UNIVERSITY OF CINCINNATI

Date: _____

I, _____,
hereby submit this work as part of the requirements for the degree of:

in:

It is entitled:

This work and its defense approved by:

Chair: _____

**Heat Transfer Augmentation for External Ice-on-Tube TES Systems
Using Porous Copper Mesh to Increase Volumetric Ice Production**

A thesis submitted to the
Division of Research and Advanced Studies
of the University of Cincinnati

in partial fulfillment of the requirements
for the degree of

MASTER OF SCIENCE

in the Department of Mechanical, Industrial and Nuclear Engineering,
University of Cincinnati, Cincinnati, Ohio.

2004

By

Victor Sanjit Nirmalanandhan

Bachelor of Technology, Calicut Regional Engineering College, Calicut, India, 2001

Committee Chair: Michael J. Kazmierczak, Ph.D., Associate Professor

ABSTRACT

Substantial thermal performance improvement in ice-on-tube TES (thermal energy storage) systems is possible by making use of porous copper mesh as a HCED (Heat Conducting Enhancement Device). HCEDs are inexpensive heat transfer augmentation devices that will result in faster rate of ice growth and larger final steady state ice build volume by reducing the controlling thermal conduction resistance of the ice layer. This will improve the competitiveness of external ice-on-tube systems as compared to other TES systems such as dynamic ice harvesters and static internal melt systems. In this study the degree of ice growth enhancement is predicted theoretically, by performing simplified calculations, and is then validated in the laboratory through carefully controlled experiments. This study shows ice volume increase between 50-90% is possible.

KEY WORDS: Thermal Energy Storage, solidification, heat transfer augmentation, porous copper mesh

To my parents with love

ACKNOWLEDGEMENTS

I would like to express my sincere gratitude to my advisor Dr. Michael Kazmierczak for suggesting this thesis topic and his continuous counsel and supervision throughout the investigation and writing of this thesis. I also extend my thanks to Drs. Frank Gerner and Raj Manglik for their valuable discussions and suggestions, and for serving on the thesis committee.

I am also thankful to the Department of Mechanical, Industrial and Nuclear Engineering at the University of Cincinnati for the financial assistance. Thanks are also extended to Doug Hurd for his assistance in developing and fabricating parts of the experimental facility.

My sincere thanks to my colleagues and friends Anand, Ashutosh, Goutham and Manish who were constant source of help, encouragement and joy through my graduate school days. Special thanks to all my friends especially to Michael and Lauren for making my stay in Cincinnati an enjoyable one; and Prija for helping me format this thesis.

Last, but certainly not the least, I like to express my special appreciation to my parents, sister, brother and cousin Roshan for their love, encouragement and support that helped me achieved this educational goal.

TABLE OF CONTENTS

List of Tables	2
List of Figures	3
Nomenclature	5
1. Introduction	7
2. Mathematical Model	12
2.1 Theoretical Predictions Showing the Effect of k_{eff}	15
3. Experimental System and Test Procedure	19
4. Properties Evaluation and Data Reduction	25
4.1 Reynolds Number Re and Velocity Correction	25
4.2 Temperature of the Tube Walls, T_w	27
4.3 Cooling Temperature Ratio, Θ_c	27
4.4 Ice Volume V_{Ice} and Total Cooling Capacity	28
4.5 Average Ice Radius r_{Ice} and Ice Thickness δ	29
4.6 Effective Thermal Conductivity of Ice/HCED matrix, k_{eff}	29
5. Experimental Results and Discussions	31
5.1 Steady State Ice Shape	33
5.2 Ice Volume	36
5.3 Enhancement of Ice Volume and Comparison to Theoretical Model	39
6. Conclusions and Future Directions	46
References	47

LIST OF TABLES

	Page No.
Table 1. Comparison of final nondimensional ice radius r^* obtained from the transient solution with that obtained from steady state energy balance, for $\Theta_c = 9.94$ and $Re = 400$, showing agreement.	18
Table 2. Experimental conditions to study the effects of Reynolds number and cooling temperature ratio on ice build and the amount of augmentation achieved using HCED.	32

LIST OF FIGURES

	Page No.
Fig. 1: Theoretical model showing predicted ice growth (r^*) as a function of dimensionless time, τ for identical operating conditions ($Re = 400$ and $\Theta_c = 10$) but for different effective thermal conductivities of ice: a) large time range; b) early times ($0 < \tau < 100$).	16
Fig. 2: Variation of thermal resistances due to conduction and convection as a function of time for $Re = 400$ and $\Theta_c = 9.94$ with $k_{eff} = 2k_{ice}$ for tube fitted with HCED.	17
Fig. 3: Schematic diagram of the experimental set up.	19
Fig. 4: Schematic of circular test tube showing basic elements, coolant flow and thermocouple location.	21
Fig. 5a: Digital picture of a bare tube and Fig. 5a: Digital picture of a bare tube and HCEDs.	22
Fig. 5b: Digital picture of a tube fitted with HCED.	22
Fig. 6: Ice shapes at steady state for different Reynolds number at $\Theta_c \approx 1.64$; Top with HCED, Bottom without HCED.	34

Fig. 7a: Digital photograph of the ice shape at $\Theta_c = 1.41$ and $Re = 1188$ (run # 5) showing minimum ice build.	35
Fig. 7b: Digital photograph of the ice shape at $\Theta_c = 4.09$ and $Re = 197$ (run # 6) showing maximum ice build.	35
Fig. 8: Ice volume as a function of Reynolds number for both bare tube and tube with HCED for nearly constant Θ_c (runs 1-5).	37
Fig. 9: Ice volume as a function of Θ_c for both bare tube and tube with HCED for constant Reynolds number (run # 1 & 6).	38
Fig. 10: % enhancement of ice volume as a function of Reynolds number at $\Theta_c = 1.64$.	40
Fig. 11: Transient ice growth curves predictions for variable $k_{eff}(r^*)$ values ($k_{eff} = 1.684k_{ice}$; $k_{eff} = 1.684k_{ice}$ for $r^* < 1.38$ and $k_{eff} = k_{ice}$ for $r^* > 1.38$; and $k_{eff} = k_{ice}$). Large dimensionless time showing comparison to steady state experimental data; b) Theoretical transient ice growth curves for variable k_{eff} at early time.	42
Fig.12: % enhancement of ice volume as a function of cooling temperature ratio, Θ_c at $Re = 198$.	43
Fig. 13: Ice shapes at $Re=189$ and $\Theta_c= 9.66$ showing ice linking.	45

