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EXPERIMENTAL STUDY OF SEVERAL ICE HEAT SINK CONCEPTS

J. Stubstad and W. Quinn

June 1974

PREPARED FOR
U.S. ARMY ENGINEER POWER GROUP
RESEARCH AND TECHNOLOGY DIVISION
BY
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HANOVER, NEW HAMPSHIRE

The findings in this report are not to be construed as an official Department of the Army position unless so designated by other authorized documents.

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PREFACE

This report was prepared by John M. Stubstad and William F. Quinn (Chief), Northern Engineering Research Branch, Experimental Engineering Division, U. S. Army Cold Regions Research and Engineering Laboratory (USACRREL). The work was performed for the Engineer Power Group, U. S. Army Corps of Engineers, under Intra-Army Order No. 32017 (USA MERDC).

The design of the model heat sink and its appurtenances was formulated by Roscoe Perham who also provided helpful advice during the test. Dr. Yin-Chao Yen, Chief, Physical Sciences Branch, Research Division, assisted greatly in formulating experimental procedures and provided consultation during the experiment. The test series involving most of the ice cube and ice block experiments were conducted by Joseph Galate; his efforts provided the basis for analysis of most of this work.

We are indebted to many people in the Plant and Equipment Office, particularly L. Bogie and A. Goerke for their very valuable contributions in putting the experimental components together into a working system. The assistance provided by the Refrigeration Section in connection with the ice manufacturing operation is greatly appreciated. The report was technically reviewed by Roscoe Perham and Donald Haynes.

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NOMENCLATURE

Symbol	*Definition*	*Units*
A	Area	ft^2
c_p	Specific heat	Btu/lbm °F
D	Diameter	ft
h	Heat transfer coefficient	Btu/hr ft^2 °F
i	Subscript referring to ice	
k	Thermal conductivity	Btu/hr ft °F
L	Latent heat of fusion	Btu/lbm
M	Mass	lbm
Nu	Nusselt number	
Q	Rate of heat transfer	Btu/hr
Q_t	Total heat storage capacity	Btu
R	Radius	ft
T	Temperature	°F
t	Time	hr
W	Mass flow rate	lbm/hr
w	Subscript referring to water	
ρ_i	Density of ice	lbm/ft^3
θ	Time	hr

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by

John M. Stubstad and William F. Quinn

INTRODUCTION

The selection and design of a system to store waste heat associated with the generation of power for hardened defense installations requires solution of a variety of engineering problems. The requirement that the heat sink be fully hardened precludes the use of such conventional approaches as cooling towers or water reservoirs on the surface. Instead, all the waste heat must be contained and stored underground for a specified period of time.

A number of research studies have been conducted to explore various methods of waste heat storage that would be compatible with the overall requirements of the installation. These studies[1-10, 13] have considered the use of water, chilled water, chilled brine, rock, ice and soil as heat sink materials. The study reported here is an extension of a previous study of ice/water heat sinks[2]. That work involved an analytical and laboratory model study of an ice heat sink system that was promising because of the large amount of heat associated with the phase change from ice to water. Such a concept can considerably reduce the amount of excavation required for chilled water heat sink systems. Another advantage of the ice sink is that the coolant water temperature tends to remain at a relatively constant low level for a prolonged time during the early stages of the heat rejection cycle, thereby permitting more efficient power generation during the critical initial period when the installation is first buttoned-up.

The previous studies of ice/water heat sinks had considered solid ice cylinders exclusively in order to maximize the heat storage per unit volume of excavation. But the U.S. Navy had considered using an ice/water slush as the heat sink medium. When the Corps of Engineers Power Group was requested by the Navy to conduct model heat sink tests using the CRREL laboratory apparatus it was decided that the heat rejection rates and coolant water flow rates used during the solid ice cylinder tests should be repeated using a variety of ice sizes ranging from 0.1-lbm cubes to 20-lbm blocks. This would permit comparisons over a substantial range of ice particle sizes.

The solid ice cylinder tests had resulted in sink outlet water temperatures which varied between 34° and 50°F while ice was still present in the sink. Since the condenser unit operates at maximum efficiency when its inlet water temperature is between 40° and 44°F, it is desirable to maintain the outlet temperature in this range for as long as possible. Although an ice/water slush heat sink would have less latent heat capacity per unit volume due to its porous structure, it was thought that as the heated coolant water was introduced at the top of the sink it would tend to stratify. Water reaches its maximum density at 39.2°F, and the sink outlet temperature should be either close to or at this temperature. This might induce a favorable situation by maintaining an essentially constant sink outlet temperature for a relatively long time. As ice

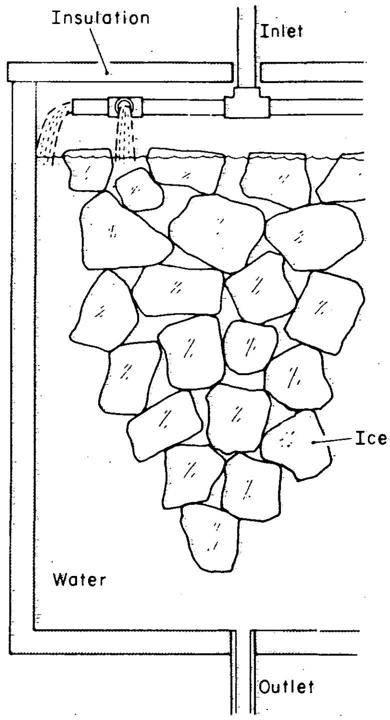


Figure 1. Typical ice block configuration.

melted at the top of the sink, a replacement volume of ice would float up and isolate the lower region of the sink, which would be the source of the coolant water (Fig. 1). Also, whereas the solid ice cylinder required initial melting at the periphery to develop an annulus for coolant water flow, the ice/water matrix would always afford many continuous flow channels through the sink. The ice for this matrix was to be manufactured by a conventional ice-making plant which would periodically add new ice to the sink to replace the small amount that had melted.

This report presents the results of the following studies: 1) comparison of data from ice cube, ice block and solid ice cylinder tests, 2) analysis of the effect of heat rejection and coolant water flow rates on heat sink performance, 3) effect of the inlet header on melting patterns, and 4) additional solid ice cylinder tests for very low and intermediate coolant water flow rates.

DESCRIPTION OF EXPERIMENT

Apparatus

Figures 2 and 3 show the heat sink test apparatus. The equipment, with only minor changes, was the same as that used in the previous experiment with solid ice cylinders[2]. The sink is a 6-ft-high, 4-ft-diam steel tank. It is cooled by circulating brine through about 200 ft of ⅝-in.-OD copper tubing wrapped around the outside of the tank. The tank is insulated by 3½-in.-thick roll-type fiberglass on the vertical surfaces and 3-in.-thick Styrofoam sheets on the bottom. Black polyethylene sheeting over the fiberglass provides a vapor barrier. The top of the sink is covered with a 48-in.-diam plywood cover which acts as an insulator and instrument support. Two Plexiglass windows and an access panel allow visual examination of the melting of the upper ice surface. Two viewports on the sides of the tank allow the vertical melting process and ice movement to be observed. When not in use, all viewports and access panels are covered with insulation.

For the ice cube and ice block tests, the water in the sink was cooled by a portable refrigeration system. This unit contained an ethylene gylcol - water solution and was able to provide a minimum refrigerant temperature of -39° to -35°F. For the solid ice cylinder tests, the CRREL -73°F (nominal) trichloroethylene system was used to freeze the sink; this considerably expedited the freezing operation.

The thermocouple system of the previous study was modified in order to measure temperature stratification within the sink. Strings of thermocouples were placed along the centerline of the tank and at the ½ and ¾ radius positions (Fig. 4). Thermocouples were monitored by an L&N 5905-NSL potentiometer as a check. One thermocouple (not shown in Figure 4) was kept in an ice/water bath to provide a calibration reference temperature during the test.

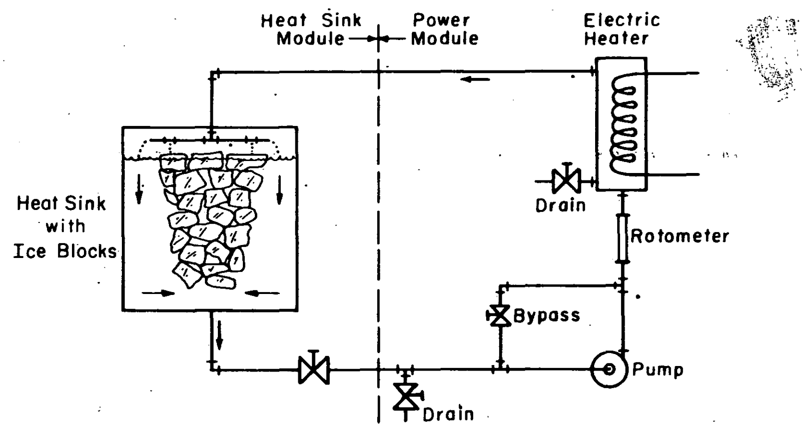


Figure 2. Schematic of heat sink experiment.

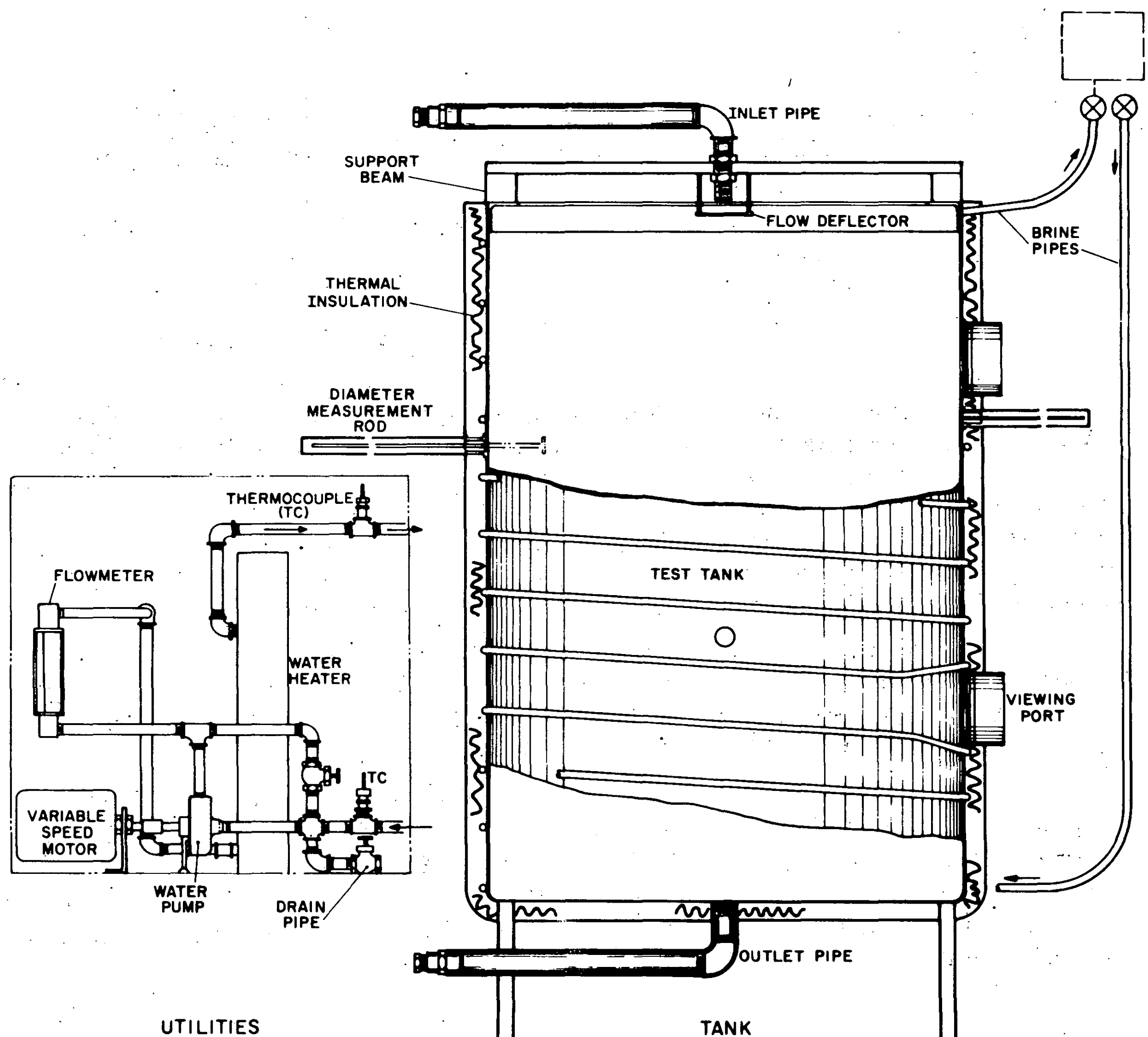


Figure 3. Model heat sink.

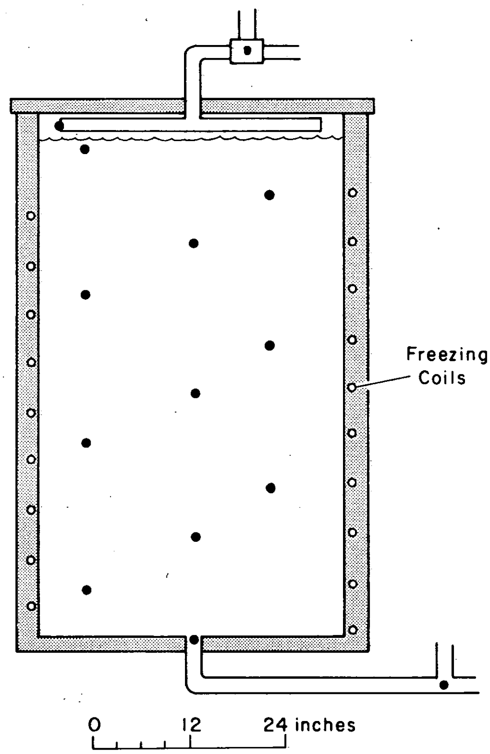
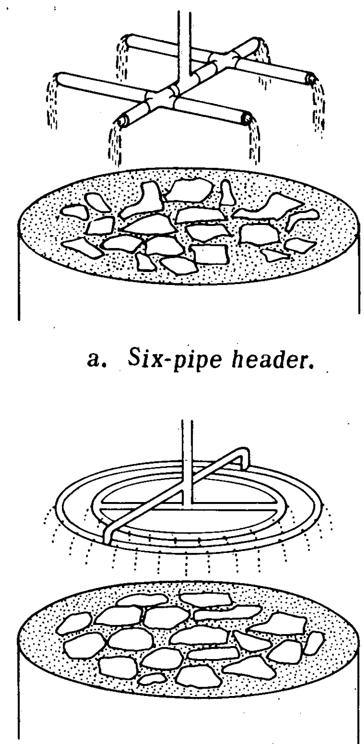


Figure 4. Location of thermocouples.



a. Six-pipe header.

b. Spray type header.

