The Heat-Transfer Characteristics of a Small Droplet Impinging upon a Hot Surface*

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This paper presents the characteristics of the breakup behavior and heat transfer of a single small droplet impinging upon a hot surface. Using the vibratory method, uniform-sized droplets of water were formed at constant intervals at room temperature and under atmospheric pressure. The breakup behavior relates well with the particular surface temperature, that is, the maximum evaporation point and the Leidenfrost point. The characteristics of heat transfer, i.e., the heat-transfer effectiveness and the heat-transfer coefficient, are also discussed.

Key Words: Phase Change, Unsteady Flow, Heat Transfer, Heat Engine, Liquid Droplet, Spray Cooling, Impingement, Thermal Conduction

1. Introduction

Large droplets break up into small droplets when impinging on a hot surface due to dynamic unbalance and vapor formed at the liquid-solid interface. They then disperse and evaporate. This phenomena can be observed on a hot iron plate which is cooled by spray; a technique so-called "spray cooling" and also on the surface of an evaporator which is installed within a combustor. The effect of surface temperature on heat flux and the heat-transfer coefficient was studied for spray-cooling. Especially in the film boiling region, these heat-transfer characteristics were found to be related to the mean droplet diameter in spray, the impinging velocity and the spray flow rate

in past studies^{(1)~(5)}. Moreover, Onaka and Fukusako reported on the modeling of liquid film formation on a hot surface⁽⁶⁾.

Seki⁽⁷⁾, et al. McGinnis⁽⁸⁾ and Holman and Ueda⁽⁹⁾ et al. presented the heat-transfer behavior of a single droplet impinging on a hot surface. Pedersen⁽¹⁰⁾ and Wakunaga and Shoji⁽¹¹⁾ investigated the behavior of heat transfer of a uniform-sized droplet array with droplet diameters of hundreds of micrometers which was impinged on a hot surface at relatively high speeds. The authors have previously reported on the deformation process, breakup behavior and dispersion of breakup droplets of a uniform-sized droplet array impinging upon a hot surface^{(12)~(14)}.

It is the object of this spray-cooling study to clarify the effects of the process of vapor formation in a liquid-solid interface on the breakup behavior and the dispersion of droplets which are impinged on a hot surface. A uniform-sized droplet array with droplets diameters of hundreds of micrometers was impinged on a hot surface. The heat-transfer effectiveness and the heat-transfer coefficient were measured using a transient technique. The relationship between the characteristics of the heat transfer and the behavior

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of deformation and dispersion of the film flow are also discussed.

2. Experimental Apparatus Procedure and Conditions

2.1 Measurement of heat-transfer effectiveness ϵ

Figure 1 shows a schematic diagram of the experimental apparatus used for the measurement of ϵ . Transverse vibrations were applied to the nozzle tip ① (200 μm inside diameter) by a signal generator ② and speaker ③. In this manner, a droplet array having uniform droplet diameters and an equi-interspace was formed. For the case of impingement of the droplet array upon a hot surface ④, the divided frequency N was calculated in the following manner: an arbitrary number n of a division of frequency was set at the frequency divider ⑤. Then, by dividing the vibration of the oscillation f of the vibrator by n , N was determined. Subsequently, a switching circuit ⑥ was operated with N , and the charging electrode ⑦ became a ground potential. Then N uncharged droplets per unit time fell straight toward the hot surface at a normal angle without being effected by a

deflection plate ⑧. Thus, N is the droplet impinging frequency.

The hot surface (8 mm in diameter, 5 mm in thickness) was made of copper, and after being buffed, a hard chrome plate was applied (50 μm in thickness). This surface was heated up to an appointed temperature in a furnace ⑨ and was cooled by air and the impinging droplet array. The record of the temperatures of the surface center was stored in a waveform memory ⑩ as the output of a C-A insulated thermocouple (0.2 mm in wire diameter) ⑪. The heat-transfer effectiveness was then calculated using a microcomputer ⑫.

An example of such a record of temperature, i. e., a cooling curve, is illustrated in Fig. 2. The value ϵ is defined by the following equation:

$$\epsilon = \frac{mc(\Delta T/\Delta t)_{\text{with}} - (\Delta T/\Delta t)_{\text{without}}}{(\pi/6) \cdot D_i^3 \cdot \rho_i \{h_{fg} + c_i(T_{\text{sat}} - T_i)\} \cdot N} \quad (1)$$

where D_i is the diameter, ρ_i the density, h_{fg} the latent heat, C_i the specific heat, T_{sat} the saturate temperature and T_i is the initial temperature of droplet, and mc the heat capacity of the surface.

