

Backward Flow of Hot Burned Gas Surrounding High-Pressure Diesel Spray Flame from Multi-hole Nozzle

Takayuki Fuyuto and Yoshiaki Hattori
Toyota Central R&D Labs., Inc.

Hayato Yamashita
Nippon Soken Inc.

Makoto Mashida
DENSO Corp.

ABSTRACT

The backward flow of the hot burned gas surrounding a diesel flame was found to be one of the factors dominating the set-off length (also called the lift-off length), that is, the distance from a nozzle exit into which a diffusion flame cannot intrude. In the combustion chamber of an actual diesel engine, the entrainment of the surrounding gas into a spray jet from a multi-hole nozzle is restricted by the walls and adjacent spray jets, which induces the backward flow of the surrounding gas. A new momentum theory to calculate the backward flow velocity was established by extending Wakuri's momentum theory. Shadowgraph imaging in an optical engine successfully visualized the backward flow of the hot burned gas.

CITATION: Fuyuto, T., Hattori, Y., Yamashita, H., and Mashida, M., "Backward Flow of Hot Burned Gas Surrounding High-Pressure Diesel Spray Flame from Multi-hole Nozzle," *SAE Int. J. Engines* 9(1):2016, doi:10.4271/2015-01-1837.

INTRODUCTION

The market continues to demand diesel engines with ever-increasing fuel efficiency, while stricter emissions regulations, in particular related to nitrogen oxide (NO_x) and particulate matter (PM), are set to be enforced in many regions around the world. Since the addition of an after-treatment system, such as a diesel particulate filter (DPF) or urea selective catalytic reduction (SCR) catalyst, would further increase costs, there is a need to reduce soot and NO_x emissions by improving the in-cylinder combustion itself, thus reducing the dependence on after-treatment systems. Premixed combustion systems have been proposed as a means of achieving low fuel consumption and low exhaust emissions [1, 2]. However, the engine operating range of these systems was limited to relatively low-load conditions. For high-load conditions in which diffusive combustion with long injection durations is dominant, some new combustion techniques have been proposed: the application of a high boost pressure and a high exhaust gas recirculation (EGR) rate [3, 4], as well as the use of high-injection pressure in excess of 200 MPa [5, 6].

The mechanism whereby soot is emitted can be divided into soot formation and soot oxidation. Many researchers and engineers have studied the effects of enhancing air utilization and optimizing the fuel distribution between the piston cavity and squish area on soot reduction. These studies can be regarded as being studies of soot

oxidation, essentially. The understanding of soot formation demands the detailed observation of rich spray combustion in the region immediately around the nozzle.

Under quasi-steady conditions with a long injection duration, the combustive diesel jet covers the region from the nozzle exit over a length L as shown in [Figure 1](#), into which the diffusion flame cannot intrude. This length L affects the air entrainment prior to combustion, and thereby strongly affects soot formation. The ϕ - T map [7] in [Figure 1b](#) shows that, if the length L is short, combustion starts in a rich region with a high equivalence ratio ϕ , such that the amount of soot formed increases.

Siebers et al. experimentally investigated the effects of ambient gas temperature, injection pressure, and nozzle diameter on the length L of a single free-spray jet in a constant-volume vessel, which was measured by taking hydroxyl (OH)-chemiluminescence images. They referred to L as the "lift-off length" [8, 9, 10]. However, the term "lift-off length" was originally defined for a lifted premixed flame in which the flow velocity of a premixed gas flow balances the flame propagation velocity [11]. Some researchers believe that "lift-off" is not an appropriate term to describe a diesel spray flame consisting of many complicated phenomena such as liquid breakup, evaporation, air entrainment, and auto-ignition [12]. Arai proposed that the length L be known as the "set-off length" [12]. In this study, the authors

chose to adopt this term. Even though many researchers have investigated the detailed mechanism that dominates the set-off length [10, 13], it has not yet been revealed completely [12].