Figure 5. Inlet headers.

During an experiment, water is continuously drawn from the bottom of the sink and pumped through a 6 kw electric immersion heater which is used to simulate the condenser module. The heated water is then reintroduced to the sink at the top of the tank. A rotary displacement pump driven by a ½ hp variable speed DC motor was used to circulate the water through the heater, water flow rates were checked by a 0.5-6.0 gpm rotameter. The heater was controlled with a variac transformer. Dependent upon the frequency with which the flow rate and heater output were checked it was possible to provide heat rejection rates within 3% of the nominal test values.

All exterior plumbing was insulated by fiberglass covered with aluminum foil. Based upon various calibration tests, it was estimated that the average heat flux into the sink from the surroundings was on the order of 600 to 700 Btu/hr; or that the sink had an overall heat transfer coefficient of approximately 0.17 to 0.18 Btu/hr ft^2 °F.

During this test series, two different inlet header systems were used. To provide data comparable to those acquired from the solid ice cylinder tests previously reported, most of the tests were run using a six-pipe system (Fig. 5a). In order to examine the effect of inlet water distribution on heat sink performance, two tests were run using a spray inlet header (Fig. 5b). This header distributed the inlet water more evenly over the upper surface, producing a more uniform melting pattern.

Procedure

Ice Cube/Ice Block Tests

The use of ice cubes and ice blocks required that the ice be made and then loaded into the sink rather than frozen in the sink as in the solid ice cylinder tests. To prevent damage to the tank and cracking of the ice, a special loading procedure was developed. Before ice was loaded into the tank, 1-in. aluminum conduit was slipped over the thermocouple strings. By using retaining plates to locate the conduit, it was possible to protect the thermocouples and at the same time keep them from shifting position in the sink during loading. After loading was completed, the conduit was removed and the ends of the thermocouple strings were securely fastened to the tank lid. Although some thermocouple movement occurred as a result of shifts in the ice mass, the net changes in position were insignificant.

A layer of water 29 in. deep cushioned the impact of the falling ice and prevented damage to the tank. In the later stages of the loading process some cracking and chipping of the ice did occur due to the impact of the falling blocks. On the average, only 10 to 15 of the 105 blocks loaded suffered any major fracture during loading.

To retard melting of the ice during loading, the water cushion was initially cooled to 32°F by the cooling coils wrapped around the tank. Since most of the tests were conducted with the water table at the upper ice surface (that is, with all the ice floating), it was necessary to add water after all the ice had been loaded. The large amount of water required to do this precluded the possibility of using tempered water. Since the temperature of the cold water servicing the laboratory ranges from a mid-winter low of 40°F to a high in late autumn of 60°F, the total starting mass of ice must be corrected to account for the sensible heat in this water. In addition, the sink was allowed to stand for at least two hours between the termination of loading and the start of the test; this permitted the ice-water matrix to reach an equilibrium temperature.

A Scotsman model SC500HA-4 ice cube maker made the ice for the ice cube tests. It produced 0.056-lb, roughly cylindrical ice cubes at a rate of approximately 600 lb/day. The size of the ice storage bin required that the machine be emptied once a day; the ice was then stored in a coldroom. When enough ice had been produced, it was removed from the coldroom and allowed to stand at room temperature to temper it from its coldroom temperature of 25°F to the desired test temperature of about 32°F.

Two procedures were used to produce the 20-lb ice blocks. Initially, 300-lb ice cakes were purchased and the cakes were cut into blocks with a chain saw. As this procedure was found to be tedious, another process for producing the blocks was developed by using the CRREL coldroom facilities. About 50 cubically shaped wooden boxes with 8¾-in. sides were lined with plastic and then filled with 22 lb of water. The boxes were placed in a 0°F coldroom for 1½ days. To facilitate removal of the ice blocks from the boxes, the boxes were allowed to stand at room temperature until the surfaces thawed. After removal, the blocks were stored in a 25°F coldroom to temper and preserve them until the required 105 blocks had been frozen.

Solid Ice Cylinder Tests

Before the ice cylinder test sequence could be started, two modifications to the tank were necessary. An air bubbling system was installed to prevent cracking of the viewports during the freezeup period and also to delay freezing at the center of the tank by providing a continuous path to a free surface. The path prevented entrapment of liquid water by ice and the development of high stresses associated with supercooled water. Also, an anchoring system was installed to keep the ice cylinder from moving during the test. The anchor consisted of several boards tied to a rope which was positioned along the vertical centerline of the tank. The rope was fastened to a tie-down anchor at the bottom of the tank and to the plywood covering at the top. In these tests the aluminum conduit was not needed to shield the thermocouples; they were merely allowed to freeze into the ice. The anchor system prevented almost all motion of the ice during the test.

Using CRREL's -73°F brine system, the solid ice cylinder was completely frozen in about four days. The average temperature gradient in the ice at the end of the freezeup period was -30°F at the tank wall to 32°F at the center of the cylinder. To decrease this gradient, the sink was allowed to remain at room temperature for at least ½ day. When the wall temperature of the ice reached 20°F, the seven 450-watt heating tapes which were wrapped around the outside of the tank were turned on. Approximately ten hours were required for the heating tapes to melt a ½-in. annulus. After the tapes were turned off the start of the test was delayed until the average reading of all the thermocouples was between 25° and 30°F and the minimum temperature was above 20°F. The delay period varied between 2 and 10 hours for the four solid ice tests and was a direct function of the temperature variation in the ice at the end of the freezeup period.

SUMMARY OF EXPERIMENTAL TESTS

Because a wide variety of tests were conducted under this experimental program, it was felt essential to develop a system for organizing and identifying the tests in a manner which would simplify a comparative analysis. The primary variables considered were:

1. Ice type (cubes, blocks, solid cylinders)
2. Inlet header configuration (six-pipe inlet, spray header)
3. Nominal heat rejection rate (8,402, 16,805, 19,105 Btu/hr)
4. Coolant water flow rate (0.65, 1.00, 1.89, 3.00, 4.00 gpm)

The symbolic test designation used in this report is explained below:

For a test number NX_i, N is the test number, which indicates the nominal heat rejection rate and coolant water flow rate. This designator does not indicate the actual sequence in which the tests were performed. The key to the meaning is:

N	Nominal heat rejection rate (Btu/hr)	Coolant water flow rate (gpm)
1	16,805	1.89
2	16,805	1.00
3	16,805	4.00
4	16,805	3.00
5	16,805	0.62
6	8,402	1.89
7	8,402	0.66
8	19.105	1.89
9	19,105	4.00
10	19,105	1.00

X is a one- or two-digit letter code indicating the type of ice and the type of header used:

	Ice type	Inlet header
S	Cylinder	Six-pipe
B	Blocks	Six-pipe
C	Cubes	Six-pipe
CV	Cubes with air voids	Six-pipe
CS	Cubes	Spray

i is a subscript used to distinguish between two tests having the same nominal heat rejection rate, flow rate and type of ice.

Because test numbers had been assigned to the solid ice cylinder tests previously discussed[2] those tests were renumbered using the symbolic test designations of this report. The following reference table compares the old and new designations.

Test (this report)	Test (ref. 2)	Nominal heat Rejection rate (Btu/hr)	Flow rate gal/min
$1S_1$	1	16,805	1.89
$1S_2$	2	16,805	1.89
2S	5	16,805	1.00
3S	4	16,805	4.00
6S	6	8,402.5	1.89
8S	3	19,105	1.89

A compilation of all tests conducted as part of the CRREL ice-water heat sink model studies is given in Table I. All these tests were conducted as part of this report series with the exception of the six tests listed above.

TEST RESULTS

Ice Cube Tests

Effect of Coolant Water Flow Rate

As discussed in the introduction, it was felt that the buoyancy of ice in water might create a buffer zone of ice cubes between the incoming heated water and the heat sink outlet. As the ice melted, the higher density water at a temperature close to 39.2°F was expected to form a reservoir at the bottom of the sink and yield relatively constant temperature outlet water. The natural buoyancy of the ice would ensure that the melting surface would always be replenished with ice and thereby maintain the buffer zone. As the rate of coolant water flow through the sink was increased, one would expect an increase in overall mixing of the inlet and outlet water. It was recognized that high water flow rates through the sink could tend to counteract this desired stratification and thus produce a more variable sink outlet temperature.

Table I. Ice/water heat sink studies
Tank size: 4 ft diameter, 6 ft high

Test No.	Flow rate (gpm)	Ice type	Actual heat rejection rate (Btu/hr)	Actual temp diff (°F)	Total mass of ice and water (lbm)	Total ice mass* (lbm)	Ratio Ice/Total mass
Nominal $\dot{Q}$ = 16,805 Btu/hr							
$1S_1$	1.89	Cylinder	16,720	17.71	4072	3805	93.4%
$1S_2$	1.89	Cylinder	18,320	19.40	4005	3955	98.8%
1B	1.89	Blocks	16,442	17.42	4185†	2080†	49.8%
1C	1.89	Cubes	16,763	17.76	4079	2122	52.0%
1CS	1.89	Cubes with spray inlet	16,597	17.58	3824	1381** (1717)††	44.9%
1CV	1.89	Cubes with air voids	16,678	17.77	2901	2297	79.2%
2S	1.00	Cylinder	15,200	30.39	3812	3319	87.1%
2B	1.00	Blocks	16,788	33.57	4200†	2100†	50.0%
2C	1.00	Cubes	16,761	33.51	4190	1834	43.7%
2CS	1.00	Cubes with spray inlet	16,644	33.28	3855	1443** (1789)††	46.4%
2CV	1.00	Cubes with air voids	16,637	33.27	3280	2072	63.2%
3S	4.00	Cylinder	16,350	8.17	3950	3442	87.1%
3B	4.00	Blocks	16,409	8.20	4170†	2100†	50.4%
3C	4.00	Cubes	16,648	8.32	4090	1790	43.7%
$4S_1$	3.00	Cylinder	16,848	11.23	4170	2920	68.9%
$4S_2$	3.00	Cylinder	16,776	11.18	4050	3542	87.5%
5S	0.62	Cylinder	20,032	64.6	3924	3494	89.1%
Nominal $\dot{Q}$ = 8,402 Btu/hr							
6S	1.89	Cylinder	8,425	8.92	4095	3695	90.2%
6B	1.89	Blocks	8,453	8.95	4200†	2100†	50.0%
7S	0.66	Cylinder	11,272	33.98	3754	3365	89.6%
Nominal $\dot{Q}$ = 19,105 Btu/hr							
8S	1.89	Cylinder	20,520	21.73	3842	3530	91.9%
8CV	1.89	Cubes with air voids	18,963	19.95	2915	2314	79.4%
9C	4.00	Cubes	19,101	9.49	3650†	2100†	57.5%
10CV	1.00	Cubes with air voids	17,983	35.77	2926	2321	79.3%

* All values are ice mass at start of test except where noted
† Approximate value only
** Ice mass at start of test (after some pre-test melting)
†† Initial ice load

Tests 1C, 2C and 3C show the effect of flow rate on ice cube heat sink performance for a nominal heat rejection rate of 16,805 Btu/hr and flow rates of 1.00, 1.89 and 4.00 gpm. The total ice mass for each test was 1834, 2122 and 1790 lbm respectively and the total water mass was 4190, 4079 and 4090 lbm respectively. Thus the variation was 332 lbm (18.5%) between maximum and minimum ice mass and 111 lbm (2.75%) between maximum and minimum water mass.

Figure 6 shows the measured sink outlet temperature-time curves for these tests. During the initial 16 to 18 hours of sink use, the flow rate has no measurable effect on the outlet temperature. A change in the rate of temperature increase appears to occur between the 12th and 14th hours. The more rapid rate of temperature change measured in the 4.00-gpm test after about the 16th hour is attributed to mixing action within the sink. Although a significant amount of ice

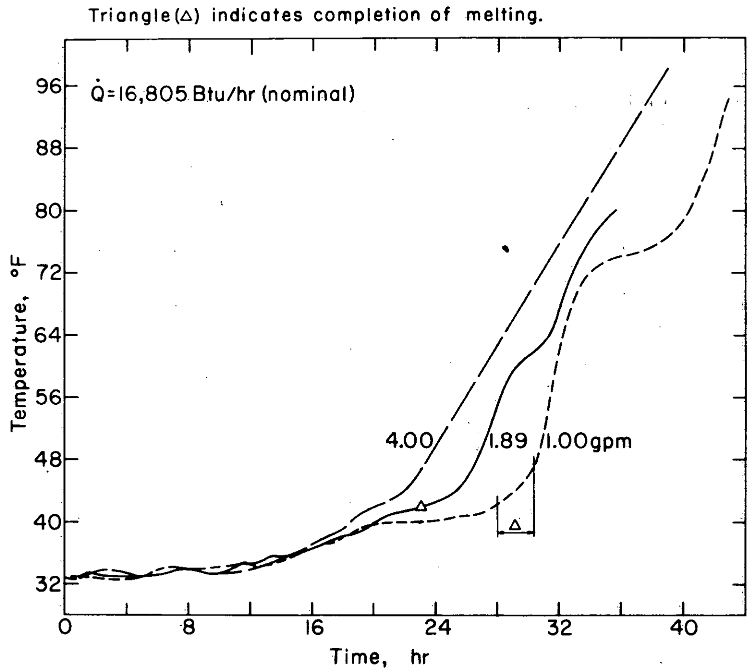


Figure 6. Effect of flow rate on ice cube heat sink outlet temperature, tests 1C, 2C, 3C.

still remained in the sink at the 16th hour, the inlet water had begun to mix with the lower temperature water at the bottom of the sink, which caused a gradual increase in its temperature. This result is felt to be due primarily to the coalescence of the remaining ice cubes into a cylindrical shape at the center of the sink. With the creation of this cohesive ice mass, the observed annular flow pattern was similar to that noted during the solid ice cylinder tests. The portion of heat rejected to the ice is decreased due to the lessened effective ice surface area and thus the outlet water temperature naturally rises. Although the ice also coagulates during the lower flow rate tests, the reduced flux of water through the sink increases the time that the high temperature inlet water is in contact with ice. Thus the reduction in the amount of heat rejected to the ice is less for the lower flow rate tests than for the 4.00-gpm test and subsequently the rate of increase in sink outlet water temperature is less.