It is necessary to use a surface with a small heat capacity and a large thermal conductivity when ϵ is calculated by eq. (1). The surface was obtained taking into account its Boit modulus B_i of $\alpha \cdot l/\lambda$ and Fourier modulus F_0 of $\alpha \cdot t/l^2$. In these relations, α is the heat-transfer coefficient, l the representative diameter, λ the thermal conductivity, a the thermal diffusivity and t the time.

2.2 Measurement of heat flux q_d and heat-transfer coefficient α_d

Figure 3 illustrates the experimental apparatus used in the measurements of q_d and α_d . Hot surfaces ⑬ and ⑭ were made of copper, and the surface conditions were the same as described previously. The surface was inclined at an angle of 30° to prevent interference between the remaining liquid film on the surface and the impinging droplets⁽¹⁵⁾. The droplet array impinged upon this surface at a normal angle. The copper was heated to a certain temperature level

Fig. 1 Schematic diagram of experimental apparatus for measurement of ϵ

Fig. 2 Cooling curve

Fig. 3 Schematic diagram of experimental apparatus for measurement of the heat flux and heat-transfer coefficient

using a heater ⑮ and a voltage regulator ⑯. Thereafter, it was cooled by air and the droplet array. The temperature drop was measured by two insulated C-A thermocouples (0.2 mm in diameter) ⑰, and then q_d and α_d computed by a microcomputer ⑱ using the information from the amplifier ⑲ and waveform memory ⑳.

One-dimensional unsteady heat conduction was assumed inside of the surface, and is shown in Fig. 4 with a schematic diagram of differential calculus. In this method, Point 1 relates to the temperature and does not lie on the surface. As a consequence, there is heat transfer from the surface to this point. Temperature θ'_1 at Point 1 after time Δt_s is expressed by the following equation:

$$\theta' = F_0 \left\{ \frac{2\delta_1}{\delta_1 + \delta_2} \theta_2 \frac{2B_i}{2 + B_i} \theta_f + \theta_1 \left(\frac{1}{F_0} - \frac{2B_i}{2 + B_i} - \frac{2\delta_1}{\delta_1 + \delta_2} \right) \right\}, \quad (2)$$

where θ_f is the temperature of the liquid film. In the above equation, B_i is calculated at the time interval of Δt_s using the dichotomizing search method, and α_d and q_d given by $\alpha_d \cdot (T_w - T_i)$ are thus obtained where T_w is the surface temperature. The sufficient condition for convergence in eq. (2) is then shown as,

$$F_0 \leq (2 + B_i)(\delta_1 + \delta_2) / \{2B_i(\delta_1 + \delta_2) + (2 + B_i)\}. \quad (3)$$

The convection heat transfer from the surface to the surrounding air was measured before the experiments, and this value was subtracted in the calculations of q_d and α_d .

2.3 Experimental conditions

Experiments were conducted at room temperature and atmospheric pressure. Droplets were made from distilled water in the temperature range from 15 to 20°C. The droplet diameter D_i ranged from 300 to 600 μm , and the impinging velocity of the droplet V_i was set at 2.5~7.0 m/s. Droplet impingement frequency N was varied in the range of 100~1000 s^{-1} and the surface temperature T_w was increased from 100 to

Fig. 4 Schematic diagram of differential calculus in the case of one-dimensional unsteady heat conduction

450°C.

Deformation and breakup behavior of a single impinging droplet within the array was observed by continuous micrographs taken with a high-speed drum camera (10000 fps).

3. Deformation and Breakup Behavior of an Impinging Droplet

The model proposed by Moriyama and Araki⁽¹⁶⁾ was applied to the radial spread of the film flow on the surface after the impingement of a droplet. Namely, the completion diameter D_{fm} and the completion thickness B_{fc} of the radial spread were calculated in this model. Figure 5 shows the changes in D_{fm} and B_{fc} of this radial spread due to the surface temperature T_w under the condition of the Weber number W_e from 140~180. W_e is equal to $D_i \cdot V_i / V_i$, where V_i is the surface tension. Since D_{fm} becomes maximum at the surface temperature of 325°C, it is found that energy loss caused by the resistance of the solid-liquid interface during the radial spread of the film is minimum at this temperature. Also, D_{fm} has an almost constant small value at temperatures of less than 175°C. The phenomena under this temperature range correspond to nucleate boiling or film boiling due to large contact resistance.

