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THIN POLYMER ICEMAKER DEVELOPMENT AND TEST PROGRAM - FINAL REPORT ON TECHNOLOGY

RICHARD W. LEIGH

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OFFICE OF UTILITY TECHNOLOGIES
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DEPARTMENT OF APPLIED SCIENCE

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ABSTRACT

We have constructed and tested a small device to produce ice in ice/water mixtures using a cold fluid as the heat sink. The device is a flexible heat exchanger constructed from a thin film of a suitable polymer. When filled with circulating liquid coolant the heat exchanger consists of an inflated series of parallel tubes; ice forms on the outside in complementary half cylinders. When the circulation is cut off, gravity drains the coolant and the static head of the water bath crushes the tubes, freeing them from the ice which floats to the surface. Brine circulation is then re-started and the cycle begins again.

Here we report recent testing of this device: it makes ice readily under water and easily sheds the semi-cylinders of ice over many cycles of operation. It produces ice at a rate of 10 kg/m^2-hour. It offers substantial benefits in simplicity and reliability over mechanical harvester ice making systems, and the potential for significant improvements in energy efficiency compared to systems which use a re-heat cycle to harvest the ice. A reliable method of leak detection has been developed. The device should be of substantial value to systems where efficiency and reliability are at a premium, such as slush ice for district cooling.

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0. Introduction and Summary

In this section we justify our interest in this device, review previous work on this and similar ice making devices and summarize the results which are presented in more detail in the following pages.

0.1 Introduction

District cooling offers several advantages over conventional, dispersed systems, including the ability to use reject heat, low grade fuels or renewable sources of energy. However, the cost of these systems is high, in large measure because the use of chilled water and the small temperature differences imposed by engineering constraints requires that large volumes of water be pumped. The use of ice slurries, in which the latent heat of fusion provides cooling, would allow much smaller piping to be used, at a substantial cost saving for the system. A delivery system based on an ice slurry is also uniquely compatible with a system of cold storage based on the latent heat of fusion.

However, when large scale energy delivery is the service at hand, rather than, say, beverage cooling, efficiency becomes of paramount importance. Although some conventional ice making systems boast efficiencies which make them practical in selected applications[1], even these are hampered by using either mechanical scrapers or a re-heating system to free the ice from the surface on which it was formed. Mechanical scrapers are subject to periodic breakdown, and re-heat systems involve an intrinsic inefficiency, since even though the heat energy can often be supplied from otherwise wasted condenser heat, the plates must then be cooled down again before the next freezing cycle can begin, resulting in a significant parasitic cooling load. In addition, existing systems do not readily make ice under water, but must rely on sprays or drip systems to distribute water uniformly over the freezing surface.

A possible answer to these problem emerged out of previous work on thin material heat exchangers in several areas of energy efficiency and efficient resource use, including solar collectors[2,3,4] and absorption chillers[5,6]. Construction techniques adapted to the strengths and limitations associated with flexible polymer membranes in these applications suggested a unique design for an ice making heat exchanger. A device was constructed based on this design and, in earlier experiments, worked well enough to establish the viability of the concept[7,8]

0.2 Background

The concept is illustrated in its original embodiment in Figures 0-1 and 0-2. The icemaker operates in a cyclic manner, with an ice-making phase and an ice-releasing phase. During the ice making phase, refrigerant somewhat below 0^oC ($32\ ^oF$) flows through a heat exchanger consisting of two layers of a hydrophobic polymer welded together to form tubes, referred to hereafter as the "pad". The pad is immersed in the water which is to be frozen. The pressure due to the refrigerant circulation pump expands the pad so that it looks somewhat like an inflated air mattress. Ice forms on the outside layer up to some selected thickness, typically about one millimeter. The pump then shuts off momentarily, and the system is configured so that with the pump off, gravity drains all coolant out of the pad into a sump. This inverse static head flattens the polymer film, pulling it away from the ice, which is freed and broken, so that the ice floats to the surface of the container of water where it may be collected. Note from the figure that ice will not form on the flats between the tubes, so no large chunks of ice are created, just a set of "semi-tubes".

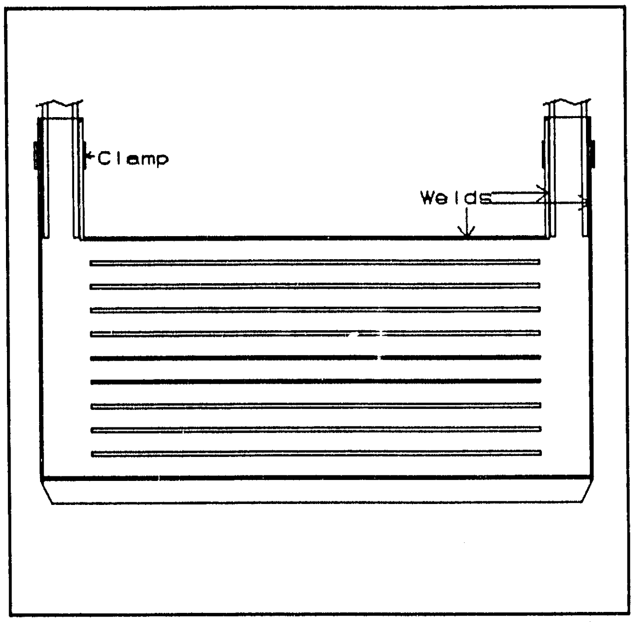

Fig. 0-1: Original ice maker pad, side view.

This device was constructed and tested at Brookhaven National Laboratory[7,8]. When coolant flow to the pad was cut off the coolant drained and the pad flattened over a period of about 15 seconds. Semi-tubes of ice broke off from central sections of the tubes as anticipated and floated to the top, available for harvesting. With coolant temperatures in the range of -6 to -3°C (22 - 27°F) and average water tank temperatures at 8°C (40°F) or below, ice formed readily on the outside of the tubes. The sections were less than a millimeter thick, about a centimeter across and 5 to 20 cm long. They were quite fragile and were easily broken by stirring, indicating that pumping them through piping should result in a slush of millimeter-sized pieces.

