



Title	Modeling wall film formation and vaporization of a gasoline surrogate fuel
Author(s)	Kobashi, Yoshimitsu; Zama, Yoshio; Kuboyama, Tatsuya
Citation	International journal of heat and mass transfer, 147, 119035 https://doi.org/10.1016/j.ijheatmasstransfer.2019.119035
Issue Date	2020-02
Doc URL	http://hdl.handle.net/2115/83975
Rights	© <2020>. This manuscript version is made available under the CC-BY-NC-ND 4.0 license http://creativecommons.org/licenses/by-nc-nd/4.0/
Rights(URL)	http://creativecommons.org/licenses/by-nc-nd/4.0/
Type	article (author version)
File Information	Manuscript_w_Figure_Table.pdf



[Instructions for use](#)

1 *Modeling Wall Film Formation and Vaporization of a Gasoline Surrogate*
2 *Fuel*

3 Yoshimitsu Kobashi^{1*}, Yoshio Zama² and Tatsuya Kuboyama³

4 ¹ Graduate School of Engineering, Hokkaido University, Japan

5 ² Graduate School of Science and Technology, Gunma University, Japan

6 ³ Graduate School of Engineering, Chiba University, Japan

7 **Abstract**

8 To simulate the wall film formation and vaporization processes in gasoline direct-
9 injection spark-ignition engines including considerations of the physical properties and
10 vapor-liquid equilibrium of multi-component fuels, spray-wall interaction sub-models
11 were implemented with the 3D-CFD software HINOCA which has been developed for
12 automotive engine cylinder simulations. The models used were the Senda model for
13 spray-wall impingement including splash, deposition, droplet-droplet interactions, and
14 droplet-film interactions; the O'Rourke model for heat transfer and film vaporization; a
15 simple film flow model considering momentum conservation; and Raoult's law for vapor-
16 liquid equilibrium. First, the model validated the calculated results for a single-
17 component fuel (iso-octane) through comparisons with experimental data in terms of wall
18 film area and heat flux between the wall and film. Second, numerical simulations were
19 conducted with a 5-component gasoline surrogate fuel which was designed taking into
20 account the average octane number, aromatic content, and distillation characteristic. The
21 results showed clear differences in the contributions of the 5 components to the wall film,
22 and the possibility that the aromatic content with higher carbon atoms could be a source
23 of soot formation.

24 **Keywords:** Gasoline Direct Injection Engine, Wall Film, Spray, Gasoline Surrogate Fuel

25
26 **Nomenclature:**

27 A Area [m^2]

28 d Diameter of droplet [m]

29 E Kinetic energy of droplet or wall film [J]

* Corresponding author at: Kita 13, Nishi 8, Kita-ku, Sapporo, Hokkaido, Japan

E-mail address: kobashi@eng.hokudai.ac.jp (Y. Kobashi)

30	E_{crit}	Energy needed to break up droplet [J]
31	K	Critical parameter which correlates deposition and splash [-] ($= Z \cdot Re^{1.25}$)
32	La	Laplace number [-]
33	m	Mass of droplet or wall film [kg]
34	M	Total mass [kg]
35	Oh	Ohnesorge number [-] ($= \sqrt{We}/Re$)
36	p	Pressure [Pa]
37	\dot{Q}	Heat flux [W/m^2]
38	Re	Reynolds number [-]
39	t	Time [s]
40	t_r	Time interval parameter [-] ($= \Delta t/\tau_{res}$)
41	T	Temperature
42	v	Velocity of droplet or wall film [m/s]
43	We	Weber number of droplet [-]
44	y	Mass fraction [-]
45	Subscripts:	
46	1	Before impingement on wall
47	2	After impingement on wall
48	<i>crit</i>	Critical
49	<i>f</i>	Wall film
50	<i>g</i>	Gas
51	<i>i</i>	Component
52	<i>inj</i>	Injection
53	<i>l</i>	Liquid
54	<i>n</i>	Normal to wall
55	<i>s</i>	Surface of wall film
56	<i>t</i>	Tangential to wall
57	<i>w</i>	Wall
58	Greek symbols:	
59	δ	Film thickness [m]
60	δ_N	Non-dimensional film thickness ($= \delta/d_p$) [-]
61	Δt	Time interval between two continuous droplets reaching a wall [s]
62	φ	Equivalence ratio [-]
63	λ	Thermal conductivity [$W/(m \cdot K)$]
64	μ	Viscosity [$Pa \cdot s$]
65	ρ	Density [kg/m^3]
66	σ_l	Surface tension of liquid [N/m]
67	τ_{air}	Shear force between airflow and wall film [Pa]

68 τ_{res} Droplet residence time on wall [s]
69 τ_w Friction stress between wall film and wall [Pa]

70 **1. Introduction**

71 Due to the improvements in thermal efficiency and engine power output, modern
72 gasoline engines have employed direct-injection spark-ignition (DISI) systems. In this
73 strategy, however, soot may be produced as the result of the formation of a liquid film
74 (wall film) on the piston surface and from pool fires fed by the wall film [1][2]. To develop
75 DISI engines which avoid the wall film formation and reduce the mass and number of
76 soot particulate, it is crucial to develop numerical analysis schemes that treat the spray-
77 wall interaction, film flow, and film vaporization processes in more detail, as well as to
78 develop an understanding of the source of the soot particles.

79 Over the past decades, a number of spray-wall impingement models have been
80 developed. Naber and Reitz [3] proposed one of the first impingement models, which
81 considers three regimes including stick, rebound, and slide of the impinged droplets. Bai
82 and Gosman [4] proposed a more detailed model which includes wider impingement
83 regimes; adhesion (stick and spread) and splash for dry walls; rebound, spread, and splash
84 for wetted walls, while employing We_{crit} as the critical Weber number for regime transition
85 criterion. Stanton and Rutland [5], assuming wetted walls, also derived a model in which
86 We_{crit} was used as the critical criterion of the regime transition. A single-droplet
87 experiment by Mundo et al. showed that the limit of adhesion and splash can be correlated
88 in terms of the Reynolds number Re and the Ohnesorge number Oh , and an additional
89 critical parameter $K (= Oh \cdot Re^{1.25})$ was also introduced [6]. The recent study [7] has
90 demonstrated that using Kuhnke's spray-wall interaction model [8], in which the effects
91 of the critical parameter K and a thermal parameter defined as the ratio of the wall
92 temperature and the droplet boiling temperature are considered, a good agreement of wall

93 film height distributions between the measurements and calculations. Zhang et al.
94 proposed a model considering the droplet/film interaction on the film dynamics, and the
95 dissipative energy loss during the expansion of the lamella formed by the deformation of
96 the deposited droplet [9]. Senda et al. [10-11] proposed a more comprehensive model
97 which considers the effects of the Weber number We and liquid film thickness on spray-
98 wall interaction behaviors as well as a We_{crit} parameter as the regime transition criterion.

99 For wall film vaporization, the model developed by O'Rourke and Amsden [12] is
100 reliable and often used in the literature, as it consists of physical processes like the wall
101 film energy balance, heat transfer from the wall to the film and to the impinging droplets,
102 and the temperature profile in the film normal to the surface. Recently, Jiao and Reitz
103 employed the model to simulate the soot formation from the wall film in a DISI engine
104 [13]. Another important aspect in the wall film vaporization is the distillation
105 characteristics of fuels as gasoline is multicomponent fuel, and the component remaining
106 in the wall film has a significant impact on the soot formation, like for example, many of
107 high boiling point gasoline components are aromatics [14] which tend to increase soot
108 emissions. A number of vaporization models have been developed for multicomponent
109 fuels [15-21]. However, there have been few studies that conducted extensive validation
110 of the models including the film evaporation process and heat flux between the wall and
111 wall film, although the heat flux modeling was studied well for single component liquid
112 [9, 22]. In addition, little has been reported on the correlation between the vaporization
113 process of wall films and the distillation characteristics of fuels. Köppler et al. proposed a
114 gasoline surrogate fuel consisting of n-hexane, iso-octane and n-decane [23]. Using this
115 surrogate fuel, Köppler et al. validated the heat transfer between the piston wall and wall
116 film [7], and showed a good correlation between the calculated mixture formation and the
117 measured particulate emissions [23]. Schulz and Beyrau demonstrated that this gasoline
118 surrogate fuel represents the wall film vaporization of real gasoline [24].

119 The purpose of the present study is to develop a simulation scheme that predicts the
120 wall film formation and vaporization of multicomponent fuel sprays with a high degree
121 of accuracy, and to investigate the potential source of soot in DISI gasoline engines. Sub-
122 models accommodating this were incorporated into the platform software HINOCA [25].
123 The calculated wall film area and heat flux between the wall and wall film in this study
124 were validated via comparisons with data obtained by measurements developed specially
125 for this study, followed by modifications to the sub-models. A gasoline surrogate fuel was
126 designed to meet the average octane number, aromatic contents, and distillation
127 characteristic of real gasoline fuel. Finally, the correlation between the formation and
128 vaporization processes of wall films and the components of the gasoline surrogate fuel
129 was examined.

130 **2. Model description**

131 The platform software employed in this study is HINOCA [25] which is an automotive
132 engine cylinder simulation software, based on fully compressible Navier-Stokes
133 equations which are filtered for Large Eddy Simulation (LES), and employs Cartesian
134 grid and immersed boundary (IB) methods. The grid is set to be equally spaced for
135 simplicity of the code. The wall adapting the local eddy viscosity (WALE) model which
136 is capable of predicting wall limiting behaviors without the calculation of the wall
137 distance [26] is also adopted.

138 Spray models are based on a DDM (Discrete Droplet Model) which describes spray
139 droplets by stochastic particles that are termed parcels, and spherical parcels having a
140 diameter equal to the nozzle hole are injected. The KH-RT model [27] was used to
141 simulate the primary and secondary breakup processes. When a spray parcel impinges on
142 a wall, it is converted into a wall film parcel. This process was modeled, and here in this

143 following section, where wall film models implemented in HINOCA are introduced.

