Newtonian Airless Liquid Jet Interaction with a High Speed Moving Surface

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Abstract

In the railroad industry a friction modifying agent may be applied to the track in the form of a liquid jet. In this mode of application the interaction between the high speed liquid jet and a fast moving surface is important. Seven different Newtonian liquids with widely varying shear viscosities were tested to isolate the effect of viscosity from other fluid properties. Tests were also done with five surfaces having different roughness heights to investigate the effects of surface roughness. High speed video imaging was employed to scrutinize the interaction between the impacting jet and the moving surface. For all surfaces decreasing the Reynolds number reduced the incidence of splash and consequently enhanced the transfer efficiency. The Weber number had a smaller impact on splash than did the Reynolds number. The ratio of the surface velocity to the jet velocity has only a small effect on the splash, whereas increasing the roughness height/jet diameter ratio substantially decreased the splash threshold.

1. Introduction

The impingement of a liquid jet on a dry surface is of great importance in many industrial applications. Examples are spray-coating and impingement cooling. One interesting application is utilization of liquid friction modifiers (LFM) in the railroad industry. LFM is a water-based suspension of polymers and inorganic solids showing non-Newtonian behaviour [2]. LFM is applied on the top surface of the rail through sprayers mounted under the locomotives or rail cars [1]. Spray nozzles are typically air-blast atomizers located 75 mm from the top-of-rail surface. Improved fuel economy and reduced rail and wheel wear are the benefits of LFM application.

In the case of air-blast atomizers the LFM is transferred via atomized particles (droplets/ligaments) to the rail surface and the concerning issues are around the interaction between falling droplets/ligaments and the moving surface with its air boundary layer [3]. For touching the top of rail surface particles should have a minimum velocity to penetrate through the boundary layer and also to avoid high deflections in case of possible cross winds; but on the other hand having a high speed impact can end up with splash or rebound which leads to undesirable reduction of transfer efficiency [4].

Newtonian droplet impact on a dry/wet stationary surface has been studied thoroughly. Examples include works by Rein [5], Rioboo et al. [6], Yarin [7], and Deegan et al. [8]. These authors have studied the effects of different fluid properties such as viscosity, density, and surface tension on the possible post-impact outcomes. Rein [5] classified the impact of a single droplet on a dry surface into three different outcomes: deposition or spreading, splash, and bouncing. Later Rioboo [6] expanded this classification to six: deposition, prompt splash, corona splash, receding break-up, partial rebound, and complete rebound. Range et al. [9] and Crooks et al. [10] reported some results with respect to the effects of surface roughness on the post impact outcomes. According to their findings, surface roughness will significantly decrease the splash threshold. Detailed discussions about Newtonian droplet impact on a dry solid surface have been published in two recent reviews by Yarin [7] and Deegan [8].

In many industrial coating applications droplets impact on a moving surface rather than a stationary one. Mundo et al. [11] have studied the impact of a Newtonian droplet on a moving surface. In reference to the splash threshold they mention that, in the frame of reference of the solid surface, the tangential speed of the droplet is less important than the normal component of velocity. In contrast, Pavarov et al. [12], Courbin et al. [13], Okawa et al. [14], and Fathi et al. [15] all have shown that the tangential speed plays an important role and have tried to include it in their proposed criteria for splash/deposition. Pavarov et al. [12] observed that in the case of impact on a spinning disk the presence of an air boundary layer lifted the droplet from the surface. Courbine et al. [13] also reported asymmetric splashes in case of impact on a moving surface; in the frame of reference of the surface, the splash is intensified in the downstream tangential direction of the droplet and diminished in the upstream direction.

Recent experiments by Dressler [16] showed that the air-blast atomizers do not perform ideally in delivering LFM to top of rail surface; in most cases some post-impact liquid ligaments/droplets are carried away from the surface by the atomizing air jet, which reduces the transfer efficiency. To avoid this problem the authors devel-

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oped a new airless, non-atomizing sprayer, which produces a high-speed liquid stream. In contrast with the vast amount of research on droplet interaction with stationary/moving surfaces we know of no previous studies of a high speed Newtonian or non-Newtonian liquid jet impinging on a high speed moving surface. Liu et al. [17] investigated the interaction between Newtonian liquid jets and a stationary surface but were mainly concerned about hydraulic jump and heat transfer issues. Hlod et al. [18] found a mathematical model for the interaction of a high viscosity slow liquid jet and a slowly moving surface; since the jet and surface speed in their study were low, nothing about splash was mentioned. Although industrial LFM are highly non-Newtonian, this initial study was focused on Newtonian test liquids to avoid the greater complexity of the non-Newtonian case.