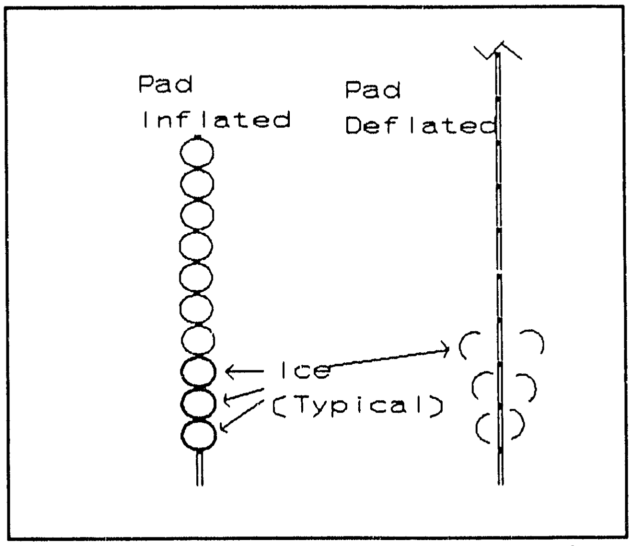

Fig. 0-2: Ice maker pad cross section, showing operation.

A problem with the initial design became apparent during operation: ice formed rapidly at the input end, where the coolant was coldest, and portions of the pad at this end were held rigid by the supply manifold and were thus unable to collapse and shed the newly formed ice. The result was a continuing ice buildup as time progressed, which gradually covered even the flexible portions of the pad and ended operation until the buildup was melted off. We refer to this problem as "parasitic ice"; design options which have overcome the problem are discussed below.

We regarded these results a successful "proof of concept". In the rest of this report we describe the results of new designs and experiments which overcame the ice buildup problem and more fully tested the potential of this concept.

In the course of patent application, we learned of several similar devices which have been developed in years past. To our knowledge, none of them have been successfully commercialized in thermal storage or transport applications, and all appear to have characteristics which may help to explain this. We briefly review these earlier efforts here:

In 1954 C.M.Green[9] patented a device which anticipates many features of the invention reported here. However, it almost certainly does not work well, due to three technical problems: first, the cold fluid ("refrigerant") is carried in tubes which are not part of the ice-making surface. This will result in poor thermal contact between the refrigerant and the ice-making surface, resulting in low efficiency. Second, the refrigerant tubes only cover a small fraction of the ice making surface. This means most of the heat will have to travel sideways through the ice-making surface. This surface must be thin so that it can flex, so it will conduct heat very poorly in this direction, resulting in further inefficiency. Third, the <u>metal</u> ice-making surface has great affinity for ice and it is very unlikely that ice will in fact free itself completely even when flexing takes place. Also, the semi-rigid metal flexes much less than do our polymer tubes, making it even less likely that the ice will reliably free itself from the surface. Our use of polymer tubes obviates all these problems.

A device using a metallic bellows was patented by Hershberg[10] and improved for ice-making by Litman[11]. This is a scheme analogous to ours, but the metal bellows will not let go of the ice easily and metal must be used due to the vacuum inside the bellows. There will also be problems in getting the water to stay on the outer surface long enough to freeze, and when it freezes it will form large lumps of ice ("crescents") which will result in thermal inefficiency due to the low thermal conductivity of the ice. This device is too complex and inefficient for energy storage applications .

Both Broadbent[12] and Moreland[13] have developed flexible polymer devices to make clear cubes or other blocks of ice, where ours is used to make thin sheets of ice or slush ice. The energy efficiency of these devices will be low, since the thick cubes mean that the low thermal conductivity of the ice will play a role, and because they are making "cubes" they have many complexities that are irrelevant to our device.

0.3 Summary of Findings

Operation of the system is straightforward. Although formation of the first layer of ice can be slow (up to one hour in a cold tank), all subsequent layers of ice form readily and harvest easily. The problem of the buildup of parasitic ice can be completely controlled by the present design, augmented by wider welds and perhaps by slight shaking of the ice maker pad during harvesting.

Although the measured efficiency is only one-half of the theoretical performance, the experiments reported here demonstrate conclusively that this device will produce ice reliably at a rate of about 10 kg/m^2-hour. In addition, a reliable method of leak detection has been developed.

1. Test Bed Development

The key to useful assessment of any heat transfer device is accurate measurement of the heat flows and energy balances involved. In this section we discuss the design, construction and instrumentation of a test bed consisting of a temperature regulated coolant loop (the heat sink), a slush storage tank (the ice repository) and the computerized system which controlled and monitored the operation of the bed. The system is shown schematically in Fig. 1-1.