NOMENCLATURE

English Letters

A	total exterior surface area of the tube, m^2
Bi	Biot number, $= \frac{hd}{k}$
C_p	specific heat at constant pressure, J/kg-K
d	local diameter, m
$\bar{d}$	average diameter, m
d^*	dimensionless diameter, $= \frac{\bar{d}_{ice}}{d_{tube}}$
Fo	Fourier number, $= \frac{\alpha_{ice} t}{d_{tube}^2}$
h	convective heat transfer coefficient, W/m ² -K
k	thermal conductivity, W/m-K
k^*	thermal conductivity ratio, $= \frac{k_{ice}}{k_{\infty}}$
L	latent heat of fusion, J/kg
l	tube length, = 0.127m
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number, $= \frac{hd}{k}$
Pr	Prandtl number, $= \frac{\mu}{\rho\alpha}$
q	heat transfer rate, W
R_{cond}	conduction resistance through ice layer, $= \frac{\ln\left(\frac{r_{ice}}{r_{tube}}\right)}{2\pi l k}$, K/W
R_{conv}	convective resistance at solid/liquid interface, $= \frac{1}{2\pi r_{ice} l h}$, K/W
Re	Reynolds number, $= \frac{u_c d}{\nu}$
r	radius, m
r^*	dimensionless radius of ice, $= \frac{r_{ice}}{d_{tube}}$
St	Stefan number, $= \frac{c_{p,ice}(T_f - T_w)}{L}$
T	temperature, °C
t	time, seconds

u_c	velocity (corrected), m/s
U_o	overall heat transfer coefficient, W/ m ² -K
U_∞	free stream velocity, m/s

Greek Letters

α	thermal diffusivity, m ² /s
Γ	convective-to-conductive heat transfer ratio at ice/water interface
ε	porosity
μ	dynamic viscosity, kg/m-s
Θ_c	cooling temperature ratio, $= \frac{T_f - T_w}{T_\infty - T_f}$
ρ	fluid density, kg/m ³
τ	dimensionless time, $St_s \times Fo$
ν	kinematic viscosity, m ² /s

Subscripts

eff	effective
f	freezing
i	condition at the ice/water interface
ice	ice layer
l	liquid
max	parameter based on maximum velocity
s	solid
tube	single tube
w	tube wall
∞	ambient condition (free stream)

Superscripts

—	average value
*	dimensionless value

1. INTRODUCTION

It is well known that the thermal performance of external ice-on-tube systems deteriorates as the ice layer thickness grows [Dorgan and Elleson, 1993; Intemann and Kazmierczak, 1997; Kazmierczak and Intemann, 1997; Saito 2002] due to the added insulating effect of the ice build. In fact, as the ice layer thickness increases, the dominate thermal resistance soon becomes the ice layer itself, becoming much larger in magnitude than either the internal convective resistance of the coolant and inner tube wall, and the external resistance due to convection between the outer surface of the ice layer and the surrounding water. In short, the thicker the ice layer, the greater its thermal resistance, which therefore significantly slows the rate of ice build. Ideally, so as to maximize the rate of ice build, the ice layer thickness should be kept as thin as possible. However this would reduce the amount of latent energy storage per tube, thus requiring a greater total number of tubes to be installed in a package to maintain the same overall total thermal storage capacity of the unit. Hence the final TES storage tank/system design that ultimately results in a compromise between the additional capital cost associated with installing more tubes against the poorer operating performance of using fewer tubes with thicker ice coating.

One of the few ways to avoid this inherit disadvantage of the ice build resistance is to reduce the resistance by reducing its thickness, so as to make the heat conduction path length shorter. Ice harvester does this by limiting the thickness of the ice growth layer to a predetermined “relatively thin” ice layer by “periodically shucking” the ice to an ice storage tank located directly below. Internal ice build systems, on the other hand,

implements this same philosophy but in a different way by always refreezing and melting the ice nearest the tube surface by utilizing the same pipe for both charging and discharging purposes [Dorgan and Elleson, 1993].

An alternate method to reduce the ice build resistance is by adding external longitudinal or circumferential “heat conducting plates” to the outer tube which then become totally encased in ice when the water freezes. These plates act as “thermal shorts” and hence enhance the freezing rate by reducing the conduction thermal resistance [Sparrow et al., 1981; Sasaguchi et al., 1986; Hirata et al., 1993]. This technique was first studied by Sparrow et al. (1981) in fluid flow of phase change materials (waxes). They studied freezing on a finned vertical tube when either conduction in the solid and natural convection in the liquid controlled the heat transfer process. They reported that the enhancement of freezing, if conduction dominated, due to finning was less than the area ratio of the finned and unfinned tube, whereas for natural-convection controlled the enhancement was very nearly equal to the area ratio. Sasaguchi et al. (1986) performed three-dimensional numerical calculations to obtain the performance of a finned-tube-type latent heat thermal energy storage (LHTES) unit and checked the validity of the calculations by comparing it against the experimental results. They studied alternate fin configurations and reported that the performance of the unit was almost the same for the finned tubes with the same surface area, even if the fin configurations were quite different. More recently Hirata et al. (1993) experimentally and analytically investigated the ice formation process in stagnant water associated with heat conductor plates located perpendicularly at some distance to isothermally heated and cooled

surfaces. Their study showed that the onset conditions, which describe whether the ice volume fraction in steady state is increased or decreased as compared to that without heat conductor plates, are dependent on the pitch and the thickness of the heat conductor plates. In conclusion, all of the previous studies agrees in the fact that fin addition is a highly successful tool in modifying the ice volume. Unfortunately, however, one serious drawback of adding external surfaces to the tube is that the tube cost becomes prohibitively high.