2. Materials and Methods

Seven different mixtures of water and glycerine were used as the Newtonian test liquids, to isolate the shear viscosity effects (Table 1). The viscosity of these mixtures varied by three orders of magnitude but the values of surface tension and density were constant to within ±13%.

An airless spray nozzle with a diameter of 648 µm was used to apply the test fluids to a rapidly moving projectile. The projectile used in this work had a 13mm thick polished steel surface, previously the top surface of a 136# designation rail, fastened to a wooden base which acted as a carrier for the impaction surface (Figure 1). This wooden base also approximated the side faces of a standard rail section to ensure that the aerodynamics of the airless spray around the rail top face mimicked a real rail.

To study the effects of surface roughness, five different grades of sandpaper with different average roughness heights were selected and attached on the top surface of the projectile (Table 2). The corresponding roughness height to jet diameter ratios fell in the range 0.02-0.65.

The projectile was fired by a high-speed linear transverse system (an air cannon) that was designed and built for similar experiments by Dressler [16]. Once the projectile leaves the air cannon barrel it passes beneath the spray nozzle. It then strikes an energy dissipation device and comes safely to a stop (Figure 2). The projectile speed can be varied, depending on the air pressure in the air cannon tank, from very low speeds (less than 1 m/s) up to speeds as great as 30-40 m/s. The main benefits of this test configuration are that it permits us to vary the surface speed independent of the jet velocity, and guarantees that impingement always occurs on a dry surface. In other words it helps us to repeat the coating process of LFM onto the rail surface while a relative velocity is present.

A Phantom V12 high speed video camera with a high-intensity halogen lamp for backlighting was used to visualize the jet impingement on the fast moving surface (Figure 3). The camera resolution was 800 x 1200 pixels with a frame rate of 6,200 pictures per second, and an exposure time of 9 microseconds.

3. Results and Discussion

Images were captured of the interaction between the surface and the liquid jet at different jet and surface speeds with various mixtures of water and glycerine. For all fluids tested, as the jet velocity increases impingement is more likely to be associated with splash than direct deposition. This finding is consistent with the single droplet impingement experiments of Yarin [4]. We also found that impingement transitions from direct deposition to splash while keeping the jet velocity constant but increasing the surface speed, Figure 4. This finding prompted the authors to introduce as a parameter the relative jet velocity, V_{rel}, which is the magnitude of the vector sum of the jet and surface velocities, Figure 5.

Jet impingement of a Newtonian, non-cavitating liquid on a surface is a function of the relative jet velocity, (V_{rel}), the jet diameter, (D), the impingement angle in the frame of reference of the surface, (α), the fluid density, viscosity, and surface tension, (ρ, μ, σ), and the surface roughness, (ε)[2]. These seven variables may be reduced to four dimensionless groups:

$$\mathrm{Re} = \frac{\rho V_{rel} D}{\mu} \tag{1}$$

$$We = \frac{\rho V_{rel}^{2} D}{\sigma} \tag{2}$$

α : Impingement angle in the frame of reference of the surface (3)

[2] It may also be a function of the surface material (e.g., as represented by the contact angle of a droplet on the surface), and the air properties, but these effects are not considered here.

$$\frac{\varepsilon}{D} : \text{Relative Roughness} \tag{4}$$

By adjusting the glycerine concentration and consequently changing the fluid viscosity and simultaneously controlling the relative jet velocity, wide ranges of different Re and We numbers were achieved through our experiments (three order of magnitude changes in Re number and almost one order in We). The fluid viscosity and surface tension were respectively measured by a HAAKE VT550 viscometer and a Du Noűy ring apparatus at a temperature of 25°C.

The relative jet velocity was calculated by vector summation of jet and projectile velocities. Projectile velocity was measured through analysis of high speed captured images. The jet velocity was obtained through flow rate and jet diameter measurements for different test liquids at different nozzle back pressures. Mass flow rate measurements were done by weighing discharged liquid from the nozzle over a span of 30 seconds. One common way of characterizing flow rate behaviour through a nozzle is by calculating the discharge coefficient at different Reynolds numbers. The discharge coefficient by definition is the ratio between the actual and ideal mass flow rate.

$$Discharge\,Coefficient : C_d = \frac{Actual\,Mass\,Flow\,Rate}{Ideal\,Mass\,Flow\,Rate} = \frac{\overset{\circ}{m}}{A_n\sqrt{2\rho.\Delta P}} \tag{5}$$

