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Experimental Investigation on Effect of Wall Roughness and Lubricant Film on the Adhered Fuel Film of N-Butanol-Diesel Blends after Spray Impingement

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Abstract: The effect of wall roughness with different lubricant film thicknesses on the characteristics of adhered fuel films of diesel-n-butanol blending fuels after spray impingement has been investigated. Four steel plates with different types of roughness (root mean square height- S_q) that were coated with different lubricant film thicknesses (h_l) were used as impinging walls. The experimental conditions included dry walls ($h_l = 0$), semi-wetted walls (SWW) with different thin oil films ($0 < h_l/S_q < 1$), and fully wetted walls (FWW) with a thick lubricant film ($h_l > S_q$). The results indicate that the adhered fuel mass ratio (ϵ) of blended fuel with 25% n-butanol (B25) was higher than that of blended fuel with 15% n-butanol (B15) under the same conditions. ϵ increased with an increase in S_q on the dry walls, but, under SWW conditions, it decreased with an increase in oil film thickness. The fuel film morphology was almost unaffected by the change in S_q , but the results implied that the roughness parameter-Skewness (S_{sk}) exerted a greater impact. The mean thickness h_a and accumulated diameter D_l of the adhered fuel film increased with an increase in h_l , but, under FWW conditions, the effect of the roughness on the adhered film's features was insignificant.

Keywords: wall roughness; lubricant film; n-butanol-diesel blends; spray impingement; adhered fuel film

1. Introduction

In conventional diesel engines, a “trade-off” relationship exists between nitrogen oxide (NO_x) and soot emissions [1]. To resolve this problem, advanced combustion modes have been proposed such as homogeneous charge-compression ignition (HCCI) and premixed charge-compression ignition (PCCI) [2,3]. Early injection is a key technology that can extend the mixing time of liquid fuel and air to achieve a homogeneous mixture and a high efficient combustion. However, because of the early spray time, the cylinder pressure and temperature are lower, which leads to an increase in spray penetration and a decrease in droplet evaporation. Part of the injected fuel will impact the cylinder liner or piston head. The adhered fuel film evaporates slowly and dilutes the lubricant film on the cylinder liner [4,5]. The mixture concentration near the wall will be too rich, which contributes to a deterioration in combustion and carbon monoxide (CO) and hydrocarbon (HC) emissions.

Therefore, the study of the fuel-injection and impingement characteristics is significant and extensive research has been conducted experimentally and numerically to reveal the mechanism of spray and impingement. Akop et al. [6,7] investigated the features of the fuel film that were adhered

on an original dry disk after impingement with different operating parameters. The adhered fuel tended to decrease with an increase in injection pressure, ambient pressure, and disk inclination angle. In addition, the adhered film thickness showed a decreasing trend with an increase in injection pressure but a decrease in the impingement distance. Panão et al. [8] studied the influence of cross-flow on the droplet impingement process visually with phase-Doppler technology. Cross-flow would weaken the impact energy that was available for secondary atomization, but the advantage was the airflow, which enhanced the fuel film evaporation and avoided the re-impact of secondary droplets on the plate. Wang et al. [9] used an ultra-high-speed imaging technique to study the process of isooctane droplets impacting on a wall. Microscopic pictures showed that droplet splashing or rebounding was affected mainly by the high spray velocity in the early stage, but then the droplets tended to stick on the wall for lower-velocity and adhered-film conditions. Zhang et al. [10] modified the widely used Estrade et al.'s impact model [11] and proposed a simplified droplet-collision model to predict the spray and splashing behavior. The improved model could provide a reliable prediction of droplet shape, velocity, and Sauter mean diameter under different ambient pressure conditions.

