial x_j} = 0 \quad (1)$$

$$\frac{\partial \rho u_i}{\partial t} + \frac{\partial \rho u_i u_j}{\partial x_j} = -\frac{\partial}{\partial x_j}(\rho \delta_{ij} - \tau_{ij}) + M_d \quad (2)$$

$$\frac{\partial \rho h}{\partial t} + \frac{\partial \rho u_i h}{\partial x_j} = -\frac{\partial}{\partial x_j}(\tau_{ij} u_j) + \frac{\partial}{\partial x_j}(\frac{\partial T}{\partial x_j}) + M_h \quad (3)$$

$$\frac{\partial Y_k}{\partial t} + \frac{\partial \rho u_i Y_k}{\partial x_j} = \frac{\partial}{\partial x_j}(\rho D_k \frac{\partial Y_k}{\partial x_j}) + M_Y \quad (4)$$

where $M_d$ is the momentum source term exerted from the droplets to the gas phase, $M_h$ is the energy source from droplets, and $M_Y$ denotes the vapor mass source term from the liquid phase (here $Y$ refers to the fuel, depending on the case).

In LES, equations (1)-(4) are spatially filtered results in additional subgrid-scale (sgs) terms from the non-linear part of the equations and they can be written in the form $\tau_{ij}(\rho, u_i, \ldots) = \tau_{ij}$. The sgs terms, which require further modeling efforts, account for the interaction between the resolved and the unresolved scales. In addition, according to the Boussinesq hypothesis, viscosity can be written as $\mu = \mu_0 + \mu_t$, where $\mu$ is the total viscosity, $\mu_0$ is the molecular viscosity obtained from Sutherland’s law, and $\mu_t$ is the turbulent viscosity calculated from
Table 2. Operating conditions for Spray A.

<table>
<thead>
<tr>
<th>Ambient conditions</th>
<th>Injector conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>O₂</strong></td>
<td>0%</td>
</tr>
<tr>
<td><strong>Pressure</strong></td>
<td>6 MPa</td>
</tr>
<tr>
<td><strong>Temperature</strong></td>
<td>900 K</td>
</tr>
<tr>
<td><strong>Density</strong></td>
<td>22.8 kg/m³</td>
</tr>
<tr>
<td><strong>Fuel injection temperature</strong></td>
<td>363/323 K</td>
</tr>
<tr>
<td><strong>Nozzle diameter</strong></td>
<td>90 μm</td>
</tr>
<tr>
<td><strong>Injection pressure</strong></td>
<td>150 MPa</td>
</tr>
</tbody>
</table>

DME: dimethyl ether.

*Diesel and n-dodecane values are given at T = 363 K, and DME and propane are at T = 323 K.

\[ \mu_t = c_1 \rho \Delta^{1/2} \] (5)

In equation (5), \( \Delta \) denotes the filter width calculated from the cell volume \( V_{cell} \) as \( \Delta = V_{cell}^{1/3} \). The present study uses a \( k-\ell \) model\(^{12} \) for the sgs where a transport equation for the sgs turbulent kinetic energy \( k_{sgs} \) is solved according to

\[ \frac{\partial k_{sgs}}{\partial t} + \frac{\partial \rho u_{sgs}}{\partial x_j} = P - \rho e_{sgs} + \left( \frac{\partial}{\partial x_j} \left[ \mu_t \frac{\partial k_{sgs}}{\partial x_j} \right] \right) \] (6)

where \( P \) is the production term calculated as follows

\[ P = \tau_{sgs} \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \] (7)

and \( e_{sgs} \) is the sgs dissipation rate of the turbulent kinetic energy

\[ e_{sgs} = c_2 \frac{k_{sgs}^{3/2}}{\Delta} \] (8)

The coefficients \( c_1 \) and \( c_2 \) have the values 0.05 and 1.0, respectively.\(^{29,33} \) A second-order accurate flux limited scheme is used for the spatial discretization, while a second-order accurate three-time level method is applied for the time integration.\(^{34} \) The mathematical closure for the system of equations is provided by the ideal gas law. Simulations have been carried out with the Star-CD 4.24 code.

**Droplet motion**

In LPT, the motion of individual droplets is tracked through the computational domain. The number of droplets in a diesel spray can be significant and, hence, it is a common practice to reduce the computational cost and group droplets with similar properties into a “parcel.” In this study, parcels have equal mass indicating that the number of the droplets within a parcel is varying depending on the droplet size. The parcel position is updated from

\[ \frac{dx_p}{dt} = u_p \] (9)

It is assumed that the force acting on a droplet is due to the aerodynamic drag, leading to the following formulation under assumption of spherical droplets\(^{19} \)

\[ \frac{du_p}{dt} = \frac{C_d}{\tau_p} \frac{u_g - u_p}{24} \] (10)

The expression for the drag coefficient \( C_d \) is given as follows

\[ C_d = \begin{cases} \frac{24}{Re_p} \left( 1 + \frac{1}{6} Re_p^2 \right) & \text{for } Re_p < 1000 \\ 0.424 & \text{for } Re_p \geq 1000 \end{cases} \] (11)

where \( Re_p \) is the droplet Reynolds number based on the droplet slip velocity.