Here, in this study, we reduce the thermal resistance of the ice layer in a similar way, but much less expensively by simply increasing the “effective” thermal conductivity of the ice layer by making use of *porous copper mesh* located in the ice build. The operating principle and underlying physics are very straightforward and are readily explained as follows. Since the thermal resistance in a solid layer is inversely proportional to its thermal conductivity, doubling the effective thermal conductivity of the ice layer, k_{eff} , then causes the thermal resistance (for the same given thickness) to be cut in half. Hence, for the same given ice build thickness as before (and thus a system having the same thermal storage capacity) a new HCED augmented ice-on-tube system will offer proportionately less conduction thermal resistance and therefore improve the rate at which the ice freezes. In short, HCED improve the thermal performance of external ice-on-tube systems by just “thermally shorting” the thick insulating ice layer, which eventually becomes the largest and most restrictive thermal resistance in the thermal network pathway in any static TES system design. This concept of using a metal matrix for heat transfer enhancement during phase-change processes was theoretically

analyzed by Khan and Morehouse (1994). They numerically modeled the heat transfer to and from constant temperature cylindrical tubes surrounded by water during the freezing and melting processes for various metal-matrix volume fractions and showed that an order of magnitude enhancement in the rate of freezing was possible for metal-matrix volume fractions of around 7%. These promising results have not, as yet, been confirmed experimentally which is the major focus of this study. Finally it should be mentioned that, Tong et al. (1996) theoretically investigated almost the same technique but for the natural convection problem and reported that solidification heat transfer can be enhanced by inserting metal matrix into a phase change material.

In terms of freezing of water on tubes without heat conduction enhancement devices, a review of the existing literature reveals that there are two key experimental studies related to freezing around a cooled pipe in crossflow. The phenomenon of freezing around a cooled pipe in crossflow was first investigated experimentally and theoretically by Okada et al. (1978). They proposed a simple non-dimensional correlation for the overall heat transfer rate from water to the ice covered pipe at the steady state. Later Cheng et al. (1981) investigated the same problem but for different (higher) Reynolds numbers and cooling temperature ratios and proposed correlations for average heat transfer and cooling capacity that can be stored in the ice layer around a cylinder. Last, it should be noted that Chiang and Kleinstreuer (1989) numerically solved the Navier-Stokes and energy equations for transient laminar two-dimensional flow past a horizontal subcooled cylinder with outward solidification, but unfortunately, they did not directly compare their results with prior experiments.

In this investigation a simplified 1-D mathematical model is developed to theoretically predict the thermal performance augmentation possible for our particular Heat Conducting Enhancement Device (HCED) made from porous copper mesh. More importantly, and the main contribution of this work, is that several carefully controlled *experiments* are reported for a range of different operating conditions. The ice volume augmentation predictions are finally compared to the experimental data set.

2. MATHEMATICAL MODEL

Modeling the radial outward solidification of phase –change material surrounding a cylinder in 1-D cylindrical coordinate system is used to help explain the theory behind this study. Cylindrical solidification with moving solid-liquid interface is referred to as the Stefan problem; the only difference being the utilization of a cylindrical coordinate system and the inclusion of a convective heat transfer term at the ice/water interface [Lunardini, 1991; Alexiades and Solomon, 1993; Lock, 1994; Khan and Morehouse, 1994].

An energy balance at the ice/water interface shows that the ice growth rate is proportional to the difference between the heat conducted away from the interface, through the ice layer to the subcooled tube and the heat entering the ice along the interface due to convection with the warm fluid stream. This energy balance is written as,

$$q_{conduction} - q_{convection} = \dot{m}L \quad (1)$$

or

$$\rho L \frac{dr}{dt} = k_s \frac{\partial T}{\partial r} - h(T_\infty - T_f) \quad \text{for } t > 0, r = R(t) \quad (2)$$

Note that a number of assumptions were made in order to keep the proposed ice growth model as simple as possible. The first was that the ice surrounding the tubes would grow uniformly in thickness; an assumption which in view of the actual experimental ice shapes (shown later) is not exactly true, but nevertheless necessary to render the model 1-

D. Thus the circumferential conduction component within the ice was neglected and the actual 2-D ice growth process thereby reduced and modeled as a 1-D radial conduction problem. This simplification is justified by previous studies (Intemann, 1996; Okada et al., 1978) that show that the error is only within 10 %. The second important assumption was that the outer convective film coefficient, h , was a given and known quantity, i.e. that it represented an input to the model. Moreover, to be consistent with the uniform ice build assumption, the outer film coefficient had to be spatially uniform, without angular dependence. Finally two additional assumptions were also deemed necessary. First, the change in the sensible energy storage within the ice is negligible. This assumption is justified for problems such as this possessing small Stefan numbers, generally less than $St = 0.1$, and constitute the quasi-steady state behavior. Second, the thermo physical properties of water and ice were assumed constant. Substituting the solution to the heat conduction equation yields

$$\rho_{ice} L \frac{d\bar{r}}{dt} = k_{ice} \frac{(T_f - T_w)}{\bar{r}_{ice} \ln\left(\frac{2\bar{r}_{ice}}{d_{tube}}\right)} - \bar{h}(T_\infty - T_f) \quad \text{for } t > 0, \bar{r} = R(t) \quad (3)$$

The left hand side of the above equation represents the heat liberated by the conversion of water to ice, while the first term on the right side represents the heat transported away from the interface by conduction through the ice layer and the second term represents the energy addition into the interface due to free stream convection.

To complete the problem formation requires the specification of the following initial condition, $r = r_{tube}$ at $t = 0$, indicating that, before the tube wall was instantaneously lowered to T_w , the tube was initially free of ice. Before solving, we

introduce the nondimensional parameters r^* , k^* , Fourier number, Fo, Stefan number, St and Nusselt number, $(\overline{Nu} = \frac{\bar{h} d_{tube}}{k_\infty})$ (defined in nomenclature list) to first nondimensionalize the governing equation. The dimensionless governing equation, after rearranging, can be rewritten as follows

$$\tau = \int_{\frac{1}{2}}^{r^*} \frac{r^* \ln(2r^*)}{1 - \Gamma r^* \ln(2r^*)} dr^* \quad (4)$$

where Γ is equal to

$$\Gamma = \frac{\bar{h}(T_\infty - T_f)}{k_{ice}(T_f - T_w)/d_{tube}} = \frac{\overline{Nu}}{k^* \theta_c} \quad (5)$$

Unfortunately, no exact closed form solution to Eq. (4) exists but numerical integration of the equation is relatively simple, provided the actual $\bar{h}$ at the interface is known. The limited knowledge regarding accurate values for $\bar{h}$ during phase change processes has always been the major stumbling block in developing and utilizing ice growth models [Intemann and Kazmierczak,1997; Kazmierczak and Intemann,1997]. As mentioned earlier, correlations for $\bar{h}$ with freezing on a single cylinder are limited to Okada et al. (1978) and Cheng et al. (1981) for low and moderate/high Re number ranges, respectively. Okada's correlation, which was found to agree well with their experimental ice growth data, and applicable for our Re range, is

$$\bar{h} = 0.57 \frac{k_\infty}{d_{tube}} \left(\frac{r_{tube}}{r_{ice}} \right)^{\frac{1}{2}} \text{Re}_{tube}^{\frac{1}{2}} \text{Pr}^{\frac{1}{3}} \quad (6)$$

and is therefore also implemented in our theoretical model to evaluate the $\overline{Nu}$.