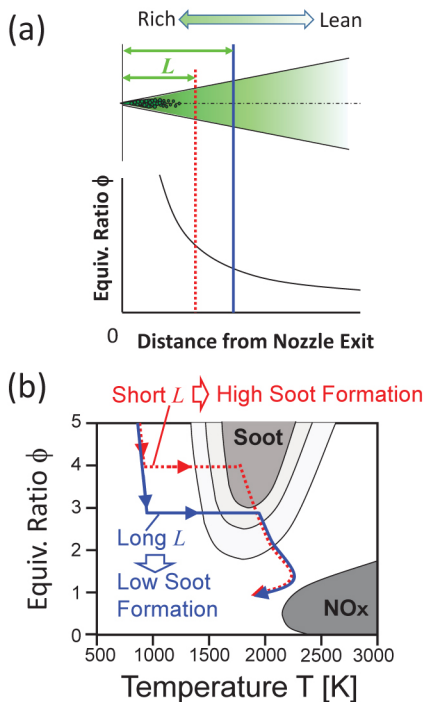


Figure 1. Relationship between set-off (also called lift-off) length L and soot formation rate on the ϕ - T map [7]

Pickett and Siebers proposed an empirical equation based on measurements of a single free spray flame in a constant-volume vessel [14]:

$$L = C \cdot T_a^{-3.74} \cdot \rho_a^{-0.85} \cdot d^{0.34} \cdot U^1 \cdot Z_{st}^{-1} \quad (1)$$

This empirical equation has been used to predict the set-off length by many researchers. In recent years, however, there have been disagreements between the predicted set-off length and the measured set-off length of a diesel spray flame injected from a multi-hole nozzle in a constant volume vessel and an optical engine.

Bruneaux et al. measured the set-off length of a spray flame injected from a 10-hole nozzle in a high-temperature and high-pressure constant-volume vessel which is similar to Siebers' vessel [15]. Their results showed that the measured L by taking OH-chemiluminescence images is shorter than the predicted L , and that the gap becomes larger as the injection pressure increases from 1000 to 2750 bar.

Using a Bowditch-type optical engine, Polonowski et al. demonstrated the reduction in the measured L with an increase in the number of nozzle orifices from 2 to 6 and then 10 by taking OH-chemiluminescence images [16]. In addition to the in-cylinder temperature rise caused by heat release up until the end of the premixed burn, they pointed out that the reason for the difference between the measured and predicted L of the spray flame from a 6-hole nozzle was the large-scale recirculation of the hot burned gas

(Figure 2). The neighboring two jet flames impinge on the cavity wall, move along the wall and then collide with each other. Burned gases are directed towards the center of the cavity between the jets, and then form a large-scale recirculation which shortens the set-off length. Although they assumed that another type of jet-jet interaction was occurring and shortening the length L in the case of a 10-hole nozzle in which the large-scale recirculation cannot be seen clearly because of the smaller interval angle, they did not show the details of the jet-jet interaction.

Chartier et al. also pointed out the large-scale recirculation of the burned gas as the reason for the difference, after capturing OH-chemiluminescence images in an optical engine [17]. They measured the value of L of a spray flame produced by an axis-symmetrical 4-hole nozzle with a 90-degree interval angle and spray flames from an axis-asymmetrical 4-hole nozzle with 45- and 135-degree interval angles, eliminating the in-cylinder temperature rise caused by heat release until the end of the premixed burn. The measured L of the spray flame between adjacent two sprays with 45-degree interval angle was shorter than the measured L of the spray flame from the axis-symmetrical 4-hole nozzle with 90-degree interval angle, which was similar to the predicted L by the empirical equation. However, this large-scale recirculation of the hot burned gas cannot explain Bruneaux's measurements in which the jet flames were found to not collide with the wall.

On the other hand, some velocity measurements in the space between the spray jets of a multi-hole nozzle in non-combusting high-pressure vessels have shown the emergence of backward flow towards the nozzle. Pawlowski et al. and Malbec et al. measured the backward velocity distributions of the gas in an interspace cross-section among the spray jets by using particle image velocimetry (PIV) [18, 19]. Toda et al. measured the velocity of the backward gas-flow surrounding a spray by using particle tracking velocimetry (PTV) and showed that the backward flow velocity increased with an increase in the number of nozzle holes from 7 to 8 (Figure 3) [20].