The parcels are advanced in time using a semi-implicit time integration method by taking five subiterations within each time step. The momentum source term \( M_d \) in equation (2) is evaluated for each cell separately by considering all the parcels within the cell. The following relation for the source term is assumed\(^{19} \)

\[ M_d = \frac{1}{2} \rho_g C_d A |u_g - u_p| (u_g - u_p) \] (12)

where \( A \) is the projected droplet area.

Concerning droplet breakup modeling, Wehrfritz et al.\(^{28} \) observed in Spray A conditions (see Table 2) that droplet breakup takes place only up to \( \sim 20D \) (nozzle hole diameter \( D \)) from the nozzle exit after which the droplet Weber number becomes too small for droplet breakup to take place. In addition, it has been numerically observed that droplet Sauter mean diameter (SMD) reduces very quickly (within \( \sim 1–2 \) mm) below 1 μm level.\(^{28,35} \) Experimental evidence in non-evaporating Spray A conditions also indicates very quick droplet SMD reduction to 1 μm level.\(^{36} \) On the contrary, detailed experimental information on evaporating droplet sizes in Spray A conditions using \( n \)-dodecane is still missing, let alone droplet sizes for other fuels (such as those studied here).\(^{1,2} \) Considering the above limitations, we apply a constant droplet size (Weber number \( We < 12 \)) at the nozzle exit without a droplet breakup model for all tested fuels with the aim to reduce ambiguity related to droplet breakup modeling with various fuels. Similar approach has been utilized in a spray LES simulation by Kaario et al.\(^{37} \) Thus, we aim to decrease the uncertainties related to the comparison between the various fuels. Neither droplet collision nor turbulent dispersion modeling is applied in the present work in similar fashion as in previous works.\(^{11,28,33,37} \)

**Droplet evaporation**

The mass transfer from the droplets due to evaporation is modeled according to Bird et al.\(^{38} \) The rate of change of the droplet mass is given as follows
\[ \frac{dm_d}{dt} = -A_d K_g p_g \ln \left( \frac{p_g - p_{r,c}}{p_g - p_{r,d}} \right) \]  
(13)

where \( A_d \) is the droplet surface area, \( K_g \) is the mass transfer number, \( p_g \) is the gas pressure, \( p_{r,c} \) is the vapor pressure in the droplet surroundings, and \( p_{r,d} \) is the vapor pressure at the droplet surface. The mass transfer coefficient \( K_g \) is modeled according to Ranz and Marshall\(^{39}\) and it is given as follows

\[ K_g = \frac{Sh D_m}{R_m T_m d} \]  
(14)

where \( Sh \) is the Sherwood number, \( D_m \) the vapor–gas mixture diffusivity, \( R_m \) the mixture gas constant, and \( T_m \) is the mixture temperature. The heat transfer at the droplet surface is derived from the droplet energy balance and the Ranz–Marshall correlations for \( Sh \) and Nusselt (\( Nu \)) number\(^{39}\) are applied in the equations for mass and heat transfer. The droplet evaporation time can be expressed as \( dm_d/dt = -m_d/\tau_e \), where the evaporation time is as follows

\[ \tau_e = \frac{6D_m Sh p_g \ln \left( \frac{p_g - p_{r,c}}{p_g - p_{r,d}} \right)}{\rho_e \rho_c^2} \]  
(15)

### Computational setup

The present study uses the ECN Spray A case as the baseline case. The Spray A experiments have been conducted with n-dodecane (\( n-C_{12}H_{26} \)). A fuel comparison is carried out for four additional fuels: diesel, methanol, DME, and propane. The non-reacting case selected here uses a mixture consisting of 0% O\(_2\) content in 900 K ambient temperature together with 150 MPa injection pressure and 90 \( \mu \)m nozzle hole diameter. Details of the operating conditions in the present fuel comparison are given in Table 2.

The most relevant fuel properties are provided in Table 3. The properties of liquid diesel have been taken from the literature\(^{40,41}\) and those of DME from Teng et al.\(^{42}\) The properties of n-dodecane, methanol, and propane are taken from the NIST database.\(^{43}\) The stoichiometric mixture fraction values have been calculated by assuming an oxidizer mixture with 21% of O\(_2\). It is seen that there is a large variation in the liquid density between the fuels ranging between 784 and 449.8 kg/m\(^3\) at the injection temperature. Figure 2(a) shows the temperature sensitivity of the liquid density for all the five fuels. Vapor pressure affects significantly the evaporation process as seen from equation (13). Figure 2(b) shows the vapor pressures of the fuels as a function of temperature. Large differences are observed also in this quantity. According to hypothesis 2, the liquid length will also be affected by vapor pressure. Due to the very low boiling point temperature of DME and propane, their injection temperature was set to \( T = 323 \) K. This is in line with liquid propane experiments comparing light fuel oil (LFO) and propane sprays in a room temperature spray bomb.\(^{44}\) In those experiments, it was necessary to decrease the injection temperature of propane due to fast vaporization inside the nozzle.