2.1 Theoretical Predictions Showing the Effect of k_{eff}

Here we first show that the rate of freezing and the final steady state ice volume accumulated on these tubes can be substantially improved by reducing the controlling thermal resistance of the ice layer by increasing its effective thermal conductivity. The effective thermal conductivity of the ice layer is increased several folds by using porous screen mesh as the HCED (Heat Conducting Enhancement Device). To show the effect of increasing k_{eff} the governing equation (Eq. 4) was solved using Matlab software for 3 representative different values of effective thermal conductivity of $k_{\text{eff}} = k_{\text{ice}}$, $2k_{\text{ice}}$ and $5k_{\text{ice}}$. Fig. 1(a) and Fig. 1(b) show the predicted ice growth over large dimensionless time range and at early dimensionless times ($0 < \tau < 100$), respectively.

Fig. 1: Theoretical model showing predicted ice growth (r^*) as a function of dimensionless time, τ for identical operating conditions ($Re = 400$ and $\Theta_c = 10$) but for different effective thermal conductivities of ice: a) Large time range; b) early times ($0 < \tau < 100$).

From the graphs it's clear that the greater the effective thermal conductivity of the ice layer the higher the freezing rate and greater final steady-state ice volume produced, with the increase being nearly linear with k_{eff} . The reduction of the conduction resistance in the ice build is responsible for the enhancement of the freezing rate and the increase in steady state ice volume produced. This is shown more clearly in Fig. 2 where, for $Re = 400$ and $\Theta_c = 9.94$, the variation of the thermal resistances due to conduction, R_{cond} , and convection, R_{conv} , are plotted individually as functions of time.

Fig. 2: Variation of thermal resistances due to conduction and convection as a function of time for $Re = 400$ and $\Theta_c = 9.94$ with $k_{\text{eff}} = 2k_{\text{ice}}$ for tube fitted with HCED.

As shown, while the thermal resistance due to convection remains approximately the same in both the cases, the thermal resistance due to *conduction* in the case of tube with HCED decreases to about half as compared to the bare tube at same τ . Eventually, as τ increases, and steady state is approached, the ice growth will cease, Fig. 1(a) and according to the Eq. (1), $q_{\text{conduction}}$ must equal $q_{\text{convection}}$. Therefore the dimensionless governing equation can be simplified, and after some algebraic manipulations reads

$$\frac{k^* \theta_c}{\ln(2r^*)} = \frac{\bar{h} r_{ice}}{k_{ice}} . \quad (7)$$

As a check, Eq. (7) was solved by trial and error method and the results perfectly match (see Table 1) the final values of r^* obtained by numerically solving Eq. (4) (i.e. transient solution) for identical conditions ($\theta_c = 9.94$ and $Re = 400$).

Effective thermal conductivity k_{eff}	Transient solution Eq. (4)	Steady state solution Eq. (7)
k_{ice}	2.08	2.1
$2k_{ice}$	3.94	3.9
$5k_{ice}$	11	11

Table 1. Comparison of final nondimensional ice radius r^* obtained from the transient solution with that obtained from steady state energy balance, for $\Theta_c = 9.94$ and $Re = 400$, showing agreement.

3. EXPERIMENTAL SYSTEM AND TEST PROCEDURE

All the experiments were performed in a closed loop water tunnel facility made out of Plexiglas and PVC piping. A schematic diagram of the experimental test rig is shown in the Fig. 3, which was used in earlier experimental study [Intemann and Kazmierczak, 1997; Kazmierczak and Intemann, 1997] for the investigation of ice formations on tube bundles.

1. Test Section with Tubes	7. Low Temperature Chiller (Tunnel)
2. Flow Conditioning Elements	8. Glycol Piping
3. Pump	9. Data Acquisition System
4. Magnetic Flow Meter	10. Flow Control Devices (Tunnel)
5. Plate Heat Exchanger	11. Low Temperature Chiller (Tubes)
6. Heated Spool Piece	12. Flow Control Devices (Tubes)

Fig. 3: Schematic diagram of the experimental set up.

The dimensions of the test section are 254mm high by 127mm wide and 774.7mm long. Water first enters through the diffuser (345.25mm long and 14° opening), then flows through the flow conditioning section (647.7mm long), the test section containing the tubes, then through the downstream section (647.7mm long) and out the reducer (345.25mm long and 14° opening) and back again after passing through the pump and the 4" SCH 80 PVC return piping. Honeycomb and baffles are provided in the flow conditioning section to produce a uniform velocity profile at the inlet of the test section.

Two copper test tubes ($d_{\text{tube}} = 17.145\text{mm}$ O.D) are set horizontally at equidistance from the center axis of the tunnel. Fig. 4 shows the construction details of the copper tubes and internal coolant flow path. In order to achieve the proposed augmentation of the ice build the upper tube is fitted with a HCED. The HCED used in this study is a copper mesh (wire gauge = 25, hole size = 2mm). The tube is hand wound with multiple layers of this copper screen mesh to a total thickness of $d_{\text{HCED}} = 47.3\text{mm}$ O.D as shown in Fig. 5. The porosity of the HCED device is $\varepsilon = 0.9$ as measured by the volume displacement method. The porosity is the ratio of the void volume to the total combined volume occupied by the solid (copper) and void matrix. The volume of the copper in the mesh is calculated indirectly by measuring the volume of the water displaced by the mesh when it is immersed in a graduated cylinder filled with water. The bottom tube in the test section serves as the reference tube without HCED. The test section has two removable windows in the front and the back. These windows hold the tubes horizontally in crossflow in the water stream. The front window serves as the access to the tubes to

measure the ice shape and the ice volume. The tunnel is insulated all over with 1" thick Amerflex insulation to minimize the heat gain through the walls of the tunnel.

Fig. 4: Schematic of circular test tube showing basic elements, coolant flow and thermocouple location.

Fig.5a: Digital picture of a bare tube and HCEDs.