Besides ambient factors and injection parameters, splashing and spreading after impingement are affected significantly by the wall conditions such as the roughness, texture, liquid film, and wall temperature [12]. Deendarlianto et al. [13] studied the influence of wall roughness on the dynamic behavior of single and multiple successive water droplets after impacting the inclined hot plate surface. The surface roughness played a significant role in the cooling process. For a hot wall that was contacted with successive water droplets, a higher roughness yielded a shorter droplet evaporation time on the plate. During the later stage of droplet contacting and plate contacting, the spreading diameter increased with an increase in wall roughness. Tang et al. [14] conducted experimental research on the impact behavior of single droplets of water, decane, ethanol, and tetradecane. The researchers were particularly interested in the effect of wall roughness. With an increase in Weber number (We) and roughness (Ra), splashing that was caused by secondary droplets was favored. The roughness could slow down the spreading process, which decreased the dimensionless maximum spreading diameter β_{max} . However, the transition from spreading to splashing was enhanced by the increase in Ra . Zhang et al. [15] compared the characteristics of microdroplet spray impingement on dry and wet walls. The existence of the liquid film changed the critical number K , which determined the boundary of the droplet splash/non-splash. The liquid film on the plate promoted microdroplet splashing. Guo et al. [16] conducted a simulation study to investigate droplet evolution after colliding with a plate covered by liquid film with the combined level set and volume of fluid (CLSVOF) method. The splashing time is advanced by accelerated droplet motion and the spreading diameter tended to decrease with an increase in liquid film thickness. The above analysis shows that most of the wall surface parametric study focused on the effect of Ra or liquid film thickness on the characteristics of a single water droplet's impingement process. Rarely have practical conditions for the spray impingement of engines been considered closely where the effects of cylinder surface roughness and lubricant film exist simultaneously. Therefore, this paper focuses mainly on the combined influence of different types of roughness with lube oil film thickness on the quality and microscopic characteristics of the adhered fuel film to analyze the spray-impingement processes under different simulated engine operation conditions. A sequence of two-dimensional measured evaluation parameters of roughness such as Sa and Sq (see Introduction, Section 2) could represent the roughness more comprehensively than the one-dimensional evaluation parameter Ra that is used in most previous papers.

To reduce excess CO and HC emissions that are caused by early injection except for increasing the mass of air intake [17], many oxygenated alternative fuels that can improve the combustion efficiency have been studied and their use has been attempted in diesel engines such as alcohol fuels [18,19], ester fuels [20], canola oil [21], and other biodiesel fuels [22]. Among these promising diesel substitutes, n-butanol exhibits unique advantages. For instance, it has a higher energy density and a lower evaporation pressure, which make it easy to evaporate and mix with air to achieve a greater amount of burning [23]. Moreover, it can be dissolved in diesel easily in any proportion without phase separation.

Additionally, it results in a lower steel corrosion [24]. For future applications, much effort has been expended to investigate the characteristics of spray, combustion, and emission for use of n-butanol in diesel engines. The microscopic spray features of neat soybean biodiesel and 20% n-butanol biodiesel blends by using the particle/droplet image analyzing (PDIA) technology, which was determined by Mo et al. [25]. The droplet diameter at the spray periphery was larger than that in the spray center because of the air-entrainment effect at the spray periphery. Fewer droplets existed near the sprayed liquid core where the effect of viscosity was important. Research by Chen et al. [26] indicated that the production of CO, HC, and soot was lower when n-butanol was used in diesel engines compared with methanol and ethanol under the same conditions. The results of Yao [27], Şahin [28], and Yilmaz [29] indicate that the use of diesel-n-butanol blending fuels could decrease the soot and CO emissions of diesel engines significantly without deteriorating in fuel economy and NOx emissions.

These achievements indicate that diesel-n-butanol blends are a promising fuel and are worthy of in-depth study [25]. In our previous work [30–32], we have studied the effect of the injection parameters, ambient parameters, and fuel characteristics on the impingement process and adhered film distribution. However, according to the analysis above, the surface condition of the impinged wall also plays a significant role in droplets' adhesion and splashing. In this work, we originally attempted to explore the effect of surface roughness's evaluated parameters and lubricant film with different thickness on the spray-impingement process. We expected to establish methods to control the fuel-adherence phenomenon. A report by Sakda [33] indicates that the appropriate volumetric fraction of n-butanol should be close to 20%, and based on Reference [32], the adhered film ratio did not changed much when the blended ratio was lower than 20%. Therefore, the volumetric ratios for n-butanol in this work were 15% and 25% and were referred to as B15 and B25, respectively. The physical properties of the two blends were estimated according to the data of References [34,35] and provided in Table 1. We also supplemented the properties of pure diesel for reference.