The injection velocity versus time profile of n-dodecane is obtained from experimental massflow profile (from CMT-Motores Térmicos (CMT) virtual profile generator). Figure 3 shows the injection velocity profiles for the studied five fuels. In the present study, the same injection pressure is used for all fuels (see Table 2). Therefore, the injection velocity for each fuel depends on its density according to \( U_i = \sqrt{2 \Delta p/\rho_i} \), where \( \Delta p \) is the pressure difference over the nozzle orifice and \( \rho_i \) is the liquid density (i refers to diesel, methanol, n-dodecane, DME, or propane). Since fuel mass flow rate is calculated from \( \dot{m}_i = \rho_i A D U_i \), where \( A D \) is the nozzle hole area, \( \dot{m}_i \) also varies between the fuels. The total injected fuel mass at \( t = 1.5 \) ms for each fuel is shown in Table 4.

Recent DNS of Spray A\(^{35}\) combining inner-nozzle simulation and the subsequent spray modeling using the volume of fluid (VOF) method suggests that at 1 mm distance from the nozzle exit the droplet sizes are very small, SMD = 0.5 \( \mu \)m being most probable. As explained previously in section “Droplet motion,” the present study utilizes a constant droplet size (i.e. \( W_e < 12 \)) at the nozzle exit without a droplet breakup model. In accordance with DNS of Spray A\(^{35}\) and LES of Spray A,\(^{26} \) we use 0.5 \( \mu \)m constant droplet size at the nozzle exit for all the simulated fuels. For reference, the total number of parcels in the present study is \( 1.2 \times 10^6 \) (at \( t = 1.5 \) ms).

### Table 3. Fuel properties.

<table>
<thead>
<tr>
<th>Fuels</th>
<th>Diesel</th>
<th>Methanol</th>
<th>n-dodecane</th>
<th>DME</th>
<th>Propane</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical formula</td>
<td>( C_{12}H_{26} )</td>
<td>( CH_3OH )</td>
<td>( C_{12}H_{26} )</td>
<td>( C_2H_6O )</td>
<td>( C_3H_8 )</td>
</tr>
<tr>
<td>Molecular weight (g/mol)</td>
<td>226</td>
<td>32</td>
<td>170</td>
<td>46</td>
<td>44</td>
</tr>
<tr>
<td>Density (kg/m(^3))</td>
<td>784</td>
<td>722.1</td>
<td>697.5</td>
<td>562</td>
<td>449.8</td>
</tr>
<tr>
<td>Vapor pressure (Pa)</td>
<td>2137</td>
<td>2.56 \times 10^4</td>
<td>1233</td>
<td>1.19 \times 10^6</td>
<td>1.71 \times 10^6</td>
</tr>
<tr>
<td>Latent heat (kJ/kg)</td>
<td>260.3</td>
<td>1046.9</td>
<td>325.9</td>
<td>356.9</td>
<td>285.3</td>
</tr>
<tr>
<td>Viscosity (kg/ms)</td>
<td>8.8 \times 10^{-4}</td>
<td>2.4 \times 10^{-4}</td>
<td>5.6 \times 10^{-4}</td>
<td>1.3 \times 10^{-4}</td>
<td>7.4 \times 10^{-5}</td>
</tr>
<tr>
<td>Critical temperature (K)</td>
<td>658</td>
<td>513</td>
<td>658</td>
<td>400</td>
<td>369.8</td>
</tr>
<tr>
<td>Critical pressure (Pa)</td>
<td>7.95 \times 10^6</td>
<td>1.82 \times 10^6</td>
<td>5.37 \times 10^6</td>
<td>4.25 \times 10^6</td>
<td></td>
</tr>
<tr>
<td>Stoichiometric mixture fraction (( – ))</td>
<td>0.0629</td>
<td>0.134</td>
<td>0.0627</td>
<td>0.100</td>
<td>0.0602</td>
</tr>
</tbody>
</table>

DME: dimethyl ether.