Fig. 5b: Digital picture of a tube fitted with HCED.

The temperature of the water entering the test section was carefully controlled with the help of the plate heat exchanger and the high capacity low temperature chiller, (see Fig. 3, components 5 and 7, respectively). Low temperature chiller circulates 50/50 mixture of glycol and water at low temperature to the cold side of the plate heat exchanger that removes the energy from the warm water passing through the hot side coming from the return loop of the water tunnel. The water tunnel flow rate is accurately controlled by adjusting the speed of the variable speed centrifugal pump using a variable speed pump controller. The flow is measured using a Foxboro magnetic flow meter device.

The freezing experiments begin by circulating 100% glycol at low temperatures through the two test tubes after the flow velocity and the temperature of the water stream are selected and made constant. The temperature of the glycol in the test tubes is controlled accurately and held constant using a separate low temperature chiller (# 11, Fig. 3). The flow rate of the glycol to the tubes are measured independently using two rotor meters and the volumetric flow rate of glycol is purposely maintained exactly the same in both tubes using control valves. The temperature of the water in the tunnel test section, wall temperature of the tubes and the inlet and outlet temperatures of glycol flowing through the tubes are measured using Copper-Constantan thermocouples. These temperatures are recorded in one minute increments using Data Acquisition System (DAS) and Labtech Notebook software.

Initially, after the freezing process starts, the ice layer continuously grows and more and more ice forms. After a certain period of time, the ice growth slows, and then it eventually stops and the ice shape become fixed. This steady state condition is achieved when the conduction heat transfer through the ice layer is exactly balanced by the outer convective heat gain caused by the water flowing over the tube. After steady state is reached, for a given set of experimental conditions (cooling temperature ratio Θ_c and Reynolds number Re), the water tunnel is quickly drained and the front window of the test section is removed. The ice shape is photographically recorded using a digital camera. The exact ice shape at the midlength of the tube is traced using two contour gauges. In order to prevent melting, the ice formation is allowed to subcool (i.e. glycol tube chiller is kept running) for approximately 10 minutes. After taking all of the ice shape measurements, the ice is allowed to melt and collected in a beaker and its volume measured.

4. PROPERTIES EVALUATION AND DATA REDUCTION

All thermophysical properties of the water stream which appear in the convection heat transfer correlation, Eq. (6), used in the theoretical model to predict ice growth (i.e., thermal conductivity, Prandtl number, etc.) are estimated at the arithmetic mean film temperature [Douglas and Churchill, 1956] of the ice-water interface and the test section inlet temperature as given by:

$$T_{film} = \frac{T_f - T_\infty}{2} \quad (8)$$

where $T_f = 0^\circ\text{C}$ which represents the freezing temperature of water and T_∞ is the set tunnel temperature. The properties used in computing the Reynolds number are estimated at the ambient inlet water temperature.

4.1 Reynolds Number Re and Velocity Correction

Freezing experiments were carried out at five different Reynolds numbers based on the bare (i.e. no ice) tube diameter. The water tunnel volumetric flow rate was measured using a magnetic flow meter, which was set to display the readings in units of LPM. The average uniform velocity into the test section is determined using the conservation of mass:

$$Q_{tunnel} = Q_{test\ section} = U_\infty A_{test\ section} \quad (9)$$

$$U_\infty = \frac{Q_{tunnel}}{A_{test\ section}} = 0.0517 Q_{tunnel} \quad [\text{cm/s}] \quad (10)$$

where Q_{tunnel} is the flow rate indicated by the flow meter in LPM and $A_{test\,section}=322.58$ cm² is the cross-sectional area of the test section.

Velocity blockage corrections were used in computing the Reynolds number. When the test tubes are inserted into the test section they obstruct the flow, thus requiring a correction to the free stream velocity (U_{∞}). The method of Vliet and Leppert (1961) has been adopted for this purpose in this study. In this method a mean area for blocked flow, A_m , is defined and the corrected velocity, u_c , is estimated from the unobstructed velocity, U_{∞} , using conversion of mass:

$$u_c = \frac{A_{test\,section} U_{\infty}}{A_m} \quad (11)$$

The mean area for blocked flow, A_m , is defined as that area, which, when multiplied by the inserted tube diameter, is equal to the net fluid volume surrounding the tube:

$$A_m d = Hld - 2 \frac{\pi d^2}{4} l \quad (12)$$

Since $A_{test\,section} = Hl$, eqn. (11) may be rewritten by using eqn. (12) as follows:

$$u_c = \frac{H}{H - \frac{\pi d}{2}} U_{\infty} \quad (13)$$

By inserting the numerical values; $H=254$ mm; $l=127$ mm; and $d=17.145$ mm, into the above expression, the following velocity correction for the test tube is obtained:

$$u_c = 1.1186 U_{\infty} \quad (14)$$

Finally, the Reynolds number based on the bare tube diameter and the corrected flow velocity is given by:

$$\text{Re}_d = \frac{u_c d}{\nu} \quad (15)$$

where ν is the kinematic viscosity of the water estimated at the test section inlet water temperature.

4.2 Temperature of the Tube Walls, T_w

Average tube wall temperature is used in the data reduction. The average test tube wall temperatures, T_w , are calculated by arithmetic averaging the four thermocouple readings, embedded in each tube.

4.3 Cooling Temperature Ratio, Θ_c

The cooling temperature ratio is an important parameter relevant in all of the freezing experiments (Okada et al., 1978; Cheng et al., 1981). It appears in the ice-water interface energy balance Eqs. (5), (7) and is defined as:

$$\Theta_c = \frac{T_f - T_w}{T_\infty - T_f} \quad (16)$$

where T_f is the freezing temperature of the water. The above expression represents the ratio of the ice subcooling (numerator) important in conduction, to the degree of water superheating important to convection (denominator). It should be noted that holding all other parameters constant, larger Θ_c values will result in thicker ice layer formations and for smaller Θ_c values the ice thickness will be thinner. This parameter is indirectly varied

during the experiments by either changing the ambient water temperature, T_∞ , and or by changing the temperature of the glycol pumped through the tube which, in turn, changes the tube wall temperature. The tube wall temperature also depends on the internal glycol flow rate which is maintained at the highest flow rate possible.

