ECONOMIC ASSESSMENT OF THE THIN POLYMER ICEMAKER

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Richard W. Leigh
Associate Professor of Physics
Pratt Institute
Brooklyn, NY 11205

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Energy Efficiency & Conservation Division
Department of Applied Science
Brookhaven National Laboratory
Upton, New York 11973

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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. We here report an economic assessment of this device. In its present form, we find it competitive with existing commercial ice making systems. The analysis also points out two areas where further technical progress could lead to a significant economic advantage for the polymer film ice maker.

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0. Introduction and Summary

A new device for the production of ice has been designed, constructed and tested as part of the District Heating and Cooling program at Brookhaven National Laboratory (Andrews90; Leigh91a,b). Here we report on an economic assessment of the potential of this device to gain a significant share of the market for energy-related ice making systems. Such systems consist primarily of slush ice production for district cooling and ice production for cold storage systems in buildings.

The study examines both the capital costs of the ice-making systems and the ongoing operating costs, taking into account the different operating efficiencies of the various systems. We compare existing alternatives to the thin polymer device, including direct contact slush ice production and metal plate heat exchangers with reheat. Since the assessment is based directly on the design for a thin polymer system which was tested in our laboratory, we also examine the potential economic benefits of various design changes and improvements which could be made to the current design.

We have not examined ice-building machinery, where the ice is grown directly on cold coils, which are subsequently used as heat exchangers to deliver heat to this ice. First, they are not suitable for producing slush ice, since the ice is inseparably bound to the pipe around which it formed, and so is unavailable even for chopping and crushing. Second, the economics of these systems are quite different, since storage capacity is proportional to heat exchanger area, and the resulting economic disadvantages of the ice-builders would unnecessarily complicate the analysis.

The results of the comparison are summarized in Table 1. The top line describes a system designed directly from the characteristics and performance of the polymer heat exchanger tested in our laboratory. The second line describes a more hypothetical thin polymer system incorporating several reasonable but as yet untested design improvements. The third line shows an equally hypothetical system based on ice formation resulting from direct contact between the ice and the evaporating refrigerant (Winters91); this system will probably never be practical due to problems involving the release of fluorocarbons to the atmosphere. The last two lines describe currently available commercial systems, differing primarily in the shape of the metal evaporator structures on which the ice is formed.

Table 1: Comparison of System Economics.

System Type	Capital Outlay ($1000)	Monthly Electric Bill ($1000)	Net Present Value ($1000)
Polymer	340	9.9	553
Advanced Polymer	298	8.1	482
Direct Contact	215-300	9.9	410-490
Flat Plate	316	10.0	524
Tubular Evaporator	372-382	8.7	550-560

The data make it clear that although the polymer system (as tested) should be competitive with the two existing systems, it is not overwhelmingly more attractive at the present time. Several factors might change this: lowering fabrication costs of the polymer system by automating the extrusion of large system components would help considerably, for example. Improving heat transfer into the brine by increasing its speed through the tubes of the pad will increase the rate of production of ice and permit a reduction in the number of pads and a proportionate decrease in capital costs. Other "incremental" imrovements are also possible.

Also, the existing machines have a significant operating expense advantage due to two more fundamental factors. First, because the refrigerant acts directly on the ice forming surface, rather than through the intermediary of a cold brine, their evaporator temperatures can be several degrees higher than that of the chiller in the polymer system, giving them relatively higher COP's. This factor also has a major impact on capital cost, since the capacity of the chiller is degraded significantly as the evaporator temperature drops, so a larger and more expensive machine must be purchased to obtain the same capacity. Second, because their plates are out of the water, they are able to operate for some fraction of the time as chillers with 40°F evaporator temperatures; this also raises the COP significantly.

These considerations led us to cost out a system which would incorporate the most important of these improvements. The line titled "Advanced Polymer" gives our estimates of the costs of constructing and operating a such a system, in which the ice making pad is operated in the air above the tank and in which the refrigerant runs directly into the polymer pad, eliminating the brine loop. The higher pressures involved call for careful design work, but the polymers are compatible with many refrigerants. The calculated costs shown in Table 1 indicate that with these design features, the polymer ice maker could outperform the existing commercial systems by a considerable margin, and that further development of the polymer ice maker along these lines would be a worth-while endeavor.