\(^{*}\)Diesel, methanol, and n-dodecane values are given at \( T = 363 \) K, and DME and propane are at \( T = 323 \) K.
The geometry of the computational domain resembles the combustion vessel at Sandia National Laboratories (the geometry of the computational domain is not exactly similar to the experimental geometry but the total volume matches that in the experiments), for which the experimental validation data are obtained. The computational domain is shown in Figure 4. Close to the nozzle exit, 31 mm cells are used in the radial directions, while in the axial direction cells are 62.5 mm long (1:2 aspect ratio). Further away from the nozzle between 10 and 30 mm (110–335D), cubical 62.5 mm cells are used. Starting from 30 mm, 125 mm cells have been utilized. Such a refinement strategy yields, altogether, 13M cells. The chosen mesh resolution is based on validation studies with different mesh resolutions in the ECN Spray A configuration.9,11,28 These studies suggest that close to the nozzle, 62.5 mm cell size is sufficient for capturing the high gradients and mixing of vapor and surrounding air. Here, we use 31 mm cells near the nozzle in order to have better description for the shear layer dynamics and, consequently, for the fuel vapor mixing. The near-nozzle resolution is highly relevant for the observed $L_{core}$. A constant time step size of $dt = 5 \times 10^{-8}$ s has been used so that $Co < 0.6$.

**Results**

**Spray A validation**

First, the LES model is validated in the Spray A conditions using n-dodecane as the liquid fuel. Figure 5 shows the liquid and vapor penetrations along with the

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Table 4. Total injected fuel mass at $t = 1.5$ ms.

<table>
<thead>
<tr>
<th>Fuels</th>
<th>Diesel</th>
<th>Methanol</th>
<th>n-dodecane</th>
<th>DME</th>
<th>Propane</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injected mass (kg)</td>
<td>$4.00 \times 10^{-6}$</td>
<td>$3.84 \times 10^{-6}$</td>
<td>$3.77 \times 10^{-6}$</td>
<td>$3.53 \times 10^{-6}$</td>
<td>$3.02 \times 10^{-6}$</td>
</tr>
</tbody>
</table>

DME: dimethyl ether.
The present numerical results on average liquid penetration are somewhat overestimated compared to the experimental data. The average liquid length between 0.2 and 1.5 ms is 11.7 mm compared to the average experimental penetration of 10.0 mm. Here, liquid penetration is defined according to the ECN guidelines by using 0.1% liquid volume fraction for the tip experimental data by considering five LES realizations. The present numerical results on average liquid penetration are somewhat overestimated compared to the experimental data. The average liquid length between 0.2 and 1.5 ms is 11.7 mm compared to the average experimental penetration of 10.0 mm. Here, liquid penetration is defined according to the ECN guidelines by using 0.1% liquid volume fraction for the tip
penetration. The average vapor penetration is noted to match well with the experimental data.\textsuperscript{45} Here, vapor jet tip is obtained as the axial location of 0.1% fuel vapor concentration value according to the ECN guidelines.

Figure 6 shows a comparison of the predicted radial mixture fraction profiles and experimental data.\textsuperscript{45} The mean values are well predicted for all downstream distances. The LES result has been first circumferentially averaged and then time averaged between 1.0 and 1.5 ms for the 17.8- to 28.8-mm axial distances and between 1.3 and 1.9 ms for the 40-mm distance. Finally, ensemble average is taken between the different injections. Root mean square (RMS) values are also compared to the experimental data in Figure 6. Relatively good comparison is seen except close to the spray axis. The deviation in the RMS values is partially related to the lower statistical accuracy in spatial averaging as the number of cells decreases when approaching the spray axis.

Figure 6(d) shows the mean gas phase velocity at the spray centerline. First, the velocity increases until \( z \approx 6D \) from the nozzle followed by a short spray core region (\( L_{\text{core}} \)) extending up to \( z = 29D \). Further away from the nozzle (experiments valid after \( z > 250D \)), the predicted mean velocity closely agrees with the experimental data. The predicted mean spray centerline velocity has been time averaged between 1.3 and 1.9 ms for each injection and then ensemble averaged between the injections.

**Fuel property comparison**

Next, the selected five fuels (diesel, methanol, \( n \)-dodecane, DME, and propane) are compared with one another. For \( n \)-dodecane, five realizations have been computed, whereas for the other fuels, three realizations are performed. First, liquid length is analyzed between the fuels after which the evaporation, vapor penetration, and mixing characteristics are shown. Finally, the equivalence ratio fields are studied for the selected fuels.