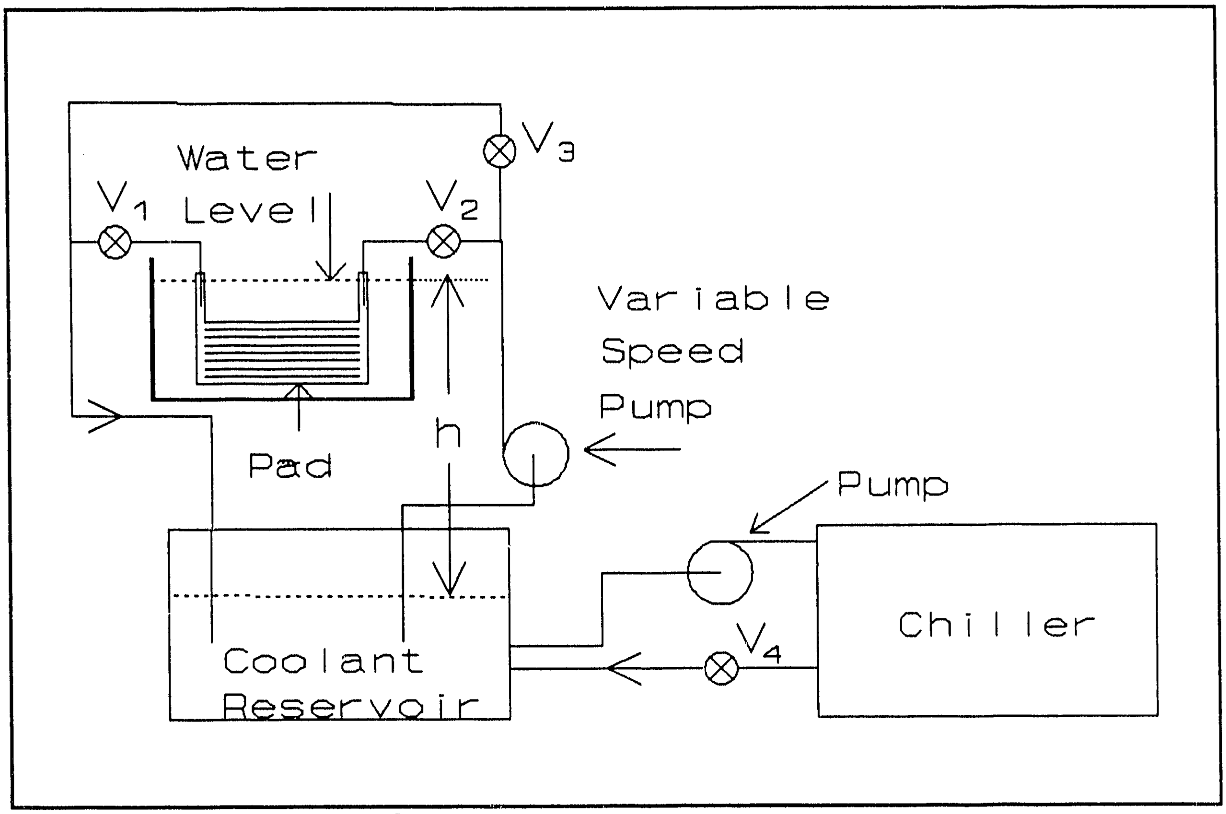

Fig. 1-1: Icemaker test bed.

1.1 Coolant Loop

The coolant loop is designed to provide cold fluid which will circulate through the ice maker and act as the heat sink as ice forms. To facilitate draining of the pad, it is also designed to maintain an equilibrium static head substantially below the surface of the water in which the pad is immersed. The pressure crushing the pad and freeing it from the ice is that due to a column of coolant of height equal to the difference between the surface of the coolant reservoir and the surface of the water, if the small difference between coolant density and water density is ignored. For the tests reported here, the static head was always 0.80±0.02 m (31.5 inches). It is indicated as "h" in Fig 1-1.

The coolant used was a near-eutectic mixture of inhibited ethylene glycol and water, based on commercial automotive "anti-freeze". Previous work[7,8] had shown that aeration of the brine interfered with use of the static head to drain the pad, and that at low temperatures the glycol would hold air bubbles tenaciously. Accordingly, the system shown in Fig. 1-1 was plumbed so that fluid entered and left the bath through submerged orifices, preventing any splashing or entrainment of air. Measurement of a density of 1.072±.001 at 22.4°C indicates a 55.8±0.8% glycol mixture[14]. Normal operating temperature was -5.5°C, where this mixture has a specific heat of 3.003±0.0030 kJ/kg-°C and density of 1.085, giving $c\rho/60$ = 54.3±0.5 W/(l/min)-°C. (This figure will be used below.)

A well-insulated 85 liter (22 gal) polypropylene tank served as a reservoir to increase temperature stability and allow any stray air bubbles a quiescent volume in which to percolate up and out. The fluid volume in the tank was typically 55 liters (15 gallons) with another 10 to 15 liters being distributed in the piping. A five ton commercial heat pump cooled the brine in the primary circuit shown in Fig 1-1. The datalogger (to be discussed below) measured the coolant temperature in the tank with a type "T" thermocouple and was programmed to turn the chiller on and off to maintain the tank temperature between -5.0 and -6.0°C. (A smaller band would have led to unacceptably rapid cycling of the chiller.)

A secondary loop withdrew coolant from the reservoir and circulated it through the ice maker. Flow was controlled by a variable speed peristaltic pump which could be turned on and off by the data logger and could be set to any pumping rate between 0 and 4 liters per minute. The actual flow rate was read by a turbine meter coupled through electronic amplification into the data logger, and capable of reading from 0.2 to 3.6 liters per minute with an observed error (after volumetric calibration) of 1.5% or less. When the pump was shut off it was effectively a closed valve, leading to draining of the ice maker and shedding of ice, so one operational cycle was obtained simply by turning the pump on, waiting for suitable amounts of ice to form, then turning the pump off and waiting for the ice to free itself.

As shown in Fig. 1-1, the tubing carrying coolant in and out of the ice maker had to rise above the surface of the water; this resulted in positive static head on the ice maker during start-up, before the drain pipe was full and exerting negative (syphon) pressure. To minimize the danger of exploding the pad, this was kept to a minimum, 17 cm on the outlet side and about 30 cm on the inlet side. Most of the tubing was rigid copper, with appropriate valving (V_1, V_2) sweated in to allow removing the ice maker without draining the system; the icemaker was connected to this rigid system by a few cm of flexible PVC tubing at each end to facilitate insertion and removal and permit height adjustments.

At first the 2.1 m long exit tube had an inside diameter of 0.76 cm (nominal 3/8 inch OD), but at the flow rates involved this gave rise to a backpressure of 7 kPa (0.7 m of water) which was excessive given the design constraints on the ice maker's polymer film. Increasing the exit tube diameter to nominal 5/8 inch tube with an inside diameter of 1.4 cm decreased this Poiseuille's Law pressure to 0.7 kPa, which was essentially negligible. This large diameter made it harder to get the tube completely filled with coolant and syphoning, so the circuit and valve V_3 were added to allow the icemaker pad to be bypassed during startup until all air had been purged from the system.

Thermocouples immersed directly in the coolant measured the coolant temperature at points inside the manifolds. Combined with the rate of flow and the specific heat, the datalogger calculated instantaneous values of the thermal power, which of course had to be averaged over several transit times of the coolant through the heat exchanger to be significant.