144 2.1 Spray-wall impingement model

145 A number of spray-wall impingement models have been proposed [3-11]. The authors
146 selected a model developed by Senda et al. [10-11] as the base model since it is
147 comprehensive with consideration of the effects of the liquid film (wall film) thickness,
148 and the diameter and Weber number of the impinging droplets $We_{1,n} (= \rho_l d_1 v_{1,n}^2 / \sigma_l, \rho_l :$
149 liquid density, $d_1 :$ diameter of droplets impinging on a wall, and, $v_{1,n} :$ velocity component
150 of droplets, normal to a wall, with, $\sigma_l :$ the surface tension). Further, the present study
151 focuses on the wall film formation, and agreement of the wall film shape between the
152 measurements and numerical simulation was demonstrated with the Senda model [28].
153 Note that the Senda model developed in the literatures [10-11] is only applicable to a
154 limited range of temperatures of droplets and walls, where the boiling-induced breakup
155 is negligible.

156 Figure 1 (a) summarizes the classification of impinging droplet characteristics in the
157 Senda model, where the non-dimensional film thickness δ_N is defined as the film thickness
158 δ divided by droplet diameter d_1 [10-11]. The model is divided into two regimes based on
159 the Weber number, $We_{1,n}$ of an impinging droplet as shown in Fig.1 (b).

160 (1) Low Weber number model ($We_{1,n} \leq 300$)

161 (1-1) Wet wall

162 Three types of breakup: rim type; cluster type; and column type are considered when
163 a droplet impinges on a wet wall, and the droplet diameter after the breakup d_2 is given
164 as a function of the non-dimensional film thickness δ_N to fit experimental results as
165 follows [29]:

$$166 \quad d_2/d_1 = 0.6478 - 0.5480 \delta_N + 1.9825 \delta_N^2 - 2.1082 \delta_N^3 + 0.6894 \delta_N^4 \quad (1)$$

167 The droplet velocity after the breakup is determined with the Weber number of the
168 bouncing droplet. The relation between the Weber number and non-dimensional film

169 thickness δ_N is determined to fit experimental results as follows [29]:

$$170 \quad We_{2,t} = 0.3818 - 0.00537 \delta_N - 0.8937 \delta_N^2 + 0.8644 \delta_N^3 - 0.2301 \delta_N^4 \quad (2-1)$$

$$171 \quad We_{2,n} = -2.1518 + 1.1493 \delta_N + 26.238 \delta_N^2 - 24.480 \delta_N^3 - 5.5650 \delta_N^4 \quad (2-2)$$

172 where $We_{2,t}$ is the Weber number, tangential to a wall, of the bouncing droplet, and $We_{2,n}$
173 is that normal to a wall.

174 (1-2) Dry wall

175 Almost all droplets stick when impinging on a dry wall, but some bounce after the
176 impingement. A subsequent droplet likely collides with the foregoing one, causing
177 breakup of the droplets. The probability of the breakup can be estimated with a time
178 interval parameter, t_r which is defined as follows:

$$179 \quad t_r = \Delta t / \tau_{res} \quad (3)$$

180 where Δt is the time interval between the two continuous droplets reaching a wall, and τ_{res}
181 is the droplet residence time on the wall, expressed as follows [30]:

$$182 \quad \tau_{res} = \pi \sqrt{\rho_l d_1^3 / (16\sigma_l)} \quad (4)$$

183 The model defines that the breakup occurs at the t_r from 0.4 to 0.6, where high breakup
184 probabilities were determined in experiments [29]. In the present study, the diameter and
185 velocity of the droplet after the breakup was calculated assigning $\delta_N = 0$ in Equations (1),
186 (2-1) and (2-2).

187 (2) High Weber number model ($We_{1,n} > 300$)

188 At $We_{1,n}$ numbers above 300, the model assumes that the droplet impinging on a dry
189 wall causes the breakup while that impinging on a wet wall leads to splashing or
190 deposition in the wall film. The critical Weber number, We_{crit} is used as a criterion that
191 distinguishes splashing from deposition, according to the experimental work of Cossali
192 et al. [31] as follows:

$$193 \quad We_{crit} = (2100 + 5880 \delta_N^{1.44}) La^{-0.2} \quad (5)$$

194 where La is the Laplace number ($= \sigma_l \rho_l d_1 / \mu_l^2$, μ_l : liquid viscosity).

195 The droplet diameter after the breakup and splash is determined according to the
196 experimental data of Mundo et al. as follows [32]:

$$197 \quad d_2/d_1 = 3.932 \times 10^2 \times K^{-1.416} \quad (\text{for rough surfaces}) \quad (6)$$

$$198 \quad d_2/d_1 = 3.903 \times 10^{10} \times K^{-5.116} \quad (\text{for smooth surfaces}) \quad (7)$$

199 where K is a non-dimensional parameter defined as:

$$200 \quad K = Oh \cdot Re^{1.25} \quad (8)$$

201 with $Oh = \mu_l / \sqrt{\rho_l \sigma_l d_1}$ the Ohnesorge number and $Re = \rho_l d_1 v_{1,n} / \mu_l$ the Reynolds
202 number. This study applied Equation (6) to wet walls, and Equation (7) to dry walls, in
203 analogy with the original model.

204 The droplet velocity after the wall impingement is critical to determine the velocity of
205 growth of a wall film as will be explained in Equation (12). The original model calculates
206 the velocity while solving the balance between the energy needed to break up the droplet,
207 E_{crit} and the kinetic energies of the droplets before the impingement, E_1 and after the
208 impingement (including droplets becoming part of the liquid film, E_f and being airborne,
209 E_2) as:

$$210 \quad E_1 - E_{crit} = E_f + E_2 \quad (9)$$

211 The film flow calculated with Equation (9), however, was inconsistent with the
212 experimental results in this study. This study then employed the film flow model proposed
213 by Kalantari and Tropea [33] which expresses the droplet velocity after the wall
214 impingement as:

$$215 \quad v_{2,n}/v_{1,n} = -1.1 \cdot We_{1,n}^{-0.36} \quad (10)$$

$$216 \quad v_{2,t} = 0.862 \cdot v_{1,t} - 0.094 \quad (11)$$

217 with $v_{2,n}$ the velocity component, normal to a wall, and $v_{2,t}$ that tangential to a wall.

218 2.2 Film flow model

219 To calculate the flow characteristics of the wall film, a momentum balance equation

220 was solved. In the model, parcels of injected fuel are deposited on one IB cell which is
 221 assumed to form a rigid wall film, and the equation is expressed as:

$$222 \quad \Delta(M_f \vec{v}_f) = \sum(m_1 \vec{v}_1 - m_2 \vec{v}_2) + \vec{\tau}_{air} A dt - \vec{\tau}_w A dt \quad (12)$$

223 where M_f and \vec{v}_f are the total mass and velocity vectors of the wall film in a cell. The
 224 first term on the right side is the momentum provided from impinging droplets to the wall
 225 film, with m_1 and m_2 the mass of the droplets before and after impingement. The A is the
 226 area of the wall film and is identical with the cross-sectional area of the IB cell. The τ_{air}
 227 is the shear force between the airflow and wall film, and the τ_w is the frictional stress
 228 between the wall film and wall.

229 2.3 Film vaporization model

230 The model proposed by O'Rourke and Amsden [12] was used to calculate the film
 231 vaporization and heat transfer to the wall film. This model considers the change in the
 232 structure of the turbulent boundary layers above the wall films because of the gas velocity
 233 normal to the wall induced by vaporization and consequent convective transport. Two
 234 assumptions are made: (1) the total transport is independent of the coordinate normal to
 235 the wall and is the sum of the transport due to turbulent diffusion and due to convection
 236 by the vaporization velocities; (2) there is a linear increase in the turbulent diffusivity
 237 with the distance from the wall.

238 The temperature profile in the wall film normal to the wall is approximated as linear,
 239 varying from the temperature at the wall T_w to the mean temperature T_f in the lower half
 240 (the half nearest the wall) of the film, and from T_f to the film surface temperature T_s in
 241 the upper half of the film, as schematically suggested in Fig.2, where \dot{Q}_s and \dot{Q}_w are
 242 the heat fluxes between the gas and liquid film, and between the liquid film and wall, and
 243 λ_l is the thermal conductivity of the liquid. The energy balance is solved considering the
 244 heat transfer between the gas and film and the wall and film, and the changes in the
 245 temperature resulting from the film movement and droplet impingement.

246 The mass vaporization rate, \dot{M}_{vap} is given by

$$247 \quad \dot{M}_{vap} = H_y \ln \left(\frac{1-y_v}{1-y_s} \right) \quad (13)$$

248 where y_v is the fuel vapor mass fraction, and y_s is the equilibrium vapor mass fraction at
249 the surface temperature. One of the authors (Kobashi) has developed a method for
250 estimating the vapor-liquid equilibrium of vaporizing multicomponent sprays with the
251 Soave-Redlich-Kwong equation of state combined with the UNIFAC method [34], but
252 Raoult's law for an ideal multicomponent mixture was employed for simplicity in this
253 study since the fuels used were with non-polar molecules, and all numerical simulations
254 were implemented at atmospheric pressure.

255 H_y is expressed as:

$$256 \quad H_y = \frac{\rho_g u_\tau}{y_c^+ S_{CL} + \frac{S_{CT}}{\kappa} \ln \left(\frac{y^+}{y_c^+} \right)} \quad \text{if } y^+ > y_c^+$$
$$257 \quad H_y = \frac{\rho_g u_\tau}{y^+ S_{CL}} \quad \text{if } y^+ < y_c^+ \quad (14)$$

258 where S_{CL} and S_{CT} are the laminar and turbulent Schmidt numbers, u_τ is the frictional
259 velocity defined as $u_\tau = (\tau_w / \rho_g)^{0.5}$, $y^+ = y u_\tau / \nu_L$, y is the distance normal to the wall, τ_w is
260 the wall shear stress, and ν_L is the kinematic viscosity. The transition between the laminar
261 and fully turbulent region, y_c^+ is 11.05, and Karmann's constant, κ is 0.433, according
262 to the literature [12].

263 The wall film thickness, δ was calculated as follows:

$$264 \quad \delta = \frac{\sum V_p}{A} \quad (15)$$

265 where $\sum V_p$ is the sum of the volume of parcels adhering to a wall defined by a Cut-Cell
266 method, and A is the cross-sectional area of the wall.

267 **3. Experimental Setup and Procedures**

268 **3.1 Measurement of wall film area**

269 In a constant volume vessel, the temporal development of the wall film area was
270 measured with the refractive index matching (RIM) method which allows photography
271 of the wall films based on the principle that the light transmission decreases when liquid
272 is deposited on a frosted glass wall [2]. A schematic of the experimental system is shown
273 in Fig.3. The fuel was pressurized by the high pressure pump and injected into the vessel
274 in which the frosted glass wall was placed. Use of continuous light emitted from the metal
275 halide lamp (Photron: HVC-UL) and a high speed digital camera (Vision Research:
276 Phantom-MIRO) allowed sequential imaging of deposition and flow of the wall film from
277 the rear of the wall. Electrical signals sent from the pulse generator were utilized to
278 determine the fuel injection period as well as the triggers for the injection and image
279 recording.