The relationship between C_d and Re is depicted in Figure 6. The discharge coefficient varies linearly with Re at really low Reynolds numbers and tends to have a constant value between 0.8 and 0.9 at high Reynolds numbers. This dependence is in accordance with the fact that viscous losses are more dominant at low Reynolds number than at higher Reynolds numbers. Lefebvre [19] has also reported a similar behaviour for round orifices discharging in ambient gas. The discharge coefficient starts to decrease at Re around 600 for water and glycerine (65 wt%) while this downturn happens at Re near to 1000 for water and glycerine (50wt%). These differences in the downward tendency of C_d for different liquids are believed to be a consequence of cavitation inception in the nozzle. Since the onset of cavitation is a strong function of local pressure and consequently velocity inside the nozzle, it can be expected that for liquids with lower viscosities cavitation inception will occur at higher Reynolds numbers. Suh et al. [20] and Payri et al. [21] reported the same result in their experiments on liquids discharging from orifices into an ambient gas.

Jet diameter was measured utilizing high speed close-up imaging of the jet at various heights from the nozzle exit. The captured images were then analyzed through an image processing code written in MatLab®. Figure 7 shows the normalized jet diameter at the nozzle exit (normalized by the nozzle exit diameter, D_0) versus the Reynolds number for two different mixtures. It is interesting that up to 10% jet expansion occurs at very low Reynolds numbers. Jet expansion for elastic non-Newtonian liquids is well known (so called “die swell”), but in our case with inelastic Newtonian liquids, the die swell was unexpected. However, Middleman [22 and 23] has observed the same behaviour experimentally and justified it on theoretical grounds.

As the jet Reynolds number increases, the die swell diminishes until at high Reynolds numbers the contraction ratio becomes independent of Re and keeps a constant value near 0.87. This finding is again consistent with the research of Middleman [22 and 23].

3.1. Smooth Surface

The splash/non-splash boundaries for Newtonian testing liquids are depicted in Figure 8 and 9. From Figure 8 (see also Figure 11 for rough surfaces) it is apparent that the splash/non-splash boundary is more strongly dependent on Re than on We . As seen in Figure 9 (see also Figure 12 for rough surfaces), the effects of α on the splash are also modest relative to the effect of the Reynolds number. The critical Reynolds number above which splash occurs is about 350 for impaction on smooth surface. Given that the Reynolds number used in this figure is based on the relative jet velocity, it is clear that the total impact energy is more important than its normal component. This finding is consistent with the insensitivity of splash to α , shown in Figure 9. It can be shown more obviously if one considers the splash/non-splash results for a single fluid: Figure 10 shows the jet velocity versus surface speed for different splash/non-splash points for water and glycerine (75%wt) mixture. The splash points are separated from deposition ones by a constant relative velocity curve, i.e. $V_{rel} = 14.9\,m/\sec$.

3.2. Rough Surfaces

By attaching sandpaper with different average roughness heights to the moving surface we have studied the effect of roughness. Interestingly, all five different roughness ratios show the same behaviour found for the smooth surface, i.e. the most important parameter is Reynolds number rather than We or α . Figures 11 and 12 summarize the results for roughness ratio equal to 0.21. For $\varepsilon / D = 0.21$ the critical Reynolds number has decreased to a lower value, around 60, relative to the smooth case. The dependence of the critical Reynolds number on the roughness ratios is depicted in Figure 13; increasing the roughness decreased the critical Reynolds number, which is in accordance with similar studies done on droplets [9, 10]. It can be observed that at lower roughness ratios (smoother surfaces) even a small increase in roughness ratio causes a significant decrease in the splash threshold but at higher values (rough surfaces) the splash threshold barely varies. Thus, the critical Reynolds number seems to decrease asymptotically to a constant value for larger roughness ratios. This finding is physically plausible because a thin jet striking a big obstacle should experience splash onset for about the same conditions, whether the obstacle is 10 times the jet diameter or 100 times the jet diameter.

4. Conclusions

An experimental investigation into Newtonian jet impingement on a high speed moving surface with different roughness heights has been performed. The experiments are qualitatively consistent with previous studies of single droplet impaction and also provide new information concerning the impaction of liquid jets on smooth/rough moving surfaces.

The key findings of the present study are:

- Increasing the surface roughness ratio decreases the splash threshold and consequently decreases the transfer efficiency. The splash threshold is more sensitive to roughness ratio changes at lower roughness ratios (e.g. 0-0.1) than at higher roughness ratios (>0.1).
- For both smooth and rough surfaces splash is more influenced by Reynolds number, defined using the jet relative velocity, than by either Weber number or the ratio of the surface velocity to the jet velocity. This confirms that the total impact energy is more important than the energy associated with the normal component of velocity. It can also be concluded that for this flow the liquid's viscosity plays a more important role than the surface tension.