Liquid length. Figure 7 indicates visually how an LES/LPT model performs relatively close to the nozzle. Figure 8 depicts the average spray centerline velocity for the various fuels. Several aspects related to Figures 7 and 8 can be pointed out. (1) Within the spray core length (\( L_{\text{core}} \)), droplets lose most of their momentum (for gas phase acceleration). (2) The maximum velocities within \( L_{\text{core}} \) correlate with the injection velocities of the fuels. (3) \( L_{\text{core}} \) correlates with the \( St \) number of the fuel: increased \( St \) number indicates reduced momentum transfer rate from droplets to gas phase and hence increased \( L_{\text{core}} \). Propane has the shortest \( L_{\text{core}} \) (\( L_{\text{core}} = 20D, \ St = 0.4 \)) followed by DME (\( L_{\text{core}} = 24D, \ St = 0.5 \)), methanol (\( L_{\text{core}} = 28D, \ St = 0.54 \)), \( n \)-dodecane (\( L_{\text{core}} = 29D, \ St = 0.52 \)), and diesel (\( L_{\text{core}} = 32D, \ St = 0.56 \)). (4) Efficient evaporation process starts after \( L_{\text{core}} \) due to the mixing of hot ambient gases into the spray. (5) The RMS velocities peak right after the spray core region (maximum mean velocity) and as such they consistently peak earlier for DME and propane. (6) When the centerline velocities are normalized by the peak velocities and by normalizing the axial location by \( L_{\text{core}} \), characteristically similar gas velocity profiles are observed (see hypothesis 3).

The spray core lengths (\( L_{\text{core}} \)) have been defined as 95% of the averaged (temporally and ensemble) maximum spray centerline velocity (Figure 8). In Appendix 1, a consistency check is provided for the definition of \( L_{\text{core}} \) along with a mesh sensitivity analysis between 31, 62, and 125 \( \mu \)m mesh resolutions. In brief, the mesh sensitivity analysis indicates that \( L_{\text{liq}} \) is relatively unchanged for diesel, \( n \)-dodecane, and methanol for \( dx \ < 125 \mu \)m. In contrast, \( L_{\text{liq}} \) fluctuates more for DME and propane between the different meshes. For \( dx \ < 125 \mu \)m, \( L_{\text{liq}}/L_{\text{core}} \approx 4.5-5 \) for diesel, \( n \)-dodecane, and methanol, while \( L_{\text{liq}}/L_{\text{core}} \approx 1.5-2.5 \) for DME and propane. This difference can be explained by much shorter \( L_{\text{liq}} \) for DME and propane. In general, the sensitivity study indicates that with the 125-\( \mu \)m cell size, the definition of \( L_{\text{core}} \) is challenging due to the slow turbulence transition process. In contrast, with the 31- and 62-\( \mu \)m cell sizes, a consistent trend is observed.

![Figure 7. Near-nozzle region velocity field with droplets at t > 1 ms. The spray core and liquid lengths are shown.](image-url)
where $L_{\text{core}}$ either stays constant or increases with increasing cell size.

Figure 9 shows the fuel concentration fields together with droplets at $t = 0.5$ ms. Large differences are observed in the liquid part of the sprays between the “long liquid length” fuels (diesel, $n$-dodecane, and methanol) and the “short liquid length” fuels (DME and propane). The relatively long liquid penetration of methanol is somewhat unexpected based on its vapor pressure curve (Figure 2(b)). However, this is related to the high latent heat of methanol that lowers the gas phase temperature during evaporation (see Figure 15). It is worth noting that the stoichiometric isoline may deviate considerably within the spray envelope depending on the fuel. While the $\Phi = 1$ contour locates on the outer shell of the spray in the diesel case, for methanol, the stoichiometric conditions are found only at the core of the spray. Such a feature should be considered, when ignition and quasi-steady flame lift-off length estimates are constructed without detailed simulations.

Figure 10(a) illustrates the obtained liquid lengths for the various fuels. The variation in the average liquid length (0.2–1.5 ms) between different LES realizations was less than 2% for all fuels. With relevance to the obtained spray core lengths for the fuels (Figure 8), Figure 10(b) correlates $L_{\text{core}}$ to liquid density. It is seen that higher liquid density implies higher $L_{\text{core}}$. This result is related to the increasing $St$ number with fuel density ($St \sim \rho_f$). As already indicated, increased $St$ number indicates reduced momentum transfer rate from droplets to gas phase and hence increased $L_{\text{core}}$.