During the 1.00-gpm flow rate test (2C), the flow process manifested itself as the net motion of convective water cells through the sink. After all the ice had melted, the temperature of the water at the top of the sink remained essentially constant for several hours. It is interesting to note that upon completion of this steady outlet temperature period, the outlet temperature suddenly increased by the temperature difference which was being maintained between the inlet and outlet. This occurrence can be noted between the 18th and 40th hours, during which time the rate of outlet temperature rise changed abruptly (also see Fig. A2c).

The 1.89-gpm test (1C) also experienced two steady temperature periods, although they were of shorter duration than in the 1.00-gpm test (see Fig. A1c). Again the outlet temperature suddenly increased by the temperature difference being maintained between inlet and outlet. This plateau temperature situation did not develop during the 4.00-gpm test.

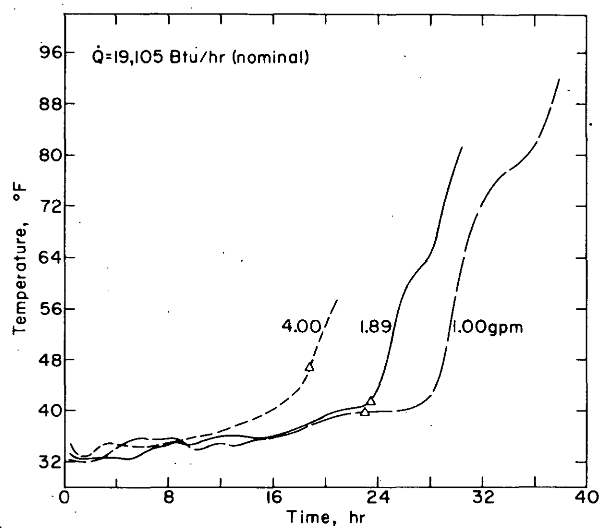


Figure 7. Effect of flow rate on ice cube heat sink outlet temperature (tests 8CV, 9C, 10CV).

The same type of behavior can be seen when the results of tests 8CV, 9C and 10CV are compared (see Fig. 7). Again the flow rates used were 1.89, 4.00 and 1.00 gpm respectively; however, a higher heat rejection rate (nominal 19,105 Btu/hr) was used. The total ice mass for each test was 2314, 2100 and 2321 lbm and the total water mass was 2915, 3650 and 2926 lbm. Although this comparison uses the results of both a standard ice cube test and two tests of ice cubes with air voids, the results of other tests which compared these two concepts indicated that the sink outlet water temperature was the same for both types of sinks during the ice melting period. (For more information, see the section on Effect of Air-Filled Voids.)

Again, for these three tests, there is no major flow rate effect on sink outlet water temperature during the initial hours of use. As for the 16,805-Btu/hr tests, the 4.00-gpm test indicates a gradual increase in the outlet temperature while ice is still present in the sink. The 1.00-gpm test exhibits increased low temperature life and subsequent step-like temperature increases after all the ice is melted. The intermediate flow rate of 1.89 gpm indicates the same step-like increases although the duration of these plateau temperature levels is less.

Effect of Heat Rejection Rate

Figure 8 shows the ice cube sink outlet temperatures for constant water flow rates but different heat rejection rates. The graphs are plotted using the results of the nominal 16,805-Btu/hr tests and the 19,105-Btu/hr tests for flow rates of 1.00, 1.89 and 4.00 gpm. For the 1.00- and 1.89-gpm tests, the increase in heat rejection rate has no significant effect on sink performance, particularly during the early stages. The actual increase in heat rejection rate was 7.3% for the 1.00-gpm tests, and 13.1% for the 1.89-gpm tests based on the time average heat rejection rates. The differences in outlet water temperature that did occur during the latter stages of the tests (after all the ice was melted) can be attributed to the different total amounts of water used in each test. For the 4.00-gpm tests, however, the increase in heat rejection rate did produce an increase in the sink outlet temperature. Again, the total water mass of the 19,105-Btu/hr test was less than that of the 16,805-Btu/hr test, and the average difference between all the 16,805-Btu/hr tests and the 19,105 Btu/hr tests was 1000 lbm of water. However, the greater mixing action of

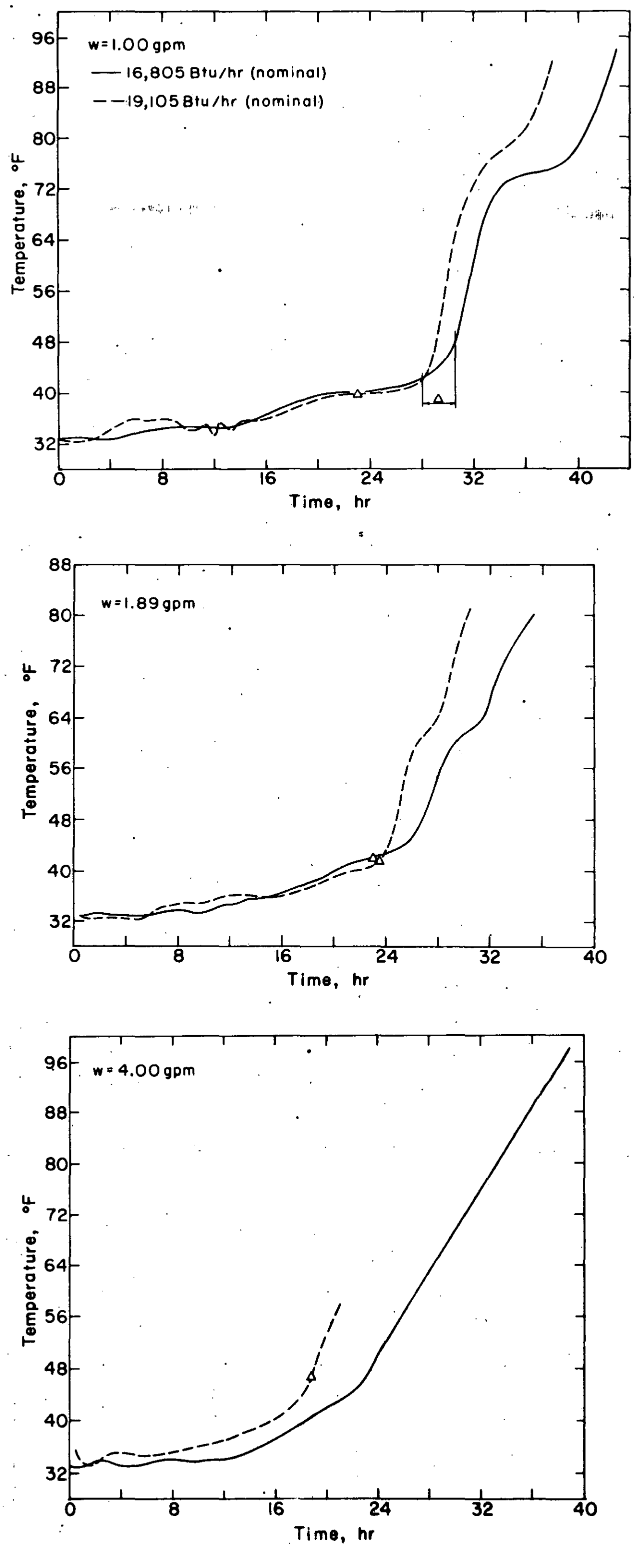


Figure 8. Effect of heat rejection rate on ice cube heat sink outlet temperature.

the 4.00-gpm test combined with the lower heat rejection rate resulted in a temperature difference between the two during the early stages. In the lower flow rate tests, the buffer zone of ice tended to isolate the water mass from the incoming heat, but in the high flow rate test, as discussed previously, this zone had a reduced buffering ability and thus the higher heat rejection rate produced higher outlet water temperatures.

It is concluded that the buffer action of ice cube heat sinks decreases as the water flow rate increases. To ensure maximum effectiveness of this concept the flow rate must be kept low, on the order of 0.5 to 1.0 gpm in the model. Using the rather simplified scaling study reported in Appendix A of ref 2, it is possible to estimate the magnitudes of the flow rates that would be required in a prototype sink. It must be noted that the scaling factors between the model and a 65-ft-diam, 110-ft-high prototype have yet to be validated and that the quantities shown below can only be used as a rough approximation. Based upon these scaled relationships, the flow rate should be kept below 300 gpm to ensure maximum effectiveness of the ice buffer zone action in a prototype ice cube heat sink.

Table II. Comparison of model and prototype sinks
Model: diameter 4 ft, height 6 ft
Prototype: diameter 65 ft, height 110 ft

$\dot{Q}$ *Model*	$\dot{Q}$ *Prototype*	*W Model*	*W Prototype*
8,402 Btu/hr	2.61×10^6 Btu/hr	0.66 gpm	205 gpm
16,805	5.22×10^6	1.00	310
19,105	5.96×10^6	1.89	590
		3.00	935
		4.00	1245

Effect of Air-Filled Voids

Of prime importance in the operation of an ice heat sink is the rate at which the ice absorbs the rejected heat. This rate of absorption (melting rate) determines the partitioning of the rejected heat between the ice and water in the sink and thus ultimately determines the relationship between the sink outlet water temperature and time. It is possible to vary this rate of absorption by either changing the surface area of the ice in the sink or by changing the heat transfer coefficient between the ice and water. The use of different ice shapes such as cylinders, blocks or cubes permitted assessment of the effect of surface area. By modifying the configuration of ice and water in the sink, it was possible to reject heat directly to the ice rather than indirectly via an ice/water matrix and thus change the coefficient of heat transfer.

At the interface of ice and water the transfer of heat and the flow of fluid in the boundary layer are complicated by the injection of meltwater, and by stratification in the bulk water. In the investigation of laminar natural convective melting of vertical ice plates, Vanier[11] found that a minimum in the heat transfer coefficient vs temperature relationship occurred at approximately the maximum density temperature of water. Tkachev[10], who studied turbulent natural convective melting of plates, also found the same phenomenon. For bulk water temperatures above 42.8°F they showed that increases in the bulk water temperature resulted in increases in the coefficient of heat transfer (see Fig. 9 and 10).

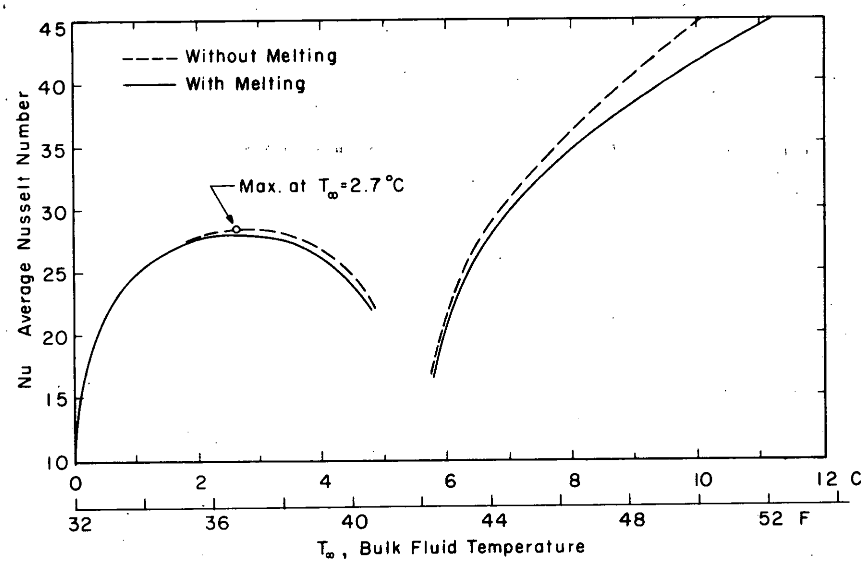


Figure 9. Average Nusselt values for 10.2-cm vertical plate (natural convection in water). (After Vanier[12].)

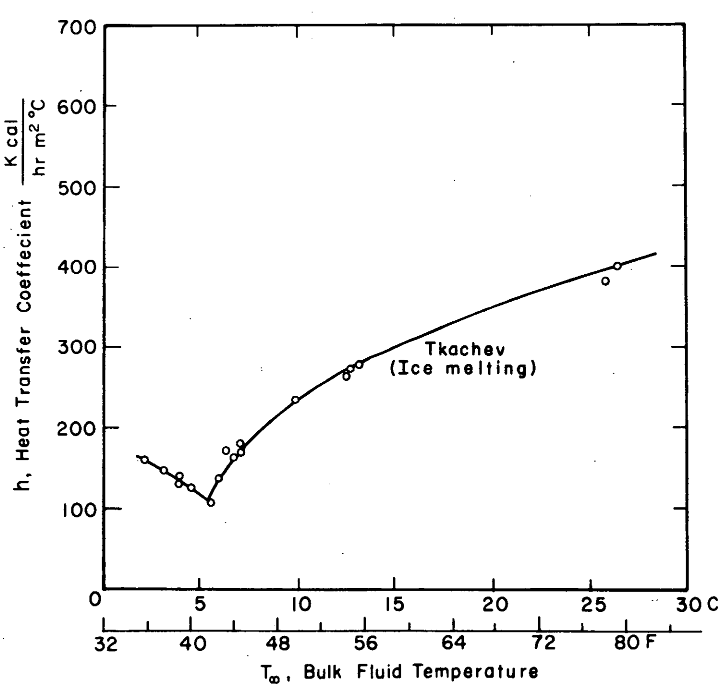


Figure 10. Correlation of the heat transfer coefficient with water temperature during melting of ice. (After Tkachev[10].)

In the ice/water matrix, the bulk water temperature is the temperature of the water surrounding the ice; this temperature normally averaged 10°F less than the inlet water temperature. If the inlet water were allowed to flow directly over the ice instead of into the ice-water matrix, it was reasoned that the "bulk water temperature" would essentially be the inlet water temperature and thus the heat transfer coefficient would be improved.

For this set of tests, the water table was kept well below the upper ice level so that a substantial zone of ice cubes was exposed directly to the inlet water. With the water table at approximately the midpoint of the tank there was an insufficient amount of water to "float" the ice, with the result that there were two distinct matrices, an ice/water matrix at the bottom and an ice/air matrix (ice cubes with air-filled voids) at the top.

