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20. ABSTRACT (Continue on reverse side if necessary and identify by block number) An experimental study of the heat transfer characteristics of a bubble-driven water jet on an ice surface was conducted. Two Lucite columns, one 1.829 m high and 0.286 m in diameter and the other 1.219 m high and 0.140 m in diameter were used. Water levels were maintained at 0.762 and 1.524 m in the large column and 0.840 m in the small column. Hypodermic needles with openings of 0.152, 0.406 and 0.838 mm were used for bubble formation. The air flow rate was varied from 7.39×10^{-8} to 9.91×10^{-7} m ³ /s. In all, 171 experimental runs were conducted. The results can be correlated by			

20. Abstract (cont'd)

$$Nu = 0.1735 (Re_i)^{0.848}$$

with a correlation coefficient of 0.84, in which Nu is defined in terms of average heat transfer coefficient h , sample diameter D_s , and thermal conductivity of water k_w . Re_i is defined in terms of diameter of the impinging water jet at the ice surface, D_i , the centerline arrival water velocity v_c and the kinematic viscosity of water ν_w .

PREFACE

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NOMENCLATURE

A	Sample cross-sectional area
C	Rate of spread of velocity profile
c	Heat capacity
D	Column diameter
D_i	Diameter of area of impingement
D_s	Sample diameter or diameter of heat transfer area
d_b	Equivalent bubble diameter
d_n	Nozzle diameter
f_b	Bubble frequency
G	Force per unit height imposed in liquid
h	Average heat transfer coefficient over heat transfer surface
h_i	Average heat transfer coefficient based on area corresponding to area of impingement
k	Thermal conductivity
L	Water depth, axial distance from nozzle tip to sample surface
L_i	Latent heat of fusion
L^*	Defined as $L + P_{atm}/\gamma_w$
M_s	Sample weight at completion of experiment
Nu	Nusselt number defined as hD_s/k_w
Nu_i	Nusselt number defined as $h_i D_i/k_w$
Nu_n	Nusselt number defined as $h d_n/k_w$
P_{atm}	Atmospheric pressure
Pr	Prandtl number defined as $c\mu/k$
Q_m	Heat exchange associated with melting
Q_s	Sensible heat change of remaining sample
R	Column radius, Rotameter setting
r	Coordinate in radial direction
Re_a	Reynolds number defined as $D_s v_c/v_w$
Re_i	Reynolds number defined as $D_i v_c/v_w$
Re_n	Reynolds number defined as $d_n v_a/v_a$
T_b	Bulk water temperature
T_f	Freezing temperature
T_i	Initial ice temperature

ΔT	Defined as $T_b - T_f$
ΔT_b	Temperature differences near melting surface at start and completion of experiment
V_a	Volumetric flow rate
v_a	Air velocity at nozzle exit
v_b	Mean bubble stream velocity
v_c	Vertical centerline water velocity
v_w	Vertical water velocity
W_i	Weight of ice melted
y	Coordinate in vertical direction
γ_w	Specific weight
μ	Viscosity
ν	Kinematic viscosity
π	Constant
ρ	Density
θ	Time

Subscripts

a	Air
b	Bubble, bottom
c	Center
f	Freezing
i	Ice, impingement
n	Nozzle
o	Origin
s	Sensible
t	Top
w	Water

HEAT TRANSFER CHARACTERISTICS OF A BUBBLE-INDUCED WATER JET IMPINGING ON AN ICE SURFACE

by

Yin-Chao Yen

INTRODUCTION

Winter, with its freezing temperatures, has posed many problems in cold regions for navigation, and for maintenance of structures in water. During long, cold winters many lakes, harbors, ports and rivers become covered with ice, which greatly shortens the navigation period. In attempts to solve this problem, air-bubbler devices have been developed which operate on the principle that warm subsurface water can be brought to the surface by a stream of air bubbles rising through the otherwise quiescent bodies of water. The first air bubbler system was installed in 1917 at the Keokuk Dam,⁴ and since that time a number of similar usages have been reported;¹⁻³ a general discussion on the use of air bubble devices as ice control devices can be found in a CRREL Monograph by Michel.⁹

In the natural environment, the necessary condition for successful operation of an air-bubbler system is the presence of the heat reservoir of the water body itself. The heat stored must be at least equal to or greater than the net heat loss at the water surface resulting from a combination of exchange processes, i.e. evaporation, condensation, and net short- and long-wave radiative and convective heat transfer.