1.2 Slush Ice Tank

Another well-insulated 85 liter (22 gal) polypropylene tank served as the slush reservoir, containing the water to be frozen and the ice maker itself, submerged in a vertical plane so that its topmost tube was 15 cm and the bottommost tube 33 cm below the surface. Removable transparent slabs could be used to partially or completely cover it as convenient, but during ice making, the top was often wide open to facilitate removal and weighing of the ice. A framework of labstands supported the plumbing, instrumentation and the ice maker itself. A small laboratory stirrer was used to prevent stratification when desired.

Four type "T" thermocouples were immersed in the water, two permanently fixed at one end of the tank 4.0 cm above the bottom and 4.0 cm below the water surface respectively ("water high" and "water low"), and two which could be brought into external contact with the top and bottom of the pad ("pad high" and "pad low"), for example, to check for supercooling.

1.3 Data Acquisition and Analysis System

The primary data taking and analysis system was a Campbell CR7-X datalogger with 64 kilobytes of memory and a wide variety of input modules for reading thermocouples voltages and both digital and analog output control channels. Four of the digital output channels were connected through solid state relays to permit control of pumps, heaters and the chiller. The Campbell was connected through a serial port to an AT microcomputer, and proprietary software from

Temperature measurements:	Time:
Coolant tank	Real time
Coolant into ice maker pad (T_{in})	Chiller elapsed time
Coolant out of ice maker pad (T_{out})	Ice maker cycle elapsed time
Water bath - high (T_{high})	
Water bath - low (T_{low})	
Pad external - high	Calculations:
Pad external - low	Power through pad
	Heat through pad this cycle
Flow measurements:	LMTD of pad
Coolant flow through chiller	
Coolant flow through pad	

Table 1-1: Datalogger Functions

Campbell was used in the AT to prepare and download programs to the datalogger and to import data into the AT for permanent storage.

During runs, the datalogger would take the data and calculate the intermediate quantities indicated in Table 1-1 every second, and display these values instantaneously and appropriately labelled on the monitor of the AT. In addition to the raw data, the datalogger calculated various quantities such as instantaneous thermal power and displayed them on the terminal in real time; these derived quantities were recalculated during analysis as discussed below.

Every 15 seconds the datalogger would send 15 second averages of these quantities into its final storage registers. The datalogger was capable of holding several hours worth of data at this rate, which at the end of the run was uploaded to the AT for manipulation in a spread sheet.

The spread sheet routinely held and analyzed up to three hours worth of data, at four points per minute, for the dozen primary measurements and several derived quantities. The derived quantities included the instantaneous thermal power, given by

$$P = \rho c f (T_{out} - T_{in}),$$

where ρ and c are the density and specific heat of the coolant discussed above and f is the volumetric flow rate. Also calculated were the integrated energy over each ice making cycle and the instantaneous log mean temperature difference (LMTD) of the pad, from

$$LMTD = (\Delta T_1 - \Delta T_2)/Ln(\Delta T_1/\Delta T_2),$$

where

$$\Delta T_1 = T_f - T_{in},\ \Delta T_2 = T_f - T_{out}$$

and

$$T_f = 0\ ^oC.$$

One might expect the LMTD to be based on the temperature of the water bath, usually 0.5-1.5^{o}C, but this would not be correct. Almost all the heat being removed by the coolant is the latent heat of fusion as ice is formed, $Q_f = mL_f$ where m is the mass of ice and L_f the heat of fusion, 334 J/g, and this takes place at 0^{o}C. A small amount of sensible heat, $Q_s = mc(T_w - T_f)$ is also removed, but with $T_w - T_f = 1^oC$, the total heat removed by the coolant is

$$Q_t \simeq Q_f + Q_s = Q_f(1+c(T_w - T_f)/L_f) = 1.013xQ_f.$$

Since our energy measurements have much larger uncertainties in them, we ignore this correction.

Using the LMTD, the instantaneous power, P, and the area of the pad, A, the conductance of the heat exchanger is calculated for each data point from U = P/(A*LMTD), for comparison with the value calculated in Section 2.2 below.

2. Icemaker Pad Design

In the previous series of experiments ice built up rapidly on those portions of the icemaker pad which were cooled by coolant but which did not flex during the drain (harvesting) cycle. In this section, we report the design, construction and analysis of a heat exchanger pad which prevents the buildup of parasitic ice by ensuring that all surfaces exposed to flowing coolant flex sufficiently during cycling to shake off the newly formed ice. This task required several cycles of design, test and refinement, although we report in detail only on the final, successful version of the ice maker, which is shown in Fig. 2-1.

2.1 New manifold and pad design and construction

The inlet and outlet manifolds, designated as "rigid tubing" in Fig. 2-1, consist of large inlet tubes (inside diameter 1.6 cm, length 50 cm) and of small nipples (inside diameter 0.64 cm, outside diameter 0.95 cm, length 3.50 cm) which screw into tapped holes in the large tubes. Since they were designed with the goal of preventing ice buildup by insulating the surface from the coolant, they were constructed of thick segments of polymer tubing. For example, the wall thickness of the large tubes was 36 times the 75μ (3 mil) thickness of the ice making surface itself, and since the thermal conductivities are comparable, we expected the outer surface of the thick tubes to stay quite close to the temperature of the water bath, 1 or 2^{o}C. (See Section 2.2 below.) There are ten nipples on each large tube, on center to center spacings of 1.74 cm. The ends of each tube are plugged.