1. System Performance Specifications

To minimize uncertain elements the study will focus on a system producing slush ice in a storage tank for cooling a commercial building or system of buildings. Distribution of this stored "cold" could then occur by conventional cold water distribution or by slush ice circulation. Since the choice of using water or ice for distribution will have little effect on the comparative economics of the ice makers, we do not include any modelling of the slush ice distribution network here. In all cases a heat pump coupled to an evaporative cooling tower will provide the heat sink for ice formation, as indicated in Figure 1-1. In some cases the circulating fluid will be the chiller's own refrigerant, in other cases an ethylene glycol-water "brine".

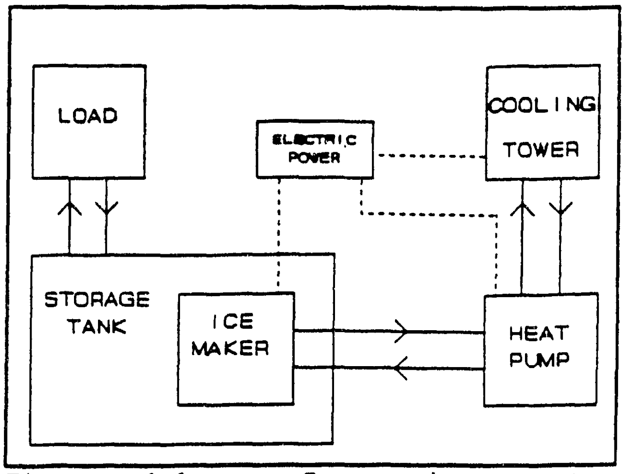

Figure 1-1: Ice maker system configuration.

For definiteness, we consider a system installed on Long Island, NY, purchasing electricity from the local utility (Long Island Lighting Company, LILCO) and meeting a peak design day load of 500 tons. The load factor on the design day will be about 0.75 for the ten hours of actual operation, requiring 3750 ton-hours of refrigeration, or the creation of 156 tons of ice.[1] However, average daily loads will be quite a bit lower. ASHRAE estimates that a properly sized cooling system will operate for 500 to 1000 equivalent full load hours in the New York area (ASHRAE85). Using the lower limit over a 15 week season of six day weeks, we find an average daily load of 2780 ton-hr or 116 actual tons of ice and, if the load factor remains at 0.75, an average daily peak of 370 tons of refrigeration. Recognizing that summers are characterized in part by heat waves, we assume that the worst case load excursion will be presented by a string of four design days in a row, during which time the system must be able to satisfy the load. These operating conditions are specified in Table 1-1.

Table 1-1: System Operating Parameters.

Design Day:		
Peak Load:	500	tons
Load Factor:	0.75	
Length:	10	hours
Seasonal Quantities:		
Length:	15 6	weeks, days/wk
Daily Peak:	370	tons
Load Factor:	0.75	
Max. Successive Design Days:	4	

These details are important because peaking capacity in these systems is very cheap, representing simply the speed at which water can be circulated through the melting ice; short-term capacity is also inexpensive, since it represents the size of the insulated tank or tanks and can be scaled up to

[1]. Note that a "ton" is the standard American unit of refrigeration power, equivalent to one ton of ice per day or 12,000 Btu/hr, a "ton-hour" is 12,000 Btu or 2000/24=83.3 lb of ice, while an "actual ton of ice" or "ton (ice)" refers to the thermal energy content of 2000 pounds of ice, or 286,000 Btu.

contain sufficient reserve ice to meet even the demands of the assumed four-day heat wave. The more expensive average capacity, corresponding to actual ice-making ability, can then be sized to the average load, resulting in substantial savings.

The costs of the electricity used for ice making by the various systems must be compared and in a storage application such as this the effect of time-of-day rates is very important. Accordingly, we have used a set of rates from the local utility (LILCO92), including demand charges, ratchets and so on, to compare the operating costs of the various options. These rates are shown in Table 1-2. Although the exact structure and amounts of the rates may be crucial in comparing storage systems such as these to non-storage based systems, or in comparing the various operating modes discussed below, the operating parameters and methods of the competitive ice makers are so similar that the economic comparisons between them will generally stand, even in regions with moderately different rate structures.

Table 1-2: Electricity Rate Structure.