Four tests were conducted using air-filled voids. Tests 1CV and 2CV were run at the standard heat rejection rate of 16,805 Btu/hr nominal and flow rates of 1.89 and 1.00 gpm respectively. Tests 8CV and 10CV were run at a nominal 19.105 Btu/hr and flow rates of 1.89 and 1.00 gpm respectively.

Figure 11 shows the sink outlet temperature-time curves for tests 2C and 2CV, each conducted at 1.00 gpm. The almost identical initial sections indicate that during this period of operation the ice is able to absorb almost the entire heat load rejected to the sink. The six-pipe inlet header was used for these tests; this resulted in the tendency for the falling inlet water streams to drill holes through the air void - ice mass. This reduced the "effective" surface area over which heat transfer occurred and tended to balance out the intended increase in the heat transfer coefficient. To take full advantage of this technique would require the random distribution of the inlet water over the ice surface, possibly by using a spray-type header.

A limitation to any serious consideration of an air void type of ice sink became obvious during the later hours of operation. After all the ice was melted there was less total water mass than would have been the case had the voids been filled with water. The reduced amount of water increased the rate of temperature change and thus decreased the overall period of useful operation.

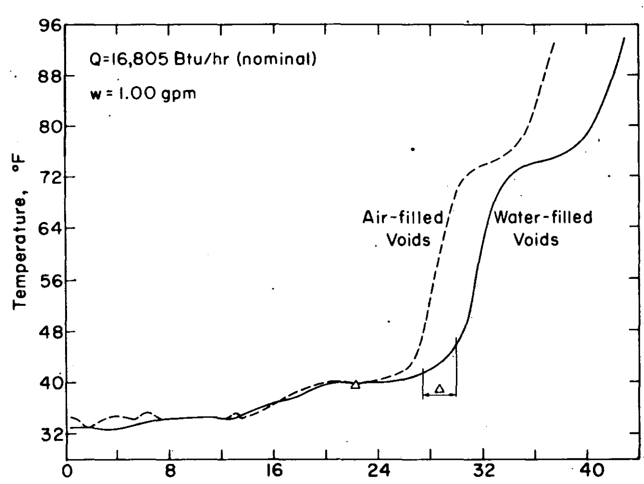


Figure 11. Comparison between outlet water temperatures obtained for air-filled and water-filled void tests (tests 2C, 2CV).

It is felt that this significant decrease in the useful life of the sink negates any possible improvement that could be made during the low temperature period of operation.

Effect of Inlet Header Configuration

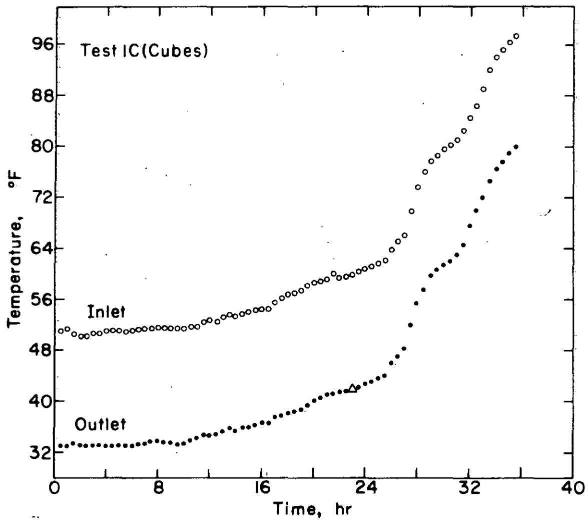
During both the ice cube and ice block tests, the ice tended to form a cohesive mass which exhibited the star-shaped melting pattern observed during the solid ice cylinder tests. The creation of this cohesive ice structure prevented development of a homogeneous ice/water upper surface; thus the desired buffer zone was not fully realized. This melting pattern is attributed to the configuration of the original inlet header (see Fig. 5a) which discharged the coolant water at six locations. These points of introduction served as the axes for the star melting patterns. When the standard header is being used the local flow through the sink varies from a turbulent type of flow in the vicinity of the inlets to a laminar flow within the sink. Since turbulent flow tends to prevent the density stratification process, it was impossible to fully develop the upper level buffer zone.

In order to increase the likelihood of creating a floating ice buffer zone at the top of the sink, a spray header was developed (see Fig. 5b) to provide uniform distribution of the inlet water over the upper ice surface. This would cause melting over the entire ice surface rather than solely at its periphery and would produce a reduced bulk flow rate of water past the ice. The spray header consisted of two concentric circular manifolds of $^{3}/_{8}$-in.-OD copper tubing with radii of 9 and 16 in. At 4-in. intervals along the circumference, $^{1}/_{16}$-in.-diam holes were drilled in the lower surface of the tubing to provide the spray outlets.

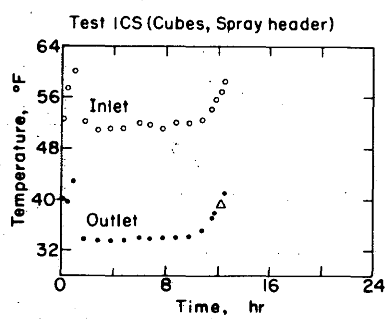
Two tests were conducted with this type of inlet header. The tests (1CS and 2CS) were run using a nominal heat rejection rate of 16,805 Btu/hr and flow rates of 1.89 and 1.00 gpm respectively. The results of these two tests were compared with those of tests 1C and 2C which used the same heat rejection rate and flow rates and the standard six-pipe header. However, a direct comparison is complicated by the different ice masses used in each test: 2122 lbm for test 1C and 1380 lb for test 1CS. This difference is attributed to the use of warm tap water for test 1CS which caused the initial ice charge of 1717 lbm to decrease to 1380 by the time the test was initiated.

The results of tests 1CS and 1C are given in Figure 12. The gradual rise in outlet temperature, particularly for test 1C after 10 hours, is indicative of the partitioning of heat storage between the ice and water. After a period of 26 hours, all of the rejected heat is being stored in the water, as evidenced by the marked increase in outlet water temperature. Test 1CS (which had about $^{2}/_{3}$ the ice mass of test 1C) indicates a very abrupt change in the outlet temperature - time curve at 10 hours, thereby reflecting that when ice was present in the sink essentially all of the rejected heat was used to melt it. Had the ice mass for test 1CS been more nearly the same as for test 1C, it is expected that the outlet temperature would have remained essentially constant at 34°F for about 15 hours, at which time it would have abruptly increased.

The results of these tests showed that the spray header did prevent the formation of a cohesive ice mass by promoting uniform melting over the upper ice surface. The header aided in producing density stratification of the water in the heat sink, resulting in essentially constant outlet water temperature while ice was still present. The distribution of the inlet water over the entire ice surface minimizes the rate of heat rejection to the water in the sink and thus results in a greater rate of melting.



a. *Six-pipe header*



b. *Spray header.*

Figure 12. Comparison between sink outlet temperatures obtained using different inlet header configurations (tests 1C, 1CS). Nominal $\dot{Q}$ = 16,805 Btu/hr, flow rate ≈ 1.89 gpm.

Ice Block Tests

A series of four ice block tests were conducted using blocks of a nominal 20 lbm. The general packing configuration resulted in a 50% ratio of ice to total melted mass. The ice to melted water ratios for the comparable ice cube tests were generally less (in the order of 44%); an exception was the 16,805 Btu/hr, 1.89-gpm ice cube test which had a ratio of 52%. Figure 13 depicts the outlet water temperature for the three ice block tests conducted at 16,805 Btu/hr.

For the 1.89- and 4.00-gpm flow rates, the time at which all ice had melted occurs when the slope of the outlet temperature curve begins to increase significantly. This sudden change in slope is what is to be expected when all of the rejected heat must be stored as sensible heat. However, the 1.00-gpm test did not exhibit this sudden outlet temperature rise upon completion of ice melting; rather it continued to maintain a low outlet water temperature at about 40°F, approximately the temperature of water at maximum density. This result is felt to be due to the density stratification of water which apparently occurs at the low flow rate of 1.00 gpm but which cannot be developed at the higher flow rates because of mixing effects. The total amounts of ice and melted water were essentially the same for each of these ice block tests.

It should be recognized that the rejection of heat at a constant rate but at various water flow rates necessitates use of varying temperature differences across the heat sink as indicated by:

$$\dot{Q} = Wc_p\Delta T .$$

The temperature difference was 8.4°F for the 4.00-gpm tests, 17.8°F for the 1.89-gpm test and 33.6°F for the 1.00-gpm tests (all values are nominal). Obviously the tests at higher temperature differences can store a larger amount of heat in the water and thus prolong the period of ice melting.

It is understood that a desirable temperature at the inlet to the condenser heat exchanger is in the order of 40°F, thus the low flow rate test results would appear to satisfy the design intent to provide relatively constant temperature water at close to 40°F for a relatively long period of time. The precise time at which all ice had melted in the 1.00-gpm test was not recorded; however, it is known that ice was present after 20 hours into the test and was not present at 23 hours. It is estimated that melting was completed at about 22 hours.

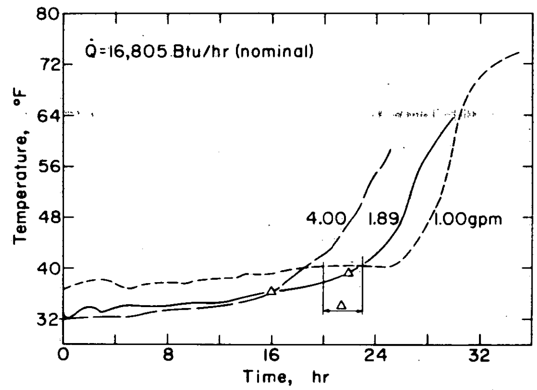


Figure 13. Effect of flow rate on ice block heat sink outlet temperature (tests 1B, 2B, 3B).

As reported in the discussion of the ice cube test results above, after all melting had taken place the outlet water temperature in the 1.00-gpm and 1.89-gpm tests exhibited a tendency to increase rapidly and then level off for a period before it rapidly increased again. This plateau temperature also developed for the 1.00-gpm ice block test at about the level of the inlet water temperature existing during the ice melting period (see Fig. A2b).

Ice Cylinder Tests

The test series previously reported[2] involved the use of three coolant water flow rates (1.00, 1.89 and 4.00 gpm). In order to make this earlier study more complete, it was decided that additional tests should be conducted at both a lower and an intermediate flow rate. The minimum rate was dictated by the operating and monitoring capability of the existing experimental apparatus. The following new tests were conducted:

Test No.	Nominal heat rejection rate (Btu/hr)	Coolant water flow rate (gpm)
$4S_1$	16,805	3.00
$4S_2$	16,805	3.00
5S	16,805	0.62
7S	8,402	0.66

The sink inlet and outlet water temperature-time curves for tests $4S_1$ and $4S_2$ are shown in Figure A3d; also included in this figure are the curves predicted using the previously developed computer program and the initial conditions of test $4S_1$. The results of previous ice cylinder tests tended to indicate that for constant heat rejection rates the sink outlet water temperature increased with an increase in the flow rate; however, test $4S_1$ at 3.00 gpm had a higher sink outlet temperature than test 3S at 4.00 gpm.

As indicated in Table I, both the ice mass (2920 lb) and ratio of ice mass to total mass (68.9%) for test $4S_1$ were significantly lower than those of the other tests; this complicated a determination as to whether the high outlet water temperature was a result either of the 3.00-gpm flow rate or the low initial ice mass. A review of tests $1S_1$ and $1S_2$ provided some insight as to the influence of the ice mass to total mass ratio; these tests had been conducted at ratios of 93.4% and 98.8% respectively. Each test was conducted at a flow rate of 1.89 gpm and at heat rejection rates of 16,720 and 18,320 Btu/hr. Although it had a higher heat rejection rate, test $1S_2$ had a slightly lower sink outlet temperature (see Fig. A1a). It was therefore decided to conduct another 3.00-gpm solid cylinder test ($4S_2$), but at an ice ratio of about 90%, which was comparable to the other tests conducted at 1.00, 1.89 and 4.00 gpm. Although there was a slight reduction in the observed outlet water temperature, the additional 3.00-gpm test still resulted in the highest sink outlet temperature. The variation in sink outlet water temperature for different flow rates is illustrated in Figure 14.

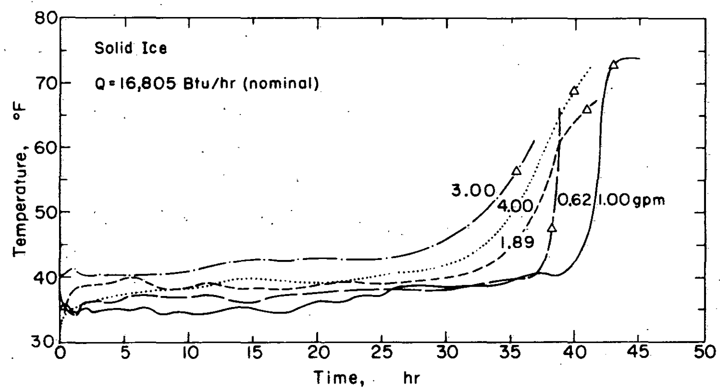


Figure 14. Effect of flow rate on solid ice cylinder outlet temperature (tests 1S, 2S, 3S, $4S_2$, 5S). Nominal $\dot{Q}$ = 16,805.

Test 5S was to be run at a heat rejection rate of 16,805 Btu/hr and a flow rate of 0.5 gpm but inaccuracies in the rotometer reading resulted in an actual flow rate of 0.62 gpm and thus an actual heat rejection rate of 20,032 Btu/hr. Unlike the previous solid ice tests in which melting had occurred predominantly in a radial direction, the direction of melting was from the top surface downwards. This resulted in the creation of a "reservoir" at the top of the tank which consisted of water at essentially the same temperature as the inlet water. Thus a significant amount of heat was stored as sensible heat in the water while a large amount of ice was still present in the sink. This phenomenon was basically the inverse of the ice buffer action sought in the ice cube tests in that it created a high temperature - low density "reservoir" at the top of the tank rather than a low temperature - high density zone at the bottom. The downward melting pattern permitted ice to remain in the proximity of the outlet for a long period of time, thus prolonging the period of low temperature outlet water. Figure A3e shows the inlet/outlet temperature-time curves for this test. The outlet temperature is shown in Figure 14.