280 The arrangement of the sprays and the wall are shown in Fig.4. The injector was
281 equipped with a 6-hole nozzle. Spray from the No.6 nozzle impinged on the wall at an
282 angle of 23° from a distance of 22 mm, while sprays of the No.2, 3, and 4 nozzles which
283 were in the same plane and impinged at the angle of 45° . As the sprays of the No.1 and 5
284 nozzles geometrically do not result in wall impingement, the numerical simulations were
285 implemented only for the sprays No.2, 3, 4, and 6. The rear view image in Fig.4 (c) is an
286 example of the RIM image with the numbers of the sprays superimposed near the
287 positions where impingement takes place. The dark areas represent regions where the wall
288 film has formed, while the thicker film is located near the position of spray impingement
289 and appears white as the wall film here became thicker than the surface roughness of the
290 frosted glass.

291 3.2 Measurement of the wall heat flux

292 The experimental setup for the measurements of the wall heat flux is shown in Fig.5.
293 As shown in Fig.5 (a), the wall was placed below the injector, and a Loex-bodied thin-
294 film thermocouple developed by Enomoto [35] was embedded in the wall. The body

295 material of the Loex has a density, heat capacity, and heat conductivity closely equivalent
296 to the wall material. The diameter of the body of the thermocouple was 3.0 mm, and the
297 diameter of the electrically-insulated center wire made of constantan was 0.15 mm. A thin
298 copper film, about 10 μm thick, was formed to establish a hot junction of the Loex –
299 constantan with a very low heat capacity on the top surface of the thermocouple, while a
300 cold junction was made inside the body of the thermocouple at the distance of 3.5 mm
301 from the top surface. Another thermocouple, of iron – constantan, was embedded near the
302 cold junction and with the cold junction immersed in ice water. Overall, it was possible
303 to measure the temperature differences between the hot and cold junctions of the Loex –
304 constantan thermocouple and to calculate the wall heat flux as the absolute temperature
305 of the cold junction was measured with the iron – constantan thermocouple. The accuracy
306 of the measurement system has been investigated by Enomoto who reported that the wall
307 heat flux measured was about 5 % higher than the true value [35].

308 The thermoelectrical power from the thermocouple was amplified and recorded with
309 the data logger. The temperature of the wall surface was controlled with a heater, based
310 on the temperature measured by the iron – constantan thermocouple. The fuel was
311 pressurized with nitrogen gas and introduced to the injector. Electrical signals sent from
312 the pulse generator were used to determine the fuel injection period as well as the triggers
313 for the injection, data logging, and image recording. The wall film images were recorded
314 with the high speed digital camera (NAC: HX-3) and the wall film was illuminated with
315 the metal halide lamp (Photron: HVC-UL), and recorded from a position diagonally
316 above the wall.

317 Figure 5 (b) shows the layout of the fuel sprays and thermocouple for the measurement
318 of the wall heat flux. The fuel injector equipped with a 6-hole nozzle was different from
319 the one used in the film area measurements. The fuel spray injected from one nozzle hole
320 orthogonally above the wall impinged on the thermocouple embedded in the piston while

321 the other sprays were masked with the metallic plate. The gap between the target spray
322 and metallic plate is large enough, and it was confirmed that the other sprays have a
323 limited impact on the target spray after the wall impingement with the movie.

324 **4. Validation of the Model with iso-Octane**

325 4.1 Wall film area

326 Table 1 details the conditions under which the wall film areas were compared in the
327 measurement and calculations. A single component fuel, iso-octane (2,2,4-
328 trimethylpentane, iC_8H_{18}) was injected from the 6-hole nozzle with nozzle diameters of
329 0.13 mm into a constant volume vessel at room temperature and atmospheric pressure.
330 The injection pressure, p_{inj} was changed as the variable.

331 Figure 6 shows the images of (a) the wall film photographed with the RIM method
332 from the rear of the frosted glass wall, and (b) the calculated wall film thickness, with the
333 injection pressure, p_{inj} of 13 MPa. In Fig.9 (a), the darkly colored areas represent regions
334 where the wall film formed, and the thick film near the location of the spray impingement
335 appear as white. In Fig.7 (b), the overall shape of the calculated wall film is very similar
336 to the photograph, the calculated thicker region presented in red corresponds to the white
337 regions in the measured image. The thicker films formed at the location far from the spray
338 impingement points, and the film thickness other than the thicker region was in the order
339 of $10\ \mu m$. These observations are in agreement with film thickness distributions of a
340 literature [36], in which the experimental conditions are similar to the present study.

341 Figure 7 shows profiles of the wall film area plotted for different injection pressures
342 p_{inj} . Here, the film area in the calculations was defined as the sum of the areas where the
343 film thickness was larger than $1\ \mu m$, corresponding to the detectable level in the
344 measurements. The measured results show that the wall film area increased with time

345 until 5.5 ms, and then it maintained a nearly constant size. The higher injection pressure
346 increased the wall film area. The calculations captured the changes in the initial spreading
347 process of the film area, and the fact that the film area increased with increasing injection
348 pressure, however the calculations underestimate the film area at the lower injection
349 pressure. The reason for this discrepancy may be due to underestimation of the impinging
350 droplet diameter at the lower injection pressures, and may be attributed to the droplet
351 breakup model and the critical Weber number.

352 4.2 Heat flux between the film and wall

353 The heat flux between the film and wall, \dot{Q}_w would be an appropriate factor to
354 validate the model as it plays an important role in the wall film vaporization as well as
355 because accurate predictions of wall film formation and movement are required in the
356 predictions.

357 Table 2 details the conditions under which the measured and calculated wall heat flux
358 values were compared. A single component fuel, iso-octane was injected from an injector
359 equipped with a 6-hole nozzle with nozzle diameters of 0.248 mm. The long injection
360 period of 4 ms was adopted to measure the heat flux in a quasi-steady state. The heat flux
361 was measured for one spray impinging on the wall as explained in Section 3.2, while the
362 calculation assumed a single-hole nozzle with the same diameter. In the calculation, the
363 heat flux between the film and wall, \dot{Q}_w was calculated as follows:

$$364 \quad \dot{Q}_w = \lambda_l \frac{T_f - T_w}{\delta/2} \quad (16)$$

365 where λ_l is the liquid thermal conductivity.

366 Figure 8 shows photographs and calculated images of the wall film at 7 ms after the
367 start of injection where the wall film is in the quasi-steady state. In Fig.8 (a), the photos
368 were taken from the diagonally upward direction at an angle of 45° as shown in Fig.5 (a),
369 making the scale of the picture longitudinally compressed. The calculated images in Fig.8

370 (b) superimposes dashed circles equivalent to the measured diameter of the wall film.
371 Unlike the situation in Fig.7, the calculations overestimated the wall film spread. This
372 may be due to differences in the wall surface roughness, or due to differences between
373 the effect of the oblique and orthogonal impingements. However, the calculations
374 captured the experimental observation that the edge of the wall film was thicker than the
375 central portion. There was boiling at the center of the wall film in both the (a)
376 photographed and (b) calculated images at T_w of 100°C which is slightly higher than the
377 boiling point of iso-octane. Further, at the T_w of 120°C, the boiling was taking place over
378 the whole of the wall film in the photograph (a), and the calculation shows less wall film
379 in (b), indicating that actually most of the wall film had vaporized.

380 Figure 9 plots the profiles of the heat flux of the measurements and calculations, where
381 the wall temperature, T_w was set at 120°C. In the calculations, the heat flux was calculated
382 for the region just below the nozzle, considering the area equivalent to the top surface of
383 the thermocouple. The direction of the heat transfer toward the wall was defined as
384 positive. Since the temperature of the fuel spray is lower than the wall temperature, the
385 heat transferred from the wall to the spray or wall film results the negative value. Both
386 the measured and calculated heat fluxes decreased rapidly immediately after the spray
387 impingement, and they gradually increased from 4 ms after the start of injection when no
388 spray reached impingement to the wall. Averaging the heat flux from 1 ms to 2.5 ms after
389 the start of injection, the measured value was -4.1 MW/m^2 , with the calculated value -3.5
390 MW/m^2 , a difference of about 15%.

391 Figure 10 plots the change in the heat flux with the wall temperature T_w as measured
392 and calculated. The heat flux decreased with the increase of the T_w . According to Schulz
393 et al. who determined the transient wall temperature and heat flux at the spray
394 impingement zone using an infrared thermography for an extensive range of influencing
395 factors, the heat fluxes during the spray impact are -3 MW/m^2 at the T_w of 80°C and -4

396 MW/m² at the T_w of 120°C [37], which correspond to the heat flux measured in the present
397 study. There were the differences in the absolute values of the heat flux between the
398 measurements and calculations, but the decreases with T_w are practically identical, with
399 similar global gradients.

400 Figure 11 plots the change in the heat flux with the distance from the nozzle to the
401 wall as measured and calculated. The heat flux increased with the increase of the distance.
402 The trend corresponds to the results of Schulz et al. [37], but it has a limited impact on
403 the heat fluxes measured and calculated within the range of the present study.

404 **5. Formation and Vaporization of the Wall Film with a Gasoline Surrogate Fuel**

405 5.1 Specification of the gasoline surrogate fuel

406 The composition of the gasoline surrogate fuel was determined to meet the octane
407 number, aromatic content, and distillation curve of the regular gasoline available in Japan.
408 It was assumed that the RON (research octane number) can be expressed as a volume-
409 averaged property, and the distillation curve was calculated following a calculation
410 method reported elsewhere [38]. It assumes an open system where the vaporized fuel is
411 vented. The heat, Q is continuously supplied from outside the system to the liquid, and
412 consumed in increasing the temperature up to and vaporizing the liquid. Considering a
413 vapor-liquid equilibrium (VLE), the composition in the liquid and vapor phases is
414 calculated and renewed at every time step, via a VLE calculation with NIST
415 SUPERTRAPP [39] which solves the Peng-Robinson equation of state [40].