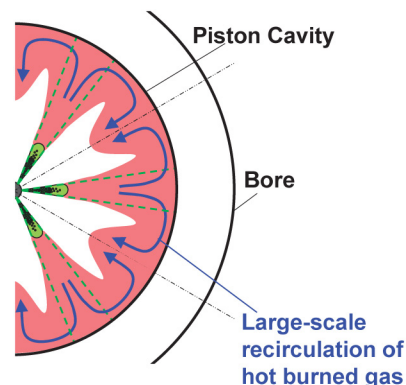


Figure 2. Large-scale recirculation of hot burned gas [16, 17]

Based on these results, the authors devised the following hypothesis shown in Figure 4; in the combustion chamber of an actual diesel engine, the entrainment of the surrounding gas into a spray jet emanating from a multi-hole spray nozzle is restricted by the walls and adjacent spray jets, which induces the backward flow of surrounding gas. Hot burned gas particles from the edge of the spray

flame by expansion of the combustion product. The induced backward flow draws the hot burned gas towards the nozzle, and forms a small recirculating flow of hot burned gas. The hot burned gas is re-entrained into the spray by this recirculating flow and shortens the set-off length L in comparison to the L of a free spray flame from a single-hole nozzle. This is one of the detailed mechanisms constituting “jet-jet interaction.”

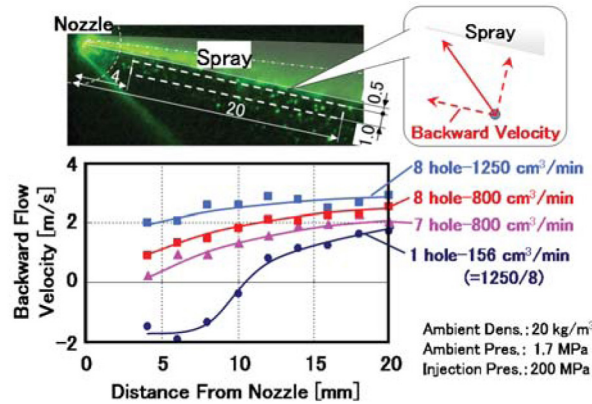


Figure 3. Measured backward flow surrounding spray jets from multi-hole nozzles [20]

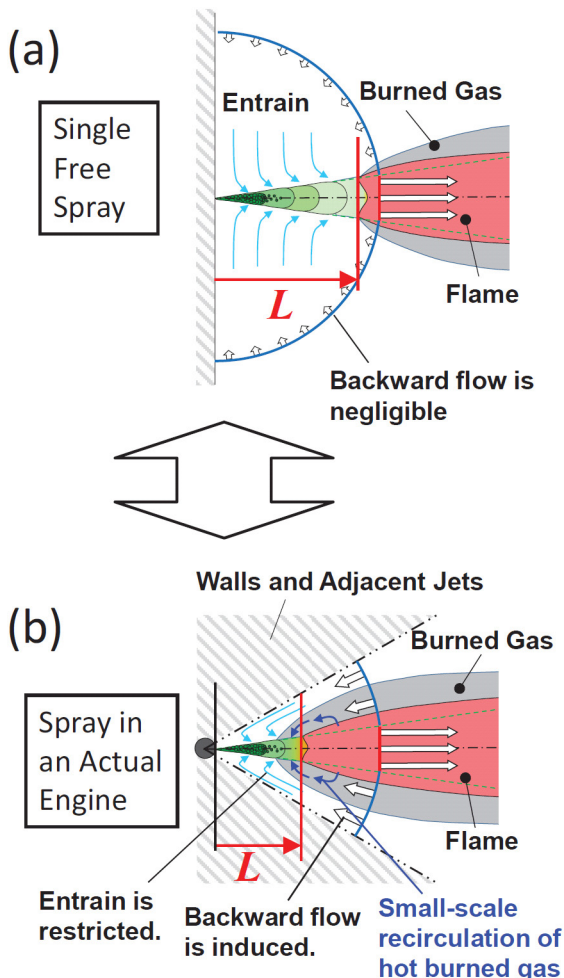


Figure 4. Authors' hypothesis: (a) shows a single free spray. (b) shows the small-scale recirculation of hot burned gas which reduces the set-off length L

First, in this study, a new momentum theory for calculating the backward flow velocity was established. Then, shadowgraph imaging with an optical engine was used to validate the emergence of the backward flow of the hot burned gas and its re-entrainment into the spray. These results constitute proof of the author's hypothesis shown in Figure 4. Finally, the effects on the injection pressure and the nozzle diameter on the set-off length were analyzed.