Although test 5S had the lowest flow rate, it also had the highest inlet water temperatures, on the order of 97° to 105°F. To obtain the desired heat rejection rate at this low flow rate, it was necessary to maintain a 64.6°F temperature difference between the inlet and outlet water temperatures. Since the unusual melting pattern could have been the result of either the high temperature inlet water or the low flow rate, or a combination of these parameters, it was decided to conduct another low flow rate test using a smaller temperature difference across the sink (and consequently a lower heat rejection rate). The maximum temperature difference used in prior tests had been 33.6°F and since it had not produced this unusual melting pattern, it was decided to conduct one test using this temperature difference and a flow rate of approximately 0.6 gpm. Test 7S had a flow rate of 0.66 gpm and a temperature difference across the sink of 33.9°F, resulting in an average heat rejection rate of 11,272 Btu/hr. The inlet/outlet temperature-time curves for this test are shown in Figure A4c. As in test 5S, melting of the ice was confined almost entirely to the upper ice surface, indicating that this melting pattern was predominantly the result of the low flow rate used. Both tests 5S and 7S exhibited a sudden increase in outlet water temperature to the level of the inlet water temperature soon after all the ice had melted, indicating that the water flux through the sink was via density-controlled convective motion of water cells in the sink. This same phenomenon had been observed during the 1.00-gpm solid cylinder test. This step-like increase had also been noted during the ice cube and ice block tests.

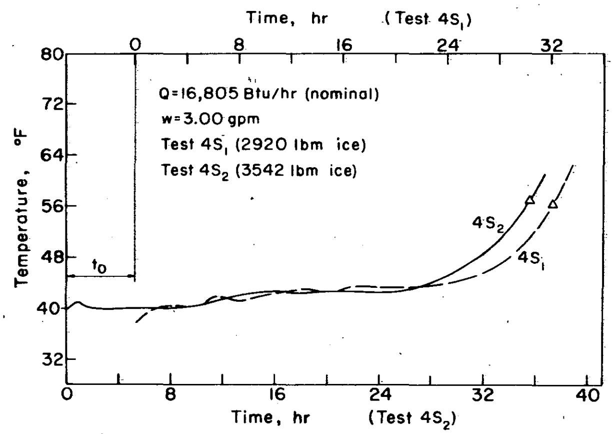


Figure 15. Effect of different initial ice masses on sink outlet water temperature (tests $4S_1$, $4S_2$).

A review of the summary curves in Figure 14 indicates that a maximum outlet temperature condition develops between flow rates of 1.89 and 4.00 gpm.

As noted previously an increase in the initial ice mass had resulted in a decrease in the sink outlet water temperature. This is illustrated by considering tests $4S_1$ and $4S_2$ which had initial ice masses of 2920 and 3542 lbm respectively. During analysis of these tests it was decided to compare the temperature-time curve for test $4S_2$, after its ice mass had reduced to 2920 lbm, with test $4S_1$. An approximation of the time lag t_0 required for test $4S_2$ to reach that ice mass can be deduced by assuming that all of the rejected heat was used to melt ice. This is calculated from:

$$t_0 = \frac{\Delta ML}{\dot{Q}} \text{, hr}$$

where ΔM = difference in the initial ice masses of the tests, lbm
L = latent heat of fusion of ice, Btu/lbm
$\dot{Q}$ = heat rejection rate, Btu/hr.

This approximation is acceptable when one considers that 144 Btu is required to melt 1 lbm of ice and only 1 Btu is needed to raise 1 lbm of water by 1°F. In test $4S_2$, during the ice melting phase, the average water temperature in the sink was about 45°F, indicating a sensible heat capacity of 13 Btu/lbm.

For tests $4S_1$ and $4S_2$, the difference in initial ice mass was 622 lbm and using the nominal heat rejection rate of 16,805 Btu/hr the calculated time lag was 5.3 hours. In Figure 15 the sink outlet temperature-time curves have been superimposed using the 5.3-hour time lag. During the ice melting period these curves become essentially identical. The deviation that did occur after all the ice had melted resulted from the different total amounts of water used in each test. The same adjustment was made for tests $1S_1$ and $1S_2$ (see Fig. 16). Since the initial mass difference was 150 lb, the time lag was only 1.3 hours. Again the curves are essentially identical during the melting phase. It can therefore be concluded that the outlet temperature - time curves for given coolant flow and heat rejection rates are unique and independent of the initial size of annulus as long as a time shift is made which provides for differences in the initial ice mass. It is therefore

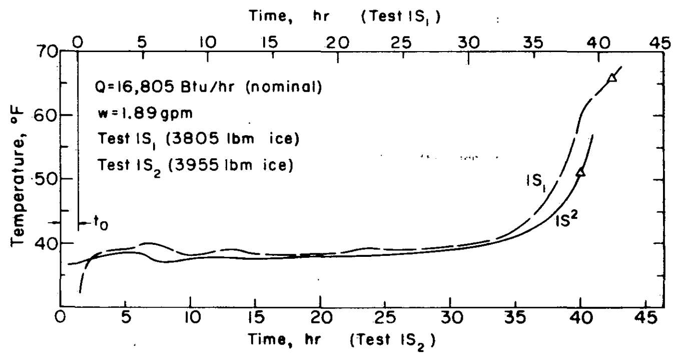


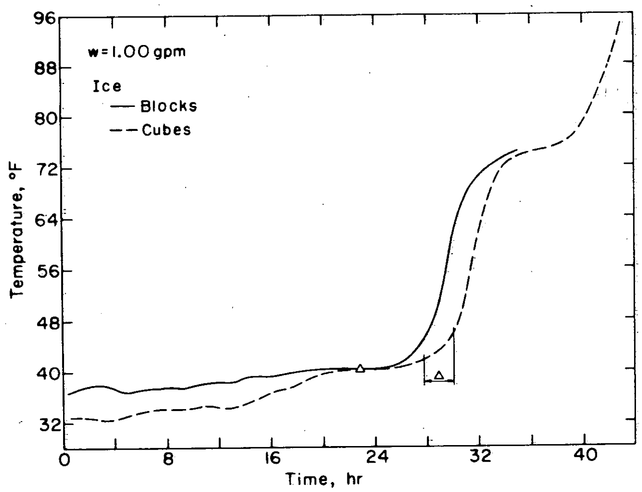
Figure 16. Effect of different initial ice masses on sink outlet water temperature (tests $1S_1$, $1S_2$).

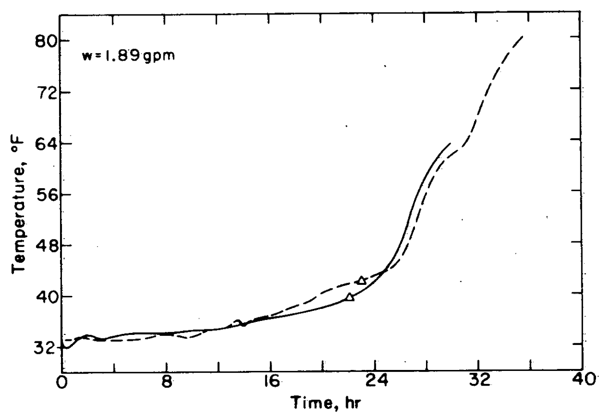
possible to directly determine the outlet temperature - time curve for any smaller initial ice mass test under the same conditions of flow rate and heat rejection rate.

COMPARATIVE ANALYSIS

Ice Cubes vs Ice Blocks

The outlet temperatures measured during the ice cube and ice block tests are shown for comparison in Figure 17. The major difference between these two concepts is the ice surface area available for melting. A 20-lbm block with dimensions of 8.5 x 8.5 x 9.5 in. has a surface area of approximately 470 $in.^2$ while the 0.056-lbm cube, in the form of a truncated cone with a height of 1½ in., upper diameter of 1⅛ in. and lower diameter of 1½ in., has a surface area of 7.9 $in.^2$. Thus while 357 cubes are required to equal the mass of a 20-lbm ice block, these cubes have a combined surface area of 2820 $in.^2$ which is approximately six times that of an equivalent mass in ice block form. The temperature-time curves are essentially the same for the 1.89- and 4.00-gpm ice cube and ice block tests; the considerable difference in surface area apparently did not influence the results. However, the influence of surface area does appear to become a factor at the 1.00-gpm flow rate. In this case the outlet temperature for the ice cube test followed a temperature pattern much the same as was experienced with the 1.89- and 4.00-gpm tests; however, the 1.00-gpm ice block test exhibited rather different characteristics. The rate of change in the outlet water temperature was much less than for the other tests. In this block test the outlet temperature was very close to the maximum density temperature, varying for the most part between 39° and 40°F. This reduced rate of change infers that a greater amount of heat was involved in ice melting. Since the same nominal temperature difference across the sink was used for both the ice block and ice cube tests run at 1.00 gpm, the higher outlet water temperature of the block test resulted in a sink inlet water temperature that was about 4°F higher than that observed in the cube test. It is of interest to note that the rate of outlet temperature change was also rather low for the 1.00-gpm solid ice cylinder test; additionally the range of outlet temperatures was very similar for the solid cylinder and the 20-lbm block tests. It would therefore appear that for the low coolant flow rate tests, the smaller effective ice surface area configurations gave more desirable outlet temperatures during the melting phase of the experiment since the sink outlet water temperatures were between 40° and 44°F, the specified range for maximum condenser efficiency.





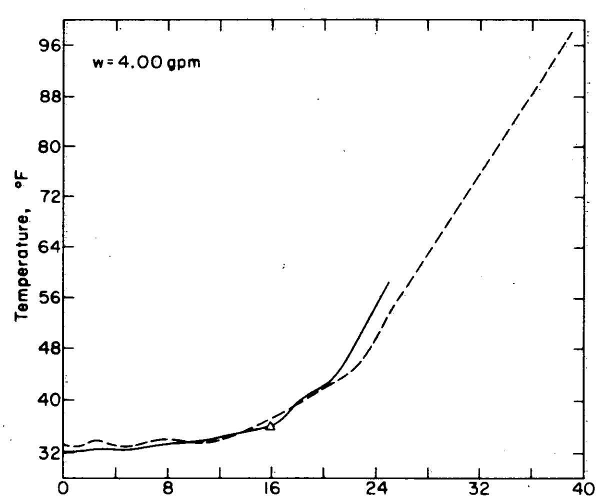


Figure 17. Comparison between ice cube and ice block sink outlet temperatures (tests 1B, 1C, 2B, 2C, 3B, 2C). Nominal $\dot{Q}$ = 16,805 Btu/hr.

Ice Cubes and Ice Blocks vs Solid Ice Cylinders

Before proceeding with a comparative analysis, it is worthwhile to consider some of the requirements inherent in the design of a hardened heat sink installation which are relevant to the selection of the ice configuration. The sink must be ready for immediate use at the moment the installation converts from a standby, or maintenance, mode to a buttoned-up, or operational, mode. The sink should be designed so that the power generating system operates at maximum efficiency. With the cube or block type of ice configuration, the sink is available for immediate use as numerous continuous flow paths for the water always exist throughout the sink. For the solid ice type of sink this criterion is satisfied when an annulus of water has been either created or maintained. With regard to the generation of power, information from the Engineer Power Group indicates that the condensing unit functions at maximum efficiency when the condenser inlet water temperature is between 40° and 44°F. In the following analysis, 44°F is considered to represent a limiting criterion. Should the sink have an outlet temperature below 40°F, the operator merely needs to bypass the sink with a predetermined amount of condenser outlet water and mix it with the sink outlet water, thereby raising its temperature to between 40° and 44° F. Such a mixing operation would optimize the power generation efficiency, thereby reducing the total heat load to the sink. Considering the option to mix condenser and heat sink outlet water, any sink outlet water below 44°F is acceptable.

The ice cube or ice block type of sink will always have a built-in porous structure which is associated with packing of the ice particles. Thus for heat sinks having the same physical shape and volume, the ice/water sinks will have less initial ice mass than the solid ice sinks. During this test program the initial ice load for the solid cylinder tests varied from 69% to 99% of the total water mass (liquid and solid) contained in the sink (see Table I); the average initial ice load was 88.4%. For the ice block tests the initial ice mass varied between 50% and 50.4% and for the ice cube tests it varied from a low of 43.7% to a high of 52.0%. The average value for all the ice cube tests was 46.1%. The tests using ice cubes with air voids have not been included in this overall average since they are not relevant to this comparative analysis. To ascertain the potential maximum packing that might be possible using ice cubes, the cubes were placed in 6-in. lifts in a 55-gallon drum. Each lift was tamped to densify the packing as much as possible. Under these ideal conditions, the maximum attainable ratio of initial ice mass to total water mass was 61.3%. It is obvious that the initial ice mass for the solid cylinder sink is significantly greater and therefore represents a large increase in the sink's capacity to store heat. To illustrate the differences in heat storage, consider a sink which is initially at 32°F and has a maximum allowable outlet water temperature of 160°F. Also, for this example it will be assumed that due to volumetric limitations the sink can contain at most a total water mass of 1000 lbm.

The total heat storage Q_t for an ice cube/ice block type of sink, assuming that 50% of the initial mass is in the form of ice, is:

$$\begin{aligned} Q_t &= (0.50)\,(1000\text{ lb})\,(144\text{ Btu/lb}) + (1000\text{ lb})\,(1.0\text{ Btu/lb}^\circ\text{F})\,(160\text{-}32^\circ\text{F}) \\ &= 200{,}000\text{ Btu.} \end{aligned}$$

If the sink is an ice cylinder type then it is possible to have 90% of the total mass in the form of ice and the heat storage is:

$$\begin{aligned} Q_t &= (0.90)\,(1000)\,(144) + (1000)\,(1.0)\,(160\text{-}32) \\ &= 257{,}600\text{ Btu.} \end{aligned}$$

Thus the capacity of the solid ice sink is 28.8% greater than that of the ice block/cube type. It may be possible to increase the solid ice mass ratio up to 95%, thus providing 32% greater heat storage than the ice cube type of sink. In addition to a comparison between solid and cube/block sinks over the full life span of the sink, it is also useful to compare the ice-water matrix to the solid cylinder concept for sink outlet water temperatures below 44°F. It is assumed that the average mixing cup temperature of the sink can be approximated as the sink outlet water temperature plus one half the temperature difference between the sink inlet water temperature and the outlet. Thus, for a 1.00-gpm test (at a nominal heat rejection rate of 16,805 Btu/hr) the temperature difference between the inlet and outlet is 33.6°F so that for an outlet water temperature of 44°F, the mixing cup temperature of the sink is approximately [44+(33.6/2)] or about 60.8°F. Using this value it is possible to calculate the heat storage of both types of sink, assuming that all the ice will be melted before the outlet water temperature reaches 44°F. Thus, in the previous expression for the heat storage we merely replace 160°F by 60.8°F so that in an ice cube type of sink (50% ice) the storage is 100,500 Btu while in the solid ice type (90% ice) it is approximately 158,100 Btu, almost a 60% increase in the storage capacity while the sink is operating at the most efficient condenser temperatures.