416 Table 3 (a) details the components selected as the composition of the gasoline
417 surrogate fuel, the properties, and the mixing fractions. The five components are needed
418 to reproduce the desired distillation curve. Considering that the aromatics have higher
419 boiling points than a range of other gasoline components [14], 1,2,4-trimethylbenzene

420 (C₉H₁₂) was chosen as the highest boiling point component, however, iso-butylbenzene
421 (C₁₀H₁₄) was used as a substitute in the calculations due to the lack of information on the
422 thermodynamic properties of 1,2,4-trimethylbenzene. Iso-pentane was chosen as the
423 lowest boiling point component since it exhibits low reactivity (high RON), and it is liquid
424 at the conditions involved here and is simple to handle. The medium boiling point
425 components: n-heptane, iso-octane, and toluene, function to adjust the RON and aromatic
426 content.

427 The properties and distillation curve of the gasoline surrogate fuel are shown in Table
428 3 (b) and Fig.12. The properties of the surrogate fuel are closely identical to the reference
429 regular gasoline (commonly available in Japan). The distillation curve appears well
430 matched, except for the disagreement at the distillation ratios from 40% to 80% and near
431 the end point.

432 5.2 Formation and Vaporization of the Wall Film

433 In this section, first, the accuracy of the present model is evaluated by comparing the
434 film area of the gasoline surrogate fuel as measured and calculated. Second, the
435 vaporization process of a wall film of the gasoline surrogate fuel is discussed with the
436 calculations.

437 The conditions under which the film area was evaluated are identical with Table 1,
438 with the injector - wall arrangement, that in Fig.3 and 4, except for the wall temperature
439 T_w of 60°C and the fuels tested. The fuel was injected with the pressure of 13 MPa into
440 ambient air at atmospheric pressure.

441 Figure 13 shows the changes in the calculated film mass distributions of (a) the
442 gasoline surrogate fuel and (b) the iso-octane. For the gasoline surrogate fuel there were
443 no significant observable changes in the distributions of the wall film mass as shown in
444 Fig.13 (a). The wall film vanished from the thinner film with (b) iso-octane since the heat
445 flux inversely increased with the film thickness as expressed in Eq.(16).

446 Figure 14 plots (a) the measured and calculated area of the wall film and (b) the
447 calculated film mass of the gasoline surrogate fuel and iso-octane at the wall temperature
448 T_w of 60°C. In Fig.14 (a), the measured film area of iso-octane (red open squares)
449 decreased due to the vaporization noted from 5 ms after the start of injection. The wall
450 film of the gasoline surrogate fuel (blue open circles) show a wider film area than that of
451 iso-octane, which decreased at a slow rate. This tendency matched with the experiment
452 of Schulz [24]. The calculations underestimated the film area of the gasoline surrogate
453 fuel (solid blue line). This is because iso-pentane which is the lowest boiling point
454 component may be assumed to form a vaporized layer on the wall immediately after the
455 wall impingement and this layer reduces the heat flux from the wall to the film in the
456 experiment, a process that was not incorporated in the calculations. Other than this
457 inconsistency, the calculations demonstrate good agreement with the measurements and
458 capture the changes in the vaporization process from 5 ms after the start of injection. The
459 changes in the calculated film mass in Fig.14 (b) reflect the results with the measured
460 changes.

461 Figure 15 plots the calculated profiles of the mass fraction of the components present
462 in the wall film of the gasoline surrogate fuel. Compared with the original mass fractions
463 detailed in Table 3, it is clear that most of the iso-pentane (solid black plot) vaporized
464 immediately after the wall impingement. For the medium boiling point components: iso-
465 octane (solid blue line), n-heptane (dashed black line), and toluene (solid gray line), the
466 wall temperature T_w of 60°C is lower than their boiling points, and the vaporization was
467 at slower rates. At 20 ms after the start of injection, the highest boiling point component
468 iso-butylbenzene (dashed red line) formed the largest quantity of the wall film as little
469 had vaporized at the wall temperature here. Considering that the aromatic contents have
470 higher boiling points among a range of gasoline components, they would present the
471 possibility to yield soot, feeding the pool fires occurring in the warm-up operations of

472 DISI gasoline engines.

473 **6. Conclusions**

474 The present study developed a CFD model to predict the wall film formation and
475 vaporization of multicomponent fuel sprays. The calculated results with the model were
476 validated by a comparison with actual measurements. Finally, the model was used to
477 understand details of the wall film behavior of a gasoline surrogate fuel. The conclusions
478 may be summarized as follows:

- 479 1. The calculations with the developed model reproduce the observed trend that the film
480 area increases with increases in the fuel injection pressure.
- 481 2. The calculated film thickness at the edge of the wall film is thicker than at the central
482 portion, similar to the observation in the experiments.
- 483 3. With iso-octane (boiling point 99.3°C), boiling occurs from the center of the wall film
484 at the wall temperature of 100°C, and the whole wall film boils at the wall temperature
485 of 120°C. The calculated results are consistent with these observations.
- 486 4. The heat flux from the wall to the wall film increases with the increase in the wall
487 temperature, and the calculations also reproduce this change well.
- 488 5. At the wall temperature of 60°C, the wall film of the gasoline surrogate fuel exhibits
489 a very low vaporization rate, compared to that of pure iso-octane. This is because the
490 highest boiling point component in the surrogate fuel remains unevaporated in the
491 wall film. As aromatics are generally the higher boiling point components in gasoline,
492 they are potential sources of pool fires and soot in DISI gasoline engines.

493 In order to additionally validate the calculated film formation and evaporation processes,
494 at higher ambient pressure, further measurements should be planned.

495

496 **Acknowledgements**

497 This work was supported by Council for Science, Technology and Innovation (CSTI),
498 Cross-ministerial Strategic Innovation Promotion Program (SIP), “Innovative
499 combustion technology” (funding agency: JST).

500

501 **References**

- 502 [1] Stevens, E. and Steeper, R., Piston Wetting in an Optical DISI Engine: Fuel Films,
503 Pool Fires, and Soot Generation, SAE Technical Paper, No.2001-01-1203 (2001).
- 504 [2] Drake, M., Fansler, T., Solomon, A. and Szekely, G., Piston Fuel Films as a Source of
505 Smoke and Hydrocarbon Emissions from a Wall-Controlled Spark-Ignited Direct-
506 Injection Engine, SAE Technical Paper, No.2003-01-0547 (2003).
- 507 [3] Naber, J. D. and Reitz, R. D., Modeling Engine Spray/Wall Impingement, SAE
508 Technical Paper, No. 880107 (1988).
- 509 [4] Bai, C. and Gosman, A. D., Development of Methodology for Spray Impingement
510 Simulation, SAE Technical Paper, No.950283 (1995).
- 511 [5] Stanton, D. W. and Rutland, C. J., Modeling Fuel Film Formation and Wall Interaction
512 in Diesel Engines, SAE Technical Paper, No.960628 (1996).
- 513 [6] Mundo, C., Sommerfeld, M. and Tropea, C., Droplet-Wall Collisions: Experimental
514 Studies of the Deformation and Breakup Process, *Int. J. Multiphase Flow*, Vol.21,
515 No.2 (1995) 151-173.
- 516 [7] Köpple, F., Jochmann, P., Hettinger, A. and Kufferath, A., A Novel CFD Approach for
517 an Improved Prediction of Particulate Emissions in GDI Engines by Considering the
518 Spray-Cooling on the Piston, SAE Technical Paper, No.2015-01-0385 (2015).
- 519 [8] Kuhnke, D., Spray / Wall - Interaction Modeling by Dimensionless Data Analysis, Ph-
520 D Thesis, University of Darmstadt (2004).
- 521 [9] Zhang, Y., Jia, M., Liu, H. and Xie, M., Development of an Improved Liquid Film
522 Model for Spray/Wall Interaction under Engine-Relevant Conditions, *Int. J.*
523 *Multiphase Flow*, Vol.79 (2016) 74-87.
- 524 [10] Senda, J., Kanda, T., Al-Roub, M., Farrell, P. V., Fukami, T. and Fujimoto, H.,
525 Modeling Spray Impingement Considering Fuel Film Formation on the Wall, SAE
526 Technical Paper, No. 970047 (1997).
- 527 [11] Senda, J. and Fujimoto, H., Multidimensional Modeling of Impinging Sprays on the
528 Wall in Diesel Engines, *American Society of Mechanical Engineers, Applied*
529 *Mechanical Review*, Vol.52, No.4 (1999) 119-138.
- 530 [12] O’Rourke, P. J. and Amsden, A. A., A Particle Numerical Model for Wall Film
531 Dynamics in Port-Injected Engines, SAE Technical Paper, No.961961 (1996).
- 532 [13] Jiao, Q. and Reitz, R. D., Modeling Soot Emissions from Wall Films in a Direct-
533 Injection Spray-Ignition Engine, *International Journal of Engine Research*, Vol.16,

534 Issue 8, (2014) 994-1013.

535 [14] Kalghatgi, G., Fuel/Engine Interactions, Chapter 2, *SAE International* (2014)

536 [15] Jin, J. D. and Borman, G. L., A Model for Multicomponent Droplet Vaporization at
537 High Ambient Pressures, SAE Technical Paper, No.850264 (1985).

538 [16] Ayoub, N. S. and Reitz, R. D., Multidimensional Computation of Multicomponent
539 Spray Vaporization and Combustion, SAE Technical Paper, No.950285 (1995).

540 [17] Lippert, A. M. and Reitz, R. D., Modeling of Multicomponent Fuels Using
541 Continuous Distributions with Application to Droplet Evaporation and Sprays, SAE
542 Technical Paper, No.972882 (1997).

543 [18] Miyagawa, H., Nagaoka, M., Akihama, K. and Fujikawa, T., Numerical Analysis on
544 Multi-Component Fuel Behaviors in a Port-Injection Gasoline Engine, SAE Technical
545 Paper, No.1999-01-3642 (1999).

546 [19] Ra, Y. and Reitz, R. D., The application of a Multicomponent Droplet Vaporization
547 Model to Gasoline Direct Injection Engines, *International Journal of Engine Research*,
548 Vol.4, No.3 (2003) 193-218.

549 [20] Kawano, D., Senda, J., Wada, Y., Fujimoto, H., Goto, Y., Odaka, M., Ishii, H. and
550 Suzuki, H., Numerical Simulation of Multicomponent Fuel Spray, SAE Technical
551 Paper, No.2003-01-1838 (2003).