THEORETICAL FORMULAS FOR BACKWARD FLOW SURROUNDING SPRAYS

Derivation of Theoretical Formulas

In the 1950s, Wakuri et al. introduced the well-known momentum theory for a non-vaporizing fuel spray in a diesel engine [21]. Today, over 50 years later, his theory is still widely used by researchers and engineers to calculate the spray penetration and the amount of air entrainment into that spray. Naber and Sibers enhanced Wakuri's theory and proposed their scaling law by introducing a characteristic length scale [22]. In this study, we have derived theoretical equations which can describe the velocity of the backward flow surrounding a spray by extending Wakuri's momentum theory.

First, Wakuri's momentum theory is explained here. A cylinder-shaped control surface is defined for a quasi-steady non-vaporizing spray (Figure 5a). The nozzle orifice exit is located at the center of the upper circle. The x-axis is defined as being along the spray axis from the orifice exit. The lower circle is the spray cross-section at a distance x . The major assumption in Wakuri's theory is that the spray has no velocity difference between the fuel and the entrained gas (air). The spray (fuel + gas) momentum balance in the x-direction and the fuel mass balance on the control surface are given by equations (2) and (3), respectively.

$$\{\rho_f A_f(x) + \rho_g A_g(x)\} U(x)^2 = \rho_f A_f(0) U_f^2 \quad (2)$$

$$\rho_f A_f(x) U(x) = \rho_f A_f(0) U_f \quad (3)$$

The left-hand side is the spray momentum or the fuel mass at a distance x , while the right-hand side shows those of the injected fuel at the nozzle exit ($x = 0$). Indexes f and g refer to fuel and gas, respectively. $A(x)$, ρ , and $U(x)$ mean the cross-sectional area of the fuel and gas, the density of the fuel and gas, and the velocity, respectively. Solving these equations, the spray velocity $U(x)$ is described in terms of $A(x)$, $A_f(0)$ and U_f .

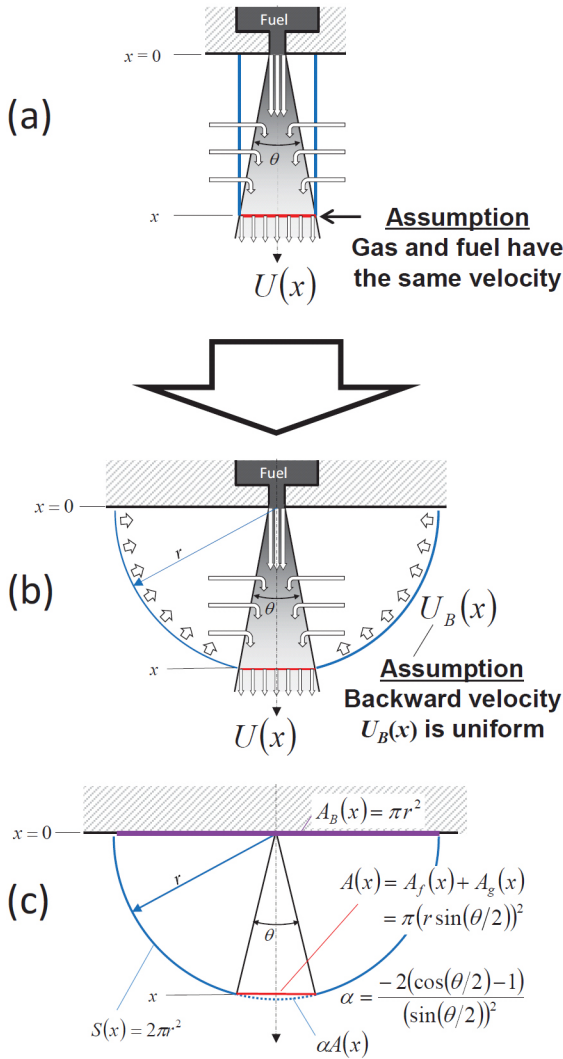


Figure 5. Schematic of the spray momentum theory: (a) Wakuri's original model, (b) the authors' model and (c) definition of the control volume geometry