It is fortunate that fresh water has its maximum density at about 4°C, since this governs the overturning process and circulation pattern during the cooling period in late fall and early winter. As a result, warmer and heavier water resides near the bottom of the water body. Because of this density stratification, conductive heat transfer, which is far less effective than convective transfer, dominates the process. Introducing air bubbles at the bottom of the water body produces extensive water mixing, and the bubbles carry a certain amount of warmer water upward to the surface, to counterbalance heat loss to the atmosphere.

Many studies have been reported in which fluid jets were used to enhance heat and mass transfer rates.^{5-6, 7, 11, 12} Most of these studies were with single and multiple air jets emerging from slots, holes and nozzles parallel to or impinging normally to the heat transfer surface. The results are usually expressed in terms of a Nusselt number as functions of a nozzle to plate distance, and a Reynolds number based primarily on the slot, hole or nozzle diameter. Sitharamayya and Raju¹⁰ reported the only experimental measurements of heat transfer between a submerged water jet and a normal plate. The average heat transfer coefficient was found to be independent of nozzle to plate distance for $L/d_n < 7$ (d_n is the nozzle diameter).

Little work has been reported regarding the hydrodynamics of the bubble-induced water jet. Based on laboratory experiments using a water tank 2.438 m square and 1.829 m deep, and some elementary analytical considerations, Baines and Hamilton² reported that the water flow characteristics of the bubble-driven jet pattern consist of a vertical jet rising from the air source and a thin, horizontal, radially spreading jet along the surface. The momentum of the vertical jet increases with height from the source. They also reported that the flow pattern is independent of the size of the nozzle opening in the range of 0.508 mm or larger, but is a function of the air discharge rate. The ratio of water entrainment to air discharge rate was found to increase with depth and decrease with increasing air discharge. In a recent study, using columns 0.914 m in

Figure 1. Schematic of the experimental set-up.

height and 50.04 mm, 76.2 mm and 145 mm in diameter, Crabtree and Bridgwater³ examined, analytically and experimentally, the liquid motion induced by a chain of bubbles rising through a viscous liquid. The velocity profile and extra pressure drop induced in the liquid by gas bubbling from an opening at the center of the column base were found to be in excellent agreement with a model of the flow which supposes the gas to be equivalent to an upward vertical line of forces along the axis of the cylinder. The most comprehensive work on the analysis of the flow induced by air-bubble systems was reported by Kobus.⁸ The analytical treatment considering the momentum-flux increase due to the buoyancy of the air, together with experimental information about the spread of the velocity profiles and the mean rising speed of the air bubble stream, lead to a complete description of the flow field.

Most recently Ashton¹ examined the bubbler system analytically in great detail with the ultimate goal of developing a procedure for predicting its effectiveness in suppressing ice formation under various field conditions.

To the best of the author's knowledge, no prior laboratory work has been done regarding the heat transfer characteristics of a bubble-driven jet impinging on an ice surface. This work is also considered to be useful in destratifying lakes and artificial water reservoirs for control and upgrading of our water resources. The present study investigated the problem experimentally. The effects of column height, diameter and air bubbling rate on the rate of heat transfer were determined.

EXPERIMENTAL APPARATUS AND PROCEDURE

Figure 1 shows the experimental set-up, which consisted essentially of a flowmeter, a heating device and a water column. Building air was first filtered through a Lucite cylinder filled with glass wool, and then passed through a pressure regulator set at $3.44 \times 10^4 \text{ N/m}^2$. A dial thermometer was also attached at the entrance of the flowmeter. From the flowmeter, the air was directed through a copper tube to a check valve rated at $1.375 \times 10^4 \text{ N/m}^2$ before being expelled at the nozzle. The nozzle was a Luer-lok hypodermic needle attached to a connector which was welded onto the end of a copper tube. The nozzle was inserted through the center of the bottom Lucite plate with the tip of the needle about 0.152 m above the plate. Two Lucite columns were used, one 1.829 m high and 0.286 m in diameter and the other 1.219 m high and 0.140 m in diameter. A Cole-Parmer thermistor temperature controller maintained the water temperature at 4°C in the area where the bubbles formed. For the small column a 310-watt Hotwatt cartridge heating unit and a thermistor probe were immersed horizontally in the water column just below the nozzle tip. For the large column, three 450-W heating units and a thermistor were positioned vertically through the bottom