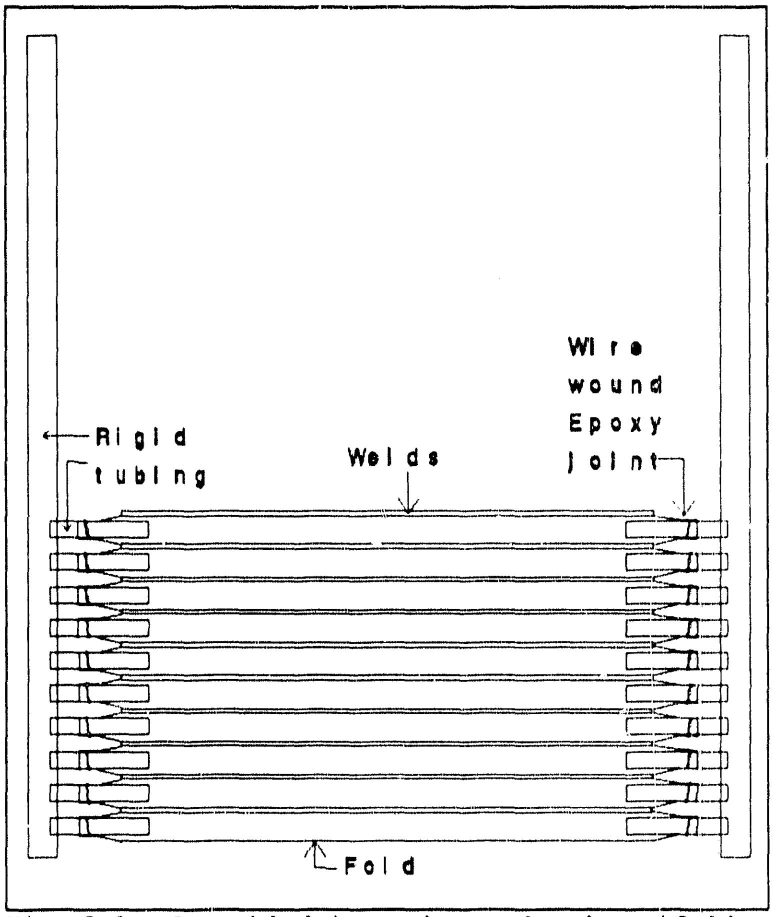

Fig. 2-1: Assembled ice-maker pad and manifolds.

The original choice of material for the manifolds was PMMA ("acrylic"), so that trapped and transiting air could be observed, but it proved too brittle for fabrication of the large tubes, which were made of PVC. With more time, we would also have re-made the nipples from PVC, since the threads broke off the brittle material on a regular basis.

The pad is constructed from a single sheet of PFA (perfluoroalkoxy) "Teflon", 75μ (3 mils) thick and originally 30 cm wide and 42 cm long. As shown in Fig. 2-2, it is folded back on itself and ten welds, at 2.1 cm intervals, then form ten tubes. Each weld is 3.2 mm wide, and is split in half at each end for about 2 cm, as shown in the detail in Fig. 2-2, to separate the tubes. Each tube is then drawn onto a nipple which has previously been smeared with epoxy, and is tied in place with a two turn wrap of fine wire. Although tedious, this sealing method worked reliably when done carefully, and no leaks developed once a given joint was successfully prepared. Of course the epoxy had not bonded to the PFA, which could readily be pulled off if the joint was disassembled. Several other methods, such as thermal welding of the film onto PFA nipples, will be preferable for automated production but for various reasons were not practical here.

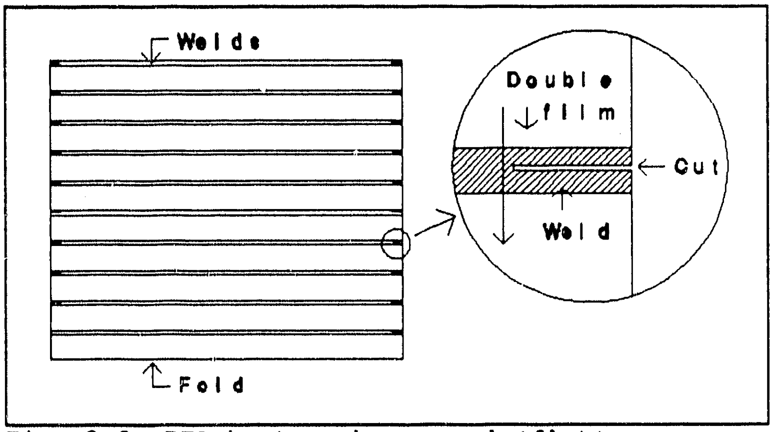

Fig. 2-2: PFA heat exchange pad (flat).

The total ice-making area of each side of the pad is then 10*(2.10-.32)*(30-2*0.5) = 516 cm^2, since 0.5 cm is lost to the joint at each end and the welds occupy 0.32 cm each. The two sides then give a total ice-making area of 1030 cm^2 = 0.103 m^2. During the experiments, we lost the use of three tubes; this reduced the total area to A' = 0.7*A = .0723 m^2, the value used for all energy data analyzed in this report.

This design is a substantial improvement over earlier efforts, where a large PFA bag either formed[7] or surrounded[8] the manifolds. The only reason for that more cumbersome design was the difficulty of making leak-free joints between the PFA film and the nipples. and once that was accomplished, the move to having only the ice-making surface made from PFA film was obvious.

2.2 Thermal Analysis

The low thermal conductivity of polymers often leads to doubt about their use in heat exchange applications. Here we briefly show that a thin film is capable of making ice at an acceptable rate, and that the low conductivity is helpful in preventing ice buildup on the rigid manifolds.

Three terms contribute to the thermal resistance between the cold refrigerant and the water to be frozen: the resistance of the boundary layer in the coolant, the resistance of the polymer film itself, and the resistance of any ice which has already formed. We follow a standard heat transfer analysis from ASHRAE handbooks[15].

The boundary layer heat transfer of course depends on the fluid flow.

Taking as nominal conditions 2.0 liters/min total flow, seven tubes of diameter 1.13 cm will give rise to a flow velocity of 4.75 cm/s. The viscosity of a 55% ethylene glycol mixture at $-5.5^{o}C$ is about 14 mPA-s[14], giving rise to a Reynolds number of 41, a very low value which ensures laminar flow and poor heat transfer. (This choice of dimensions resulted from a desire to be conservative with respect to over-pressuring the thin polymer films; in future designs the flow should be sped up considerably, as discussed in Chapter 4.) The thermal conductivity of the coolant is 0.41 W/m-°C and the specific heat is 3400 J/kg-°C[14], so the Prandtl number works out to 116 and the heat transfer coefficient to h = 380 W/m^2-C[15].