552 [21] Zhang, L. and Kong, S.-C., High-Pressure Vaporization Modeling of Multi-
553 Component Petroleum-Biofuel Mixtures Under Engine Conditions, *Combustion and
554 Flame*, Vol.158, Issue 9 (2011) 1705-1717.

555 [22] Hosain, M. L., Fdhila, R. B. and Daneryd, A., Heat Transfer by Liquid Jets Impinging
556 on a Hot Flat Surface, *Applied Energy*, Vol.164 (2016) 934-943.

557 [23] Köppl, F., Jochmann, P., Kufferath, A. and Bargende, M., Investigation of the
558 Parameters Influencing the Spray-Wall Interaction in a GDI Engine – Prerequisite for
559 the Prediction of Particulate Emissions by Numerical Simulation, *SAE Int. J. Engines*
560 6 (2) (2013) 911-925.

561 [24] Schulz, F. and Beyrau, F., Systematic Investigation of Fuel Film Evaporation, SAE
562 Technical Paper, No.2018-01-0310 (2018).

563 [25] Mizobuchi, Y., Development of Automotive Engine Combustion Simulation
564 Software “HINOCA”, (written in Japanese) *Journal of the Combustion Society of
565 Japan*, Vol.58, No.186 (2016) 191-196.

566 [26] Ducros, F., Nicoud, F. and Poinso, T., Wall-Adapting Local Eddy-Viscosity Models
567 for Simulations in Complex Geometries, *Proceedings of the Conference on Numerical
568 Methods for Fluid Dynamics* (1998) 293-299.

569 [27] Beale, J. C. and Reitz, R. D., Modeling Spray Atomization with the Kelvin-
570 Helmholtz/Rayleigh-Taylor Hybrid Model, *Atomization and Sprays*, Vol.9, Issue 6
571 (1999) 623-650.

572 [28] Tomoda, T., Kawauchi, M., Kubota, M., Nomura, Y. and Senda, J., Modeling of Wall
573 Impinging Behavior with a Fan Shaped Spray, SAE Technical Paper, No.2003-01-
574 1841 (2003).

575 [29] Al-Roub, M., Farrell, P. V. and Senda, J., Near Wall Interaction in Spray Impingement,
576 SAE Technical Paper, No.960863 (1996).

577 [30] Ueda, T., Enomoto, T. and Kanematsu, M., Heat Transfer Characteristics and
578 Dynamic Behavior of Saturated Droplets Impinging on a Heated Vertical Surface,
579 Bulletin of the JSME, Vol.22, No.167 (1979) 724-732.

580 [31] Cossali, G. E, Coghe, A. and Marengo, M., The Impact of a Single Drop on a Wetted
581 Solid Surface, Experiments in Fluids, Vol.22, Issue 6 (1997) 463-472.

582 [32] Mundo, C., Sommerfeld, M. and Tropea, C., Droplet-Wall Collisions: Experimental
583 Studies of the Deformation and Breakup Process, International Journal of Multiphase
584 Flow, Vol.21, No.2 (1995) 141-173.

585 [33] Kalantari, D. and Tropea, C., Spray Impact onto Flat and Rigid Walls: Empirical
586 Characterization and Modeling, International Journal of Multiphase Flow, Vol.33,
587 Issue 5 (2007) 525-544.

588 [34] Kobashi, Y., Kimoto, Y. and Kato, S., Experimental Validation of a Droplet
589 Vaporization Model for Ethanol-Blended Fuels and its Application to Spray
590 Simulation, SAE Technical Paper, No.2014-01-2733 (2014).

591 [35] Enomoto, Y. and Furuhashi, S., Study on Thin Film Thermocouple for Measuring
592 Instantaneous Temperature on Surface of Combustion Chamber Wall in Internal
593 Combustion Engine, Bulletin of JSME, Vol.28, No.235 (1985) 108-116.

594 [36] Schulz, F. and Beyrau, F., The Effect of Operating Parameters on the Formation of
595 Fuel Wall Films as a Basis for the Reduction of Engine Particulate Emissions, Fuel,
596 Vol.238 (2019) 375-384.

597 [37] Schulz, F., Schmidt, J., Kufferath, A. and Samenfink, W., Gasoline Wall Films and
598 Spray/Wall Interaction Analyzed by Infrared Thermography, SAE Int. J. Engines 7(3)
599 (2014) 1165-1177.

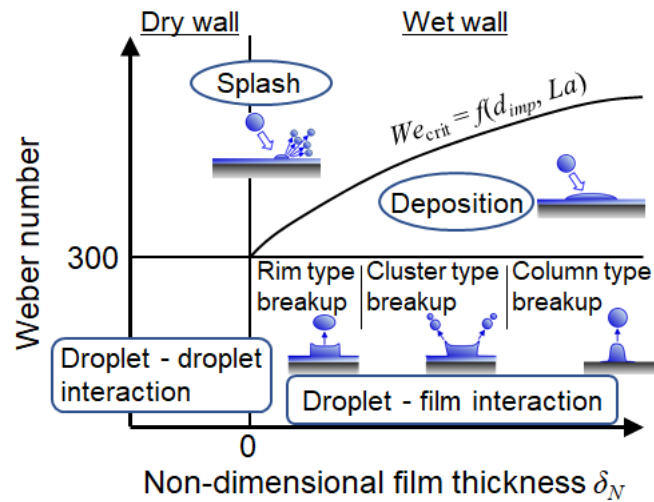
600 [38] Senda, J., Higaki, T., Sagane, Y., Fujimoto, H., Takagi, Y. and Adachi, M., Modeling
601 and Measurement on Evaporation Process of Multicomponent Fuels, Journal of
602 Engines, Vol.109, Section 3 (2000) 347-358.

603 [39] Friend, D. G. and Huber, M. L., Thermophysical Property Standard Reference Data
604 from NIST, International Journal of Thermophysics, Vol.15, No.6 (1994) 1279-1288.

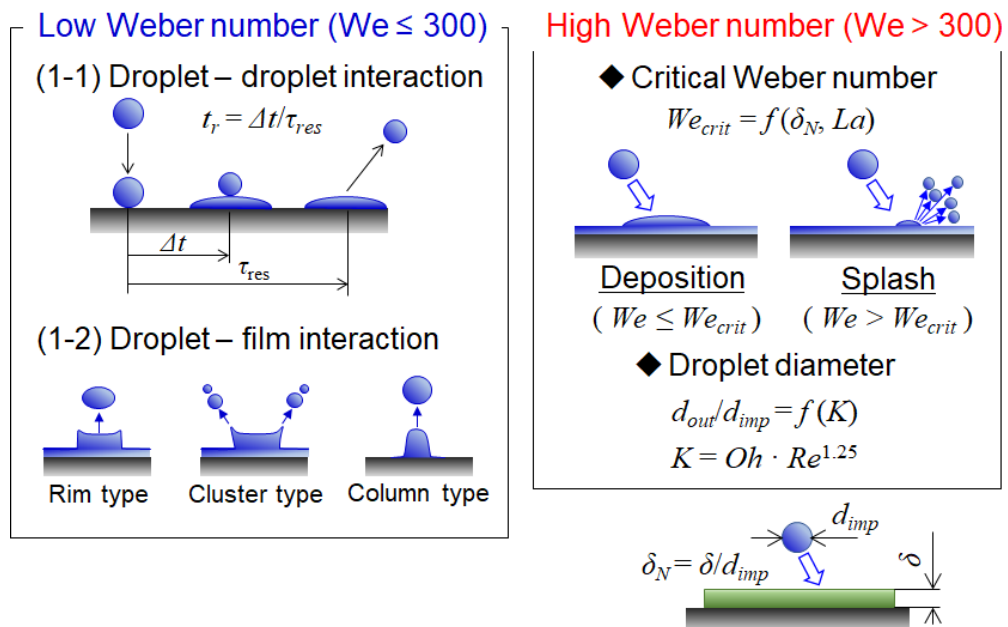
605 [40] Peng, D.-Y. and Robinson, D. B., A New Two-constant Equation of State, Ind. Eng.
606 Chem. Fundam., Vol.15, No.1 (1976) 59-64.

607
608
609

610
611
612
613
614
615
616
617
618
619
620
621
622
623
624
625
626
627
628
629
630
631
632
633
634
635
636
637
638
639
640
641
642
643
644
645
646
647



(a)



(b)

Fig.1 Droplet / wall impingement model proposed by Senda [10-11]

- (a) classification of impinging droplet behavior with Weber number and non-dimensional film thickness
- (b) droplet impingement model for low and high Weber numbers

648
 649
 650
 651
 652
 653
 654
 655
 656
 657
 658
 659
 660
 661
 662
 663
 664
 665
 666
 667
 668
 669
 670
 671
 672
 673
 674
 675
 676
 677
 678
 679
 680
 681
 682
 683
 684
 685

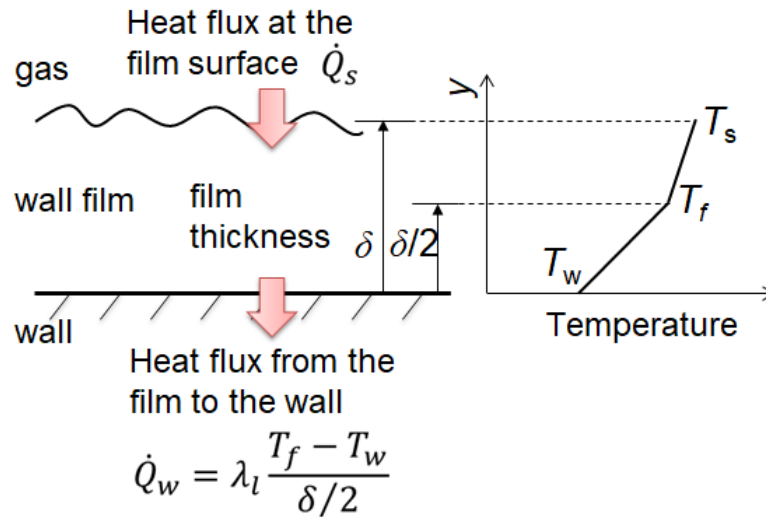


Fig.2 Illustration of the heat transfer and linear temperature assumptions in the wall film model [12]

686
687
688
689
690
691
692
693
694
695
696
697
698
699
700
701
702
703
704
705
706
707
708
709
710
711
712
713
714
715
716
717
718
719
720
721
722
723