plate projecting into the column. Eighteen thermocouples were installed at various heights, extending various distances inward from the column wall to measure the water temperature distribution within the column. Step-like notches 6.35 mm deep were cut into the top of the column to support the ice and maintain it at a proper immersion level during the experiment. Two drain holes 12.7 mm in diameter were located at the desired water level to provide for overflow resulting from the melting. The water circulation in the column resulted purely from bubble motion. The whole column was insulated to minimize the effect of room temperature variation.

Test specimens

The test specimens were cylindrical ice blocks ranging from 0.102 to 0.165 m high and 0.254 to 0.267 m in diameter for the large column; and 0.130 m high and 0.114 m in diameter for the small column. These samples were made from distilled water which had been frozen in an open Lucite cylinder with a copper bottom. Tee-type Lucite supporting devices were partially immersed in the center of the containers during the freezing period and were used in transporting the ice blocks and supporting them at the top of the column during the experiments. The ice was frozen from the base upwards by using cold plates with glycol at -51°C as the circulating medium. Ice was formed in this manner to reduce the amount of air bubble entrainment in the ice specimens. When completely frozen, the sample-making device was placed in a warm room in an inverted position until the sample slid out of the container. The ice sample was then placed in a coldroom maintained at -10°C to condition it to the same initial temperature before the initiation of an experiment. Copper-constantan thermocouples were used to measure the temperature of the ice sample during each experiment. They were imbedded in holes drilled in the samples and frozen into place, one on top and the others along the sides. The first was located 38.1 mm from the melting surface (bottom). The second and third were 63.5 and 88.9 mm high initially.

Prior to any experiment, the water level in the column was restored by pouring in an appropriate amount of ice cubes. At the same time, the air flow rate was fixed to a specific Rotameter setting. The experimental preparation was carried out until all the ice cubes melted. Then the ice sample, with its dimensions and weight recorded, was lowered into the column and adjusted so that the bottom of the sample was about 6.35 mm below the water level. The time of initiation of the experiment was recorded.

Bubble frequency and velocity

Bubble frequency and velocity were determined by filming the rising bubbles at various flow rates. A 10.0×10.0 mm grid was placed behind the column. A tensor lamp was projected on the grid and a floodlight was placed at the top of the column to provide adequate lighting. A Kodak Cine-1600 camera on a tripod was used to film the bubbles. The shutter was set at $\frac{1}{2}$ open, the lens opening was set at $f/2.8$, and the film speed was at 64 frames/second. The film used was Eastman Plus-X Negative.

ANALYSIS OF EXPERIMENTAL RESULTS

Three nozzle sizes, 0.152, 0.406 and 0.838 mm in diameter (d_n), were used in this study. The air flow rates covered the range from 7.39×10^{-8} to $9.91 \times 10^{-7} \text{ m}^3/\text{s}$. During the experiments, the water temperature throughout the column was found to be almost uniform at 4°C with a decrease of a fraction of a degree from the center toward the wall. The coldroom was maintained at 0.0 to 0.2°C . The rising air bubbles frequently struck the center of the cylindrical ice block and moved radially toward the periphery. Frequently a large bubble formed and covered much of the melting surface, thus retarding heat transfer. This was because the bottom ice surface became slightly concave due to faster melting in the center impinging region. The remedy was to tip the ice sample a little at regular intervals to facilitate the removal of these large bubbles.

Figure 2 shows the variation of bubble frequency f_b with volumetric air discharge rate V_a . It can be observed that f_b values at low V_a are somewhat erratic regardless of the column height L , sample diameter

Figure 2. Bubble frequency f_b as a function of V_a .

D_s and nozzle diameter d_n . However, in general the f_b values increase with decreasing values of d_n and can be satisfactorily expressed in a form of exponential function [$f_b = A \exp(BV_a)$] with more or less identical decay constant B (with the exception of $d_n = 0.152$ mm and $D_s = 0.260$ m). It can also be noted that there is no significant variation in f_b values for the large column with different values of L . Furthermore the f_b values in the small column were found to depend on d_n : for $d_n = 0.838$ mm, f_b is lower than that from the large column; on the other hand for $d_n = 0.406$ mm, the opposite is true; for $d_n = 0.152$ mm, f_b is found to be dependent on V_a and is lower and higher than that from the large column in the low and high V_a range respectively.