	Demand ($/Kw)	Energy (¢/kWhr)
Off-Peak 00:00-07:00	0.00	7.00
On-Peak 10:00-22:00 (Mon-Sat)	20.25	9.84
Intermediate (All other hours)	4.83	8.61
Service:	$1.00 per day	

In addition to producing slush for distribution, ice offers a very attractive medium for cold storage, and we will assume that use is made of this storage capacity. Storage can be operated in three modes. In "full storage" the ice maker operates only when inexpensive, off-peak electric power is available, creating ice which is then used for cooling during the day. In the "partial storage" mode, the ice maker runs all the time, making ice with excess capacity at night and cooling with a combination of stored ice and operating chiller capacity during the day. The third option, "intermediate storage", makes ice both during the times of off-peak and "shoulder" or "intermediate" electric rates, but shuts off when the expensive, "on-peak" rates are in effect. These options result in different optimal sizes and economics for the system components, which we examine in Appendix A. There we find that partial storage is optimal for the broadest set of likely circumstances, and we assume this operating mode for the rest of the study. For buildings with cooling seasons of 26 weeks or more (due either to climate or to high internal loads), intermediate storage might well be optimal, but we will not examine this option further here.

Table 1-3: Component Capacities - Partial Storage.

Circulation:	
Water=	1200 gal/min, 45 to 55°F
Ice =	150 gal/min @ 50% ice, 40°F return
Tank Capacity:	
Ice=	5520 ton-hr
Volume=	16,000 ft^3 @ 50% ice
Ice maker Capacity:	
Production=	116 tons (ice)/day
Capacity =	116 tons

The characteristics of a system which will meet the specified load in the partial storage mode of operation are shown in Table 1-3. The "circulation" is that necessary to meet the peak load, while the tank capacity is that necessary to meet that load for the specified "heat wave" of four design days in a succession. The ice maker capacity is then that needed to meet the average load, with ice made for storage when there is no cooling demand from the building.

In the sections that follow we will analyze several systems which will meet these specifications. The baseline thin polymer system and the competitive systems discussed below have many elements in common, such as the cold water or slush ice distribution network, building cold air ducts or wiring systems to supply power. For all these components we will assume that both capital and associated operating costs are the same for all of the systems we are comparing, and may therefore be ignored.

2. Baseline Thin Polymer Ice Making System

In this section we prepare a baseline design for an ice-making system using the thin polymer heat exchanger, estimate its operating efficiencies and work up expected capital and operating costs. For consistency with the experiments we are extrapolating from, in this system a reciprocating compression chiller, with the condenser coupled to a wet cooling tower, will produce cold brine to circulate through the ice-making heat exchanger. At the end of the section we also discuss the impact of using other polymers in the heat exchanger and of making various possible design changes on overall system costs.