The PFA film used for the ice making surface has a thermal conductance of about 990 W/m^2-°C for the 75μ (3 mil) film used here[16]. The conductivity of the ice is 1.6 W/m-°C, and the conductance will of course vary with thickness, from an initial value of infinity down to 1600 W/m^2-°C at a (presumed) maximum thickness of 1.0 mm. Since the three heat transfer coefficients add as reciprocals, it is clear that the poor heat transfer out of the coolant will dominate, and with a 0.5 mm thick layer of ice we can expect overall heat transfer at a rate of 256 W/m^2-o^C, which combined with the latent heat of fusion of ice of 334 J/gm and a coolant temperature of $-5.5^{o}C$ leads us to expect ice to build at an average rate of 0.25 mm/minute.

We next show that ice will form on the rigid tubes so slowly it can be easily managed. The thermal conductivity of the PVC is $(3.5 - 5.0)x10^{-4}$ cal/cm-s-°C[16], so for a thickness of 0.26 cm and coolant at $-5.5^{o}C$, we expect heat transfer at a rate of 31-44 mw/cm^2. (The thermal resistance of the polymer is here so large that we can ignore the thermal resistance of the water or the ice.) The heat of fusion of water is 334 J/g, leading to a maximum expected ice build up of 3 - 5 mm per hour. This level is perfectly acceptable for these experiments, and can be reduced as needed in commercial systems by adding increased insulation to the rigid tubing.

3. System Testing

In this section, the core of this report, we present the results of performance and energy balance testing of the icemaker pad and new manifold described in Chapter 2.

3.1 Formation of Ice

Operation of the system is straightforward: the coolant reservoir is chilled to -5.5°C (or other chosen temperature) by the chiller in about twenty minutes. At this point cold fluid is circulated through the pad and the data described in Chapter 1 is displayed on the monitor and recorded in the datalogger. Using this small pad and the coolant to lower the fresh water tank to near 0°C would take several hours, so it is pre-cooled with shaved ice to 1 or 2°C. Stratification appears (with the cold water on top, at these temperatures) and the stirrer is used to defeat it to speed melting of the pre-cool ice.

There then follows a tedious period where the pad is cooling the tank of water, which is at 1 or 2°C, which can last for an hour. Heat transfer is slow, about 100 W/m^2-°C, 2.5 times less than the predictions of Chapter 2. The external thermocouples normally indicate 0 to +0.5°C, but have been seen to dip as low as -0.7°C, with no ice formation, if they are in direct contact with the pad. This may indicate a slight degree of supercooling, but it is more likely that the thermocouple itself is being cooled by the pad to slightly below water temperature.

Finally ice forms on the rounds; when it appears to be one millimeter thick the pump is shut off and the pad drains and flattens, and the freed ice floats to the surface as expected. Initially, the draining process took 30 to 40 seconds, but we early found that it could be speeded up to 10 to 15 seconds by opening valve V_3 in Fig. 1-1 so that coolant was being drawn from both ends of the pad. This was routinely done thereafter.

Once the first cycle of ice formation occurred, the succeeding layers of ice formed rapidly (a few minutes per millimeter) and harvested readily. We have no explanation of why the first layer of ice takes so long to be initiated, but the behavior is consistent with some sort of cleansing or conditioning of the polymer surface by that first layer. We tried cleaning the pad with soapy water and acetone before immersing it, but this did not speed up initial ice formation significantly.

3.2 Testing for parasitic ice

The next criterion for successful operation was the ability of the pad to shed <u>all</u> the ice that formed on it, cycle after cycle. If there is any buildup of residual ice, as there was on the first generation system, the entire system will rapidly become inoperable. We conducted multi-cycle tests on three different days.

On the first day, after eight or nine cycles there was no sign of parasitic ice. The ice was, however, often coming off with the semi-cylinders connected to each other, indicating ice formation on the welds. This behavior was present during all subsequent tests and indicates that it may be desirable to use wider welds so that the two semi-cylinders cannot "grow" together as they thicken enough to reach across the weld. During the eight or nine cycles the laboratory stirrer had been on, and since it was mounted on the same framework as the ice maker, it provided a gentle shaking of the whole apparatus as well as stirring the water. It was shut off, and two more harvests were carried out, again leaving no parasitic ice. The shaking from the stirrer probably sped the harvest up, but was not necessary.

On the second day, more than twenty cycles were carried out. After one of them a slab of several semi-cylinders became stuck to something low down on the pad, but it floated free without intervention after the next cycle. After this many cycles we did begin to observe a thin layer ice building up on the inlet and outlet manifolds, but it did not appear to connect to the harvestable ice or to interfere with harvesting.

On the third day we made ice for two hours and thirty minutes, running through numerous cycles. For the last half hour there was a tendency for slabs to stick on one side of the pad that had inflated in a slightly convex overall shape. Either the stirrer or slight shakes of the frame were enough to dislodge them, and might be a good insurance policy in an installation. However, with wider welds insuring separate semi-cylinders, they probably would not have gotten stuck in this manner.

We conclude that the problem of parasitic ice can be completely controlled by the present design, augmented by wider welds and perhaps by slight shaking of the ice maker pad during harvesting.

3.3 Adiabatic Energy Balance

On each of the three days the datalogger kept detailed records, as discussed in Chapter 1. In this section we present the results as recorded in these data, and compare them to the amount of ice actually harvested, where this was done. As we will see, the device operates approximately as predicted in Section 2.2.

On the first day, no ice was actually weighed, but steady ice formation was observed in the tank, and the apparent thickness was roughly consistent with the recorded thermal data, which are displayed in Tables 3-1, showing the detailed data leading up to the 6th harvest of that day. The various data were mostly discussed in Section 1-3; "ET" is the elapsed time since the run began, "Ref" is the datalogger reference temperature, essentially room temperature, and "t" shows the expected thickness of ice buildup, calculated directly from Q, the integrated energy that cycle, L_f and A.