Figure 11(a) indicates a positive correlation between the fuel density and the average liquid length. The
observation is qualitatively in line with the experimental results by Kook and Pickett\textsuperscript{22} and Naber and Siebers.\textsuperscript{46} Based on the classical understanding, the entrained ambient mass ($\dot{m}_a$) per fuel mass ($\dot{m}_f$) is inversely proportional to fuel density according to $\dot{m}_a/\dot{m}_f \sim \sqrt{\rho_a/\rho_f}(\tan(\Theta)/D)z$, where $z$ is the axial distance from the nozzle and $\Theta$ is the spray angle.\textsuperscript{46} Thereby, a higher fuel density decreases $\dot{m}_a/\dot{m}_f$ leading to a longer liquid length when ambient entrainment (mixing) limited vaporization is assumed. The scaling law by Siebers\textsuperscript{14} indicates liquid length dependence on fuel–air density ratio as $L_{\text{liq}} \sim \sqrt{\rho_f/\rho_a}$. However, under-resolved near-nozzle LES/LPT simulations do not necessarily capture such scaling. Indeed, in contrast to the square-root behavior, here we observe more likely a linear scaling. In addition to resolution aspects, the deviation can be explained by the much lower density ratio range used in the present study $20 < \rho_f/\rho_a < 34$ compared to that used by Siebers\textsuperscript{14,13} $\rho_f/\rho_a < 220$. Based on the present numerical results, a higher fuel density leads to increased $L_{l_1}$, $L_{l_2}$, $L_{\text{core}}$, and $L_{\text{liq}}$. In the present model, $L_{l_1}$ is solely dependent on $St$ which further affects $L_{l_2}$, $L_{\text{core}}$, and $L_{\text{liq}}$. Thereby, consistent with Siebers et al.,\textsuperscript{14} the higher the density, the longer the $L_{\text{liq}}$. However, the new LES/LPT-specific aspect here is the intertwined character of the four length scales which emerge from $St$.

With relevance to hypothesis 1, Figure 11(b) shows the correlation between $L_{\text{core}}$ and $L_{\text{liq}}$. It is noted that
$L_{\text{liq}}$ is always larger than $L_{\text{core}}$ confirming hypothesis 1. Since droplet evaporation is weak within $L_{\text{core}}$ due to low entrainment, it is expected that $L_{\text{liq}} > L_{\text{core}}$. In particular, a linear fit to the data indicates that $L_{\text{liq}} = 10.8L_{\text{core}} - 189 [z/D]$. Thereby, in the present model setup, the spray core length has a significant effect on the liquid length. The liquid lengths in Figure 11(a) and (b) have been time averaged between 0.2 and 1.5 ms.

**A numerical test on virtual fuels.** According to Kook and Pickett, the effect of liquid density on liquid length is not fully understood. In general, assessing this in experiments is challenging as the density and other properties of fuels are typically interlinked. On the other hand, in numerical simulations it is possible to assess variation in only a specific quantity. The idea here is to emulate a virtual diesel and DME in order to quantify the sole effect of density on the liquid length.

Next, such a virtual fuel test is used to assess the liquid density effect on liquid length. Two test cases are carried out. In the first case, diesel liquid density was scaled down to that of DME (“low density diesel”). All other properties were as in the original diesel fuel. In the second case, liquid density of DME was scaled up to that of diesel (“high density DME”). Again, all other properties were as in the original DME fuel. Obviously, consistent with the endeavor to maintain a constant injection pressure, the density changes affected the resulting injection velocity according to $D\frac{U_{\text{inj}}}{\sqrt{\rho_1/\rho_2}}$, where subscript 1 refers to the original liquid density and 2 to the new density.

Figure 12(a) shows the resulting liquid length as a function of time, while Figure 12(b) indicates the effect of liquid density on the average liquid length. A positive correlation is observed between $L_{\text{liq}}$ and density, as already noted in Figure 11. However, it is observed that the “low density diesel” does not have the liquid length

![Figure 12](image_url)

**(a) Liquid penetration with modified liquid density. (b) Effect of changing liquid density on diesel and DME liquid length.** Red symbols indicate a model prediction by Naber and Siebers.

![Figure 13](image_url)

**(a) Ensemble-averaged vapor penetration for the different fuels. It is observed that the vapor penetrations increase with higher $L_{\text{core}}$. (b) Scaled vapor penetration for the different fuels showing similarity between the penetrations after $L_{\text{core}}$.**
of DME, nor does the “high density DME” have the liquid length of diesel. In fact, the change in liquid length only accounts between 15% and 33% of the difference between the liquid length of diesel and DME. The implication is that other properties, such as vapor pressure and boiling point, play a more significant role in defining the liquid length of a fuel (hypothesis 2). This is consistent with various previous observations.20–22

In addition, the importance of vapor pressure for the evaporation process can also be noted from equation (13). The accuracy of the zero-dimensional (0D) model by Naber and Siebers46 is noted to be within 10% in predicting the liquid length correctly. This can be considered to be a relatively good agreement since the changes in density were rather significant.

Evaporation and mixing. Next, we analyze the differences in evaporation and mixing between the fuels. The analysis is started by considering vapor jet penetrations with focus on proper normalization. It is of particular interest to find a similarity relationship between the different fuels. Figure 13(a) illustrates the average penetrations for the fuels. It is noted that the higher the $L_{\text{core}}$, the higher the final vapor penetration at $t = 1.5 \text{ ms}$. In Figure 13(b), the vapor penetration has been scaled with $L_{\text{core}}$. Time has been normalized with $t_w = W/U_L$, where $W$ is the spray width and $U_L$ is the axial velocity at $L_{\text{core}}$. Hence, the vapor penetrations of the various fuels are seen to scale with their respective spray core values. Similarly, in conventional gas-jet studies, a self-similar solution is obtained by normalizing the statistics by the steady potential core, that is, virtual origin.30 Importantly, related to hypothesis 3, $L_{\text{core}}$ seems to have a governing role in the final vapor jet penetration. Within the spray core, the droplet laden jet velocity is faster compared to the situation after the spray core (Figure 8). Thereby, the longer the spray core length, the greater the distance from the nozzle where the mixing-induced slower velocity region starts. When considering the absolute vapor jet penetrations between the fuels, Figure 14(a) indicates a positive correlation between the $L_{\text{core}}$ and the $L_{\text{vap}}$. However, we note that the trend remains relatively weak.