4.4 Ice Volume V_{ice} and Total Cooling Capacity

Steady state ice volume measurements are based on the total amount of water that was collected at 20°C from the melted ice formation around the test tubes. The ice volume is then calculated from mass conservation:

$$V_{ice} = V_{water} \frac{\rho_{water}}{\rho_{ice}} \quad (17)$$

where $\rho_{water}=999 \text{ kg/m}^3$ and $\rho_{ice}=917 \text{ kg/m}^3$ are the densities of water and solid ice at 20°C and 0°C, respectively. The ice volume is then nondimensionalized by dividing with the volume of the test tube:

$$V^* = \frac{V_{ice}}{V_{tube}} \quad (18)$$

It is important to note that this quantity also represents the dimensionless total cooling capacity stored in the steady state ice layer formed around the test tube. This fact is readily evident from the definition of the total cooling capacity stored in the ice:

$$E_{cc} = \rho_{ice} L V_{ice} \quad (19)$$

where L is the latent heat of fusion. In this equation the sensible energy storage is neglected, being usually very small by comparison to the latent heat of fusion.

4.5 Average Ice Radius r_{ice} and Ice Thickness δ

Finding the resulting average ice radius, r_{ice} , is an important goal of this study. It is used to quantitatively compare the degree of ice build enhancement using HCED and is also used in calculating the Nu number used in the theoretical model. Assuming ice is uniformly deposited around the tube, the diameter of the tube with ice formation is determined from the ice volume as follows:

$$d_{ice} = \sqrt{\frac{4V_{ice}}{\pi l} + d_{tube}^2} \quad (20)$$

The dimensional and dimensionless ice radii therefore are:

$$r_{ice} = \frac{d_{ice}}{2} \quad (21)$$

$$r_{ice}^* = \frac{r_{ice}}{d_{tube}} \quad (22)$$

where the ice radius was nondimensionalized by dividing it with the diameter of the bare tube. The difference between the radius of ice and that of the tube gives the thickness, δ , of the ice layer as

$$\delta = r_{ice} - r_{tube} \quad (23)$$

4.6 Effective Thermal Conductivity of Ice/HCED matrix, k_{eff}

The conduction resistance of the ice build depends on the “effective” thermal conductivity of the ice/HCED matrix. Since the arrangement of the pores in the copper screen is neither parallel nor series, the effective thermal conductivity is calculated using the correlation for the geometric mean of parallel and series arrangements as proposed by Batchelor and O’Brien (1976) as

$$\frac{k_{eff}}{k_f} = \left(\frac{k_s}{k_f} \right)^{1-\varepsilon} \quad (24)$$

where ε is the porosity, k_s is the thermal conductivity of the solid metal of the mesh and k_f is the thermal conductivity of the stagnant fluid filling the voids, or in our case, ice. Hence k_{eff} depends on thermal conductivities of both copper and ice as long as the ice growth is constrained within the HCED. As the ice layer grows outside the HCED, its thermal conductivity is no longer enhanced. Instead, it is the same as the thermal conductivity of pure ice alone. Note that this change takes place at $r^*=1.38$ for our given mesh thickness. For $k_s = 401$ W/m-K, $k_f = 2.2$ W/m-K and $\varepsilon = 0.9$, $k_{eff} = 1.684k_{ice}$ for $r^* < 1.38$ and $k_{eff} = k_{ice}$ for $r^* > 1.38$. This step change in the effective thermal conductivity can be easily taken into account in the mathematical model (Eq. 4) by inserting the appropriate values of thermal conductivity [i.e. $k_{eff}(r^*)$] during the numerical integration process. Note that the theoretical solutions presented previously in Figs. 1-2, showing the basic concept, assumes that the copper mesh extends throughout the entire ice build.

5. EXPERIMENTAL RESULTS AND DISCUSSION

A total of 7 experiments (Table 2) were performed in order to understand the influence of the Reynolds number and cooling temperature ratio on the thermal performance of the tubes, with and without the HCED. To study the effect of Reynolds number, the cooling temperature ratio is fixed at approximately $\Theta_c = 1.64$ and the experiment is run for 5 different Reynolds number in the range of 200 to 1200 (runs 1-5). In addition, the Reynolds number is fixed at approximately 200 and two more experiments are conducted for a total of 3 different cooling temperature ratios to study the effect of Θ_c (runs # 1,6,7). The tests experimental conditions and the final steady state ice volume obtained, with and without HCED, are summarized in Table 2.

Exp #	Re	Θ_c	T_∞ (°C)	T_w (°C)		Ice volume V_{ice} (ml)		% enhancement Eq. (25)
				Bare	HCED	Bare	HCED	
1	198	1.83	2.92	-5.94	-5.36	134	247.3	84.55
2	378	1.70	2.96	-5.1	-5.04	91.5	174.3	90.48
3	519	1.78	2.76	-4.72	-4.91	76.3	142.7	86.96
4	886	1.46	3.10	-4.42	-4.53	65.4	105.7	61.67
5	1188	1.41	2.93	-4.05	-4.15	48.0	70.8	47.73
6	197	4.09	2.74	-11.08	-11.21	215.5	406.4	88.38
7	189	9.66	1.45	-14.64	-14.02	*	—	—

* The ice build on the bare tube and tube with HCED linked together

Table 2. Experimental conditions to study the effects of Reynolds number and cooling temperature ratio on ice build and the amount of augmentation achieved using HCED.

5.1 Steady State Ice Shape

The resulting final ice shapes are a consequence of the local convective heat transfer coefficient and the local energy balance at the ice-water interface. If the internal energy removal rate (from cold tube) is larger than the external convective heat gain (warm water stream), then ice forms, the amount being determined by the magnitude of this difference and by the value of the latent heat of fusion. In the early stage of freezing, the conduction heat transfer resistance does not limit the ice growth and the initially very thin ice layer removes large amount of energy from the adjacent water and the ice builds rapidly. Here the ice forms a concentric ring with the growth rate is controlled by the inner and outer convection resistances and latent heat balance. However as the ice layer thickens the thermal resistance of the growing ice layer increases up to the point to where the heat conduction limits the ice growth [Intemann and Kazmierczak,1997; Kazmierczak and Intemann,1997] and the influence of the local variations in the local convective heat transfer becomes significant. The ice layer thickness then starts to vary with angular location around the tube until steady state is finally reached for the given set of experimental conditions.