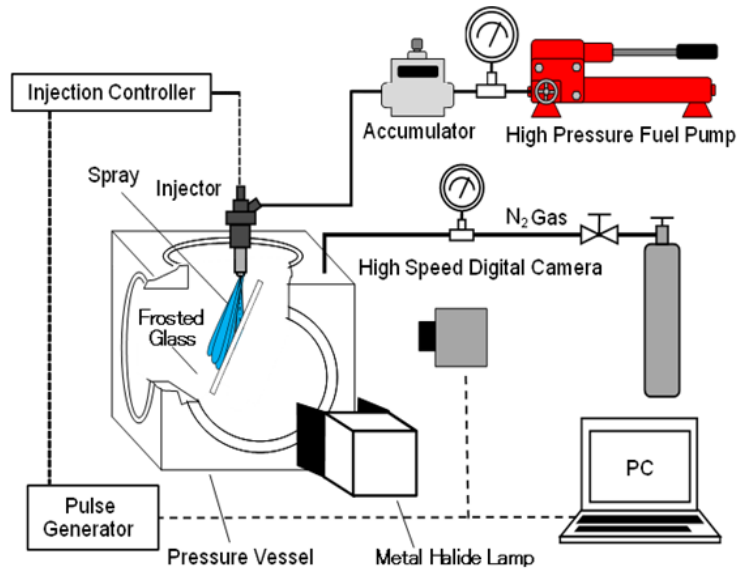


Fig.3 Experimental setup for film area measurements

724
725
726
727
728
729
730
731
732
733
734
735
736
737
738
739
740
741
742
743
744
745
746
747
748
749
750
751
752
753
754
755
756
757
758
759
760
761

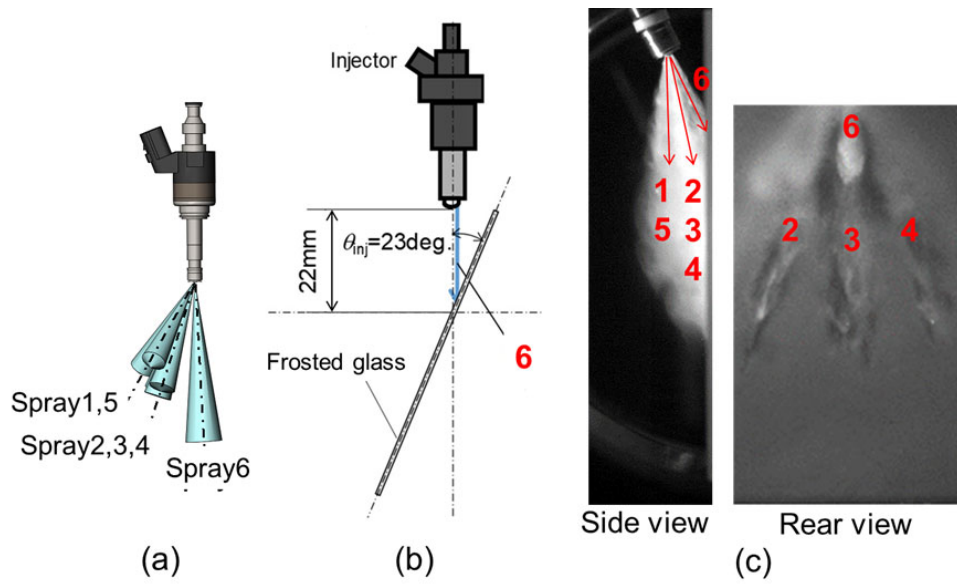
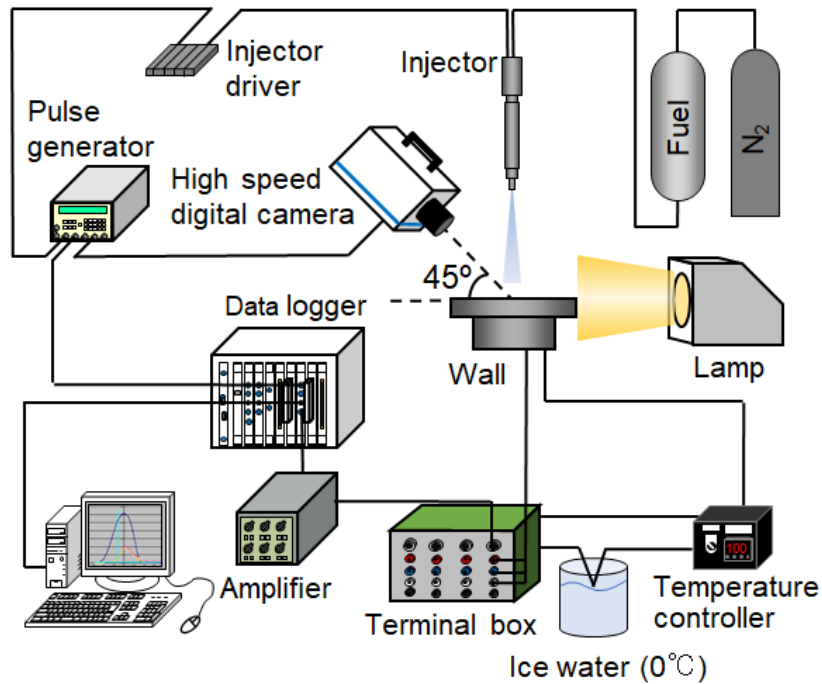
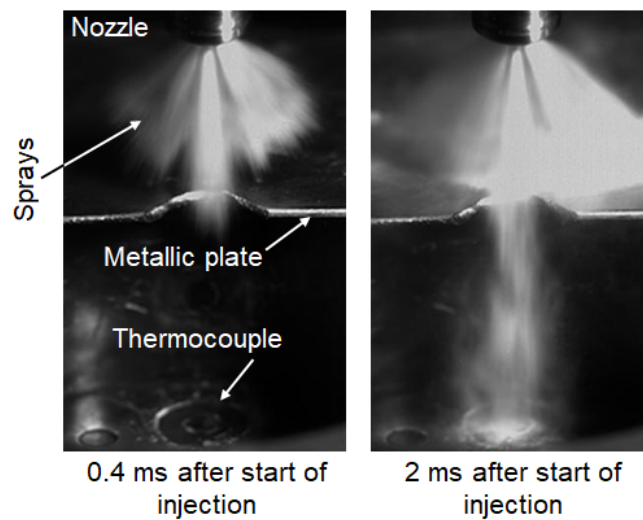


Fig.4 (a) & (b) Arrangement of the fuel sprays for the film area measurements and (c) images of recorded spray tracks



(a)



(b)

792 Fig.5 Experimental setup for visualization of the wall film and measurements of the wall
793 heat flux

794 (a) overall system

795 (b) layout of the fuel sprays and thermocouple

796 (one spray orthogonally impinged on the thermocouple while the other sprays
797 were masked with the metallic plate)

800
801
802
803
804
805
806
807
808
809
810
811
812
813
814
815
816
817
818
819
820
821
822
823
824
825
826
827
828
829
830
831
832
833
834
835
836
837

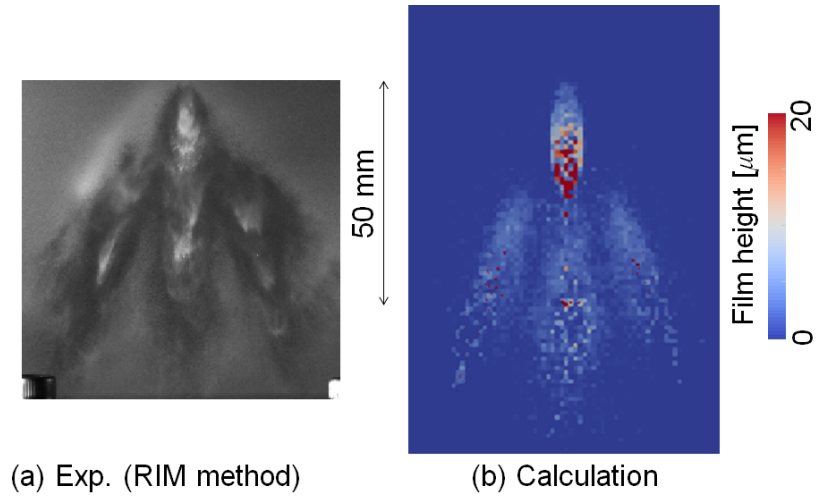


Fig.6 Wall film images (a) taken with RIM method and (b) calculated
($p_{inj} = 13 \text{ MPa}$, $T_w = 20^\circ\text{C}$)

838
839
840
841
842
843
844
845
846
847
848
849
850
851
852
853
854
855
856
857
858
859
860
861
862
863
864
865
866
867
868
869
870
871
872
873
874
875

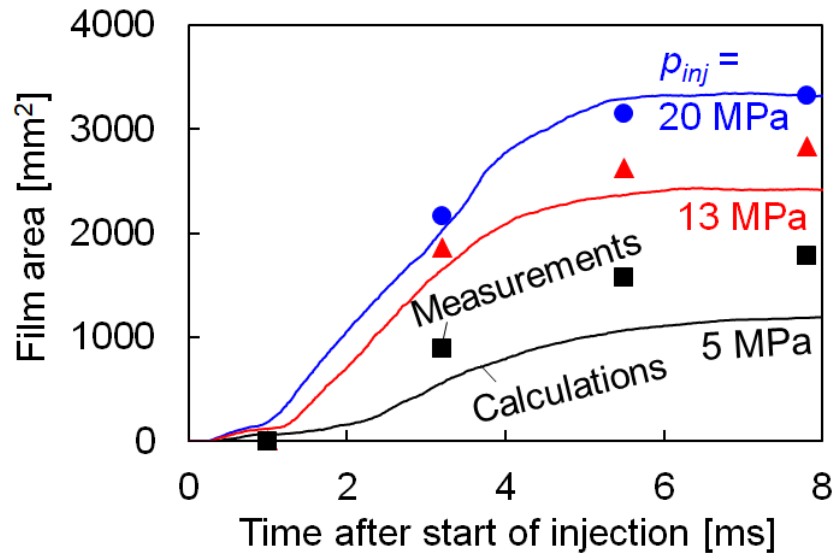


Fig.7 Plots of wall film areas for the measurements and calculations ($T_w = 20^\circ\text{C}$)

876
877
878
879
880
881
882
883
884
885
886
887
888
889
890
891
892
893
894
895
896
897
898
899
900
901
902
903
904
905
906
907
908
909
910
911
912
913