ET (min)	hr-mi	sec	Tank	PadIn	PadOut	WtrHi	WtrLo	EPadHi	EPadLo	LMTD	Ref	PadFlw (L/min)	PPad (W)	U Pad W/m2C	Q/cycl (kJ)	t (mm)
			Temperatures (C)													
22[illegible].5	1843	45.7	-4.8	-3.53	-2.40	1.02	1.12	1.12	1.15	2.93	29.21	0.04	2	11	0.0	0.00
227.8	1844	0.7	-4.8	-3.28	-2.13	1.03	1.13	1.12	1.16	2.66	29.21	0.71	43	222	0.0	0.00
228.0	1844	15.7	-4.9	-3.37	-2.10	1.03	1.13	1.12	1.16	2.68	29.21	2.09	143	735	0.7	0.03
228.3	1844	30.7	-5.1	-4.03	-3.18	1.03	1.13	1.13	1.15	3.59	29.21	2.09	95	367	2.8	0.12
228.5	1844	45.7	-5.4	-4.31	-3.48	1.04	1.13	1.13	1.15	3.88	29.21	2.08	93	331	4.2	0.18
228.8	1845	0.7	-5.6	-4.52	-3.69	1.03	1.13	1.11	1.15	4.09	29.21	2.08	93	313	5.6	0.23
229.0	1845	15.7	-5.6	-4.71	-3.84	1.03	1.13	1.11	1.13	4.26	29.21	2.08	97	315	7.0	0.29
229.3	1845	30.7	-5.7	-4.93	-3.98	1.04	1.13	1.12	1.16	4.44	29.21	2.07	106	329	8.5	0.35
229.5	1845	45.7	-5.8	-5.14	-4.16	1.03	1.13	1.11	1.14	4.63	29.21	2.07	109	324	10.1	0.42
229.8	1846	0.7	-6.3	-5.29	-4.34	1.03	1.13	1.12	1.13	4.80	29.21	2.06	105	303	11.7	0.48
230.0	1846	15.7	-6.6	-5.52	-4.48	1.03	1.13	1.12	1.14	4.98	29.21	2.05	115	319	13.3	0.55
230.3	1846	30.7	-6.6	-5.62	-4.68	1.03	1.13	1.11	1.16	5.13	29.21	2.05	104	281	15.0	0.62
230.5	1846	45.7	-6.5	-5.60	-4.69	1.03	1.13	1.12	1.14	5.13	29.21	2.06	101	272	16.6	0.69
230.8	1847	0.7	-6.5	-5.58	-4.67	1.03	1.14	1.13	1.15	5.11	29.21	2.06	100	271	18.1	0.75
231.0	1847	15.7	-6.4	-5.50	-4.64	1.03	1.14	1.10	1.15	5.06	29.21	2.06	96	262	19.6	0.81
231.3	1847	30.7	-6.3	-5.43	-4.55	1.04	1.14	1.12	1.16	4.98	29.21	2.07	97	270	21.0	0.87
231.5	1847	45.7	-6.2	-5.27	-4.45	1.04	1.14	1.12	1.14	4.85	29.21	0.75	34	98	22.5	0.93
231.8	1848	0.7	-6.1	-4.09	-4.09	1.03	1.14	1.11	1.15	4.09	29.21	0.04	-0	-0	23.0	0.95

Table 3-1: Detailed heat transfer data during ice-making.

Note that although "U Pad" can fluctuate quite a bit, it is essentially consistent with the 256 W/m^2-^oC predicted in Section 2.2. Before the start of ice building, U was both smaller (often less than 100 W/m^2-^oC) and more erratic, reflecting both the thermal resistance of the water bath and the effects of eddies and fluctuations from convection or stirring. In the interest of brevity, data tables of the sort shown in Table 3-1 will not be repeated, but they lie behind all the secondary data to be presented below. In all, eight such harvests were recorded on the first day, all calculated to produce ice between 0.5 and 1.2 mm thick, which was consistent with rough visual observation.

Table 3-2 summarizes the ice harvests for the second day of tests, showing the time of each ice harvest, the duration, the average U during ice building, the total energy carried away by the coolant, and the expected ice production if all this energy represented latent heat of fusion. The calculated thicknesses are omitted, since they are impossible to measure accurately.

For one individual harvest, and then for all succeeding harvests, the ice was scooped off the top, thrown into an empty beaker and subsequently measured on a calibrated electronic balance. For the single harvest which occurred at 16:23 hours, The Q of 13.5 kJ implies that a mass of 40 gm was produced. The corresponding mass measurement gives 64 gm. Since it is hard to tell how much of the water draining off the ice is stray liquid and how much is melting ice, it is possible that an extra 24 gm of water was picked up, although it seems unlikely. For the rest of the harvests, the agreement is much better: summing the remaining Q's in Table 3-2 gives a total of 314 kJ, which corresponds to 939 gm of ice, and the measured mass turned out to be 931 gm. We will see below that agreement <u>this</u> close is probably fortuitous, but it is none-the-less satisfying. Combining the total (weighed) ice production with the duration over which the it was produced and the area of the pad, we get an overall rate of 0.18 kg/m^2-minute, which includes harvesting time.

Harvest #	Time (hh:mm)	Duration (mins)	PadFlow (l/min)	Q (kJ)	-- m(ice) in gm - calc	meas
1	16:00	?	3.3	?	?	?
2	16:03	2.75	3.3	10.8		
3	16:09	5.75	3.3	13.2		
4	16:16	7.25	3.3	30.1		
5	16:19	3.25	3.3	12.4	37	
6	16:23	4.00	3.3	13.5	40	64
7	16:28	4.75	3.3	26.3	79	
8	16:31	2.75	3.3	12.8	38	
9	16:34	2.75	3.3	14.1	42	
10	16:37	3.25	3.3	15.3	46	
11	16:41	4.50	3.3	25.5	76	
12	16:46	5.00	3.3	16.9	51	
13	16:50	3.25	3.3	14.8	44	
14	16:55	5.25	3.3	24.3	73	
15	16:59	4.25	3.3	18.2	54	
16	17:03	3.75	3.3	18.1	54	
17	17:06	3.50	3.3	17.6	53	
18	17:13	7.00	3.3	23.1	69	
19	17:17	3.25	3.3	14.1	42	
20	17:20	3.50	3.3	14.0	42	
21	17:26	5.75	3.3	20.3	61	
22	17:33	6.80	3.3	24.8	74	
		------			------	------
Totals (#'s 6-22)=		73.3			939	948
Measured (weighed) performance=					.179 kg/m2-min	

Table 3-2: Ice-making performance, second day of tests.