Earlier, it has been proposed that vapor penetration would not be related to liquid density.22 The argument is that the fuel spray momentum flux $M \sim D p A_D$ is not dependent on liquid density. However, the average vapor penetrations in Figure 14(b) are noted to be positively correlated with liquid density. This observation is not fully in line with the experimental findings in Kook and Pickett22 which were based on a more narrow liquid density range (755–870 kg/m$^3$) compared to the present study. Similar to Figure 14(a), the correlation
found here is relatively weak. It is also noted that the injection-to-injection fluctuations are higher for the vapor penetration ($\leq 4\%$) than for the liquid length ($\leq 2\%$) as expected.

Figure 15 illustrates the mean spray centerline temperature of the gaseous phase. The low local temperatures close to the nozzle are noted to be approximately $450-550$ K lower than the average gas phase temperature of $900$ K. The lowest near-nozzle temperatures are observed with DME and propane. This is due to their very fast evaporation rate leading to a rich and cool mixture which is consistent with Figure 9 as well. Apart from the near-nozzle region, the very high latent heat of methanol is clearly seen as lower centerline temperature compared to the other fuels. Right after the liquid length ($z > 125D$), methanol has $70-130$ K lower gas phase temperature compared to the other fuels. $n$-dodecane has the second highest latent heat which is reflected in equivalent centerline temperature. The low heat of evaporation of propane and low fuel concentration after the near-nozzle region is noted to result as the highest gas phase temperature.

**Equivalence ratio.** Next, we examine the equivalence ratio ($\Phi$) fields of the five fuels. Figure 16 shows the equivalence ratios from the spray centerline cross-section at $t = 1.5$ ms. The broad range of $\Phi$ values is clearly visible as propane has maximum $\Phi > 30$, DME has maximum $\Phi \sim 15$, while diesel, methanol, and $n$-dodecane have much lower maximum values $\Phi < 6$. The trends for DME and diesel are not shown here for brevity.
addition, it is interesting to look at the axial extent of the rich mixture region, here marked with $\Phi = 1$ iso-contour line, because it could potentially influence the combustion process of these fuels. For example, it has been observed that high temperature ignition, for typical liquid fuels, takes place between $1 < \Phi < 2$. For $n$-dodecane and diesel, the rich mixture area extends up to the vapor tip region of the gas jet. Interestingly, almost the same situation is seen for propane. The fuels containing oxygen, DME, and methanol have much shorter rich mixture penetration length.

Figure 17(a) shows the mean spray centerline mixture fraction, while Figure 17(b) presents the mean spray centerline equivalence ratio. Very high $\Phi$ values are noted close to the nozzle for DME and propane. This is related to their high vaporization rates. Diesel and $n$-dodecane have a second peak close to their $L_{\text{liq}}$ ($z \sim 125D$). Methanol has the lowest $\Phi$ all the way starting from the injector up to the vapor tip, while DME has the second lowest $\Phi$ after $z \sim 100D$. These differences could potentially have significant effect on combusting sprays, such as ignition phenomena, flame lift-off length (FLOL), or soot formation.\textsuperscript{57,48}

**Conclusion**

Here, diesel, methanol, $n$-dodecane, DME, and propane sprays were numerically compared using LES/LPT. The numerical setup was based on the ECN Spray A target conditions and the fuel was changed by keeping the injection pressure constant. As an obvious consequence, the injection velocity and fuel mass flow rates were affected. The modeling work presented here targets (1) to yield insight for modelers encountering complex spray cases with various fuels in their daily practices and (2) to provide a basis for potential experimental work on such fuel comparison. First, the LES/LPT model was validated in the ECN Spray A target condition with $n$-dodecane. Second, fuel comparison was carried out focusing mostly on the liquid lengths as well as on the vapor penetrations between the different fuels. The fuel-air mixing and the resulting equivalence ratio fields were also studied due to their paramount importance to, for example, emission formation under reacting conditions. Finally, also the injection-to-injection effects were analyzed. The main conclusions of the present study can be summarized as follows:

1. With relevance to hypothesis 1, a strong link between liquid length ($L_{\text{liq}}$) and spray core length ($L_{\text{core}}$) was confirmed to exist in the present numerical model according to $L_{\text{liq}} = 10.8L_{\text{core}} - 189 \ [z/D]$. In addition, $L_{\text{liq}}$ was shown to be positively correlated with liquid density consistent with previous studies.