Acknowledgement

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Nomenclature

D	diameter[m]
Re	Reynolds Number
V	velocity[m.s^{-1}]
We	Weber Number
α	Jet impingement angle in solid surface frame of reference[rad]
σ	surface tension[N.m^{-1}]
ε	Roughness Height [m]
μ	viscosity[Pa.s]
ρ	density[kg.m^{-3}]
Wt	Weight
C_d	Discharge Coefficient
m°	Mass Flow Rate[kg.s^{-1}]
A_n	Nozzle Area[m^2]

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Table 1: The composition and properties of Newtonian test liquids at 25℃ [2]

Glycerin[wt%]	Water[wt%]	Viscos-ity[mPa.s]	Surface Ten-sion[mN/m]	Den-sity[g/cm^3]
0	100	0.9	72.1	1.00
50	50	5.1	68.8	1.13
65	35	12.5	67.5	1.17
75	25	28.0	67.1	1.20
80	20	46.7	66.6	1.21
90	10	154	61.9	1.23
99.5	0.5	806	61.9	1.26

Table 2: The properties of different sand-papers used as rough surfaces

Sandpaper Sur-face Number	Grit Size	Average Roughness [micrometer]	Roughness Ratio
1	40	425	0.65
2	60	265	0.41
3	100	190	0.21
4	220	68	0.10
5	600	16	0.02

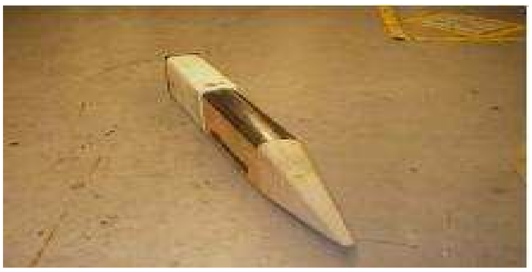

Figure 1. Projectile

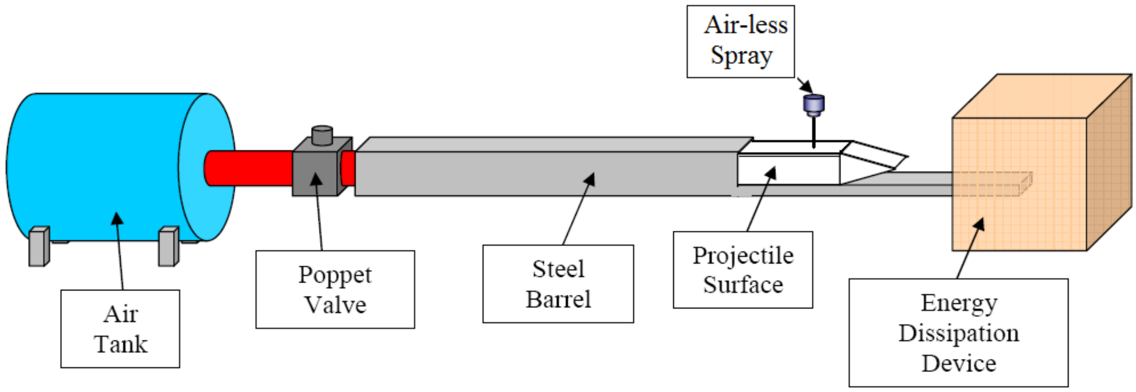

Figure 2. Experimental set-up: linear transverse system [16]

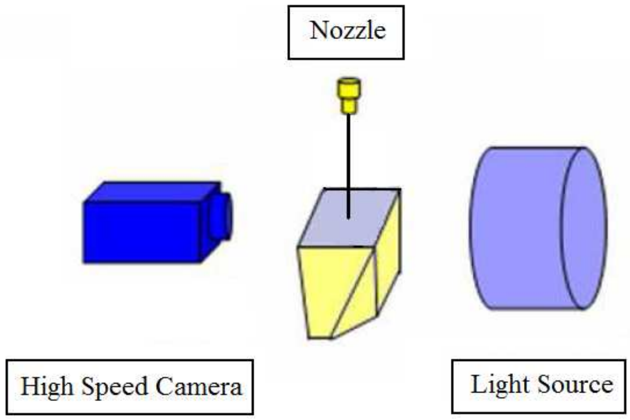

Figure 3. High speed camera and light source position relative to the projectile and the nozzle