In the case of bubble velocity v_b , in general it increases with V_a except that for $d_n = 0.152$ mm v_b decreases slightly with increasing V_a . Furthermore, it demonstrates no significant variation with the different D_s and L (see Fig. 5). The v_b values have a wider spread for $d_n = 0.838$ mm than for $d_n = 0.406$ mm (see Fig. 3 and 4). However, in both cases v_b values are lower in the small column. For the large column, the shorter column has higher v_b values. The bubble size is more or less uniform for a given d_n and a given volumetric air flow rate V_a . However, the bubbles are not exactly spherical and mostly spherical-capped. To a large extent, the bubble shape depends on V_a .

The small column experiments were conducted for a duration of 1 hour to eliminate or reduce the extent of an uneven melting surface due to excessive melting in the jet impinging region. For the large column, a period of 1½ hours or longer was usually used. To ensure that the experimental data were representative of steady state conditions, a series of experiments were conducted for a specific air flow rate for various durations. The weight loss of the ice sample (or the extent of melting) was found to vary linearly with the duration of the experiment (see Fig. 6).

Figure 3. Bubble velocity v_b as a function of V_a ($d_n = 0.838$ mm).

Figure 4. Bubble velocity v_b as a function of V_a ($d_n = 0.406$ mm).

Figure 5. Bubble velocity v_b as a function of V_a ($d_n = 0.152$ mm).

6 HEAT TRANSFER CHARACTERISTICS OF A BUBBLE-INDUCED WATER JET IMPINGING ON ICE

Figure 6. Ice melted as a function of θ with V_a as the parameter.

The heat transfer coefficient between the melting ice and the bubbling water was evaluated in the following manner. The ice sample was weighed before and after the experiment, and the weight loss was used to determine the total heat used to melt the ice. The temperature change of the remaining ice sample was also recorded at various distances from the melting surface during the experiment.

The temperature change was a result of the combined effects of heat conducted through the melting surface, and radiation exchange and natural convection between the ice sample and the surroundings. The latter effects were determined by suspending the ice sample in the column but not in contact with the water, and recording the temperature rise of the ice for a given experimental duration.

To compute the total upward heat transfer, the enthalpy change of the remaining ice sample and the amount of heat associated with phase changes were evaluated. The net enthalpy change can be approximated by

$$Q_s = M_s c_i (\Delta T_m - \Delta T_{cr}) \quad (1)$$

where M_s is the weight of the ice sample at the completion of the experiment, c_i is the heat capacity of ice, ΔT_m is defined as $(\Delta T_b + \Delta T_t)/2$ where ΔT_b and ΔT_t are the differences between the initial and final ice temperatures near the bottom melting and top ice surfaces, respectively. ΔT_{cr} is the temperature rise of the

Figure 7. Relationship between Nu_n and Re_n for $d_n = 0.152$, 0.406 and 0.838 mm ($D_s = 0.110$ m, $L = 0.840$ m).

ice sample resulting from the combined effects of natural convection and thermal radiation. The heat exchange associated with the melted ice is

$$Q_m = W_i [L_i + (T_f - T_i)c_i] \quad (2)$$

where W_i is the weight of ice melted, L_i is the latent heat of fusion, and T_i is the initial ice temperature. With Q_s and Q_m determined, an average heat transfer coefficient based on the sample cross-sectional area can be computed:

$$h = \frac{Q_s + Q_m}{A \Delta T \theta} \quad (3)$$

where A is the cross-sectional area of the ice sample. ΔT is the difference between the bulk water temperature and the melting point of ice and θ is the duration of the experiment.

