Int. J. Heat Mass Transfer. Vol. 25, No. 3, pp. 422–424, 1982
Printed in Great Britain

0017-9310/82/030422-03 $03.00/0

SURFACE EFFECT ON DROPLET DEPOSITION IN TWO-PHASE FLOW

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(*Received* 6 *April* 1981 *and in revised form* 17 *August* 1981)

NOMENCLATURE

$\bar{c}$, $= W_d/(W_g/\rho_g + W_d/\rho_d)$, bulk concentration [kg/m^3];
D, test section inner diameter [m];
f, smooth tube friction factor for turbulent flow;
d_{10}, characteristic arithmetic mean drop diameter [m];
k_d, drop deposition velocity [m/s];
N_0, mass rate of deposition of droplets [kg/(m^2.s)];
Re, $= \rho_g V_g D/\mu_g$, tube Reynolds number;
Re_f, liquid film Reynolds number, $4\Gamma/\mu_l$;
r_0, tube radius [m];
r_0^+, $\frac{1}{2}Re\sqrt{(f/2)}$;
U, velocity distribution in the gas phase [m/s];
u^*, $= \sqrt{(\tau_w/\rho_g)} = U\sqrt{(f/2)}$, friction velocity [m/s];
V_g, superficial gas velocity [m/s];
W_d, droplet mass flow rate in the test section [kg/s];
W_g, gas mass flow rate in the test section [kg/s];
y, normal distance from wall [m];
y^+, $= yu^*/\nu$, dimensionless distance.

Greek symbols

Γ, mass flow rate of liquid per unit width of tube [kg/sm];
μ_g, dynamic viscosity of gas [N.s/m^2];
μ_l, dynamic viscosity of liquid [N.s/m^2];
ρ_d, droplet density [kg/m^3];
ρ_g, gas density [kg/m^3];
ν_g, kinematic viscosity of gas [m^2/s];
τ, $= d_{10}^2\rho_d/(18\mu_g)$, drop relaxation time based on Stokes drag [s];
τ_w, wall shear stress [N/m^2];
τ^+, $= \tau u^{*2}/\nu_g$, dimensionless drop relaxation time.

INTRODUCTION

HEAT TRANSFER in a dispersed turbulent flow takes place via a variety of mechanisms: heat transfer from wall to the gas as in single phase flow by convection, heat transfer from the gas to the droplet by convection and vaporization, heat transfer from the wall to the deposited droplets by conduction and boiling or evaporation, and heat transfer from the wall to the droplets and vapor by radiation if the wall temperature is very high [1].

One major uncertainty in the prediction of dispersed flow heat and mass transfer exists in the determination of wall to droplet heat transfer, which can be significant for a wide range of wall temperatures [2]. This is further compounded by the fact that the mechanism of the droplet deposition is a complex process that makes a reliable estimation of the droplet transfer rates onto the wall very difficult. In such situations, it is desirable to evaluate the deposition rates experimentally. Unfortunately a direct measurement of the deposition rates of droplets suspended in a turbulent gas stream through a heated tube may not be possible since the droplets evaporate as they come in contact with the wall.

One alternative means in such situations is to obtain the deposition data for an adiabatic tube, and consider that information in the heat transfer calculations. It is usually convenient and often more informative to conduct the deposition experiments in an adiabatic transparent tube since it is possible to examine the nature of the liquid layer formed adjacent to the wall due to the deposited droplets, and also obtain pertinent information regarding the droplet diameters and velocities. Therefore many of the droplet deposition experiments reported have used transparent tubes [3–5]. In order to check if the data for the transparent tube is consistent with the data for the actual heat transfer surface which is usually metallic, it is necessary to obtain the data for the two surfaces and compare them to evaluate any surface effects on the deposition rates.