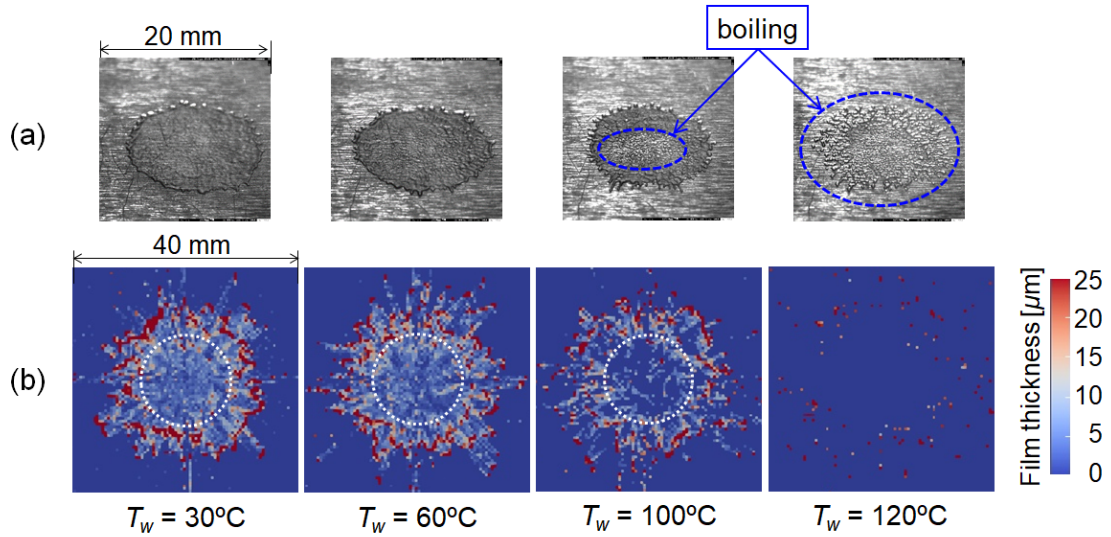


Fig.8 Images of the (a) photographed and (b) calculated wall films at different wall temperatures T_w . Dashed white lines equivalent to the measured wall film diameters are superimposed on (b) ($t_{inj} = 7$ ms)

914
915
916
917
918
919
920
921
922
923
924
925
926
927
928
929
930
931
932
933
934
935
936
937
938
939
940
941
942
943
944
945
946
947
948
949
950
951

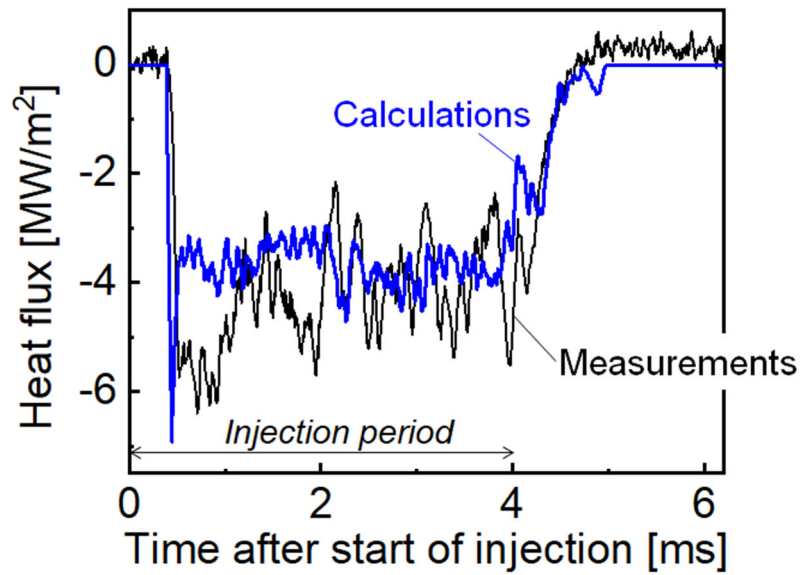


Fig.9 Plot of profiles of the heat flux for the measurements and calculations ($T_w = 120^\circ\text{C}$)

952
953
954
955
956
957
958
959
960
961
962
963
964
965
966
967
968
969
970
971
972
973
974
975
976
977
978
979
980
981
982
983
984
985
986
987
988
989

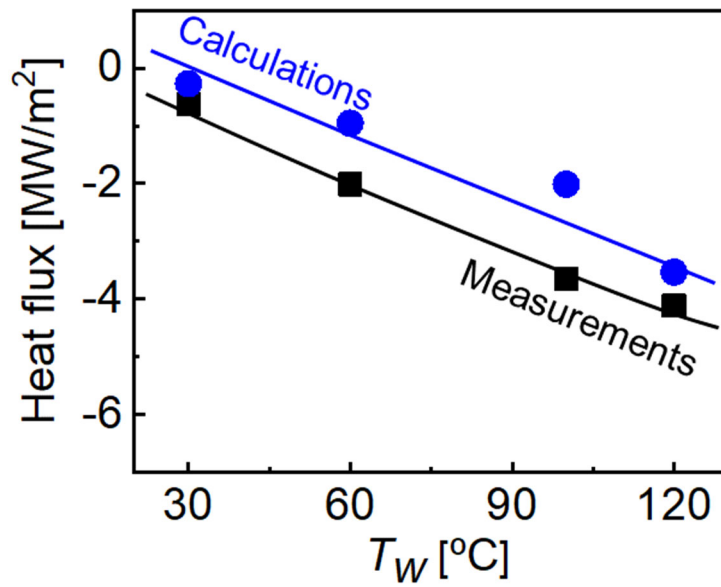


Fig.10 Plot of the changes in the heat flux versus the wall temperature T_w for the measurements and calculations

990
991
992
993
994
995
996
997
998
999
1000
1001
1002
1003
1004
1005
1006
1007
1008
1009
1010
1011
1012
1013
1014
1015
1016
1017
1018
1019
1020
1021
1022
1023
1024
1025
1026
1027

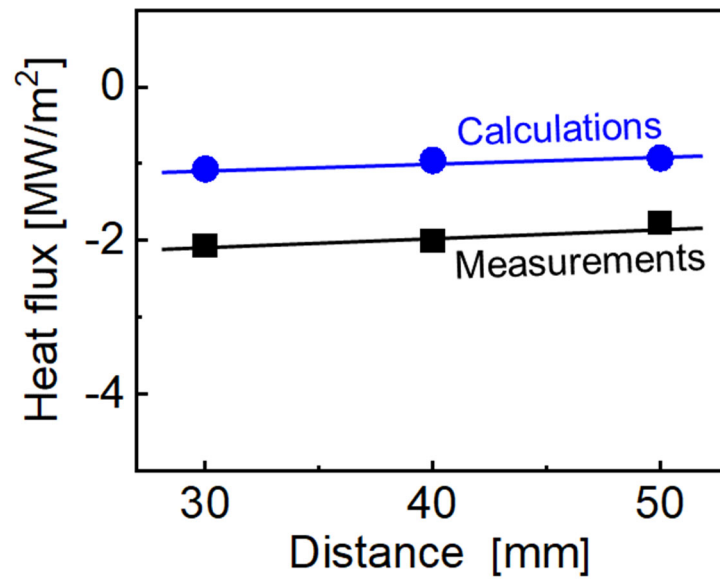


Fig.11 Plot of the changes in the heat flux versus the distance from injection nozzle to wall for the measurements and calculations

1028
1029
1030
1031
1032
1033
1034
1035
1036
1037
1038
1039
1040
1041
1042
1043
1044
1045
1046
1047
1048
1049
1050
1051
1052
1053
1054
1055
1056
1057
1058
1059
1060
1061
1062
1063
1064
1065

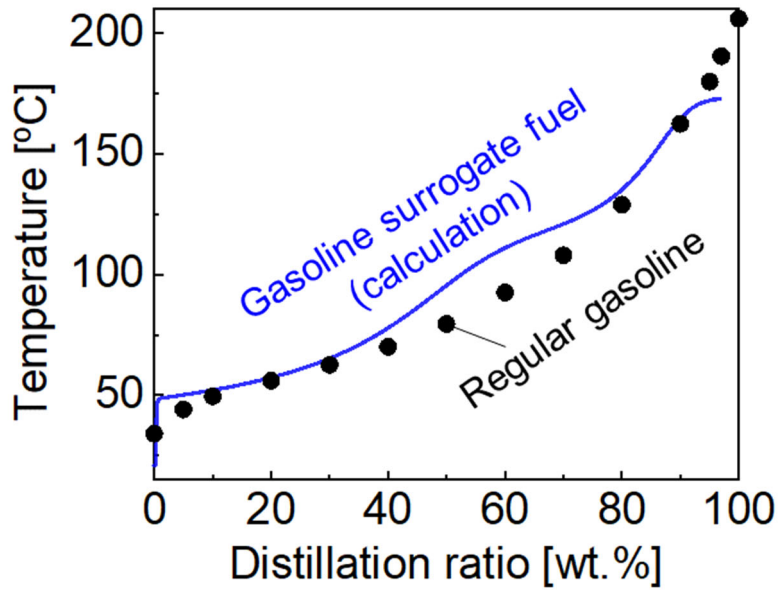


Fig.12 Plot of the temperature versus distillation ratio curves for the gasoline surrogate fuel and regular gasoline

1066
1067
1068
1069
1070
1071
1072
1073
1074
1075
1076
1077
1078
1079
1080
1081
1082
1083
1084
1085
1086
1087
1088
1089
1090
1091
1092
1093
1094
1095
1096
1097
1098
1099
1100
1101
1102
1103