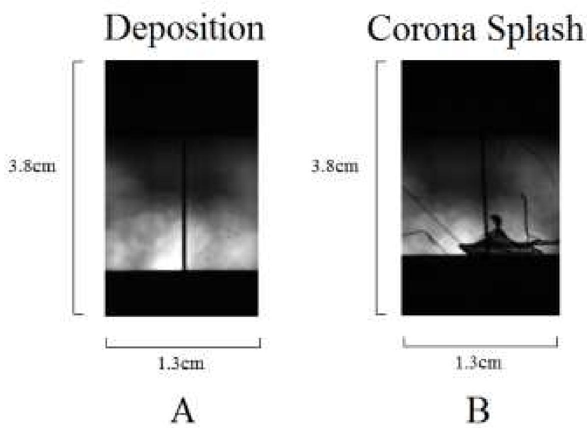

Figure 4. Projectile is traveling from left to right with a velocity equal to 10.8 m/s (A) and 13.9 m/s (B). In both (A) and (B) the jet velocity is 17.5 m/s.

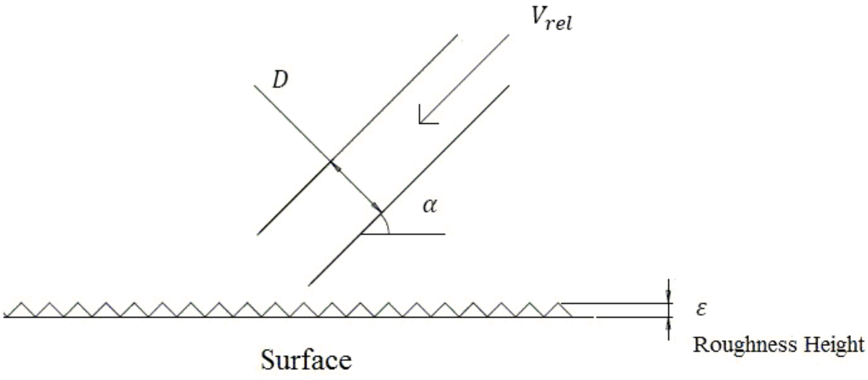

Figure 5. Relative Jet Velocity

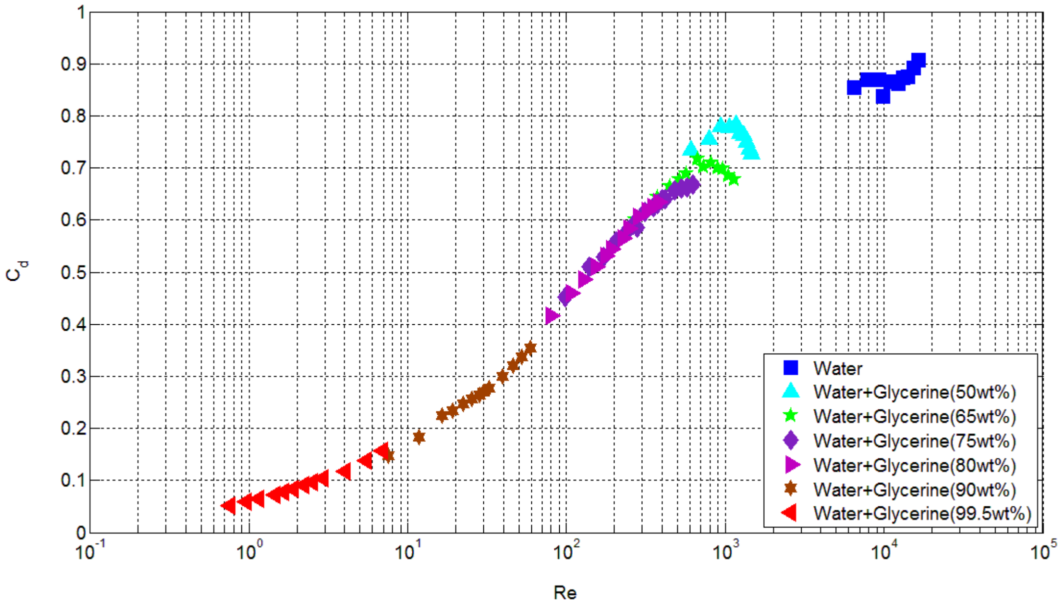

Figure 6. Discharge coefficient versus Reynolds number for different test liquids