The ice shapes, at mid length, at steady state for all Reynolds numbers tested for similar Θ_c (runs 1-5) were measured separately for both tubes using contour gages. All of the recorded ice shapes are superimposed in Fig. 6. Fig. 6 compares the ice shapes obtained from bare tube (lower figure) and the tube fitted with HCED (upper figure) at a constant cooling temperature ratio, $\Theta_c \approx 1.64$ and different Reynolds numbers. The ice shapes obtained here in our experiments for all Re numbers are in accordance with the

description of the ice shape, first reported by Okada et. al (1978) and later by Cheng et. al (1981) as follows; the shape from the front stagnation point to about 120° is an eccentric circle of which the center is shifted backwards in the direction of the flow from the center of the cooled pipe, and the ice shape of the back side forms a concentric circle with the pipe. More importantly, comparing the experimental ice shapes, at fixed Re, observe that the tube with the HCED has significantly greater ice volume.

Fig. 6: Ice shapes at steady state for different Reynolds number at $\Theta_c \approx 1.64$; Top with HCED, Bottom without HCED.

Fig. 7a: Digital photograph of the ice shape at $\theta_c = 1.41$ and $Re = 1188$ (run # 5) showing minimum ice build.

Fig. 7b: Digital photograph of the ice shape at $\theta_c = 4.09$ and $Re = 197$ (run # 6) showing maximum ice build.

Figures 7a and 7b are digital photographs of the steady state ice shapes for the representative “minimum ice volume” and “maximum ice volume” experimental conditions, runs number 5 and 6 respectively. Note that the ice build is constrained within the HCED in run 5 (Fig. 7a) but grew outside the HCED in run 6 (Fig. 7b).

5.2 Ice Volume

The total amount of ice volume obtained in the case of bare tube and the tube with HCED are plotted as a function of Reynolds number, Re , and cooling temperature ratio, Θ_c , in Fig. 8 and Fig. 9, respectively. It is clear that the ice volume is greater in the tube with HCED than bare tube for identical operating conditions. Because of the higher effective thermal conductivity in the ice build, the thermal resistance due to conduction decreases in the case of the tube with HCED as compared to the bare tube. This in turn increases the ice volume as explained earlier in the discussion of Fig. 2. The degree of the enhancement of the ice volume decreases as a function of increasing Reynolds number as depicted in Fig. 8. This is because at higher Reynolds number the thickness of the ice build is less, and the corresponding conduction resistance is relatively smaller (as compared to the total of all the thermal resistances), so therefore, reducing it by using HCED has less impact. On the other hand, better ice formation augmentation using HCED is obtained at high cooling temperature ratio, Θ_c (Fig. 9). Here, larger Θ_c means more ice build and larger conduction resistance and therefore the HCED is better able to improve the ice growth volume.

Fig. 8: Ice volume as a function of Reynolds number for both bare tube and tube with HCED for nearly constant Θ_c (runs 1-5).

Fig. 9: Ice volume as a function of Θ_c for both bare tube and tube with HCED for constant Reynolds number (run # 1 & 6).

5.3 % Enhancement of Ice Volume and Comparison to Theoretical Model

The % enhancement of the ice volume is defined as,

$$\% \text{ enhancement} = \frac{V_{HCED} - V_{bare}}{V_{bare}} \times 100 \quad (25)$$

and is examined more closely in Figs. 10-12 from both the experimental data and the theoretical model point of views. The variation of the % enhancement of ice volume with the Reynolds number (Fig. 10) exhibits a local maximum. In general the ice volume is greater for lower Reynolds number and decreases with increasing Reynolds number, but the % enhancement shows a local maximum between $Re = 300$ to $Re = 600$. Note that there is a significant deviation between the experimental results and the mathematical model predictions, but however, both show a local maximum. The deviation between the experiment results and the mathematical model at higher Reynolds number can be attributed to the mathematical model's inability to take into account the change in the external convective heat transfer coefficient when the ice growth is constrained within the HCED. As explained in the previous section, the ice volume decreases with increasing Reynolds number. For $Re > 500$, the ice growth is less and it does not grow outside the HCED. In fact the ice growth is constrained totally within the HCED (similar to Fig. 7a). As a result of this, the HCED is divided into two regions, the inner region embedded in the ice and the other region outside the ice build. The portion of the HCED without any ice now acts as a large number of external fins for convective heat transfer with the fluid stream. This affects the outer film coefficient around the ice layer and increases the convective heat gain to the ice formation. Since Okada's correlation for h that is used as a part of the mathematical model (Eq. 6) doesn't include the extended surface around the

ice, therefore the mathematical model over predicts the ice growth. Because of this fact the experimental curve lies below the theoretical curve in Fig. 10 for $Re \geq 500$.

Fig. 10: % enhancement of ice volume as a function of Reynolds number at $\Theta_c \approx 1.64$.

The experimental results and the theoretical predictions for % enhancement compare much better for Reynolds number less than 400. In this case the ice always grows outside the HCED (see Fig. 7b) and therefore the uncertainty in the external h value due to extended surfaces discussed above is avoided. The mathematical model here predicts the % enhancement at low Reynolds numbers much more accurately. The detailed theoretical predictions of r^* variation with dimensionless time, τ , at the lowest Reynolds number tested ($Re = 198$) along with the experimental steady state data (Run # 1) is shown in Fig. 11. Note here that since the ice build extends beyond the HCED that a step variation in k_{eff} was considered. At $r^* = 1.38$ the ice grows outside the HCED and the effective thermal conductivity changes. As shown in Fig. 11 solving the mathematical model for $k_{eff} = 1.684k_{ice}$ everywhere would over predict the ice growth. Solving it for $k_{eff} = k_{ice}$ everywhere, on the other hand, would under predict the ice growth or, more precisely, predict the ice growth volume without the HCED. Using variable $k_{eff}(r^*)$ values, ($k_{eff} = 1.684k_{ice}$ for $r^* < 1.38$ and $k_{eff} = k_{ice}$ for $r^* > 1.38$) in the mathematical model is necessary to accurately predict the ice volume in this case (or whenever the ice build size exceeds the actual dimensions of the selected HCED device installed).

Fig. 11: Transient ice growth curves predictions for variable $k_{\text{eff}}(r^*)$ values ($k_{\text{eff}} = 1.684k_{\text{ice}}$; $k_{\text{eff}} = 1.684k_{\text{ice}}$ for $r^* < 1.38$ and $k_{\text{eff}} = k_{\text{ice}}$ for $r^* > 1.38$; and $k_{\text{eff}} = k_{\text{ice}}$). a) Large dimensionless time showing comparison to steady state experimental data; b) Theoretical transient ice growth curves for variable k_{eff} at early time.