Earlier deposition experiments by the authors using an acrylic tube have indicated that, in the deposition test section two distinct flow regimes of the liquid film formation on the wall are visually observed (Fig. 1). In region A downstream of the inlet the deposited droplet flow rate was not sufficient enough to form a continuous liquid film around the periphery of the test section, resulting in a rivulet-type flow. Some droplets may rebound and the formation of this flow regime is dependent upon the tube surface mostly through the liquid–wall contact angle. The length over which region A persists can depend upon factors including Re, droplet flow rate and droplet size. Region B is characterized by a thin continuous liquid film covering the tube surface due to the increased film flow rate. The droplet deposition in region B is therefore not dependent upon the surface. In view of these factors, it was speculated that the average deposition rates for the entire test section may differ for different tube surfaces. To the authors' knowledge, such a study has not been previously reported in the literature. This aspect prompted the authors to study the surface dependence of the deposition rates. Surface roughness can also influence the deposition rates, and its effects are eliminated here by considering smooth tubes only.

The purpose of the present note is to report new deposition data for air–water droplet system in a vertical unheated stainless steel tube. The results will be compared with data for an acrylic tube [5] under similar conditions to evaluate the effect of tube material on droplet deposition in two-phase flow.

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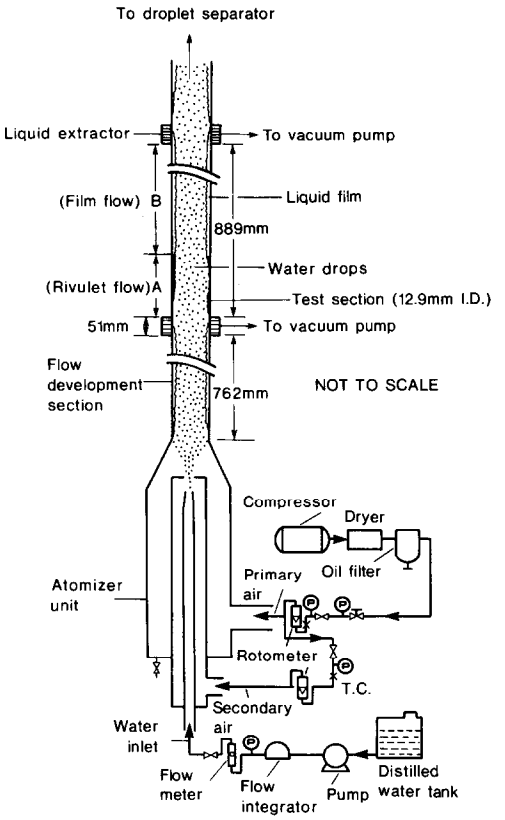

FIG. 1. Schematic of experimental set-up for deposition.

EXPERIMENT

A schematic of the experimental deposition test set up is shown in Fig. 1. The vertical stainless steel (type 304) test section is smooth and is 12.95 mm i.d., 3.05 mm thick and 889 mm long. Details of the experiment are given in [6]. The system considered is air–water droplet flow. A flow development section of about 60 tube diameters is used to ensure that the turbulent air-flow is fully developed at the test section inlet. The water droplets are generated by an atomizer in which a secondary air stream atomizes the liquid jet. Extraction units connected to vacuum pumps are used to remove the liquid film formed on the wall. At the inlet, the film thickness is made zero, and the flow rate of liquid extracted at the outlet of the test section yields the average deposition rates for the test section length. This technique is similar to that used by Cousins and Hewitt [3].

The measured deposition data are usually deduced in terms of a deposition velocity k_d as

$$k_d = N_0/\bar{c} \tag{1}$$

For particles or droplets characterized by the dimensionless drop relaxation time, τ^+ in excess of about 40 which is of interest in many two-phase flows, it has recently been shown [5] that the principal resistance to the droplet radial transport resides in the turbulent core. It was shown that [5] for Stokes particles, the dimensionless deposition velocity

$$k_d/u^* = f(\tau^+, Re) \tag{2}$$

and that for longer particles not obeying Stokes law, the density ratio ρ_d/ρ_g may additionally influence k_d/u^*, for which no satisfactory deposition theory presently exists. The friction factor f used for evaluating u^* is calculated from the well known correlations for smooth tubes

$$f = 0.0791\, Re^{-0.25}, \; 3 \times 10^3 < Re < 10^5 \text{ (Blasius)}$$
$$= 0.046\, Re^{-0.2}, \; 10^5 < Re < 10^6. \tag{3}$$