Table 1. Physical properties of the fuels (20 °C).

Fuel Abbreviation	Density (kg/m ³)	Dynamic Viscosity ($\times 10^{-3}$ pa·s)	Surface Tension (mN/m)
B15	849.8	3.627	26.03
B25	845.1	3.491	25.57
Pure diesel	856.5	3.847	26.67

2. Materials and Methods

2.1. Experimental Apparatus and Materials

To investigate the characteristics of the adhered fuel film after spray impingement, the spray-impingement and film-thickness measurement system were used, which is shown in Figure 1. Figure 1a shows the constant-volume vessel with a common-rail injection system for fuel spray and wall impact including the high-pressure common rail system, the constant-volume vessel-injection system, and the electronic control unit. The injector with a single hole is mounted in the central top of the vessel and the angle between the spray direction and the plate's horizontal plane could be adjusted by rotating the plate 15°–90°. Therefore, the plate surface will be impacted by the injected fuel and covered by the adhering fuel film. The blended mixture in the tank were premixed and transported to the injector by a high oil pump and the injection pressure can be controlled between 0 MPa–300 MPa by adjusting the solenoid valve opening. More details about this part of the system can be found in Reference [30]. After the injection, the plate surface with a fuel film will be measured by using the oil film sensor from Infralytic, which was illustrated in Figure 1b. This instrument is portable and uses infrared filter technology, which can determine the fuel film thickness on the metal surface accurately and rapidly. Its working principle is based on the Lambert–Beer law [36], which can calculate the liquid film thickness by using light absorbance after a light beam has passed through the film. Details of this device are described in References [31,35].

Figure 1. Spray impingement and film-thickness measurement system: (a) schematic diagram of spray-impingement test rig [30]; (b) wall film thickness measurement instrument [31].

To study the effect of wall roughness on the fuel film's characteristics, four circular cold-rolled steel plates with different surface roughness types ($S_q = 0.24 \mu\text{m}$, $1.00 \mu\text{m}$, $1.84 \mu\text{m}$, $2.59 \mu\text{m}$) have been used. The plate shape is identical and their diameter and thickness are 95 mm and 3 mm, respectively. One surface of each plate is ground by the grinding machine and a small area has been cut to measure the wall roughness by using a 3D optical profiler. The plates and corresponding measured area's microtopography are shown in Figure 2 with each number following ξ . For most previous papers analyzed above (Section 1), researchers always focused on the linear evaluation parameter of surface roughness (R_a or R_q), but it could only reflect the one-dimensional micro-texture height distribution. In this work, we used mainly the root mean square height S_q to analyze the influence of roughness on the fuel film. It is R_q 's areal expansion with two-dimensional form and it can represent the practical roughness of the plate surface in most cases so that it will have a real impact on droplets' spreading during the impingement process. Numbers that follow ξ are arranged according to the value of S_q from smallest to largest, which means that the ξ_1 plate is smoothest and ξ_4 is rougher than the others. The results of the wall roughness parameters of these plates are shown in Figure 3. Table 2 provides the formula definition where $z(x, y)$ is the height of the measured points in the (x, y) two-dimensional planes and "A" is the area of the measured region. Result of Figure 3 shows that S_a and S_q both have the same development trend for their averaging characteristics. The averaged performance also influences the maximum height and S_z has a similar changing trend with S_a and S_q . The differences of S_{sk} among ξ_1 , ξ_3 , and ξ_4 are small, but which of ξ_2 is lower obviously than others. The relationship of the overall concave–convex extent of the wall and S_{sk} is that, if $S_{sk} < 0$, the surface microstructure shows a high height distribution, so it implies that the convex extent of ξ_2 is relatively significant. Additional details on these roughness parameters are given in Reference [37].