Of course, such increased capacity can also mean a 30% reduction in the excavation required for the sink complex and thus substantial savings in the initial construction cost. Also, prolonging sink operation in the more efficient temperature range reduces the gross amount of heat rejected to the sink.

A comparison of the experimental results of the solid ice cylinder and ice cube tests is shown in Table III. The time required for the sink outlet water to reach 44°F was used as a basis for this comparison. As noted above, this temperature is the maximum outlet water temperature for optimum condenser efficiency. Selection of a temperature in this lower range minimizes the influence of different total water masses which have a more dominant effect on the temperature-time relationship during the later phase of the experiment. As indicated in the table, the outlet water from the cylinder took about 40% longer to reach 44°F. The initial ice mass in the solid cylinder tests averaged about 85% greater than that for the ice cube tests.

The greater heat storage capacity of the solid ice sink tends to mask any differences between the solid cylinder and ice cube/block tests which were caused by the considerably different ice surface areas exposed to meltwater. To gain some insight as to the influence of these surface areas, it is possible to compare these two concepts on the basis of equivalent heat storage capacities. As noted in the discussion of the cylinder test, the temperature-time history of the sink was found to be unique for a given flow rate and heat rejection rate; this unique relationship permitted examination of the effects of different initial ice masses. If the difference in the initial ice masses is ΔM_i, then the time required to melt this additional amount of ice, t_m, is given by:

$$t_m = \frac{\Delta M_i L}{\dot{Q}}$$

where L = latent heat of fusion of ice, 144 Btu/lb
$\dot{Q}$ = heat rejection rate, Btu/hr.

Table III. Comparison of solid ice and ice cube sinks
(Equivalent physical size sinks)

Nominal Heat Rejection Rate = 16,805 Btu/hr

Test no.*	Flow rate (gpm)	Initial ice mass (lbm)	Total water mass (lbm)	Ice mass / Total water mass (%)	Time to reach 44°F (hr)	Increase in time (%)	Difference in ice mass (%)
1C	1.89	2122	4079	52.0	25.5	-	-
$1S_1$	1.89	3805	4072	93.4	34.7	36.1%	79.3%
$1S_2$	1.89	3955	4005	98.8	37.8	48.2%	86.4%
2C	1.00	1834	4190	43.7	29.5	-	-
2S	1.00	3319	3812	87.1	40.5	37.3%	81.0%
3C	4.00	1790	4090	43.7	22	-	-
3S	4.00	3442	3950	87.1	32	45.5%	92.3%

*S = solid ice test, C = ice cube test.

If the difference in the total mass of ice and water is ΔM_W, then the time required (t_S) to raise the sink outlet temperature of this additional water to 44°F is given by:

$$t_S = \frac{\Delta M_W C_p (44-32)}{\dot{Q}} .$$

Thus for a solid cylinder sink having a larger initial ice mass and a larger total mass of ice and water, the equivalent time, t_e, for the sink outlet to reach 44°F may be determined from:

$$t_e = t_0 - (t_m + t_S)$$

where: t_0 = measured time for the larger ice mass test to reach 44°F.

A solid cylinder sink having a large initial ice mass and a smaller total mass of ice and water would have an equivalent time of:

$$t_e = t_0 - (t_m - t_S).$$

The equivalent times for the 16,805-Btu/hr nominal heat rejection tests are shown in Table IV. The solid ice tests have been "adjusted" to the initial ice mass of a comparable ice cube test. The table does point out that the significant time differences are due to variations in initial ice mass, not combined ice/water mass. Elimination of the influence of these ice mass differences reveals that the greater ice surface area in the ice cube sink increases the time it takes for the outlet water to reach 44°F by about 5 to 20%. It is felt that this improvement is the result of the buffering action of the ice/water matrix which creates a zone of low-temperature, high-density water in the lower region of the sink.

It must be remembered that the above analysis was contrived to separate the effect of differences in ice surface area from differences in initial ice masses. The determining factor in the selection of ice configuration should be volumetric efficiency. The solid ice type of sink has a higher total heat storage capacity as well as a longer period of time during which the sink outlet temperature is below 44°F.

Table IV. Comparison of an ice cube sink and a solid ice sink having equivalent heat storage capacity

Nominal Heat Rejection Rate = 16,805 Btu/hr

	Test no.						
	1S₁	*1S₂*	*1C*	*2S*	*2C*	*3S*	*3C*
Difference in initial ice masses (lbm)							
$\Delta M_i = M_{solid} - M_{cube}$	+1683	+1833	0	+1485	0	+1652	0
Time required to melt ice mass difference (hr)							
$t_m = \frac{144\ \Delta M_i}{\dot{Q}}$	+14.42	+15.7	-	+12.72	-	+14.16	-
Difference in total masses (lbm)							
$\Delta M_w = M_{solid} - M_{cube}$	-7	-74	-	-378	-	-140	-
Time required to raise temperature of total mass difference to 44° F (lbm)							
$t_s = \frac{\Delta M_w C_p (44-32)}{\dot{Q}}$	-000	-0.05	-	-0.27	-	-0.10	-
Measured time for sink to reach 44°F (hr)							
t_o	34.7	37.8	25.5	40.5	29.5	32.0	22.0
Equivalent time for solid ice sink to reach 44°F (hr)							
t_e	20.3	22.1	-	28.1	-	17.9	-
Percentage decrease in time for solid ice sink to reach 44°F	20.4%	13.3%	-	4.7%	-	18.6%	-

When the relative performance of ice cubes, blocks and cylinders is compared, an important feature that appears is a maximum sink outlet temperature with respect to flow rate for the blocks and cylinders. As stated previously, during the melting period in the ice cube test there was no noticeable flow rate effect on sink outlet temperature. On the other hand, in the ice block test series the 1.00-gpm test had the highest observed sink outlet temperature while the outlet temperatures for the 1.89- and 4.00-gpm block tests were lower. Finally, in the ice cylinder tests it was found that the outlet temperature increased to a maximum at approximately 3.00 gpm and then began to decrease again. Since the ratio of exposed surface area to total mass is lowest for the solid ice cylinder and highest for the ice cubes, it is felt that the flow rate at which the maximum outlet water temperature occurs could possibly be inversely related to this ratio. Thus, a maximum sink outlet water temperature in an ice cube sink would occur at extremely low flow rates. Similarly, for the ice cylinder type of sink this might indicate that as the flow rate was increased above 4.00 gpm, the outlet temperature would tend to become independent of the flow rate, as appears to be the case for the ice cube and higher flow rate ice block tests.

APPROXIMATION OF THE COEFFICIENT OF HEAT TRANSFER

Although previous investigations into the melting of ice have determined relationships for the Nusselt number and the coefficient of heat transfer during the melting process, these studies were conducted on small samples under well controlled, rather idealized test conditions. Extrapolation of these results to the rather complex geometry, melt patterns, and water flow patterns in the experimental heat sink is of questionable validity. The melting process in the model sink is also subject to tank wall effects and variations in the bulk water temperature, as would be the case for prototype sinks.

A rather gross approximation of the coefficient of heat transfer and Nusselt number was obtained for the solid ice cylinder tests by use of some simplifying assumptions. The differential amount of heat rejected to the ice, dQ_i, during any finite time period is given by:

$$dQ_i = hA\Delta T_m d\theta$$

where: h = heat transfer coefficient
A = area of the melting surface
ΔT_m = temperature difference between the melting surface and the bulk water temperature
$d\theta$ = length of the time period.

If the ice is initially at 32°F then this amount of rejected heat may also be expressed in terms of the amount of ice melted so that:

$$dQ_i = \rho_i LAdR$$

where: ρ_i = density of ice
L = latent heat of fusion
dR = change in radius due to melting during the time period $d\theta$.

Using these two relationships and solving for the heat transfer coefficient yields:

$$h = \frac{\rho_i L}{\Delta T_m} \frac{dR}{d\theta}.$$

Thus to determine the heat transfer coefficient it is necessary to determine the rate of radius change and the bulk water temperature. As a first order approximation, it is assumed that the bulk water temperature was approximately the algebraic average of the inlet and outlet water temperatures observed during the ice melting period. Thus, the average bulk water temperature T_B is given by

$$T_B \approx \tfrac{1}{2}(T_{in} + T_{out})$$

where T_{in} = average inlet water temperature during the melting period
T_{out} = average outlet water temperature during the melting period.

Therefore, the mean temperature difference between the ice and the water ΔT_m is:

$$\Delta T_m = T_B - 32°F.$$

The rate of radius change $dR/d\theta$ may be approximated by considering the time required to melt all the ice in the sink. That is

$$\frac{dR}{d\theta} \approx \frac{\Delta R}{\Delta\theta}$$

where ΔR = total change in radius from the initial value to zero
$\Delta\theta$ = time required to melt all the ice in the sink.

Using these approximations the solid ice test results were analyzed to determine the coefficient of heat transfer. Table V and Figure 18 show the results of this analysis. The overall Nusselt number applicable over the entire melting process was determined from the following relationship:

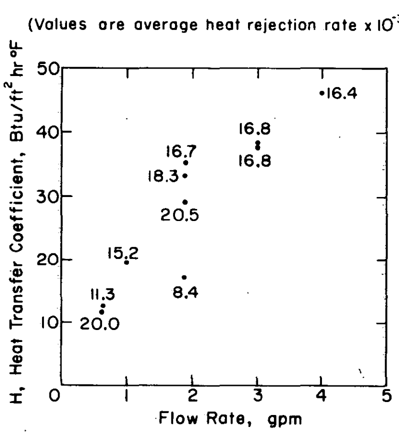
$$Nu = \frac{hD}{k}$$

where: Nu = Nusselt number based on diameter
h = heat transfer coefficient
k = conductivity of ice.

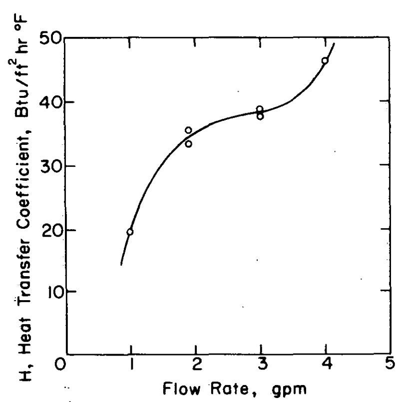
The calculated overall Nusselt numbers for the solid ice tests are shown in Figure 19 and in Table V. The time-averaged heat rejection rate for each test has been noted in Figures 18a and 19a since it is anticipated that the rate would influence both the Nusselt number and the coefficient of heat transfer. To isolate the effect of heat rejection rate, Figures 18b and 19b were prepared depicting those tests having a time-averaged heat rejection rate which was within the range of ± 10% of 16,805 Btu/hr. The curves drawn through these points should be considered tentative and serve only to indicate general relationships with respect to coolant water flow rates. The inflection in the curves which occurs at the 3.00-gpm flow rate may be indicative of a change in the mode of heat transfer. Since the 3.00-gpm tests also had the highest observed sink outlet water temperature it is felt that this flow rate might represent a transition from natural to forced convection.

Table V. Heat transfer coefficients and Nusselt numbers for solid ice tests

Flow rate (gpm)	*Test no.*	*Time to melt all the ice (hr)*	*Bulk water temp (°F)*	*Mean temp diff between ice and water (°F)*	*Heat rejection rate (Btu/hr)*	*Heat transfer coeff (Btu/ft² hr °F)*	*Nusselt no.*
0.62	5S	37.5	69.75	37.75	20,032	11.6	134
0.66	7S	56	54.9	22.9	11,272	12.8	148
1.00	2S	39	53.2	21.2	15,200	19.6	229
1.89	6S	72	45.3	22.3	8,425	17.0	199
1.89	8S	28.5	51.7	19.7	29,520	39.2	338
1.89	$1S_1$	32	46.5	14.5	16,720	35.4	410
1.89	$1S_2$	34	46.5	14.5	18,320	33.3	386
3.00	$4S_1$	28	47.5	15.5	16,848	37.8	438
3.00	$4S_2$	29	46.75	14.75	16,776	38.4	445
4.00	3S	32	43.2	11.2	16,350	45.8	536

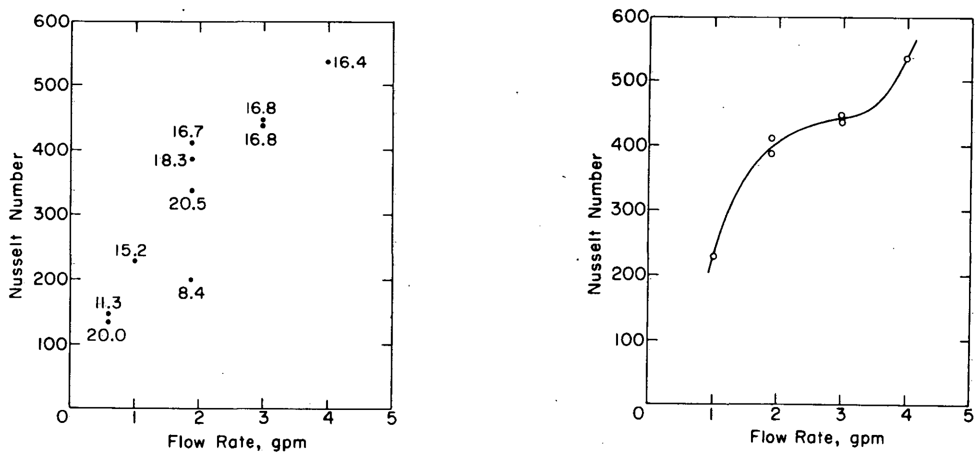


a. *Various heat rejection rates.*



b. *Nominal heat rejection rate of 16,805 Btu/hr.*

Figure 18. Relationship between flow rate and heat transfer coefficient (solid ice cylinder).



a. Various heat rejection rates

b. Nominal heat rejection rate of 16,805 Btu/hr.