Figure 6 indicates the relation between the residence time τ of a droplet on the surface and the surface temperature T_w . The parameter is W_e . The first-order vibration period of a free oscillating droplet τ_r ⁽¹⁷⁾ and the completion time t_c of the formation of the film⁽¹⁴⁾ are also shown in this figure. τ is markedly smaller than τ_r and is slightly larger than t_c . τ has a minimum value at T_w of 175°C and becomes almost constant in the range over 250°C for the range of Weber numbers investigated.

The residence time and the apparent contact area of the film on the surface are important to clarify the heat-transfer process. In this study, an average con-

Fig. 5 Trends of completion diameter D_{fm} and completion thickness B_{fc} of film as a function of surface temperature T_w

tact area $\bar{A}$ during the residence time τ was estimated as shown in the following equation:

$$\bar{A} = \int_0^\tau \dot{A} dt / \tau. \quad (4)$$

Figure 7 indicates the relation between $\bar{A}$ and T_w . The tendency of $\bar{A}$ is almost the same as that of D_{fm} and τ . $\bar{A}$ becomes minimum at $T_w = 175^\circ\text{C}$ and maximum at $T_w = 325 \sim 350^\circ\text{C}$.

Figure 8 shows the schematic breakup form of an impinging droplet observed in micrographs at each surface temperature under the conditions of $W_e \geq 120$. The relation between the Sauter mean diameter d_{32} of breakup droplets caused by the destruction of the film and the T_w is shown in Fig. 9⁽¹⁴⁾. The detailed characteristics of the breakup form and the behavior of breakup droplets were reported in a previous study⁽¹²⁾. From these figures are derived the conditions of the liquid-solid interface corresponding to T_w of 125°C , $150 \sim 225^\circ\text{C}$ and $250 \sim 400^\circ\text{C}$, namely, the states of

Fig. 6 Relation between residence time τ and surface temperature T_w

Fig. 7 Relation between average contact area $\bar{A}$ and surface temperature T_w

nucleate boiling, transition boiling and film boiling, respectively.

4. Heat-transfer Characteristics of an Impinging Droplet

Figure 10 illustrates the experimental results of the evaporating lifetime of a water droplet 2 mm in diameter, which is in a the spheroidal state on the hot surface. The shape of the surface has the configuration shown in Fig. 10 to prevent evaporating droplets from falling from it. The surface is made of copper and its surface treatment is the same as de-

Fig. 8 Schematic breakup form of an impinging droplet

Fig. 9 Relation between Sauter mean diameter d_{32} of breakup droplets and surface temperature T_w ($W_e = 130 \sim 180$)

Fig. 10 Relation between the lifetime of a droplet and the surface temperature T_w ($D_i = 2 \text{ mm}$, $V_i = 0 \text{ m/s}$)

scribed previously. The maximum evaporating point T_M and the Leidenfrost point T_L exist at the surface temperatures of 145°C and 230°C, respectively.

4.1 Heat-transfer effectiveness ε

Figures 11 and 12 respectively show the heat transfer Q and the heat-transfer effectiveness ε as functions of the surface temperature T_w and droplet impinging frequency N . Q and ε increase when temperature T_w becomes larger in the region of nucleate boiling, and then decrease in the transition boiling region when T_w is greater than T_M . Thereafter, Q increases slightly and is approximately constant in the region of the film boiling above the Leidenfrost point T_L . Accordingly, a similar correlation is confirmed between these characteristics of Q and ε and the usual boiling curve of a stationary droplet present on the hot surface. Though Q is rightly increasing with an increase in N at each T_w , ε decreases as N increases. This is particularly evident in the temperature region less than T_M . This causes the interference between the remaining liquid film formed on the surface and a succeeding impinging droplet. The effect of this interference is great in the range below T_M due to the large residence time shown in Fig. 6⁽¹⁵⁾. T_M lies in the range from 130~150°C and T_L is nearly equal to 240°C in Figs. 11 and 12. The results correspond well with those