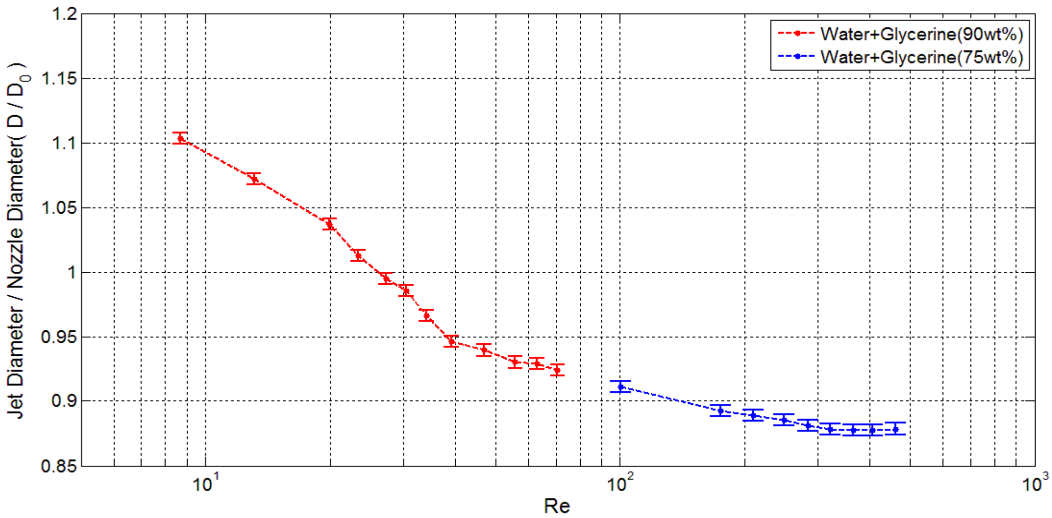

Figure 7. Normalized jet diameter versus Reynolds number at the nozzle exit

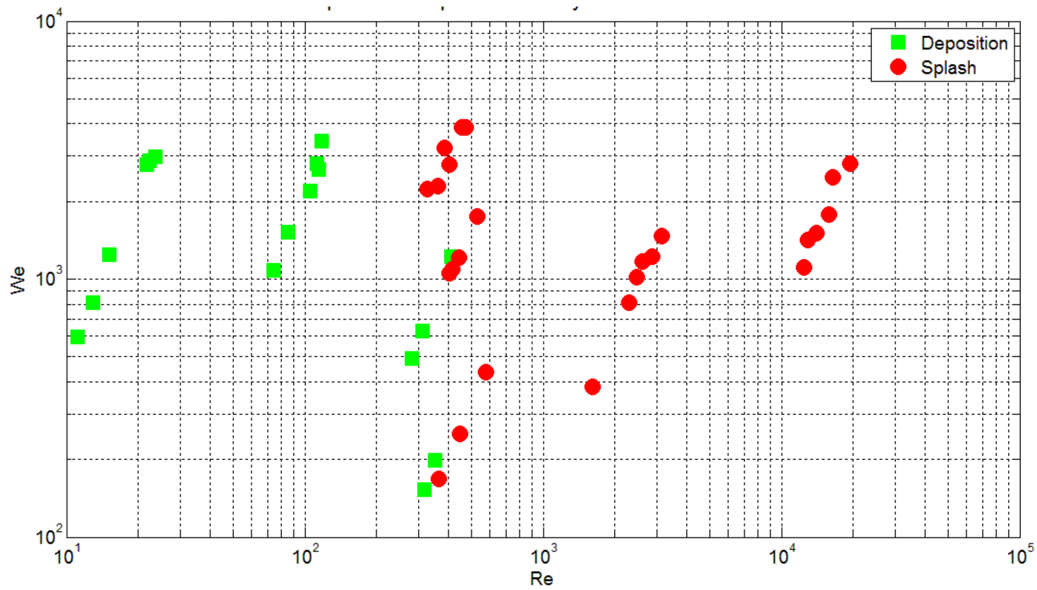

Figure 8. Splash/non-splash boundary for smooth surface

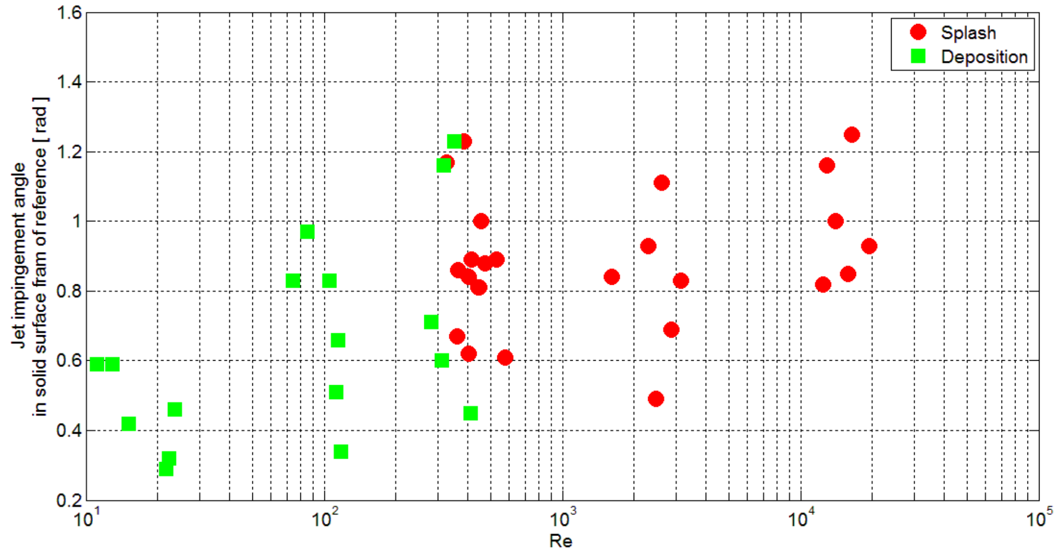