Due to the lack of information on flow characteristics of the bubble-driven water jet, d_n and the air velocity v_a at the nozzle exit were used to define a Reynolds number $Re_n (= d_n v_a / \nu_a)$, where ν_a is the kinematic viscosity of air evaluated at 4°C . Similarly, a Nusselt number also based on d_n was defined, i.e. $Nu_n = h d_n / k_w$, where k_w is the thermal conductivity of water at 4°C . Table I summarizes the experimental parameters covered in this study. The results from the small column with ice sample diameter $D_s = 0.110$ m and nozzle to ice surface distance $L = 0.840$ m are shown in Figure 7. The experimental value lies fairly well along a straight line on a log-log plot for each d_n . A cross plot of the results shows that for a given Re_n , the Nu_n variation with d_n is as a power function $Nu_n \propto (d_n)^{1.4}$. The results for a large column with $D_s = 0.260$ m and $L = 1.524$ are shown in Figure 8. As demonstrated in Figure 7, the data also can be fairly represented by straight lines on a log-log plot. Quite a few experiments were also carried out for $L = 0.762$ m for $d_n = 0.152$ and 0.406 m. There seems to be a trend showing a higher Nu_n value for a given Re_n , but it is

Figure 8. Relationship between Nu_n and Re_n for $d_n = 0.152$, 0.406 and 0.838 mm ($D_s = 0.260$ m, $L = 0.762$ and 1.524 m).

Table I. Variation of experimental parameters.

Column height L (m)	Sample diameter D_s (m)	Nozzle opening d_n (mm)	L/d_n	D_s/d_n
0.840	0.110	0.838	1000	133
		0.406	2063	273
		0.152	5500	729
1.524	0.280	0.838	1818	310
		0.406	3750	641
		0.152	10000	1708
0.762	0.260	0.838	909	310
		0.406	1875	641
		0.152	5000	1708

felt that there are insufficient data to warrant a conclusion. A cross plot of this result also gives a Nu dependence on d_n by the relation $Nu_n \propto (d_n)^{1.4}$. Subsequently, using $Nu_n/(d_n)^{1.4}$ as ordinate, all the data (64 for $D_s = 0.110$ m and 107 for $D_s = 0.260$ m) were plotted in Figure 9. The least square analysis gave correlation coefficients 0.976 for $D_s = 0.110$ m and 0.946 for $D_s = 0.260$ m, with corresponding slopes of 0.367 and 0.405 respectively. Since the effect of L is not significantly demonstrated in our limited data, the effect of D_s was subsequently evaluated by a cross plot of Figure 9. The dependence of $Nu_n/(d_n)^{1.4}$ on D_s was found to be $Nu_n/(d_n)^{1.4} \propto D_s^{-1}$. As shown in Figure 10 all the results can be well represented by

$$Nu_n = 99.89(d_n)^{1.4} (D_s)^{-1} (Re_n)^{0.391} \quad (4)$$

with a correlation coefficient of 0.947, in which d_n and D_s are in centimeters. In terms of D_s and expressing D_s and d_n in meters, eq 4 can be transformed to

$$Nu = 630(d_n)^{0.4} (Re_n)^{0.391} \quad (5)$$

in which $Nu = hD_s/k_w$.

Figure 9. Relationship between $Nu_n / (d_n)^{1.4}$ and Re_n .

Figure 10. Relationship between $Nu_n / (d_n)^{1.4} (D_s)^{-1}$ and Re_n .

DISCUSSION

To the best of the author's knowledge, no work of a similar nature involving the hydrodynamics and the characteristics of heat transfer between an air-bubble driven jet and a surface held at normal has been reported. In fact, very little study has been done on the flow characteristics on the water jet induced by the rising bubble stream.

In Crabtree and Bridgwater's study using glycerol as the fluid and at a much slower bubble rise velocity, on the order of 40 to 55 mm/s, they developed the following liquid velocity function:

$$v = \frac{G}{2\pi\mu} \left[\left(\frac{r^2}{R^2} \right) + \ln \left(\frac{R}{r} \right) - 1 \right] \quad (6)$$

in which G is the force per unit height imposed on the liquid, r is the radial distance from the column axis, μ is the liquid viscosity, and R is the column radius. This experiment was compared with their experimental data and was found to be in excellent agreement. From eq 6, it can be calculated that at $r = 0.45R$, $v = 0$. Therefore there are two flow regions, an upward flow when $r \leq 0.45R$, and a downward flow when $r > 0.45R$. However, it is doubtful that this finding can be used to describe the flow characteristics of the bubble-driven jet.