Fig. 11 Relation between heat transfer Q and surface temperature T_w ($D_i=482\ \mu\text{m}$, $V_i=5.0\ \text{m/s}$, $W_e=164$)

Fig. 12 Relation between heat-transfer effectiveness ε and surface temperature T_w ($D_i=482\ \mu\text{m}$, $V_i=5.0\ \text{m/s}$, $W_e=164$)

of the evaporating lifetime in Fig. 10. Therefore, each evaporating type in Fig. 8 coincides with the so-called boiling phenomena; namely, the RB -type is nucleate boiling, the two types of B and N are transition boiling and the N -type is film boiling.

Figure 13 shows the changes in T_M and the heat-transfer effectiveness ε_M at T_M as a function of N . The relation between the Leidenfrost point T_L and N , and also that between the heat-transfer effectiveness ε_L at T_L and N are shown in Fig. 14. As N increases, ε_M has a lower value because of the increase in the interference between the remaining film and the impinging droplet. Then, T_M rises a little with the increase in N since the temperature which brings about the interference decreases. Thereafter, ε_L decreases slightly and T_L becomes constant at 240°C with increasing N . Consequently, these results suggest that there is little evidence of interference near T_L due to the short residence of the film.

4.2 Heat-transfer coefficients α_e , α_d and α_s

In the measurement of heat-transfer effectiveness, heat transfer per one droplet $\dot{Q}$ is given by Q/N . Therefore, the mean heat-transfer coefficient α_e

Fig. 13 Maximum evaporating point T_M and heat-transfer effectiveness ε_M at T_M as a function of droplet impinging frequency N ($D_i=482\ \mu\text{m}$, $V_i=5.0\ \text{m/s}$, $W_e=164$)

Fig. 14 Leidenfrost temperature T_L and heat-transfer effectiveness ε_L at T_L as a function of droplet impinging frequency N ($D_i=482\ \mu\text{m}$, $V_i=5.0\ \text{m/s}$, $W_e=164$)

during the residence time τ is assessed approximately in terms of the average contact area $\bar{A}$;

$$\alpha_e = \dot{Q} / (\bar{A} \cdot \tau (T_w - T_i)) \quad (5)$$

Figure 15 displays the dependence of α_e on the superheating degree of ΔT_{sat} which is given by $(T_w - T_{\text{sat}})$. The effective heat-transfer area is larger than $\bar{A}$ as a result of the secondary impingement of breakup droplets and the film grown on the surface at T_w below 150°C . In consequence, α_e is slightly larger than that of the previous study.

Figure 16 shows the relation between the heat-transfer coefficient α_d calculated by Eq. (2) and ΔT_{sat} for a surface diameter D_s relating to the heat transfer on the surface. In this case D_s is equal to 8 mm. The maximum value of α_d occurs in the range of ΔT_{sat} from 10 to 30°C ; this range is slightly lower than for the results of T_M in Figs. 11 and 12. The Leidenfrost

point T_L exists in the range from 250°C to 300°C in the case of $N=100\text{s}^{-1}$, and has a tendency to rise slightly with an increase in N . This result coincides with the relation between the Leidenfrost point and the spray flow rate in normal spray-cooling⁽⁴⁾. The increase in N causes an increase in the flow rate to the surface, and consequently, α_d becomes large. Toda⁽¹⁸⁾ conducted an experiment with a stationary method and Shoji⁽¹⁹⁾ carried out that with a transient technique to represent the spray-cooling of a horizontal surface. The results from these reports plotted in Fig. 16 are similar to those in this study despite the differences in the experimental conditions.

Figure 17 shows the comparison of the heat-transfer coefficient α_d with that of α_s obtained by Shoji⁽¹⁹⁾ for the conditions of N equal to 890 and 201s^{-1} . α_d and α_s are almost identical for all ranges of ΔT_{sat} .