Figure 9. Splash/non-splash boundary for smooth surface

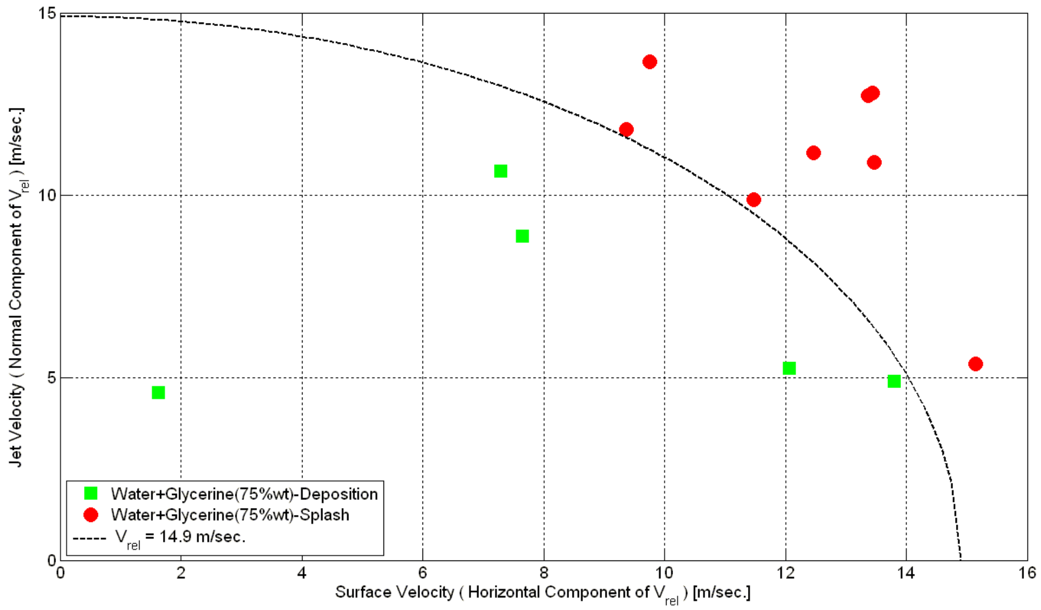

Figure 10. Splash/non-splash boundary for smooth surface

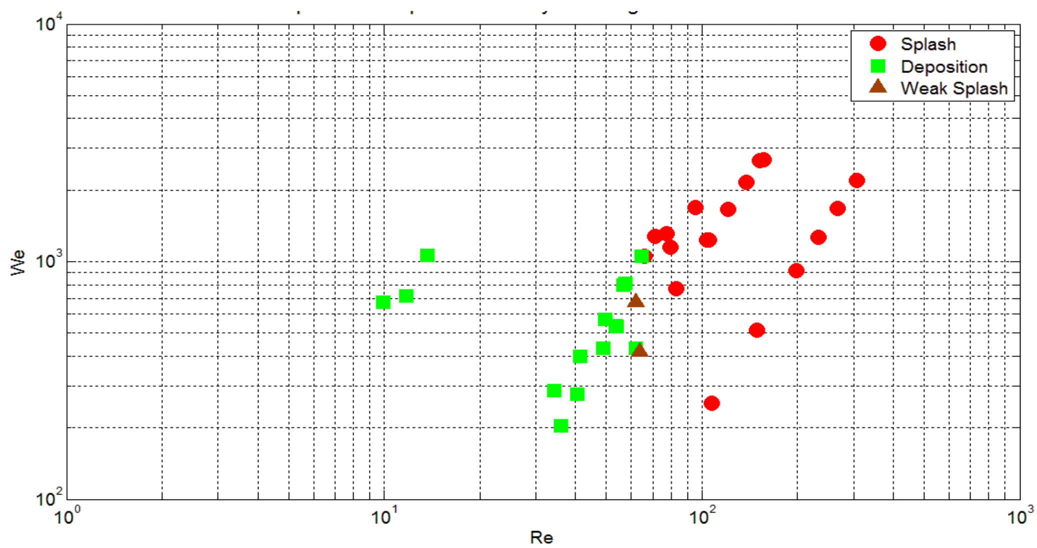

Figure 11. Splash/non-splash boundary for the surface with roughness ratio equal to 0.21