RESULTS AND DISCUSSION

Figure 2 shows the variation of measured k_d/u^* with τ^+ at $Re = 52{,}500$ for two test surfaces. Both the test surfaces are assured clean since the moisture and oil content in the air from the compressor were eliminated by passing the air through a dryer and an oil filter, and the liquid used is distilled water. Visual observation of the acrylic tube before and after the experiment also shows that the test surface is clean. The dark circles correspond to the data for stainless steel tube, and the open circles to those reported earlier for acrylic tube of 12.5 mm i.d. and 889 mm long [5]. The length of the rivulet flow region A as observed in the acrylic tube is less than about 30–40% of the test section length for the conditions shown in Fig. 2. Eleven runs were conducted using the steel test section. Absence of entrainment (generation of droplets and their dispersion into the core from the liquid layer on the wall) is confirmed using the correlation of Minh and Huyghe [7], which suggests entrainment of less than 0.5% for $Re = 52{,}500$. The estimated droplet diameter d_{10} ranges from 14 to 59 μm for the data in the steel tube and from 12 to 46 μm for those in the acrylic tube. The liquid flow rates W_d in the steel tube and the acrylic tube are in the range of $(15-34) \times 10^{-5}$ kg/s and $(16-28) \times 10^{-5}$ kg/s respectively. Droplet concentration in the experiment varies from run to run, and corresponds to a range of $(20-34) \times 10^{-3}$ kg/m^3 for the steel tube, and $(25-40) \times 10^{-3}$ kg/m^3 for the acrylic tube. The droplet τ^+ is in the range of about 300–8000. The results for both the two tube materials suggest that the deposition velocity decreased with the particle size [5]. The predicted k_d/u^* from the theory of [5] is also shown for comparison. The deviation of the theory with the data for larger droplets having τ^+ in excess of about 2000 is attributed to the inadequacy of the assumption of Stokes law, and is discussed in detail in [5]. It is seen that the measured k_d/u^* for stainless steel tube are in general slightly lower than those for the acrylic tube. However, the difference in the deposition velocity for the two surfaces is not significant, as compared to the experimental uncertainty in k_d/u^* which is estimated to about 12%. This suggests that there is no appreciable effect of wall on the deposition rates for the conditions and the size range of particles studied here.

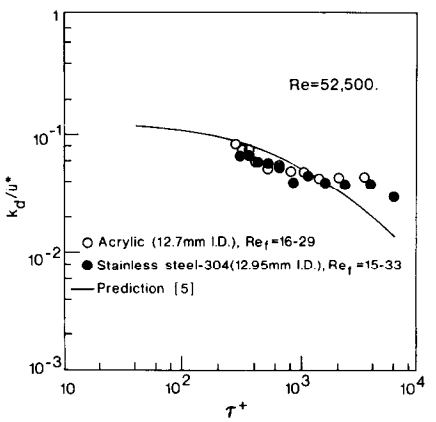

FIG. 2. Comparison of deposition data for stainless steel and acrylic tubes at $Re = 52{,}500$.

In reality, when deposition occurs there exists a sticking probability which is related to the adhesion of particles at the immediate vicinity of the wall, even though there is no drifting motion due to field effects [8]. The adhesive forces are either electrical or liquid (viscosity and surface tension) in nature. The observed mean deposition velocity therefore includes the probability that some droplets do not merge into the wall layer, but rebound and are removed by turbulent diffusion. There is, however, the possibility that the rivulent flow in a steel tube might be disturbed differently in an acrylic tube, even though the two flow rates were the same. Since sticking on a rivulet might be different from sticking on a dry surface, the overall deposition rate in the rivulet flow of an acrylic tube might be different from that in a steel tube. Because the entrainment in the present case is insignificant, and also due to the agreement of the deposition data for the two test surfaces, it can be inferred that the sticking probability for these conditions is close to unity. It therefore appears that the deposition rates for dry wall dispersed flow, rivulet dispersed flow, and annular dispersed flow are all represented by the data of Fig. 2, for the conditions studied here.