Figure 2. Impinging plates with microscopic surface images.

Figure 3. Results of the surface roughness parameters of test plates.

Table 2. Definition and formula of roughness parameters.

Roughness Parameter	Definition Formula
Arithmetic mean height (Sa)	$Sa = \frac{1}{A} \iint_A z(x, y) dx dy$
Root mean square height (Sq)	$Sq = \sqrt{\frac{1}{A} \iint_A z^2(x, y) dx dy}$
Skewness (Ssk)	$Ssk = \frac{1}{Sq^3} \left[\frac{1}{A} \iint_A z^3(x, y) dx dy \right]$
Maximum height (Sz)	$Sz = \max_A z(x, y) + \left \min_A z(x, y) \right $

2.2. Experimental Conditions

Table 3 shows the operational conditions of this experiment. To minimize the evaporation of liquid film to obtain details of the film features, the injection and measurement process occurs under room pressure and temperature. The blended fuel will be sprayed by a high pressure and impinged vertically on the plate (angle between injection direction and the horizontal plane of the plate was fixed in 90°). Three wall conditions will exist including the dry wall condition that indicates the influence of wall roughness on the film characteristics and the fully wetted wall (FWW) condition means that each plate surface will be covered by an original lubricating oil film with a thickness ($3\text{--}4 \text{ g/m}^2$, $\sim 3.3\text{--}4.4 \text{ }\mu\text{m}$) that is larger than Sq. The semi-wetted wall (SWW) condition entails that the plate will be covered by original lubricating oil with a thickness of 0 g/m^2 (dry wall), $1\text{--}2 \text{ g/m}^2$, $2\text{--}3 \text{ g/m}^2$, and $3\text{--}4 \text{ g/m}^2$, and the wall roughness is fixed at a maximum $Sq = 2.59 \text{ }\mu\text{m}$ (ξ_4 plate), which can be transferred to a change in the dimensionless number.

$$\varphi' = h_l / Sq \quad (1)$$

Table 3. Experimental conditions.

Experiment Parameter	Conditions
Fuel	B15, B25
Ambient temperature	20 °C
Ambient pressure	0.1 MPa
Injection pressure	80 MPa
impingement distance	60 mm
Wall condition	Dry/FWW/SWW

2.3. Adhered Film Characteristic Parameters

2.3.1. Adhered Fuel Mass Ratio

After fuel impinging, some droplets will splash off the wall and some fuel will adhere to the plate, which affects the evaporation speed and mixture concentration distribution near the wall directly. In our previous study [30], we defined the adhering fuel ratio (ε) on the dry wall condition as Equation (2).

$$\varepsilon = m_{ad}/m_{inj}, \quad (2)$$

where the mass of the single injected fuel (m_{inj}) and adhered fuel (m_{ad}) were measured by using a precision balance (0.01 mg). However, with the existence of the original lubricant film, the mass loss of lube oil (Δm_l) in the impingement process cannot be ignored. Therefore, under the wet wall conditions, Equation (2) can be improved by the equation below.

$$\varepsilon = (m_{ad} - \Delta m_l)/m_{inj}. \quad (3)$$

2.3.2. Longitudinal Section Profiles of Adhered Fuel Film

A fusion deformation of liquid film in a short time after adhering will cause a distortion in measurement results. As shown in Figure 4, we make the longitudinal section profile in the middle of the film with 20 points of film thickness that represent the three-dimensional accumulation of adhered fuel film to show the integral film thickness distribution. The entire measurement length is 50 mm in the Y-axis direction (see Figure 1b).

Figure 4. Thickness distribution of adhered fuel film represented by a longitudinal section profile in the middle of the film.