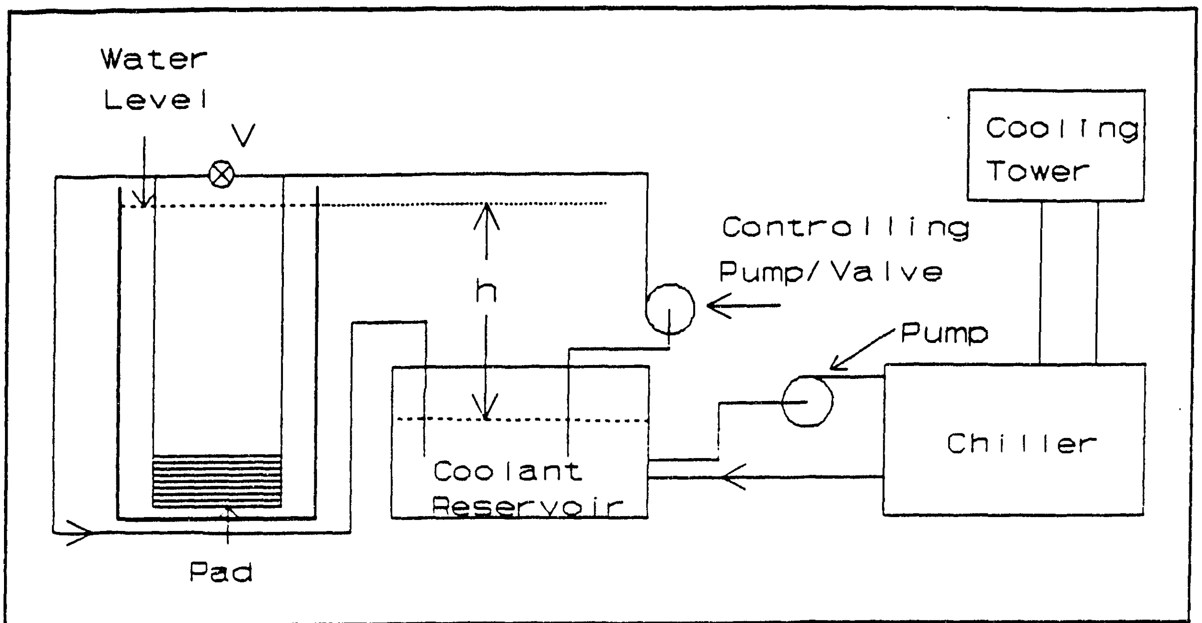

Figure 2-1: System Schematic Showing Typical Icemaker Pad.

Figure 2-1 presents a schematic diagram of the system, showing one typical ice making pad out of 525. The pad resides just off the bottom of the storage tank, so that the volume above it can be nearly completely filled with ice. The controlling pump/valve circulates brine during ice building, then shuts off and seals off the circuit to allow the pad to collapse during harvesting. Analysis will show that the differential pressure crushing the pad will always correspond to "h", the height difference between the top of the brine reservoir and the top of the ice water in the storage tank.

Valve V is a check valve connecting the two sides of each individual pad, so that when its right side is under positive gauge pressure (i.e., when brine is being pumped), it is closed, and when its right side is under negative gauge pressure (when the pump/valve is off and ice is being harvested), V is open to speed drainage of the pad.

Each pad is connected to the electronic leak detection circuit (Leigh91b); when a pad develops a leak, detection of minute electric current through the hole triggers closure of the pump/valve. Static head then drains brine from the leaky pad, and perhaps draws a small amount of system water into it, mitigated by the tendency of the crushed pad to partially seal itself. (This system may prove unnecessary once the reliability of the pads has been established.)

The brine in the reservoir is continually re-chilled to 24°F (-4.5°C) in a liquid-to-liquid heat pump, which is in turn connected to an evaporative cooling tower. Since the leak detection system depends on isolation of the brine from electrical ground, the evaporator of the chiller must be isolated from the rest of the machine by two short lengths of heavy polymer tubing; otherwise it is a completely standard unit, operating with its evaporator at 20°F (-6.5°C). We here assume it is a single unit of 116 tons capacity (at 20°F evaporator temperature), but it could equally well be made up of two or more units in parallel, which would allow more efficient operation in periods of modest load.

Some assumptions on sizing are needed for conceptual design of the system. With a packing density of 50% for the ice after it has floated to the top, 16,300 ft^3 are needed for the slurry. In addition, the pads will occupy a two foot deep volume at the bottom of the tank, so for the 35 ft by 40 ft configuration assumed here, we require a total tank volume of 19,100 ft^3. This will give a total depth of 14 ft in the slurry tank, and a five or six foot deep brine tank will provide a more than adequate crushing head of eight or nine feet of water. For simplicity in construction, we assume that the pads are four times as large as those on which our performance estimate is based (Leigh91b), each having a heat exchange area of 9.0 ft^2 and a polymer area of 11 ft^2, due to the area taken up by welds. Then, to get 116 tons capacity, our observed performance of 2.0 lbs/ft^2-hr (10 kg/m^2-hr) requires a total of 525 pads.