It can be concluded from the above results and discussion that for the conditions studied in the present work representing drop sizes and flow Reynolds number typical of many two-phase flow applications, the data for adiabatic acrylic tube may be used directly in the heat transfer analysis of dispersed two-phase flow in metallic tubes provided that there is no significant droplet entrainment. It should, however, be recognized that in some cases the condition of adiabatic tube may be much different from the condition of a heated tube due to non-uniform evaporation of droplet, vapor generation near the tube surface and change of surface tension at high temperatures, etc. Therefore, in such situations, the conclusion of this experiment may not be able to be applied to heated tubes directly without justification.

Acknowledgement—This work was performed with support of the National Science Foundation under grants ENG78-06211 and ENG80-05681.

REFERENCES

1. E. N. Ganic and W. M. Rohsenow, Dispersed flow heat transfer, *Int. J. Heat Mass Transfer* **20,** 855–866 (1977).
2. K. Mastanaiah and E. N. Ganic, Heat transfer in two-component dispersed flow, *J. Heat Transfer* **103,** 300–306 (1981).
3. L. B. Cousins and G. F. Hewitt, Liquid phase mass transfer in annular two-phase flow: Droplet deposition and liquid entrainment, AERE-R 5657 (1968).
4. Y. Hagiwara, K. Suzuki and T. Sato, An experimental investigation on liquid droplets diffusion in annular-mist flow, *Multiphase Transport* (edited by T. N. Veziroglu). Hemisphere (1980).
5. E. N. Ganic and K. Mastanaiah, Investigation of droplet deposition from a turbulent gas stream, *Int. J. Multiphase Flow* **7,** 401–422 (1981).
6. K. Mastanaiah, Experimental and theoretical investigation of droplet deposition and heat transfer in air–water dispersed flow, Ph.D. Thesis, University of Illinois at Chicago Circle (1980).
7. T. Q. Minh and J. Huyghe, Some hydrodynamical aspects of annular dispersed flow: Entrainment and film thickness, Symposium on Two-Phase Flow, Paper C2, Exeter, England (1965).
8. S. L. Soo and S. K. Tung, Deposition and entrainment in pipe flow of suspension, *Powder Technol.* **6,** 283–294 (1972).

Int. J. Heat Mass Transfer. Vol. 25, No. 3, pp. 424–425, 1982
Printed in Great Britain

0017-9310/82/030424-02 $03.00/0
Pergamon Press Ltd.

MINIMUM MASS CONVECTIVE ANNULAR FIN

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(*Received* 16 *April* 1981)

INTRODUCTION

IN A RECENT paper Mikk [1] has considered the problem of minimizing the mass of an annular fin on a cylindrical base of specified radius that rejects heat to the surroundings by convection at a specified rate. He asserts that, contrary to the results of Schmidt [2] and Duffin [3], the temperature at the tip of the minimum mass fin is not the same as that of the ambient fluid. In this paper we call attention to a logical flaw in his analysis that invalidates his conclusions. Moreover, if this flaw is corrected, his analysis then leads immediately to the earlier results.

We will use Mikk's notation and refer to equations in his paper without further explanation.

NOMENCLATURE

l, fin height [m];
q, dimensionless heat flux Q/Q_1;
Q, heat flux per meter of fin base [W/m];
r, radius [m];
v, dimensionless volume $2\lambda\alpha^2\vartheta_1^3 V/Q_1^3$;
V, fin volume per meter of fin base [m^2];
α, heat transfer coefficient [W/(m^{2}K)];
δ, fin thickness [m];
ε, dimensionless parameter $Q_1/2\alpha r_1\vartheta_1$;
ϑ, excess of fin temperature over temperature of ambient fluid [K];
θ, dimensionless temperature ϑ/ϑ_1;
λ, thermal conductivity [W/(m·K)];
ρ, dimensionless radius r/l;
σ, dimensionless parameter $\alpha l^2/\lambda\delta_1$;
φ, radius ratio r_2/r_1;
χ, dimensionless parameter $Q_1 l/\lambda\vartheta_1\delta_1$;
Δ, dimensionless fin thickness δ/δ_1.

Subscripts

1, fin base;
2, fin tip.