2.3.3. Average Thickness (h_a) and Accumulated Diameter (D_l) of Adhered Film

As shown in Figure 4, the adhered fuel film accumulates around the impacting central point and presents a “mountain” shape. In Reference [31], it has made some similar definitions about thickness and diameter of adhered film, but it didn’t consider the effect of the original lubricant film thickness. Therefore, we should re-define the metrical parameters. The red line is the original lubricant film with a mean thickness of h_l ($h_l = 0$ under dry wall conditions). To investigate the effect of roughness and wet walls on the evaporation and spreading of the adhered fuel film, we have defined two quantity factors to understand the adhered film, which include the average thickness (h_a) and accumulated diameter (D_l) as the formula below.

$$h_a = \frac{1}{n} \sum h_i, h_i \geq m \quad (4)$$

where n is the number of measured points, h_i is the film thickness at each point, and m is a reference thickness (g/m^2), which is the lower limit of the measurement points with a range of:

$$m = \begin{cases} 1.5, & h_l = 0 \\ 3.0, & 1 \leq h_l \leq 2 \\ 4.0, & 2 \leq h_l \leq 3 \\ 5.0, & 3 \leq h_l \leq 4 \end{cases} \quad (5)$$

D_l is a measured length of the line that is intercepted by the two sides of a film profile when the thickness is m .

3. Results and Discussion

3.1. Development of Adhering Fuel Mass Ratio

Figure 5 shows the adhered fuel mass ratio (ϵ) of the n-butanol/diesel blended fuels with 15% and 25% blending ratios at different roughness types for dry walls (a); FWW (b); and SWW (c). Regardless of whether the wall is dry or wet, the adhering ratio of B25 is higher than that of B15. This result may be consistent with Reference [32], which shows that higher n-butanol ratios imply a lower spreading radius and higher impinged spray heights interact with air. Due to the normal pressure and temperature, the masses for evaporation of the two blends that were sprayed before impinging were not significantly different. Due to the high injection pressure and the same injection pulse for the defined fuel-injected volume, the lower density of B25 implies a lower momentum for wall impact so that the rebounded mass will be less than that of B15. Combined with the decrease in viscosity and surface tension to lower the spreading resistance, an explanation can be provided for the phenomenon that, although trends for the two blended fuels are similar under dry wall or FWW conditions (Figure 5a,b), differences in ϵ between these walls become insignificant for B25.

Figure 5. Adhering fuel mass ratio (ϵ) of B15 and B25 with different wall conditions: (a) dry walls with different roughness; (b) FWW with different roughness; (c) SWW with different ϕ' .

The effect of roughness on ε for B15 is relatively clear when the fuel impinges on dry walls. As shown in Figure 5a, the adhered fuel ratio presents an increasing trend with an increase in S_q and this trend can contribute to the reason for the liquid fuel drops impinging on the wall. The increase in roughness enhances liquid osmosis on the wall surface and hinders the evaporation of secondary smaller droplets. However, Figure 5b shows that, when stripes on these walls are fully covered with lube oil, the adhered ratio remains almost unchanged, which means that, when the wall film is thicker than S_a or S_q , the influence of roughness on the droplet splash decreases or has no effect. A similar conclusion for a single water droplet impacting is provided in Reference [38].

After comparing the results of the dry wall and FWW conditions, it is important to research the regulation of ε when the thickness of the original oil film is smaller than S_q . Figure 5c shows the development in an adhered fuel ratio for B15 and B25 after spray impinges on the wall under SWW conditions. Since it is inconvenient to contrast the results between B15 and B25 from φ' , which will change as h_l changes in each procedure, a general term " φ " was used in Figure 5c to represent each lubricant film thickness. Additionally, φ_1 to φ_4 implies an increase in φ' . Differences in the results are not obvious especially for higher n-butanol blends, but, in general, these exhibit a downward trend for ε with an increase in φ' . The presence of lubricant film increases the viscosity resistance for the fuel droplet splash so that the change in ε can be small. Equation (3) shows that, with an injection pressure of 80 MPa, sprayed fuel has a large impact momentum and the lubricating film will bounce off the wall, increase the film quality difference Δm_l , and decrease ε . Therefore, the findings of Figure 5c can be proof that, under similar engine conditions, a lower ε does not imply an improvement in fuel and air mixture, but, rather, it means that more lube oil will be consumed and deteriorate the combustion.

