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TECHNICAL NOTES

Extended studies of spray cooling with Freon-113

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INTRODUCTION

A PREVIOUS paper by one of the authors [1] has presented spray cooling heat transfer data for horizontal sprays on vertical constant heat flux surfaces with subcooled Freon-113. Reference [1] has shown that the Weber number defined as the ratio of inertial force to surface tension force has a large effect on the spray cooling process. In the paper, a non-dimensional generalized correlation for heat flux was presented for droplet velocities from 5.4 to 28.0 m s^{-1} and droplet diameters from 210 to 980 μm. While the results of the previous paper were satisfactory, it was desirable to extend the range of applicability of the empirical correlation to include a wider range of droplet sizes and velocities. The present study furnishes this information.

EXPERIMENTAL APPARATUS AND PROCEDURE

The same experimental apparatus and procedure described in ref. [1] was used for this study primarily so that the results would be comparable. The apparatus consisted of a supply reservoir of Freon-113, heat exchanger, stainless steel chamber, centrifugal pump, vacuum pump, positive displacement flow meter, copper tubing, and a spray nozzle. Figure 1 illustrates the test section assembly for the experiments. The current carrying foil was constructed of 0.0101 cm (0.004 in.) thick stainless steel 302 shim stock with a test section area of 58.1 cm^2 (7.62 × 7.62 cm). Five chromel–constantan, 0.00127 cm (0.0005 in.) thick, cement-on, foil thermocouples instrumented the back surface of the test surface through the Teflon insulation. A random thermocouple pattern was used to provide maximum coverage of the test section. All of the nozzles used in this study were commercially available full cone type.

Before every experiment, the vacuum pump was utilized to evacuate air from the test chamber. The centrifugal pump was operated in conjunction with the heat exchanger to bring the Freon-113 down to the testing temperature. The flow rate was controlled by a throttling valve located prior to the nozzle. The test section surface temperature was varied by increasing the voltage input by means of a step-down transformer. All the experimental runs progressed from low power to high power. The heat flux was determined by measuring the voltage drop across the test section and the current along with the average temperature of the test surface.

EXPERIMENTAL RESULTS AND DISCUSSION

A series of experiments were performed with different nozzle diameters and droplet velocities. The liquid Freon-

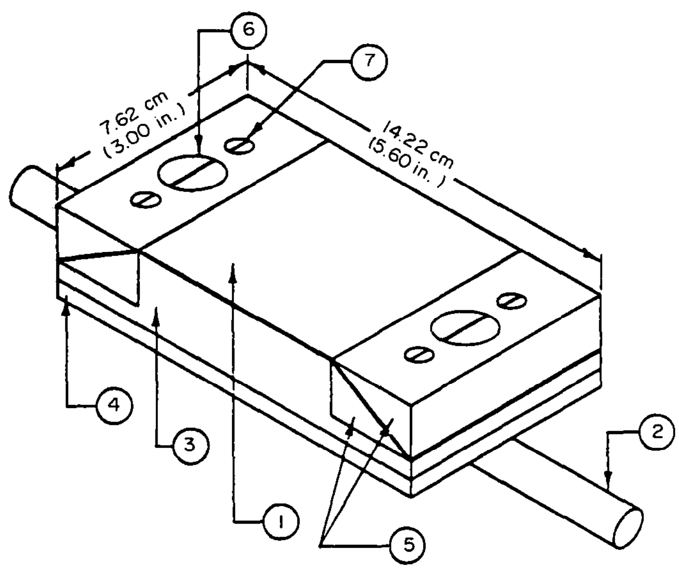

FIG. 1. Test section assembly: 1, stainless steel foil; 2, support shaft; 3, Teflon insulation; 4, support plate; 5, bus bar; 6, brass bolt; 7, flat head brass screw.

NOMENCLATURE

c specific heat [kJ kg^{-1} °C^{-1}]
D constant of equation (6)
d nozzle orifice diameter [m]
d_p droplet diameter [m]
h_{fg} enthalpy of vaporization [kJ kg^{-1}]
p pressure [Pa]
Δp pressure difference across nozzle [Pa]
q heat flux [W m^{-2}]
T temperature [°C]
ΔT temperature difference, $T_w - T_f$ [°C]
v droplet breakup velocity [m s^{-1}]
We Weber number, $\rho_l v^2 d_p \sigma^{-1}$
x distance from nozzle to heat surface [m].

Greek symbols
β nozzle spray angle [deg]
γ exponent in equation (6)
μ viscosity [kg m^{-1} s^{-1}]
ρ density [g cm^{-3}]
σ surface tension [dyn cm^{-1}].