Figure 19. Relationship between flow rate and Nusselt number (solid ice cylinder).

CONCLUSIONS AND RECOMMENDATIONS

The primary purpose of this study was to compare the relative performance of an ice/water matrix type heat sink with an annular flow, solid ice sink. As noted in the discussion, the ice cube or ice block type of sink will always have a built-in porous structure which is associated with the packing of the ice particles. Thus for heat sinks having the same physical shape and volume, the ice/water sinks will have less initial ice mass than the solid ice sinks. Ice masses in the order of 95% of the total heat sink mass are possible for solid ice cylinder sinks, whereas a comparable ratio for the ice/water matrix sink is about 50 to 55%. Such increased capacity can be translated into 30% reduction in the excavation required for a solid ice sink relative to an ice-water sink. Also, prolonging sink operation in the more efficient condenser inlet temperature range between 40° and 44°F reduces the total amount of heat rejected to the sink. The determining factor in the selection of ice configuration should be volumetric efficiency. The solid ice type of sink has a higher total heat storage capacity; it also provides a longer period of time during which the sink outlet temperature is below 44°F.

The results of the study did point out that the ice/water matrix configuration can provide a buffering action which creates a zone of low temperature, high density water in the lower region of the sink. The buffering action is attributed to the increase in surface area available for melting and the positioning of the ice near the inlet to the tank. The buffering isolated the sink's outlet water temperature from variations in the coolant water flow rate. Density stratification occurred at the lower coolant water flow rates for all three types of sinks (cubes, blocks and solid). Stratification increased the period of low temperature operation and represents a significant improvement in the performance of the sink, regardless of the type of sink used. The significant change in the melting pattern that results from the use of low water flow rates cannot be predicted using the computer program developed in the previous ice heat sink study[2] which involves ice cylinder melting in both the radial direction and upward from the bottom. Although the program does predict the sink outlet water temperature with a high degree of accuracy for flow rates greater than 1.00 gpm in the scale model, it was found to be unsuitable with water flow rates in the order of 0.6 gpm. Since the use of low flow rates appears to improve sink performance significantly, it is felt that the computer program should be modified to predict sink outlet water temperature for all flow rate conditions.

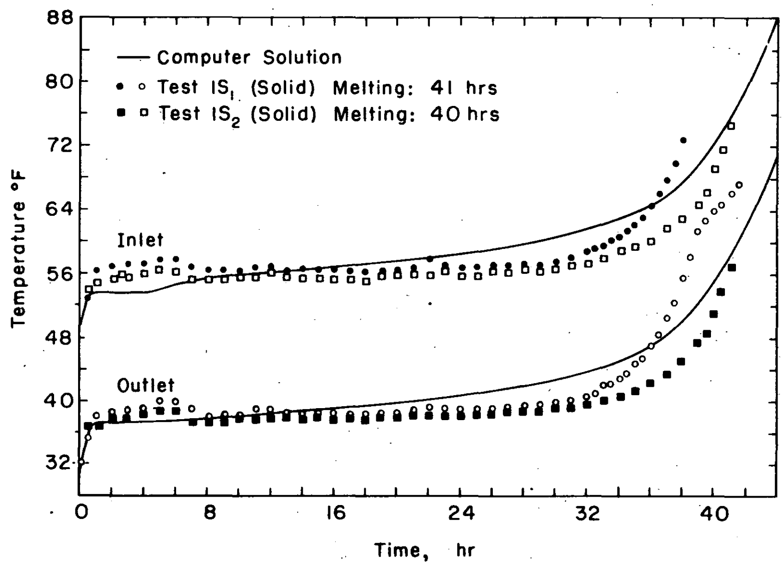
The existence of an inflection in the relationship between the Nusselt number (heat transfer coefficient) and the water flow rate indicates that a transition in the mode of heat transfer will probably develop within the range of flow rates most likely used. It is felt that this transition represents a change from natural to forced convection. Since the coolant water flow rates associated with this transition zone tend to exhibit the highest sink outlet temperatures, a better understanding of its causes appears warranted.

Although many characteristics of ice heat sink operation and performance have been studied and appraised for a rather small ice cylinder, it is necessary that these relationships be assessed for a larger model cylinder.

LITERATURE CITED

1. Becker, E. and J. L. Brown (1970) An investigation of a multiple hole ice-water heat sink. USACRREL Technical Note.
2. Brown, J. L. and W. F. Quinn (In prep.) An annular flow ice-water model heat sink study. USACRREL Special Report.
3. Dill, R. S. and A. B. Peavy (1956) Some observations on the use of underground reservoirs as heat sinks. National Bureau of Standards Report 4795.
4. Hubbard, R. III (1967) Tunnel condenser concept (U). Nuclear Power Field Office, Ft. Belvoir, Virginia (CONFIDENTIAL).
5. Parson, Brinckenhoff, Hall and MacDonald (1958) Feasibility study - icing of underground reservoirs (U) (SECRET).
6. Perham, R. E. (1969) Model ice heat sink. USACRREL Technical Note.
7. Quinn, W. F. and M. Greenberg (1967) A study of the steam sink concept. USACRREL Draft Report (SECRET).
8. Quinn, W. F., H. W. C. Aamot, and M. Greenberg (1971) Analysis and test of a rock-steam condenser heat sink concept. Winter Annual Meeting of the ASME on Environmental and Geophysical Heat Transfer, HTD, vol. 4, p. 70-77.
9. Tien, C. (1960) Analysis of a sub-ice heat sink for cooling power plants. U. S. Army Snow, Ice and Permafrost Research Establishment (USASIPRE) Research Report 60.
10. Tkachev, A. G., A. E. C. - Translation 3405 (1953) Translated from a publication of the State Power Press, Moscow, Leningrad.
11. U. S. Army Cold Regions Research and Engineering Laboratory (1966) Studies of underground heat sink concept (U). USACRREL Draft Report (SECRET).
12. Vanier, C. R. (1967) Free convection melting of ice. Syracuse University, M. S. thesis.
13. Zehnder, A., Y. C. Yen, R. Perham, and W. F. Quinn (1973) An analytical study of a coiled pipe heat sink. USACRREL Special Report 195.

APPENDIX: EXPERIMENTAL DATA, COOLANT WATER TEMPERATURES VS. TIME



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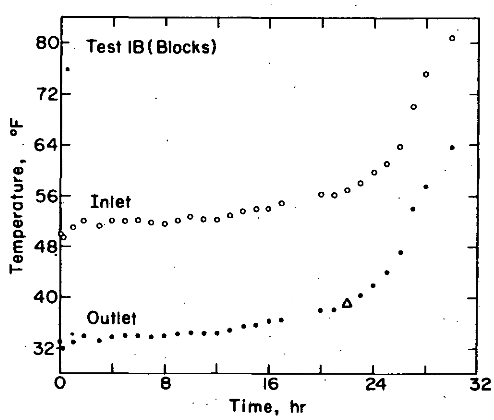

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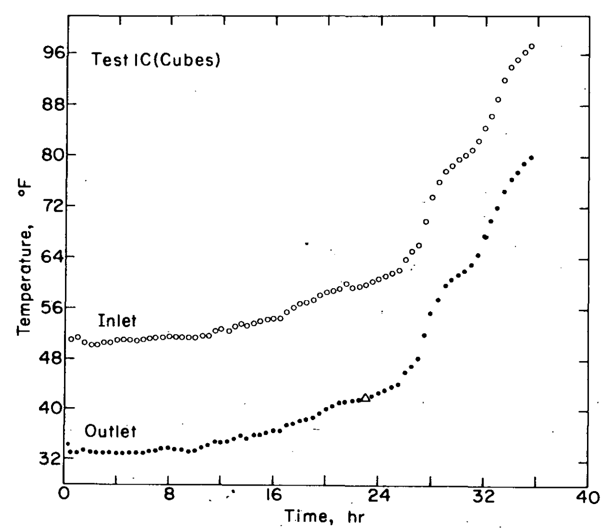

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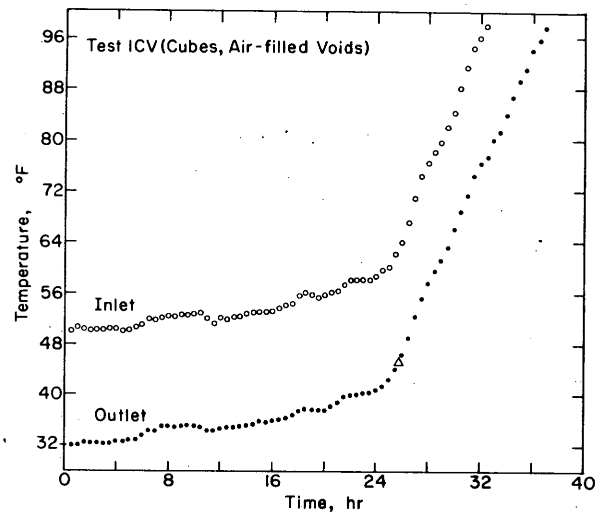
Figure A1. Inlet and outlet water temperature vs time curves for various ice configurations. Nominal $\dot{Q}$ = 16,805 Btu/hr, flow rate = 1.89 gpm.



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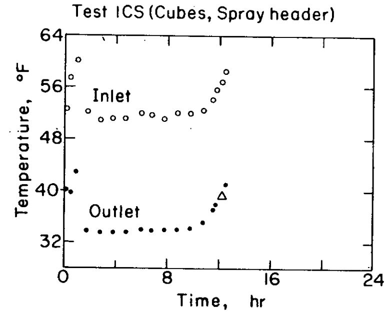

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Figure A1 (Cont'd).

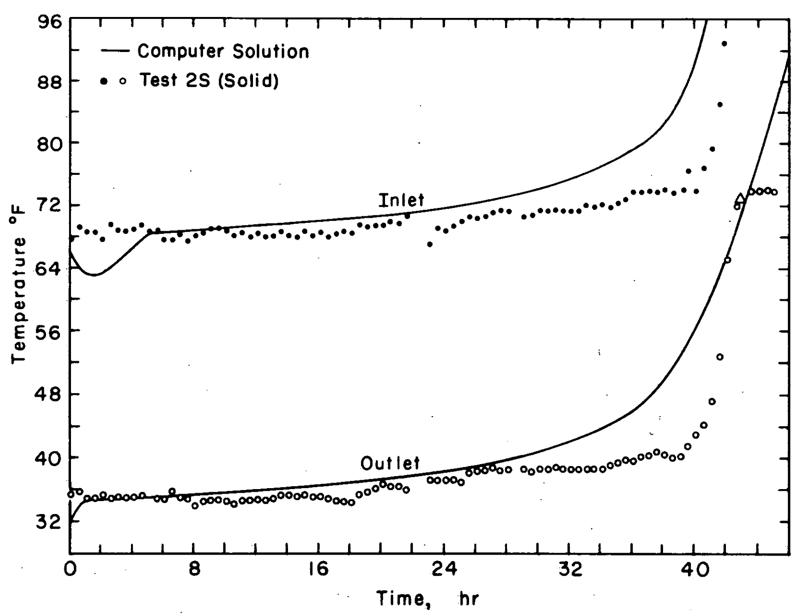

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Figure A2. Inlet and outlet water temperature vs time curves for various ice configurations. Nominal $\dot{Q}$ = 16,805 Btu/hr, flow rate = 1.00 gpm.

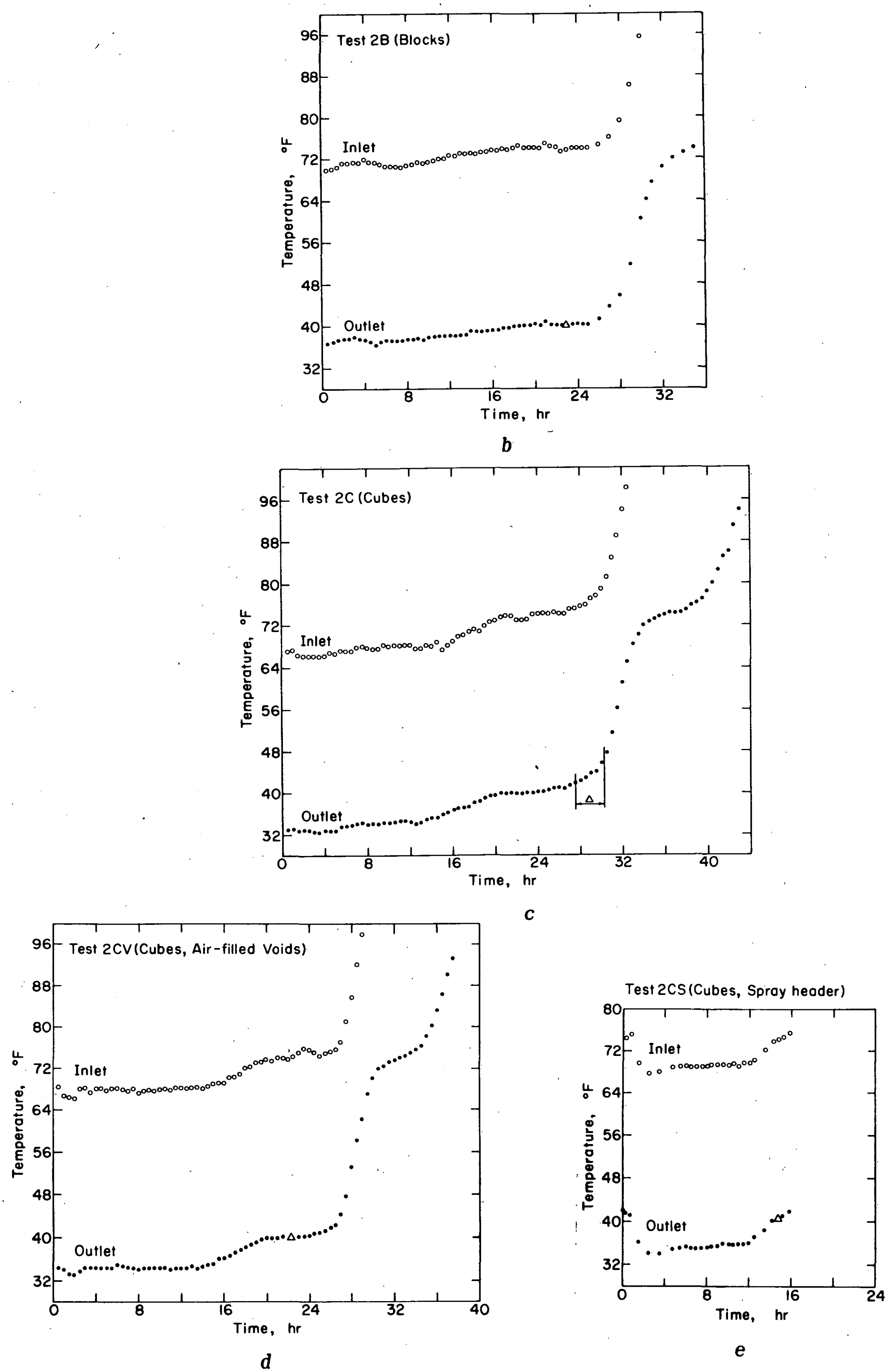
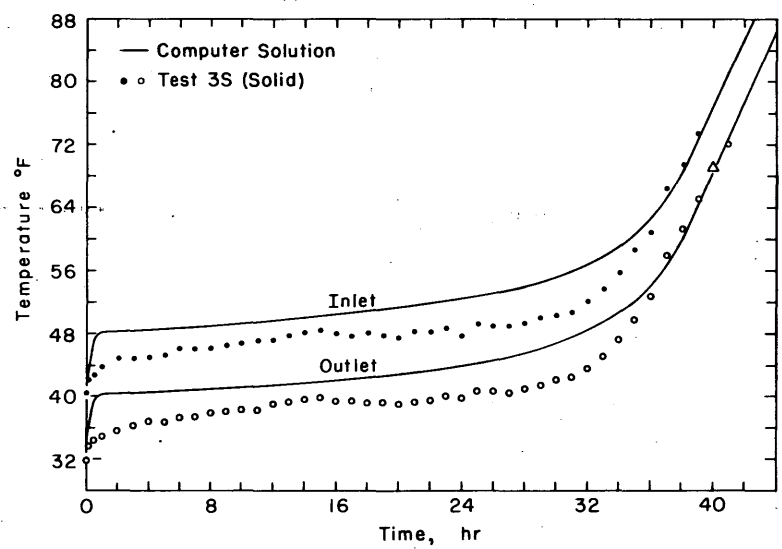