3.2. Adhered Fuel Film-Thickness Distribution

3.2.1. Dry Wall Conditions

After fuel has impinged on the wall and splashed, the adhered liquid will gather and form a "mountain-like" film, which evaporates and mixes with environmental air to form combustible gas. At the same time, the distribution of fuel film influences the speed of evaporation. Figure 6 shows longitudinal section profiles of the adhered film for B15 and B25 blends with a change in roughness of the dry wall conditions. Most film peaks appear as a "W" shape, which means that a concave-ring exists around the central point of the impinged beam. Ring formation results because, after the liquid fuel impact on the wall, adhered fuel will accumulate at the impinging point and try to spread. Air has an entrainment effect on the spreading film edge, forms a resistance, and raises the edge. The liquid fuel surface tension makes the film edge rebound to the center at the same time. At this point, the B25 peak widths with a lower surface tension are larger than those of B15. Few differences exist among the height of these film peaks except on the ξ_2 wall. As described in Reference [39], this result implies that the impact of wall roughness on the impinging characteristic cannot be only required by average measured parameters such as R_a . In this work, we have found that the particularity of ξ_2 is that the skewness of the height distribution, S_{sk} , is lower than the other three plates and it means a higher convex extent is analyzed above. Therefore, with a fuel-droplet impact, the bounced mass will increase and the adhered film will be thinner. It also means that the roughness parameter S_{sk} of the surface has an important influence on the distribution of adhered film during the impingement process under a dry wall condition.

Figure 6. Longitudinal section profiles of adhered fuel film for (a) B15 and (b) B25 blends under dry wall conditions.

3.2.2. FWW Conditions

Figure 7 shows the longitudinal section profiles of the adhered fuel film for B15 and B25 blends with FWW conditions. The plates are covered by lubricating oil with a fixed thickness (3–4 g/m²). The different dotted lines represent the original lube oil film thickness for each wall. Compared with the dry wall conditions, the concave-ring phenomenon still exists on FWW for the same reason as mentioned above. However, adhered peaks have a wider and flatter shape and the difference between these films is smaller. In combination with the analysis for the adhered fuel mass ratio, if the thickness of the original oil film is larger than the roughness parameter Sq, the lube oil's fusion effect plays a leading role in fuel spreading, which changes the peak shape rapidly and tends to spread over a larger area.

Figure 7. Longitudinal section profiles of adhered fuel film for (a) B15 and (b) B25 blends under FWW conditions.

3.2.3. SWW Conditions

Figure 8 shows the longitudinal section profiles (blue lines) of the blends after impinging on the ξ_4 wall with a change in original oil film thickness (1–2 g/m², 2–3 g/m², 3–4 g/m²). The red lines include the lubricant film thickness before impact. The shape of the fuel film peaks with a 1–2 g/m² oil film thickness, which is almost the same as that on the dry wall. After the oil films change to 2–3 g/m²

and $3\text{--}4\text{ g/m}^2$, the shape of the adhered film peaks become smoother and flatter, which may indicate that when the original oil film is thicker than S_q by more than half, the fusion effect between the liquid fuel and the lubricating oil will be prominent and rapid. The effect of original lubricant film becomes larger than wall roughness. Therefore, it can be found that many automobile engine manufacturers use surface treatment to change the cylinder surface topography such as by using small pits to hold more oil for a better lubrication performance. However, in this situation, the oil film should be thick, which will yield a more fuel–lubricant mixture and results in an increased dilution and consumption of lube oil.

Figure 8. Longitudinal section profiles of adhered fuel film under SWW conditions: (a–c) B15 with different lubricant film thicknesses; (d–f) B25 with different lubricant film thicknesses.