The % enhancement of ice volume plotted as a function of cooling temperature ratio at $Re = 198$ (runs # 1,6) is shown in Fig. 12. Observe that the experimental results closely match with the theoretical predictions since the HCED is within the ice build at these Θ_c and Re values. Note that the experimental results and theoretical predictions show only a slight % enhancement increase with Θ_c beyond $\Theta_c > 2$ (85% $\rightarrow$ 88%), since it appears that the relative size of the HCED installed is too small (see Fig. 7b) for such high Θ_c operating values. (It is believed that greater % augmentation may have been achieved using a larger HCED.)

Fig. 12: % enhancement of ice volume as a function of cooling temperature ratio, θ_c at $Re = 198$.

In closing, note that our original intent was to vary Θ_c even further, up to $\Theta_c = 10$ in the experiments, but it was found that we couldn't collect data for $\Theta_c > 4.09$ since at higher Θ_c values the ice growth from the bare tube and the tube fitted with HCED grew together (see Fig. 13). Hence the individual ice volume from bare tube and tube with HCED could not be discerned, although it appears that the larger top half of the ice build and asymmetry of the ice shape does in fact imply ice volume improvement due to the HCED.

Fig. 13: Ice shapes at $Re=189$ and $\Theta_c = 9.66$ showing ice linking.

6. CONCLUSIONS AND FUTURE DIRECTIONS

HCEDs improve the thermal performance and competitiveness of external ice-on-tube TES systems. To the best of our knowledge no previous experimental work on thermal performance augmentation of TES systems using HCEDs of this type (porous mesh) has been reported in the literature. A mathematical model was developed that showed that the rate of freezing and the steady state ice volume of a TES system can be increased by increasing the effective thermal conductivity of the ice layer by using inexpensive HCED made from rolled copper screen mesh. The higher value of the effective thermal conductivity reduces the controlling thermal conduction resistance of the ice layer, and hence improve the overall thermal performance of the TES ice system. The thermal performance of the tube fitted with HCED was quantitatively compared against that of the bare tube in the laboratory. Experiments showed that 50% - 90% enhancement in the final steady-state ice volume can be achieved by using the porous copper mesh as shown. Since the effective thermal conductivity of the HCED and the ice matrix is a function of the porosity and the packing arrangement, we believe that the thermal performance can be improved further by varying the mesh properties (i.e porosity, wire and layer thicknesses) of the HCED. Therefore further research is warranted to determine the optimum design of the HCED to maximize the ice build enhancement for different operating conditions.

REFERENCES:

- Alexiades, V. and Solomon, A.D. 1993. *Mathematical Modeling of Melting and Freezing Processes*. Hemisphere Publishing Corporation, Washington, DC, USA.
- Batchelor, G. K. and O'Brien, R. W. 1976. Thermal or Electrical Conduction Through a Granular Material. *Proc. Roy. Soc (London)* A355:313-333.
- Cheng, K. C., Inaba, H., and Gilpin, R. R. 1981. An Experimental Investigation of Ice Formation around and Isothermally Cooled Cylinder in Crossflow. *J. Heat Transfer* 103:733.
- Chiang, H. and Kleinstreuer, C. 1989. Solidification Around a Cylinder in Crossflow. *Int. J. Heat and Fluid Flow* 10 (4):322.
- Dorgan, C. E. and Elleson, J. S. 1993. Design Guide for Cool Thermal Storage. ASHRAE. Inc.
- Douglas, W. J. M. and Churchill, S. W. 1956. Recorrelation of Data for Convective Heat Transfer between Gases and Single Cylinders with Large Temperature Differences. *Chem. Eng. Prog. Symp* 52:23-28.
- Hirata, T., Ando, S. and Ishikawa, M. 1993. Effect of Heat Conductor Plates on Ice Formation near a Wall. *Proceedings of the 4th International symposium on thermal Engineering and sciences for cold region* 65-70.
- Intemann, P. A. and Kazmierczak, M. 1997. Heat Transfer and Ice Formations Deposited Upon Cold Tube Bundles Immersed in Flowing Water –I. Convection Analysis. *Int. J. Heat Mass Transfer* 40(3):557-572.

- Intemann, P.A. 1996. Steady and Transient Forced Convective Freezing Experiments on Subcooled Tube Bundles in a Cross Flow of Water, Ph.D. Dissertation, University of Cincinnati, Cincinnati, Ohio.
- Kazmierczak, M. and Intemann, P. A. 1997. Heat Transfer and Ice Formations Deposited Upon Cold Tube Bundles Immersed in Flowing Water –I. Conjugate Analysis. *Int. J. Heat Mass Transfer* 40(3):573-588.
- Khan, J.A. and Morehouse, J.H. 1994. A Metal Matrix for Heat Transfer Enhancement during Phase-Change Processes. *ASHRAE Transactions*, 100(II): 707-714.
- Lock, G.S.H. 1994. *Latent Heat Transfer- An Introduction to fundamentals*. Oxford University Press, UK.
- Lunardini, V., 1991. *Heat Transfer with Freezing and Thawing*. Elsevier Science Publishers, B.V.
- Okada, M., Katayama, K., Terasaki, K., Akimoto, M. and Mabune, K. 1978. Freezing around a Cooled Pipe in Crossflow. *Bull. JSME* 21(150):514.
- Saito, A., 2002. Recent advances in research on cold thermal energy storage. *International Journal of Refrigeration* 25(2):177-189
- Sasaguchi, K., Yoshida, M. and Nakashima, S. 1986. Heat Transfer Characteristics of a Latent Heat Thermal Energy Storage Unit with a Finned Tube (Effects of Fin Configuration), *Bull. JSME* 29 (255):2986.
- Sparrow, E. M., Larson, E. D. and Ramsey, J. W. 1981. Freezing on a Finned Tube for Either Conduction-Controlled or Natural –Convection-Controlled Heat Transfer. *J. Heat and Mass Transfer* 24:273-284.

Tong, X., Khan, J. A. and Amin, M. R. 1996. Enhancement of Heat Transfer by Inserting a Metal Matrix into Phase Change Material. *Numerical Heat Transfer, Part A*, 30:125-141.

Vilet, G. C. and Leppert, G. 1961. Forced Convection Heat Transfer from an Isothermal Sphere to Water. *J. Heat Transfer* 83:163-175.