Subscripts
c chamber
l liquid
v vapor
w test surface.

113 supplied to the nozzles was subcooled an average of 30°C. An arithmetic average of the thermocouple readings was used to determine the surface temperature for the heat transfer calculations by

$$T_w = \Sigma T(i)/n \quad i = 1\text{–}n. \tag{1}$$

The droplet breakup velocity, developed in ref. [1], is given by

$$v = (v_l^2 + 2\Delta p/\rho - 12\sigma/\rho d_p)^{0.5} \tag{2}$$

where v_l is the fluid velocity in the line upstream of the nozzle. Surface tension values were calculated from the relationship presented in the Dupont Technical Bulletin [2]:

$$\sigma = 3.14749(\rho_l^4 - \rho_v^4) \tag{3}$$

with the surface tension σ in dyn cm^{-1} and the density in g cm^{-3}. The droplet diameter is best described by an empirical correlation of mass median diameter originally developed by Longwell [3] and later simplified by Bonacina and Comini [4] into the form

$$d_p = 9.5d/(\Delta p^{0.37} \sin(\beta/2)) \tag{4}$$

where d_p has the same units as d. Freon-113 properties were taken from the Dupont Technical Bulletin [5] and the ASHRAE Thermodynamic Properties of Refrigerants tables [6]. Properties of the working fluid were evaluated at the film temperature which is defined as the average of the local surface and the spray temperature

$$T_f = (T_w + T_l)/2. \tag{5}$$

Data were obtained for various combinations of parameters such as droplet diameter and droplet breakup velocity. Table 1 is a summary of the experimental test variables.

Figure 2 presents the overall heat transfer as a function of the temperature difference ($T_w - T_l$). As in ref. [1], the cooling curves indicate that the heat transfer capability increased as the flow rate was increased.

A cross plot of ($q/c_l\Delta T$) vs Weber number showed the slope of the line on a log–log plot to be approximately 0.6. On the basis of these plots, the data were correlated by a non-dimensional equation of the form

$$qx/\mu_l h_{fg} = D(We)^{0.6}(c_l \Delta T/h_{fg})^{\gamma}. \tag{6}$$

Previous results presented in ref. [1] determined the constants of equation (6) to be $D = 10.55$ and $\gamma = 1.46$. Figure 3 compares the non-dimensional spray cooling heat flux data plotted with the line having these constants.

The data in Fig. 3 fall within the experimental uncertainty of equation (6); however, altering the constants D and γ produces a more accurate representative equation. Values of $D = 8.5$ and $\gamma = 1.54$ where determined to provide a more accurate correlation equation. Figure 4 presents the non-dimensional spray cooling heat flux data plotted with a line given by equation (6) with these constants. These correlation

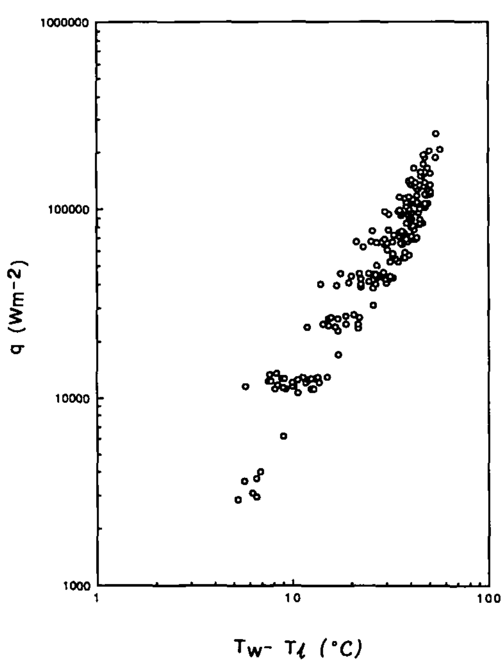

FIG. 2. Spray cooling data; overall heat transfer (q) vs temperature difference ($T_w - T_l$).