Figure A2 (Cont'd). Inlet and outlet water temperature vs time curves for various ice configurations. Nominal $\dot{Q}$ = 16,805 Btu/hr, flow rate = 1.00 gpm.



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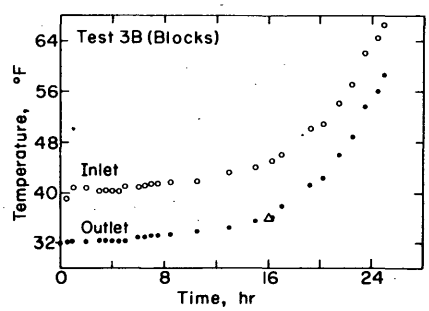

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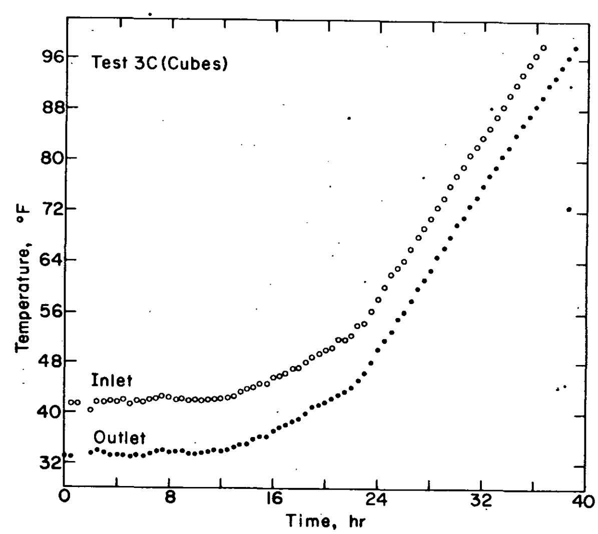

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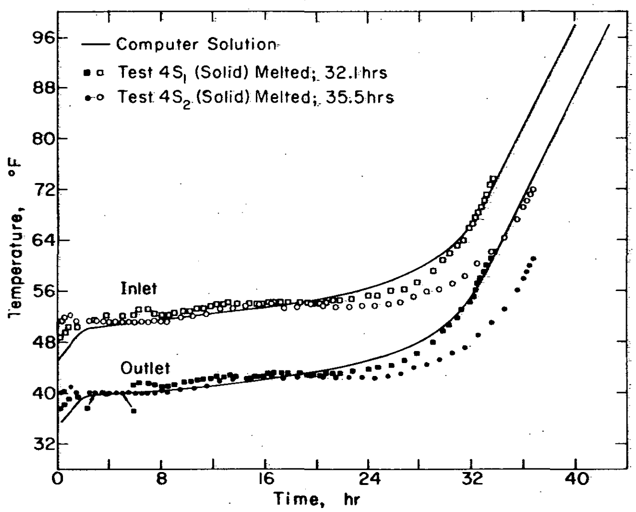
Figure A3. Inlet and outlet water temperature vs time curves for various ice configurations and flow rates. Nominal $\dot{Q}$ = 16,805 Btu/hr.



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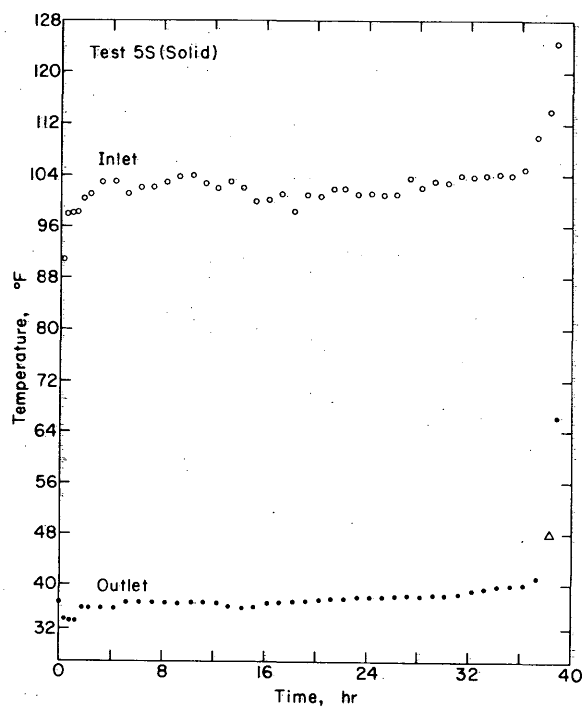

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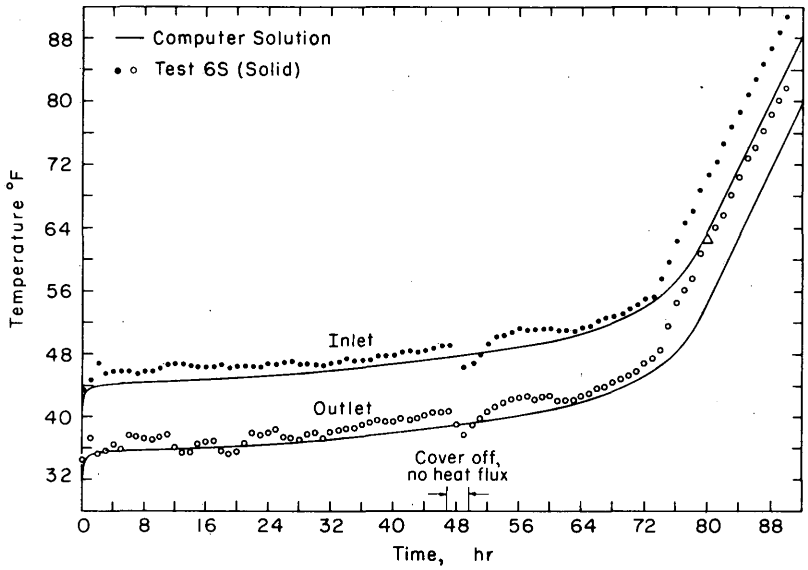
Figure A3 (Cont'd). Inlet and outlet water temperature vs time curves for various ice configurations and flow rates. Nominal $\dot{Q}$ = 16,805 Btu/hr.



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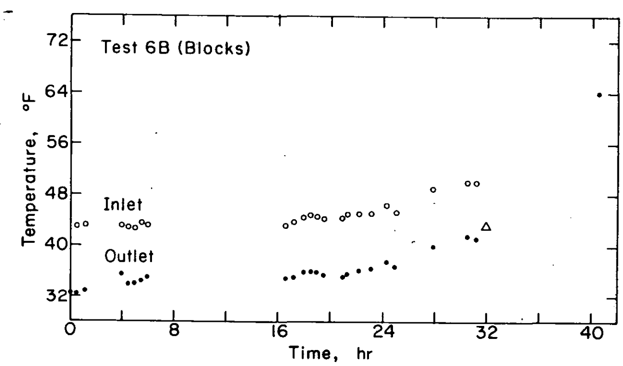

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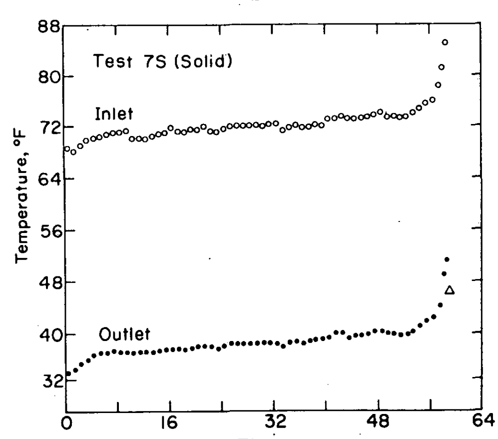

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Figure A4. Inlet and outlet water temperature vs time curves for solid ice cylinders and ice blocks. Nominal $\dot{Q}$ = 8402 Btu/hr.

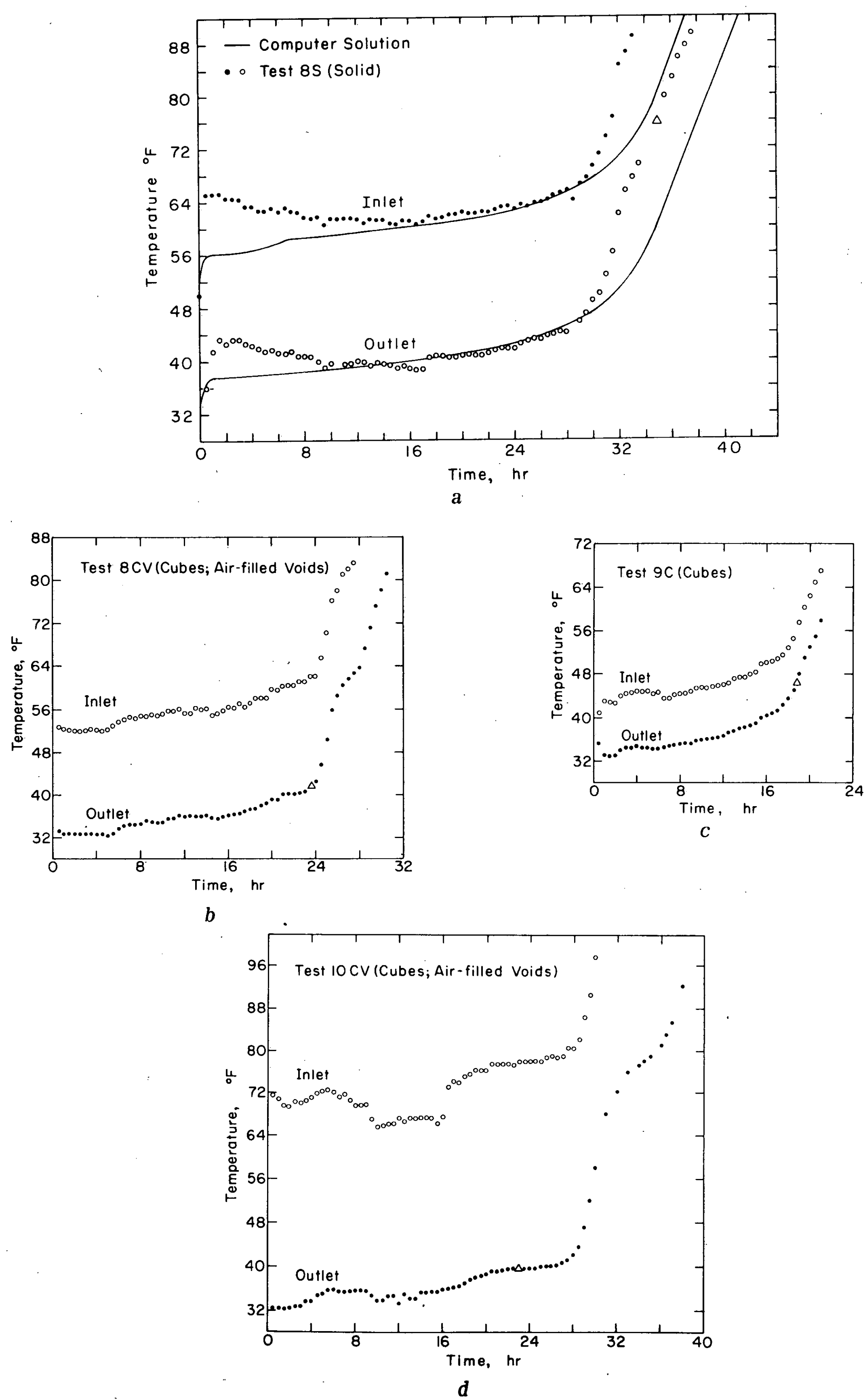

Figure A5. Inlet and outlet water temperature vs time curves for various ice configurations. Nominal $\dot{Q}$ = 19,105 Btu/hr.

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13. ABSTRACT

An analytical and laboratory model study was conducted on the potential use of ice as a medium for storage of waste heat developed by a hardened underground defense installation during the time it was operating on a closed cycle system. The use of ice is an attractive concept because of its high heat storage capacity per unit volume and its associated low coolant water temperatures. Three ice configurations were studied; a solid ice cylinder with annular flow, an ice block-water mixture, and an ice cube-water mixture. All of these configurations are shown to be feasible with the solid ice cylinder yielding the best overall performance.

14. Key Words

Cooling
Cooling systems
Electric power plants
Heat sinks
Military facilities
Underground facilities