2. A separate sensitivity test on virtual diesel and DME revealed that the virtual diesel with DME density did not yield the DME $L_{\text{liq}}$. Thereby, only a relatively weak dependency between the liquid density and $L_{\text{liq}}$ was observed. At maximum, 33% of the observed differences in $L_{\text{liq}}$ could be explained by the liquid density. This implies that thermodynamics plays a more important role in spray evaporation than density.

3. Vapor jet penetration ($L_{\text{vap}}$) was noticed to scale with $L_{\text{core}}$ according to $L_{\text{vap}}/L_{\text{core}} = A\sqrt{1/\tau_n}$, where $A = 2.37$ confirming hypothesis 3. Therefore, the vapor jet tip penetrations were similar between the fuels in non-dimensional form.

4. The evaporative cooling effect of various fuels at the spray centerline is shown here for the first time. The local gas phase temperature for methanol was noted to be 70–130 K lower compared to the other fuels.

5. The study revealed significant differences in the local equivalence ratio fields for the fuels. Diesel, $n$-dodecane, and propane exhibited relatively similar equivalence ratio fields after the near-nozzle region. In contrast, DME and methanol showed much lower average equivalence ratios within the mixture.

6. Injection-to-injection variations were observed, and the variations were lower for liquid length ($< 2\%$) compared to vapor penetration ($< 4\%$). Fuels with longer vapor penetration/liquid length (diesel, $n$-dodecane, and propane) had also higher variation in vapor penetration/liquid length.

It should be noted that the present results are of numerical character. The discussed length scales are clearly linked to one another, but the relationship is inherently dependent on (1) near-nozzle modeling assumptions and (2) grid resolution. We note that presently very little is known on near-nozzle gas and liquid velocities. Here, an attempt was made to shed light on certain features of LES/LPT spray models. The present numerical model explains consistently the trends in fuel property variation for all the fuels. Based on the study, we note that experimental evidence of axial velocity profiles for $z/D < 250$ would be highly valuable for various fuels.

**Declaration of conflicting interests**

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

**Funding**

The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This work was supported by the European Union HERCULES-2 project within Horizon 2020 research and innovation program under grant agreement no. 634135. The present study has also
been financially supported by the Academy of Finland (grant nos 289592 and 318024) and by the King Abdullah University of Science and Technology, Office of Sponsored Research (OSR) under Award OSR-2017-3319. The simulations presented above were performed using computer resources within the Aalto University School of Science “Science-IT” project.

**ORCID iDs**

Ossi Tapani Kaario [https://orcid.org/0000-0001-6765-0807](https://orcid.org/0000-0001-6765-0807)

Heikki Kahila [https://orcid.org/0000-0002-4103-7833](https://orcid.org/0000-0002-4103-7833)

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**Figure 18.** Effect of mesh resolution on the averaged maximum spray centerline gas phase velocity. Black marker represents the 31-μm mesh, green marker the 62-μm mesh, and red marker the 125-μm mesh.
Figure 19. (a) Effect of mesh density on $L_{\text{liq}}/L_{\text{core}}$. (b) Mesh comparison of $L_{\text{liq}}$.

Table 5. Spray core ($L_{\text{core}}$) [z/D] as a function of mesh density and 90%/95% $U_{\text{max}}$ definition.

<table>
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<th>Mesh/smallest cell size</th>
<th>$L_{\text{core}}$, 95% $U_{\text{max}}$</th>
<th>$L_{\text{core}}$, 90% $U_{\text{max}}$</th>
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<tr>
<td></td>
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<td>62 $\mu$m</td>
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<td>32</td>
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<tr>
<td>Methanol</td>
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<td>28</td>
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<tr>
<td>n-dodecane</td>
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<td>29</td>
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<tr>
<td>DME</td>
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<td>40</td>
</tr>
<tr>
<td>Propane</td>
<td>20</td>
<td>43</td>
</tr>
</tbody>
</table>

DME: dimethyl ether.

Table 6. Liquid length ($L_{\text{liq}}$) [z/D] as a function of mesh density.

<table>
<thead>
<tr>
<th>Mesh/smallest cell size</th>
<th>31 $\mu$m</th>
<th>62 $\mu$m</th>
<th>125 $\mu$m</th>
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<td>143.3</td>
<td>153.3</td>
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<tr>
<td>Methanol</td>
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<td>127.8</td>
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<tr>
<td>n-dodecane</td>
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<tr>
<td>Propane</td>
<td>34.4</td>
<td>64.4</td>
<td>44.4</td>
</tr>
</tbody>
</table>

DME: dimethyl ether.