The work of Sitharamayya and Raju¹⁰ on heat transfer between a submerged jet of water and a flat surface held normal to the flow was much closer to the conditions of the present study. They present their final correlation as

$$(\text{Nu}_n, \text{Pr}^{-0.33}) (D_s/d_n)^2 = 32.49(\text{Re}_n)^{0.523} + 0.266[(D_s/d_n) - 8] (\text{Re}_n)^{0.828} \quad (7)$$

for Re_n ranging from 2,000 to 40,000 and for $L/d_n < 7$, and $8 \leq (D_s/d_n) \leq 58$ with both Nu_n and Re_n defined based on d_n as in the present analysis in which D_s is the diameter of the heat transfer surface and Pr is the Prandtl number. Because the parameter L/d_n used is much smaller than in the present case, this work cannot be used for comparison (see Table I).

The work of Kobus⁸ was conducted in an $8 \times 280 \times 4.70$ m basin and again only concerned the hydrodynamic aspect of the problem. He found that the pattern of vertical velocity induced by an orifice discharging air into water can be presented by Gaussian distribution curves with a linear spread in the vertical except for the regions near the orifice and close to the free surface. The velocity distribution was expressed as

$$\frac{v_w}{v_c} = \exp \left[\frac{-r^2}{2C^2(L + y_o)^2} \right] \quad (8)$$

in which v_w and v_c are vertical and arrival or axial velocity, C is the rate of spread of velocity profiles, L is the vertical distance above the air source, and y_o is the distance from the orifice to an analytical origin resulting from an analytical extension of the established flow pattern into and beyond the limits of the initial region. This origin does not coincide with the location of the air source because of the markedly different conditions in the initial region. The expression for v_c was derived as

$$v_c = \frac{1}{C(L + y_o)} \left[\frac{-P_{\text{atm}}}{\pi \rho_w v_b} \ln \left(1 - \frac{L}{L^*} \right) \right]^{1/2} \quad (9)$$

where v_b is mean rising speed of air bubble stream, L^* is the sum of L and P_{atm}/γ (γ is the specific weight of water). From his experimental study, he reported that both spread coefficient C and v_b can be represented by

$$C = C_c (V_a)^{0.15} \quad (10)$$

and

$$\nu_b = C_b (V_a)^{0.15} \quad (11)$$

From his experimental results the values of C_c and C_b are found to be $C_c = 0.133(\text{m}^{-0.45} \text{s}^{0.15})$ and $C_b = 1.84(\text{m}^{0.55} \text{s}^{-0.85})$.

In the work reported by Gardon and Akfirat,⁷ the following correlation was given for single round air jets impinging centrally on circular targets of diameter D_s :

$$\text{Nu} = 0.78(\text{Re}_a)^{0.55} \quad (12)$$

Provided $\text{Re}_a > 4000$, $L/d_n > 12$, and $1 < D_s/d_n < 24$ in which Nu is defined as hD_s/k_a and Re_a as $D_s \nu_c / \nu_a$. By analogy with other turbulent heat transfer, it is expected that $\text{Nu} \propto \text{Pr}^{1/3}$. Assuming $\text{Pr} = 0.7$ for air and $\text{Pr} = 11.7$ for water at 4°C , eq 12 can be transformed to apply to water flow in the form

$$\text{Nu} = 1.99(\text{Re}_a)^{0.55} \quad (13)$$

Applying Kobus's results to a confined cylindrical column, the diameter at impingement is

$$D_i = 2C(L + y_o) \quad (14)$$

or

$$D_i = 2C_c(L + y_o)(V_a)^{0.15} \quad (15)$$

where the value of y_o as reported by Kobus is taken as 0.8 m, V_a is the volumetric air flow rate at nozzle exit at atmospheric conditions. With ν_b evaluated from eq 14, the value of ν_c can be computed from eq 9.

With the value of ν_c and D_i thus determined and the given value of ν_w , the Reynolds number ($\text{Re}_i = D_i \nu_c / \nu_w$) describing the flow characteristics of the bubble-induced water jet can be evaluated. Corresponding Nusselt numbers based on D_i ($\text{Nu}_i = h_i D_i / k_w$) were also computed. The log-log plot of Nu_i vs Re_i is shown in Figure 12.