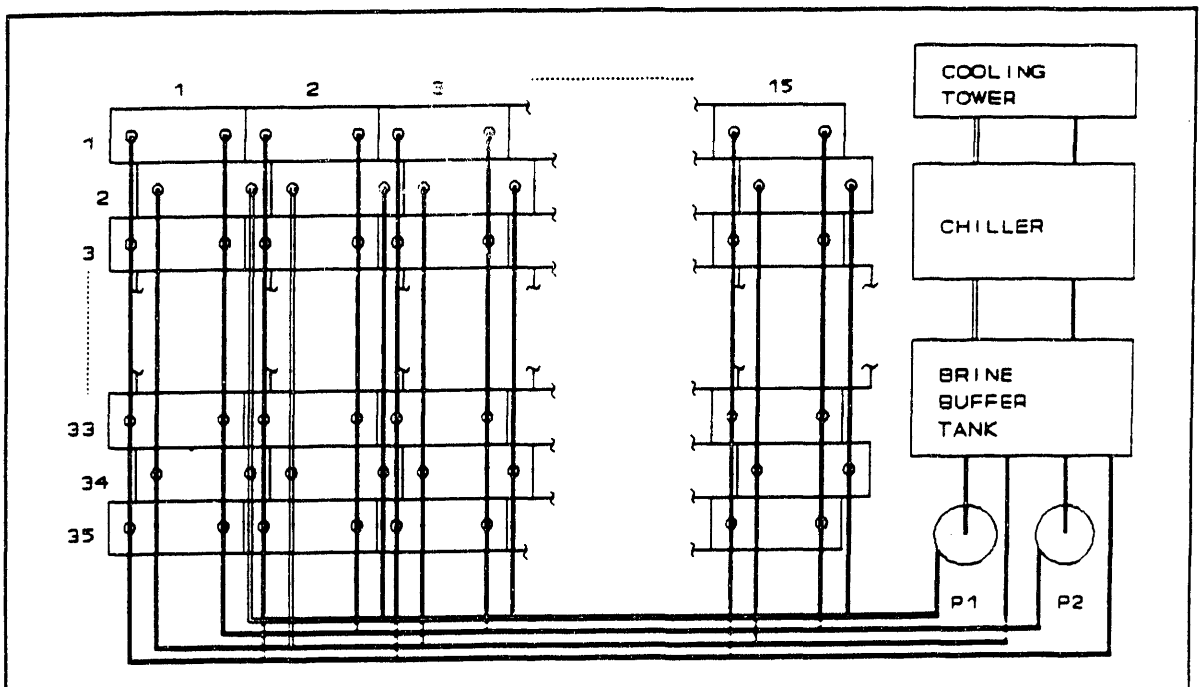

Figure 2-2: Ice Making System Piping Layout.

Figure 2-2 shows a possible layout for piping over the 35 ft by 40 ft tank, with two independent circuits supplying brine to alternating rows of pads, which are arranged in a 16x35 matrix in the ice tank. By alternating the harvests, interference is minimized between ice from neighboring pads as it floats to the top. Note that some of the piping and the valves from Fig. 2-1 have been omitted from Fig. 2-2 for clarity.

Table 2-1 Gives a component-by-component breakdown of our estimates of the capital costs of the polymer heat exchanger. The estimates were derived from a

Table 2-1: List of Components, Materials and Costs for Polymer Film Ice Maker.

Item	Quantity /Size	Units	Unit Cost ($/unit)	Item Cost($)	System Cost($)
Pads:	525				
PFA Film	10.8	sf	1.16	12.53	
PVC Tube	18	ft	0.15	2.70	
Nipples	80		0.10	8.00	
Check Valve	1		11.00	11.00	
Leak Detection	1		1.00	1.00	
Insulation	18	ft	0.33	5.94	
Assembly				40.00	
Pad Total=				81.17	42,614.25
Piping:					
2" PVC	2100	ft	1.50	3,150.00	3,150.00
6" PVC	160	ft	8.25	1,320.00	1,320.00
Fittings	2300		2.50	5,750.00	5,750.00
Assembly	500	hrs	30.00	15,000.00	15,000.00
Brine Pumps/valves	2 @ 110	gpm	1000.00		2000.00
Buffer Tank	350	cf	2000.00		2000.00
Glycol	1225	gals	1.50		1,837.50
Chiller	116(169)	tons	123,000.00		123,000.00
SYSTEM TOTAL:					196,671.75

variety of sources, including industrial suppliers (duPont92, Grainger92), construction cost estimators (Means85) and engineering studies (Schneider86, Christian78). Where necessary costs have been inflated to 1992 dollars using the Consumer Price Index. The cost estimates for the chiller were based on 40°F evaporator operation, and have been inflated by 46% to account for the de-rating that comes at 20°F operation (ASHREA89). Clearly this estimate is not a detailed design study, and some factors may have been omitted or under-estimated.

However, several other areas may permit further reductions in the costs, such as a reduction in the cost of the plumbing achieved by fabricating complex header assemblies in one piece. It is interesting that the cost of the PFA film is a minor part of the whole, and there is little reason to try to economize by using a less expensive polymer, given that the rate of pad failure would very likely increase. We believe the final estimate, $196,000, is within $20,000 of the actual capital costs once the systems are in production.