On the third day of tests, we measured both the rate of ice production and coolant pump flow, to see if the expected dependence could be observed. The data are summarized in Table 3-3, which is even more aggregated than Table 3-2: here we have already added together the several harvests and indicate only the total production and production rate found at each pump speed. For each run we show both the "calculated" ice production, meaning the mass of ice expected from the total heat transferred out of the coolant for the harvests involved at each flow rate, and the "measured" production, which refers to the actually weighed mass of ice. The theoretical ice building rate, derived completely from the heat transfer analysis methods of Section 2.2, is also shown.

Agreement is clearly poor between weighed and calculated production rates at the lower flow rates. Taking the actual mass of the ice produced, we find an overall production rate at the three higher flow rates of 0.16 kg/m^2-minute, completely consistent with the previous day's run. The last column makes it clear that the ice building rates we observe are about one half of what we projected theoretically in Section 2.2. At this time we have no explanation for this discrepancy. Our data is so scattered that we cannot assert that the one-third power law is being observed, but the general trend is certainly for increased heat transfer with increased flow velocities. The calculation assumes an average ice thickness of 0.5 mm, and does not include the dynamic increase in

Pad Flow (l/min)	Number of Harvests	-Total mass of ice- Calc. (gm)	Weighed (gm)	Duration (mins)	---- Production Rates ---- Calc. (gm/min)	Weighed (gm/min)	Theor. (gm/min)
3.3	5	187	189	17.25	10.8	11.0	20.3
3.0	5	172	188	16.75	10.3	11.2	19.9
2.5	4	118	198	16.00	7.4	12.4	19.1
2.0	3	110	148	16.00	6.9	9.3	18.2
1.5	4	247	174	24.00	10.3	7.3	17.1

Table 3-3: Ice-building rate versus coolant flow rate.

thickness and thermal resistance as buildup occurs, but it is unlikely that this would produce an error of a factor of two.

Even if there are factors which may allow the efficiency to be increased dramatically, the experiments reported here demonstrate conclusively that this device will produce ice reliably at a rate of 0.16 - 0.17 kg/m^2-minute, or about 10 kg/m^2-hour.

3.4 Optimization of Operating Parameters

In the course of running these experiments we noted various constraints and dependencies in the operating parameters that will be useful in later design efforts. With respect to coolant temperature, the system might well run more efficiently if the coolant could be used at a somewhat higher temperature, say -3 to -4°C, rather than -5.5°C as it was here. If this is true (it will depend on the chiller being used), it will still be necessary to start out with the coolant at -5°C or -6°C, as we tried several times to produce ice with warmer coolant and never saw the initial layer of ice form. We did not try raising the coolant temperature after initial ice formation.

The experiments reported in the previous section demonstrate that higher coolant flow rates are better, although rough calculations indicate that reasonable flow rates will give rise to at most a 50% increase in heat transfer rate. The next prototype should certainly include the possibility of measuring this.

Our experience with cycle time, and the calculation of Section 2.2 indicating that only a small part of the thermal resistance is due to the ice, indicates that if the ice thickness is held to about 1.0 - 1.5 mm, the cycle time is not a key variable for efficiency. Since letting the ice get much thicker would interfere with harvesting, we conclude that the cycle time is not a critical design issue, and can be set to any convenient values giving about 1.0 mm of ice at harvest time.

4. Advanced Design Concepts

In this task we assessed and tested various advanced design alternatives, including the use of alternative polymers, the possibility of blow molding the heat exchanger pads, and the possibility of the direct use of compression refrigerants inside the icemaker pad in place of the coolant. We also developed an operational leak detection system.

4.1 Alternative polymers

In the course of building and testing the ice maker we have tried constructing pads from a variety of polymers selected on the basis of strength, thermal conductivity and price. These have included FEP Teflon (fluorinated ethylene-propylene) in several thicknesses, PET (polyethylene teraphthalate, Mylar) and two varieties of PVDF (polyvinylidene fluoride). In the end none of them could survive the stress of pressurization the way PFA did, and this is consistent with its very high ratings for "elongation at break", or "stretchability"[16].

4.2 Blow Molding of Icemaker Pad

As part of this project we had intended to try to overcome the lack of "stretchability" in less expensive polymers by "blow-molding" them to shape. In this technique, a heat exchanger pad would be prepared just as was described in Section 2.1 above, but would then be heated to its softening point and then inflated. In this way it would take on the inflated, rather than the deflated shape, so that when inflated at low temperatures, stresses would be evenly distributed through the material rather than concentrated at a few highly stressed points as they are now. Unfortunately, a shortage of time for more pressing tasks coupled with an unexpected move of the laboratory space made it impossible to pursue this option.

4.3 Leak Detection

There is no doubt that the polymer membrane comprising the "pad" will rupture occasionally, and when this happens the ethylene glycol in the coolant must not be allowed to mix with the water in the ice bath, since this would then require draining the entire system and refilling it with fresh water, a time-consuming and expensive operation. We have demonstrated that it is quite easy to detect such leaks electronically, probably even localizing them to a particular pad in a multi-pad array, which can then be quickly drained and disconnected by electrically controlled valves.

The key is simply that both the water and the coolant, unless totally de-ionized, are modest electrical conductors, while the polymer membrane is a near-perfect insulator. We demonstrated this by dipping the probes of a digital multi-meter's ohm-meter into the ice tank and into a prepared PFA bag filled with coolant. With the bag intact the meter indicated "overload", corresponding to over 30 megohms resistance. When a tiny pinhole was punched in the bag the resistance immediately dropped to 0.2 megohms, a readily detectable level for an automated system. The only problem with implementing this system will be the necessity of avoiding a common ground between the coolant and the water/ice circuit, which would short-circuit the detection voltage.

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