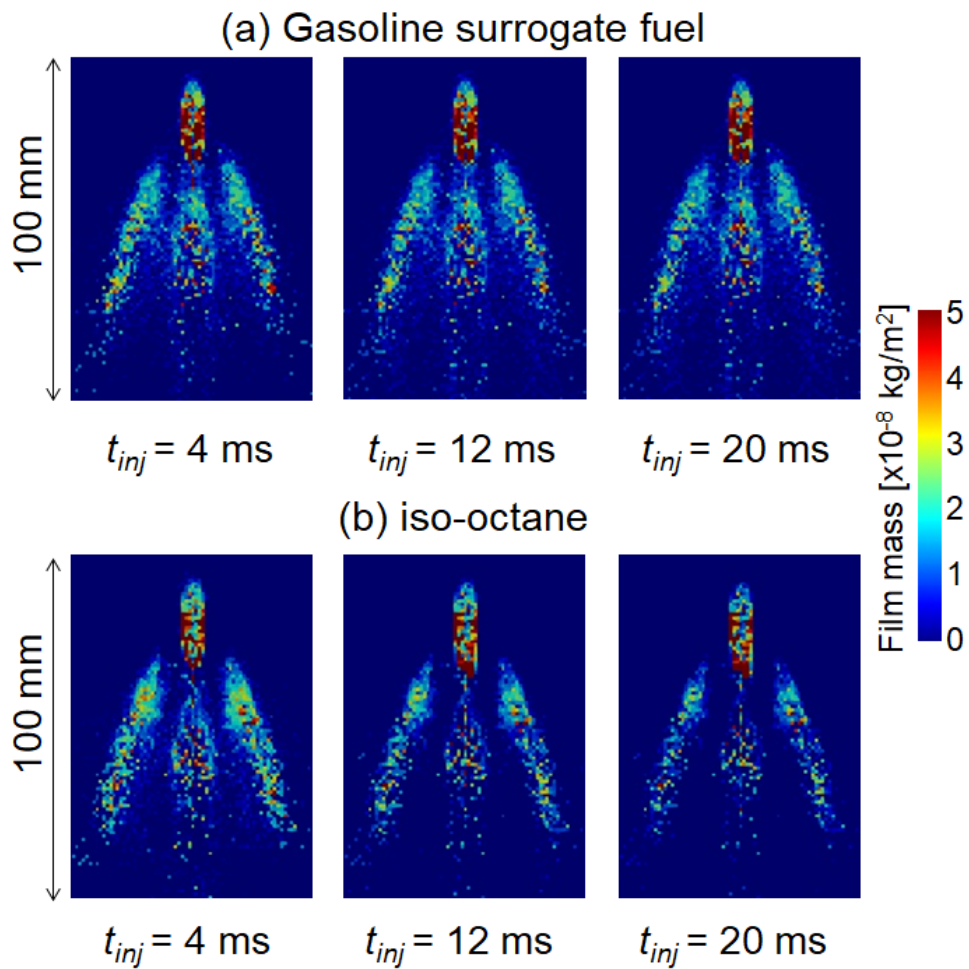


Fig.13 Calculated film thickness distributions of (a) the gasoline surrogate fuel and (b) iso-octane ($T_w = 60^\circ\text{C}$)

1104
1105
1106
1107
1108
1109
1110
1111
1112
1113
1114
1115
1116
1117
1118
1119
1120
1121
1122
1123
1124
1125
1126
1127
1128
1129
1130
1131
1132
1133
1134
1135
1136
1137
1138
1139
1140
1141

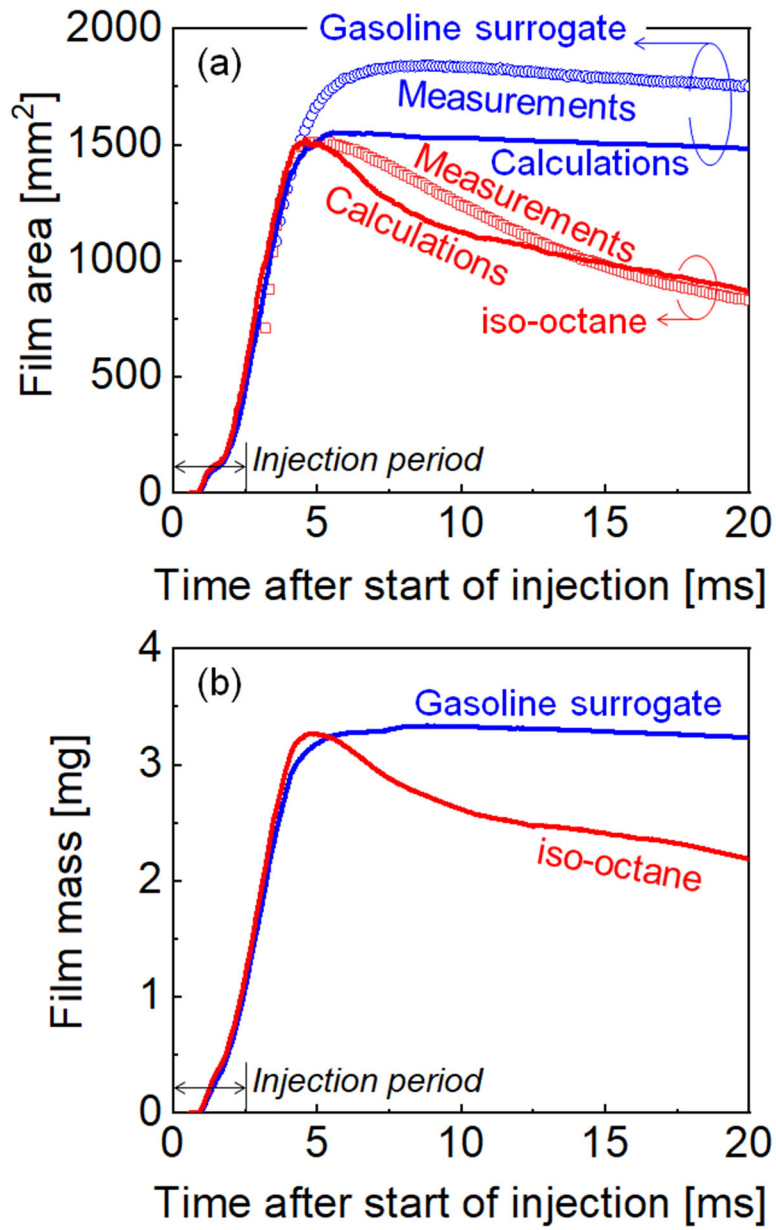


Fig.14 Plots of (a) the measured and calculated film areas and (b) the calculated mass of the wall film of the gasoline surrogate fuel and iso-octane ($T_w = 60^\circ\text{C}$)

1142
1143
1144
1145
1146
1147
1148
1149
1150
1151
1152
1153
1154
1155
1156
1157
1158
1159
1160
1161
1162
1163
1164
1165
1166
1167
1168
1169
1170
1171
1172
1173
1174
1175
1176
1177
1178
1179

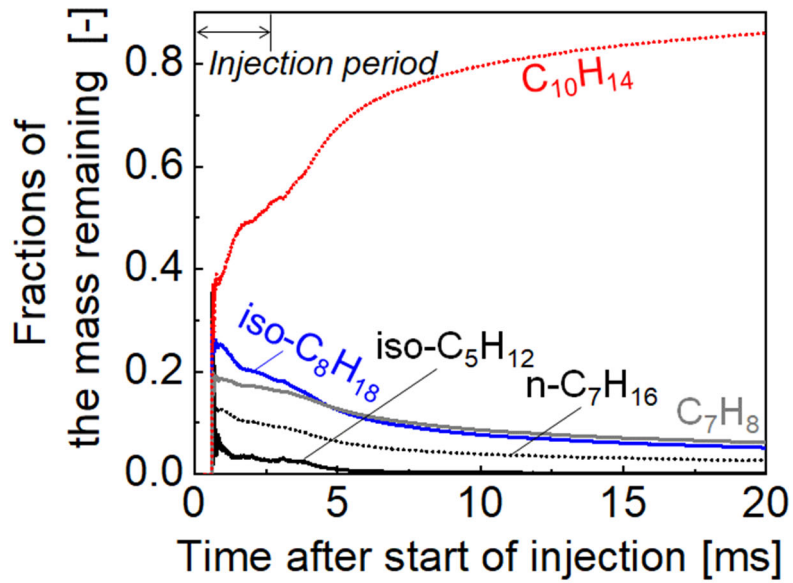


Fig.15 Calculated profiles of the mass fraction of the components present in the wall film of the gasoline surrogate fuel ($T_w = 60^\circ C$)

1180
1181
1182
1183
1184
1185
1186
1187
1188
1189
1190
1191
1192
1193
1194
1195
1196
1197
1198
1199
1200
1201
1202
1203
1204
1205
1206
1207
1208
1209
1210
1211
1212
1213
1214
1215
1216
1217

Table 1 Conditions for the film area measurements and calculations with iso-octane

Ambient pressure	[MPa]	0.10
Ambient temperature	[°C]	20
Wall temperature, T_w	[°C]	20
Nozzle diameter	[mm]	0.13
Injection pressure	[MPa]	5, 13, 20
Injection period	[ms]	2.5
Fuel		iso-octane (C ₈ H ₁₈)

1218
1219
1220
1221
1222
1223
1224
1225
1226
1227
1228
1229
1230
1231
1232
1233
1234
1235
1236
1237
1238
1239
1240
1241
1242
1243
1244
1245
1246
1247
1248
1249
1250
1251
1252
1253
1254
1255

Table 2 Conditions used in evaluating the wall heat flux of iso-octane

Ambient pressure	[MPa]	0.10
Ambient temperature	[°C]	20
Wall temperature, T_w	[°C]	30, 60, 100, 120
Nozzle diameter	[mm]	0.248
Injection pressure	[MPa]	10
Injection period	[ms]	4
Impinging distance	[mm]	30, 40, 50
Impinging angle	[°]	90
Total Injection quantity	[mg]	109
Fuel		iso-octane (C_8H_{18})

1256
 1257
 1258
 1259
 1260
 1261
 1262
 1263
 1264
 1265
 1266
 1267
 1268
 1269
 1270
 1271
 1272
 1273
 1274
 1275
 1276
 1277
 1278
 1279
 1280
 1281
 1282
 1283
 1284
 1285
 1286

Table 3 Gasoline surrogate fuel proposed in the present study
 (a) boiling point and RON of single component fuels and their mixing fraction
 (b) properties of gasoline surrogate fuel, compared with regular gasoline referenced

	Boiling point [°C]	RON [-]	Volume fraction [vol.%]	Mass fraction [wt.%]
iso-pentane (C ₅ H ₁₂)	28.0	96	45	39
n-heptane (C ₇ H ₁₆)	98.4	0	10	10
(a) iso-octane (C ₈ H ₁₈)	99.3	100	20	20
Toluene (C ₇ H ₈)	110.6	111	10	12
1,2,4-trimethylbenzene (C ₉ H ₁₂)*	169.4	108	15	19

* replaced by iso-butylbenzene (C₁₀H₁₄) in the present study

	Gasoline surrogate fuel	Reference regular gasoline
(b) RON [-]	90.5	90.8
Aromatic content [vol.%]	25.0	25.4
Density [kg/m ³]	711	730
Viscosity [μPa·s]	404	N/A