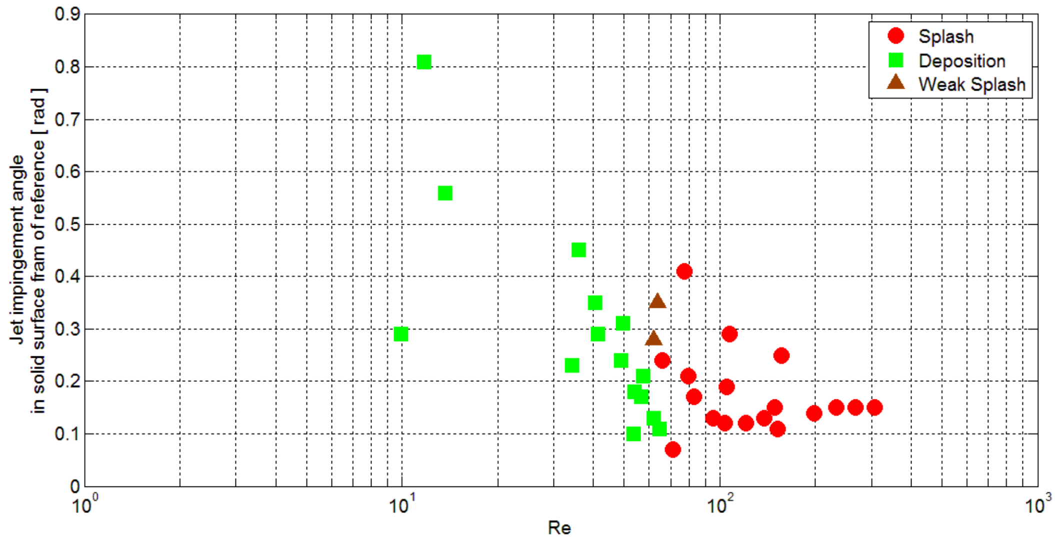

Figure 12. Splash/non-splash boundary for the surface with roughness ratio equal to 0.21

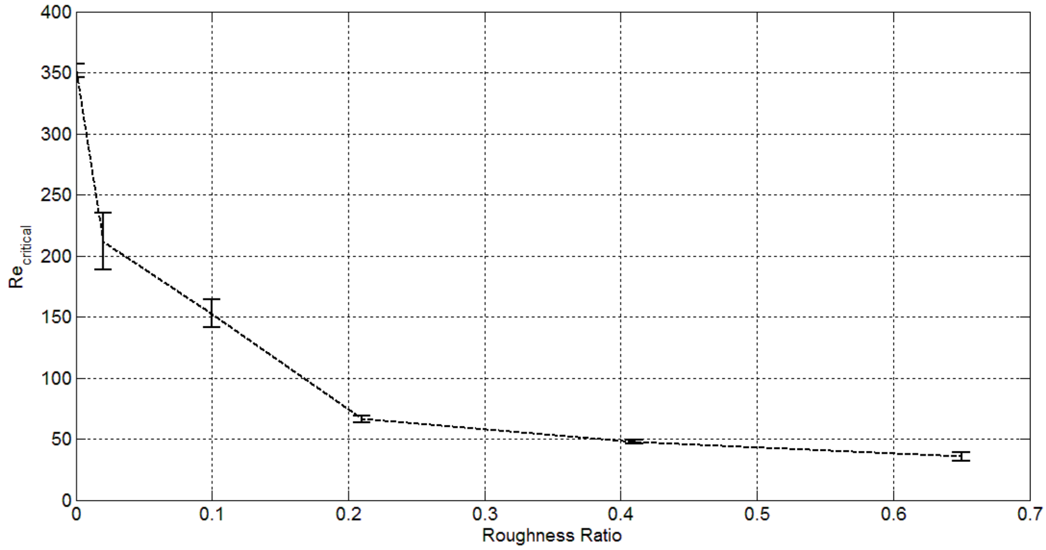

Figure 13. Critical Reynolds number versus roughness ratio