3.3. Average Thickness (h_a) and Accumulated Diameter (D_l)

Figure 9 shows the development of average thickness (h_a) and accumulated diameter (D_l) of the adhered film after spray impingement of B15 and B25 blends under three wall conditions. To provide reasonable contrast results, the weak impact of the density of lubricating oil is ignored and units of h_a are converted from (g/m^2) to (μm) . Under almost all conditions, h_a and D_l of B25 are higher than those of B15 and the reason for this result can be added from the results of the adhering fuel ratio ϵ for the lower rebounded mass of B25. Under dry wall conditions shown in Figure 9a, the variation in D_l increases slowly with an increase in S_q . A larger S_q means a larger resistance for spreading adhered

fuel, which causes more fuel to gather at the impact point. Due to the larger accumulated diameter with more adhered fuel mass, the change in h_a with a different wall roughness is small, which implies that the roughness at this work scale is not the major factor that affects the adhered film's average thickness. Under FWW conditions, according to the above analysis, the influence of roughness is not clear after the lube oil has almost covered the wall surface texture. For SWW conditions, h_a and D_l mostly increase with a rise in φ' . As the original lubricant film thickness increases, the fusion effect increases gradually and then promotes the spreading of adhered fuel away from the impingement point, which increases D_l . An increased h_l improves the initial measured value m in Equation (4), increases the average film thickness h_a , and emphasizes the important influence of splashing lube oil mass Δm_l on the decrease in ε , which is shown in Figure 5c.

Figure 9. Mean thickness h_a and accumulated diameter D_l of adhered fuel film with different wall conditions: (a) dry walls with different roughness; (b) FWW with different roughness; (c) SWW with different φ' .

4. Conclusions

An experimental study was conducted to investigate the effect of surface roughness and lubricant film on the characteristics of the adhered fuel film of n-butanol-diesel blending fuels after spray impingement. The lubricant film thickness can be changed. We mainly focus on the development of the adhered mass ratio, the thickness distribution, and the morphological characteristics of the adhered fuel film. The main conclusions are below.

- (1) The adhered fuel film mass ratio ε of B25 is larger than B15 for a lower splashing mass of B25 that is caused by the smaller impinged momentum. However, gaps of ε between different surface roughness types became inconspicuous with an increase in n-butanol blending ratio. ε increases

with a rise in roughness S_q under dry wall conditions, but decreases with an increase in lube oil film thickness under FWW conditions.

- (2) The thickness distribution of the adhered fuel film does not change much with different S_a or S_q , but the film profile of the ξ_2 wall showed a lower shape than others, which signifies that S_a or S_q were not the only parameters that were used to evaluate the effect of roughness. This plate has a lower S_{sk} , which determined the overall concave–convex degree of the wall. The influence of roughness was insignificant for a lubricant film thickness that was greater than S_q .
- (3) The mean thickness h_a and accumulated diameter D_l of B25 were higher than those of B15 under almost all conditions and stable developments of h_a and D_l resulted in dry walls and FWW. Both were enhanced by an increase in φ' , which implies that, under thinner lubricant film conditions ($\varphi' < 1$), the wall roughness is an important factor that cannot be ignored for an impacting process.

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Abbreviations

S_q	surface roughness parameter-root mean square height (μm)
h_l	mean thickness of original lubricant film (g/m^2)
SWW	semi-wetted walls
FWW	fully wetted walls
B15/B25	15%/25% volumetric blended ratio of n-butanol in blending fuels
ε	adhering fuel mass ratio (%)
S_{sk}	surface roughness parameter-skewness
h_a	mean thickness of adhered fuel film (μm)
D_l	accumulated diameter of adhered fuel film (mm)
R_a	one-dimensional surface roughness parameter-Arithmetic mean height (μm)
R_q	root mean square deviation of a line (μm)
S_a	two-dimensional surface roughness parameter-Arithmetic mean height (μm)
ξ_1 – ξ_4	four walls with increasing surface roughness S_q
A	area of the measured region for surface roughness
φ'	$\varphi' = h_l/S_q$
Δm_l	the mass loss of lube oil in the impingement process (mg)
Y	the measurement direction of the adhered fuel film thickness (mm)
δ	liquid film thickness (g/m^2)
φ	conditions with different original lubricant film thicknesses

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