It also shows clearly that the shorter the column height, the higher the values of Nu_i for the same D_s . This is from the fact that for the same air flow rate, the value of D_i is lower for the lower water level (see eq 15), and the value of Nu_i is inversely proportional to the first power of D_i . In all cases, the calculated values of D_i are much smaller than D_s . The ratio of D_i/D_s was found to vary from 0.135 to 0.201 at $L = 0.762$ m and from 0.200 to 0.300 at $L = 1.524$ m for the large column, and from 0.335 to 0.492 at $L = 0.840$ m for the small column for volumetric discharge rates ranging from 7.39×10^{-8} to $9.91 \times 10^{-7} \text{ m}^3/\text{s}$ (see Fig. 11). In computing the value of the heat transfer coefficient in Nu_i shown in Figure 12, the area corresponding to D_i and the total amount of heat exchange ($Q_s + Q_m$) were used. Since the ratio of D_i/D_s varies with both V_a and D_s , and represents only about 4 and 9% of the sample surface area A for the large column, and 25% of the small column, it is expected that results from the small column were lower than those of comparable water level but with larger diameter. However, from observations at the completion of an experiment, the melting ice surface was more or less even, indicating uniform melting across the sample surface for low to medium V_a . For higher V_a , the resultant melting surface was slightly concave upward, exhibiting a faster melting rate near the jet impingement region. To have a better comparison with eq 13, one would have to evaluate the variation of melting rate in the radial direction and thus calculate the h values associated with the impingement area. This could have been estimated by examining the melting profile at the completion of the experiment with weight measurement to check the integrated melting determined from the profile.

Figure 11. Variation of D_i/D_s as a function of V_a . Figure 12. Relationship between Nu_i and Re_i .

Figure 13. Relationship between Nu and Re_a .

Also the conditions given in eq 13 are not exactly met. In our case, the Reynolds numbers are ranging from 1.4 to 5.5×10^3 with a majority of Re_i less than 4.0×10^3 . In regard to the condition $1 < D_s/d_n < 24$, we have the range $42 < D_i/d_n < 510$. However, as can be noted in Figure 12, the value of Nu_i has a dependence on Re_i with an exponent close to 0.55 as in eq 13. Furthermore, using v_c to define the Reynolds number, the effect of d_n disappeared.

Since the topography of the melting surface was quite smooth and even at the completion of the experiment (especially for low V_a values), the heat transfer coefficient based on the sample area was used in defining Nu , and D_s instead of D_i in defining Re_a . The results are shown in Figure 13 along with eq 13. As shown in Figure 12, d_n has no effect in such a correlation. Furthermore the effect of L for the same column diameter D is insignificant. However, the effect of D_s is clearly demonstrated. The results indicate that generally the experimental values are lower than those predicted from eq 13, but with a much stronger dependence of Nu on Re_a ; therefore at higher Re_a the experimental values are higher than those from eq 13.

In light of the restriction imposed in eq 13, the agreement was found to be good. Since the log-log plots of the data for the two D_s are indeed linear and parallel, a cross-plot showing Nu vs Re_a results in a dependency of $Nu \propto (D_s)^{-1.2}$. Thus all the data can be correlated as shown in Figure 14 with a correlation coefficient of 0.983 as

$$Nu = 0.00010(D_s)^{-1.2} (Re_a)^{1.311} \quad (16)$$

in which D_s is expressed in meters. Unfortunately, this correlation is still dimensional and is inferior to the nondimensional one for practical use.

As previously stated the value of D_i depends on L as well as V_a . Therefore it is believed that Re_i instead of Re_a will best describe the flow characteristics of the induced jet flow. Figure 15 shows a log-log plot of Nu versus Re_i . The least-square analysis of the data can be fairly represented by

Figure 14. Relationship between $Nu/(D_s)^{-1.2}$ and Re_a .