Table 1. Summary of experimental test variables

Variable	Range
Liquid flow rate	6.3–56.8 cm^3 s^{-1}
Nozzle diameter (max. free passage)	0.0635–0.127 cm
Heat source to nozzle distance	0.1842–0.1937 m
Droplet breakup velocity	11.4–28.5 m s^{-1}
Droplet diameter	96–343 μm
Surface temperature	20–80°C
Spray temperature	4–10°C
Heat flux	0.28–25.2 W cm^{-2}
Weber number ($We = \rho_l v^2 d_p \sigma_l^{-1}$)	2400–11 775
Reynolds number ($Re = \rho_l v d_p \mu_l^{-1}$)	4875–12 850
Prandtl number ($Pr = c_l \mu_l k^{-1}$)	6.0–10.7

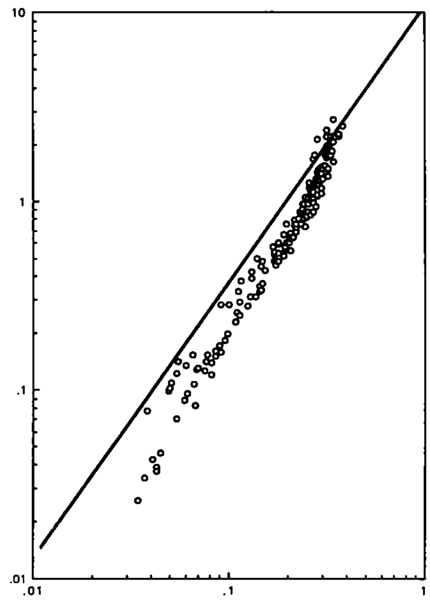

FIG. 3. Non-dimensional correlation of spray cooling data plotted with equation (6) using constants $D = 10.55$ and $\gamma = 1.46$.

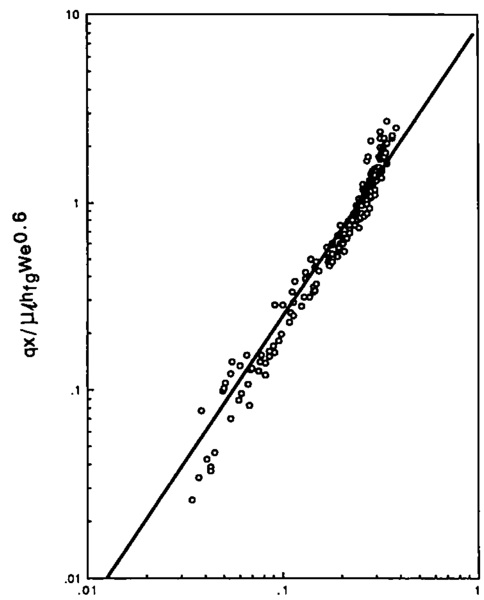

FIG. 4. Non-dimensional correlation of spray cooling data plotted with equation (6) using constants $D = 8.5$ and $\gamma = 1.54$.

constants lead to a non-dimensional equation for droplet velocities from 11.4 to 28.5 m s^{-1} and droplet diameters from 96 to 343 μm which is

$$qx/\mu_l h_{fg} = 8.5(We)^{0.6}(c_l\Delta T/h_{fg})^{1.54}. \tag{7}$$

It should be noted that the scatter of the data for lower values of the temperature difference $(T_w - T_l)$ result mainly from the experimental uncertainty in the measurements of the temperatures. Small uncertainties in T_w and T_l magnify the uncertainty in ΔT when ΔT is small.

The difference between the correlation constants in ref. [1] ($D = 10.55$ and $\gamma = 1.46$) and the newly determined constants ($D = 8.5$ and $\gamma = 1.54$) may reflect an inability of equations (2) and (4) to accurately represent average droplet diameters and breakup velocities over such a wide range of nozzle sizes and flow rates.

Finally, if all of the data presented in ref. [1] are included with the data presented herein, a new set of constants is determined to be $D = 9.5$ and $\gamma = 1.50$. A non-dimensional correlation equation for droplet velocities from 11.4 to 28.5 m s^{-1} and droplet diameters from 96 to 980 μm is determined to be

$$qx/\mu_l h_{fg} = 9.5(We)^{0.6}(c_l\Delta T/h_{fg})^{1.50}. \tag{8}$$

It would be useful to compare this correlation with the results of other investigations of spray cooling. Unfortunately, the present authors are unaware of other correlations for comparison. Plots of raw heat flux versus temperature difference are frequently given, but without sufficient information to calculate the dimensionless variables in equation (8). Previous studies by one of the authors [7] for individual droplet heat transfer rates have shown a dependence of heat transfer on the Weber number to the 0.34 power, but would not be expected to apply to the more complicated flow system involved in spray cooling.

CONCLUSIONS

All of the data of ref. [1] and the present study may be expressed in the form of equation (8) for droplet diameters ranging from 96 to 980 μm and droplet breakup velocities from 5.4 to 28.5 m s^{-1}.

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