The various technical factors are summarized in Table 2-2. With 95°F leaving condenser water temperature giving a 100°F condensing temperature, and 20°F evaporator, standard engineering estimates for an R-22 chiller (ASHRAE89) indicate a compressor power requirement of 0.99 kW/ton. Although the system's performance would improve at cooler temperatures, we have used the design point efficiency as a seasonal average, since more accurate modelling would also require hourly modelling and analysis of transients; we have made the same approximation for all systems under evaluation. Cooling tower power of 9 kW is used for all systems with cooling towers in this study, and the brine circulation requirement will be less than 1.0 kW, based on our estimate of 220 gal/min and a head of fourteen feet.

Table 2-2: Technical Characteristics of Polymer Ice Maker.

Capacity (tons):	116
Compressor Power(Kw):	114
Auxiliary Power (Kw):	10
Tank Size (ft^3):	19,100

The economic factors are then brought together in Table 2-3, where we combine the capital cost element of Table 2-1 with our $6.33/$ft^3$ estimate for the storage tank (Landry91) and the various operating costs, derived from the system efficiencies. The electric bill is calculated as in Appendix A, and is added up over a five year planning horizon, with future expenses discounted at six per cent per year. The key numerical results are carried back to Table 1 of the Summary, where they are compared to the four other systems examined in the sections that follow.

Table 2-3: Economic Characteristics of the Polymer Ice Maker.

CAPITAL COSTS	
Ice Maker ($1000):	197
Installation ($1000):	(Inc.)
Tank ($1000):	132
Tower ($1000):	14
OPERATING COSTS	
Electricity ($/mnth):	9920
Maintenance ($/mnth):	1000
PRESENT VALUE	
(Over 5 yrs, $1000):	553

The thin polymer ice making system described above has been based directly on the system tested in our laboratory (Leigh91b) and on conservative engineering estimates, so we can assert with assurance that it will work. There are several design improvements which should make the system considerably more cost-effective, and we discuss here both "incremental" and "substantial" changes. We then use the more substantial improvements to develop a set of costs for an "advanced" polymer ice maker which show that continued development can produce a system which is not merely competitive with existing commercial devices, but which will clearly be economically superior.

Two incremental design changes could improve the economics of the existing system substantially. First, the current rate of ice making, 2 lbs/ft^2-hr (10 kg/m^2-hr) is lower than is predicted by standard heat transfer analysis, about 3 lbs/ft^2-hr (Leigh91b). The source of the discrepancy is not clear, but the same analysis shows that the low heat transfer rate is dictated by the low speed of brine flowing through the tubes (Reynolds number is 40 to 50); by pumping the brine through faster we can increase the heat transfer and ice building rate up to a theoretical limit of twice the current value. This means we could cut the number of pads, almost certainly not by 50%, but by some substantial fraction. For reference, Table 2-1 shows that a 50% cut in the number of pads would cut the system cost by over $35,000 (20%), since almost all components other than the chiller would scale down with the number of pads.

Second, the low evaporator temperature condemns the chiller to inefficient and expensive operation. If the ice making pads were suspended in the air above the tank, with water running over them as it froze, it would be possible to operate the system in "chiller mode", at a 40 - 45°F brine temperature. This mode, which is (only) useful at those times when there is a substantial cooling load, is described in detail in Chapter 4. Both of the commercial systems described below can make use of it and derive substantial saving in electricity consumption by doing so, and the option of moving the pads into the air above the tank will permit the thin polymer system to achieve substantially lower operating costs.

Finally, a substantial design change would improve the performance and economics of the polymer ice maker dramatically. The use of the intermediate ethylene glycol-water brine pushes the evaporator temperature in the chiller down about 5°F below the temperature required to make ice. This both lowers the efficiency of the chiller and increases its cost, since the derating that comes with a low evaporator temperature must be compensated for by buying a larger machine. The obvious answer is to run the chiller refrigerant directly into the ice maker pads, making them into the evaporator. This is not a simple design problem, since the polymer membrane cannot withstand high pressures and chemical compatibility must also be determined. Also, the choice of refrigerant is difficult, given current concerns about the ozone layer; the vapor pressure of ammonia at sub-freezing temperatures (ASHRAE89b) indicates that a system design is difficult but not impossible.