Figure 15. Relationship between Nu and Re_i .

$$Nu = 0.1735(Re_i)^{0.848} \quad (17)$$

with a moderate correlation coefficient of 0.84. As indicated in Figure 15, the Nu values from $L = 1.524$, $D_s = 0.260$ m are generally and consistently somewhat lower than those from $L = 0.762$ m of the same D_s . On the other hand, the results from $L = 0.762, 0.840$ and $D_s = 0.260$ and 0.110 m are more or less in good agreement with each other throughout the whole Re_i range. It is hypothesized that in the shorter column the induced water flow is most likely in the developing stage (without the experimental verification) and thereby may possess more turbulent flow characteristics and thus promote a higher rate of heat transfer. However, for the time being, in view of the fact that the values of v_c and D_i (in defining Re_i) are computed from eq 9 and 15, developed from a much larger scale field study ($8 \times 280 \times 4.70$ m), the nondimensional correlation (eq 17) of the present results is considered to be exceptionally good and is far more practical than dimensional correlations as given in eq 4 or 5 and 16. With the assumption that due to lack of experimental evidence, there is no difference between the diameters of the target surface and the area of the impinging air jet, eq 13 can be rewritten as

$$Nu = 1.99(Re_i)^{0.55} \quad (18)$$

and is also shown in Figure 15. In general, the agreement is considered to be exceptionally good, in spite of the fact that the conditions given in eq 13 were quite different from those of this study.

Due to the lack of understanding and theoretical analysis regarding the hydrodynamics and temperature field in a confined cylindrical column, no analytical expression can be given for comparing with the present experimental results. To ensure the rightness of the nondimensional correlation, in future work it is believed it would be a worthy undertaking to study the velocity and temperature field of the bubble-induced water jet in a confined cylindrical column.

SUMMARY

An experimental study was conducted of the heat transfer characteristics of a bubble-driven water jet impinging on an ice surface. Two Lucite columns were used, one 1.829 m high and 0.286 m in diameter and the other 1.219 m high and 0.140 m in diameter. Water levels were maintained at 0.762 and 1.524 m in the large column and 0.840 m in the small column. Hypodermic needles with openings of 0.152, 0.406 and 0.838 mm were used for bubble formation, covering a volumetric air flow rate of 7.39×10^{-8} to 9.91×10^{-7} m³/s. To simulate winter environmental conditions in northern North America, the water temperature at the bottom of the column was thermostatically maintained at its temperature of maximum density, i.e. about 4°C. Cylindrical blocks of ice having diameters slightly less than those of the columns were supported at the column top while the melting surface was maintained at the proper depth below the water level. The experimental results can be represented by

$$Nu = 630(d_n)^{0.4}(Re_n)^{0.391}$$

with a correlation coefficient of 0.947 for $8 < Re_n < 10^3$. In the above expression d_n is the nozzle diameter expressed in meters, and the Nusselt number Nu (hD_s/k_w) is defined in terms of an average heat transfer coefficient h over the sample's cross-sectional area, the sample diameter D_s and the thermal conductivity of the water k_w . The Reynolds number Re_n [$(d_n v_a)/\nu_a$] is defined in terms of the air flow characteristics at the nozzle exit where v_a and ν_a are the velocity and kinematic viscosity respectively. The effects of both water level and column diameter were found to be absent in the correlation.

Due to lack of understanding and theoretical analysis of bubbling hydrodynamics and heat transfer, an attempt was made to compare the results with the limited work reported on heat transfer between a round

air jet and a plane surface held normal to the jet flow (prior to its use it was modified to water flow). The least-square analysis expression

$$Nu = 0.00010(D_s)^{-1.2} (Re_a)^{1.311}$$

was found to represent the data extremely well with a correlation coefficient of 0.983. The Reynolds number Re_a is defined as $D_s v_c / \nu_w$ where ν_w is the kinematic viscosity of water and v_c is the centerline vertical water jet velocity (which is computed from an expression developed from a much larger scale field study by Kobus). The correlation provides results of the same order of magnitude as those predicted from the pure water jet [i.e. $Nu = 1.99(Re_a)^{0.55}$].

Even though the above correlations provide a good representation of data, the expressions nevertheless are dimensional and inconvenient to use. A nondimensional expression was developed by replacing D_s in Re_a with the diameter at impingement D_i (also computed from Kobus's work). The least-square analysis provides

$$Nu = 0.1735(Re_i)^{0.848}$$

with a somewhat lower correlation coefficient of 0.84. This lower coefficient was apparently reflecting the water flow characteristics existing in the column at different water levels. For lower water levels, the induced water flow may still be in the developing stage, exhibiting a more turbulent nature and thus providing a higher heat transfer rate.

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