Fig. 15 Relation between mean heat-transfer coefficient α_e and superheating degree ΔT_{sat} of the surface

Fig. 17 Comparison of heat-transfer coefficient α_d from Eq. (2) with that of α_s from the equation given by Shoji

Fig. 16 Relation between heat-transfer coefficient α_d from Eq. (2) and superheating degree ΔT_{sat} of the surface

Fig. 18 Relation between heat-transfer coefficient α_d from Eq. (2) and superheating degree ΔT_{sat} of the surface

The dependence of α_d on D_s is illustrated in Fig. 18. α_d indicates a lower value in the case of $D_s=15$ mm than one in the case of $D_s=8$ mm. The heat-transfer coefficient obtained here was calculated assuming the application one-dimensional unsteady heat conduction. The radial heat conduction was not a factor in this case, and consequently, there was a difference in the internal temperature gradient between the two cases of D_s . For a surface of $D_s=15$ mm, α_d is lower because the temperature drop at the impinging point of the droplet array becomes smaller due to the bigger heat capacity and the heat conduction from the radial direction of the surface.

5. Conclusions

The following conclusions can be drawn from the experiments and calculations presented here.

(1) The surface temperature ranges of $T_w \leq 125^\circ\text{C}$, $T_w = 150 \sim 225^\circ\text{C}$ and $T_w \geq 250^\circ\text{C}$ correspond to nucleate boiling, transition boiling and film boiling, respectively. The relation between the breakup behavior of the impinging droplet and the boiling phenomena at the liquid-solid interface is clarified.

(2) In the temperature range below 150°C , the heat-transfer effectiveness decreases as the droplet impingement frequency increases due to the interference between the remaining liquid film and the impinging droplet. The effectiveness is kept almost constant in the film boiling region.

(3) The heat-transfer coefficient is calculated by applying differential calculus for the equation of one-dimensional unsteady heat conduction. The effects of the differences between differential calculus methods and of the variation of the surface diameter relating to heat transfer are discussed. The heat-transfer coefficient becomes large with increasing impingement frequency.

References

- (1) Tanaka, S., Mitsubishi Jyuko Giho, (in Japanese),

- Vol. 12, No. 2 (1975), p. 192.
 (2) Yanagi, K., Setoguchi, K. and Hayashi, K., Mitsubishi Jyuko Giho, (in Japanese), Vol. 9, No. 6 (1972-11), p. 792.
 (3) Shoji, M., Proc. 15th Jpn. Heat Transf. Symp., (in Japanese), (1978-5), p. 187.
 (4) Ishigai, S., Proc. 16th Jpn. Heat Transf. Symp., (in Japanese), (1979-5), p. 316.
 (5) Yanagi, K., Proc. 9th Jpn. Heat Transf. Symp., (in Japanese), (1972-5), p. 463.
 (6) Onaka, I. and Fukusako, T., Proc. 12th Jpn. Heat Transf. Symp., (in Japanese), (1975-5), p. 217.
 (7) Seki, M., Kawamura, H. and Sanogawa, Y., Proc. 9th Jpn. Heat Transf. Symp., (in Japanese), (1972-5), p. 459.
 (8) McGinnis, F. K. and Holman, J. P., Int. J. Heat Mass Transf., 12 (1969), p. 95.
 (9) Ueda, T., Enomoto, T. and Kanetsuki, M., Bull. JSME, Vol. 22, No. 167 (1979), p. 724.
 (10) Pedersen, C. O., Int. J. Heat Mass Transf. 13 (1970), p. 369.
 (11) Wakunaga, T. and Shoji, M., Proc. 18th Jpn. Heat Transf. Symp., (in Japanese), (1981-5), p. 394.
 (12) Senda, J., Sato, Y. and Takeuchi, K., Proc. of the 2nd Int. Conf. on Liquid Atomization and Spray Systems, (1982-6), p. 397.
 (13) Senda, J., Sato, Y. and Takeuchi, K., Internal Combustion Engine, (in Japanese), Vol. 21, No. 268 (1982-9), p. 9.
 (14) Senda, J., Yamada, K., Takeuchi, K. and Miki, H., Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 52, No. 481 (1981-9), p. 3372.
 (15) Senda, J., Yamada, K. and Takeuchi, K., ASME-JSME Ther. Eng. Joint Conf., (1983-3), p. 165.
 (16) Moriyama, A. and Araki, K., Tetsu-to-Hagane, (in Japanese), 64 (1978), p. 533.
 (17) Rayleigh, L., Proc. Royal Soc., 29 (1879), p. 71.
 (18) Toda, S., Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 38, No. 307 (1972), p. 581.
 (19) Shoji, M., Proc. 12th Jpn. Heat Transf. Symp., (in Japanese), (1975), p. 213.