An advantage of Wakuri's momentum theory is that the spray velocity $U(x)$ and the equivalence ratio at x can be described as a function of x . However, the backward flow of the gas surrounding the spray is not taken into account, assuming that the gas is entrained perpendicularly into the control volume through the cylindrical surface. Therefore, the authors have extended Wakuri's model so that we can calculate the backward flow velocity as a function of x , while retaining the original advantage of Wakuri's model. The control surface was changed from a cylindrical shape to a spherical surface (Figure 5b). Assuming that the backward gas flow velocity $U_B(x)$ is uniform over the spherical surface, the gas mass balance is given by:

$$\rho_g A_g(x) U(x) = \rho_g \{S(x) - \alpha A(x)\} U_B(x) \quad (4)$$

where $S(x)$ means the surface area of the half sphere, which has a radius $r = x / \cos(\theta/2)$. In a quasi-steady condition during a long injection period, the left-hand side means the mass of the gas flowing

out from the spray, which is equal to the mass of the gas flowing backwards into the control volume through the spherical surface, that is, the right-hand side.

Furthermore, a term for the negative momentum arising from the backward gas flow has been added to the spray momentum equation (2):

$$\begin{aligned} & \{\rho_f A_f(x) + \rho_g A_g(x)\} U(x)^2 \\ &= \rho_f A_f(0) U_f^2 - \rho_g (U_B(x))^2 \{A_B(x) - A(x)\} \end{aligned} \quad (5)$$

$A_B(x)$ is the projected area of the spherical surface in the spray axis (x -direction). Eliminating $U_B(x)$, $A(x)$, $A_f(x)$ and $A_g(x)$ from equations (3), (4), (5) gives a quadratic equation in $U(x)$, from which $U(x)$ can be solved explicitly. The detailed procedure for solving these equations is introduced in the Appendix. The ratio of $U_B(x)$ to $U(x)$ is described as the ratio of the passing area of the spray to the passing area of the backward gas flow.

$$\frac{U_B(x)}{U(x)} = \frac{A_g(x)}{S(x) - \alpha A(x)} \approx \frac{A(x)}{S(x) - A(x)} \quad (6)$$

Calculation of Backward Gas-Flow Velocity

The spray velocity $U(x)$, the backward flow velocity $U_B(x)$, and the ratio of $U_B(x)$ to $U(x)$ at $x = 20$ mm of a single-hole free spray, multi-hole free sprays, and a multi-hole spray in an actual combustion chamber were calculated using the derived equations. The results are shown in Figure 6. Other conditions assumed for the calculation are listed in the table in Figure 6. Under the condition of a free spray being emitted from a single-hole nozzle, the backward flow velocity is approximately 1% of the spray velocity, and is negligible. If the number of holes is increased from 1 to 10, the spherical surface of the control volume for one spray becomes 1/10. The backward flow velocity increases to approximately 12% of the spray velocity, and the spray velocity is reduced a little by the backward flow. Under the conditions existing in an actual diesel engine, assuming that the motion of the piston around the top dead center (TDC) is negligible, the lower part of the control volume is closed by the conical bottom surface of the piston cavity. The direction of the inlet gas flow into the control volume becomes almost opposite to that of the spray. The spray velocity is reduced by the negative momentum of the backward gas flow, to a value approximately 10% less than that of the single-hole free spray. The backward flow velocity increases to approximately 17% of the spray velocity.

As described above, the new momentum theory taking the gas mass balance into account showed that the strong backward flow of the gas surrounding a spray jet is induced in an actual combustion chamber where the gas entrained into the spray jet is restricted by the wall and adjacent spray jets during the injection period. The increase in the backward flow velocity of the gas surrounding a spray jet with the increase in the number of holes in a nozzle, as measured by PIV and PTV in non-combusting constant volume vessels [18, 19, 20] can be explained by these theoretical equations.