Such a system design is beyond the scope of this report, but by making various simplifying assumptions we have been able to estimate the cost and performance of a machine based on these advances. We assume the evaporator temperature is increased to 25°F, lowering the power requirement to .91 kW/ton (ASHRAE89) and resulting in the technical characteristics displayed in Table 2-4. (As before, we have assumed R-22 is used as the refrigerant.)

Table 2-4: Technical Characteristics of Advanced Polymer Icemaker.

Capacity (tons):	116
Compressor Power(kW):	106
Auxiliary Power (kW):	10
Tank Size (ft^3):	19,100

With respect to capital costs, a system with refrigerant piped directly to the pads would have various economic advantages and disadvantages over the system described in Table 2-1. The large plastic brine piping would be replaced by much smaller metal refrigerant tubing; the buffer tank, brine pumps and ethylene glycol inventory are all eliminated, but the refrigerant inventory is increased and refrigerant pumps may be necessary to flood the evaporator (the pads). What we have assumed is simply that all these changes will cancel each other out, and that the system cost per pad, exclusive of the chiller, will remain what one can easily derive from Table 2-1, $140 per pad.

Table 2-5: Economic Characteristics of Advanced Polymer Ice Maker.

CAPITAL COSTS	
300 Pads ($1000):	42
Chiller ($1000):	110
Tank ($1000):	132
Tower ($1000):	14
OPERATING COSTS	
Electricity ($/mnth):	8103
Maintenance ($/mnth):	1000
PRESENT VALUE	
(Over 5 yrs, $1000):	482

Because the pads are now being cooled by evaporation by 25°F refrigerant, heat transfer is much higher than it was for the brine. Previous estimates (Leigh91b) put the heat transfer coefficient from the brine to the polymer at 65 - 80 Btu/hr-ft^2-°F; for refrigerant in a partially flooded horizontal tube evaporator (ASHRAE89c), the corresponding values are 400 - 1000 Btu/hr-ft^2. Although the heat transfer coefficients for the polymer (175 Btu/hr-ft^2-°F) and ice (560 Btu/hr-ft^2-°F at an average thickness of 0.5 mm) are unchanged, the overall heat transfer for a 25°F evaporator is now 755 Btu/hr-ft^2, which corresponds to a theoretical ice building rate of 5.3 lbs/hr-ft^2. Since in the tested machine we observed 2/3 of the theoretical rate, we will allow the same margin here and expect an ice building rate of 3.5 lbs/hr. Since this is an increase of 76% over the observed rate for the brine-based machine, we should be able to produce ice at a rate of 116 tons from 300 pads instead of 525, a significant saving.

Also, the increased evaporator temperature means that a smaller chiller will still produce the requisite 116 tons capacity. Where for a 20°F evaporator, we needed to upsize by 46% to a machine rated at 169 tons and costing $123,000, with a requirement of 116 tons at 25°F, we need only upsize by 31%, to 150 tons and $110,000[2].

Finally, we assume that the pads are suspended in the air above the tank, and incorporate the "chiller mode" option, with a 40°F evaporator employed eight hours per day, as explained in Section 4, below. This results in about $16,000 savings in the electric bill over the five year planning period, which is incorporated into the data in Table 2-5. The result of these improvements is a system which is substantially less expensive than the system based directly on our laboratory experience. It is compared to existing machinery in the Summary.

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2. There is clearly a need to optimize on evaporator temperature: the colder it is, the fewer pads are required, but the bigger and less efficient the chiller will be. We have done this, using the techniques described here, and find the optimal evaporator temperature to be 24-25°F; we do not include the details since the engineering foundation is still undeveloped.

3. Direct Contact Ice Making System

It is possible to design an ice-making system based on direct-contact heat exchange (Winters91). In this system the refrigerant liquid is introduced directly into the system water and fuses small grains of ice as it vaporizes, producing a slurry. The refrigerant vapor is recovered from the top of the vessel containing the slurry and returned to the compressor. The direct contact ensures very effective heat exchange, and the refrigerant vapor can be as warm as 27°F (-3°C); also, the slurry particles are small (30 to 60 mils, or 0.8 to 1.6 mm), facilitating pumping and manipulation of the slurry.

The system has one dramatic drawback: a small amount of refrigerant remains entrained in the water or slurry leaving the vessel in which it was formed. Although most of this water will probably be returned, it cannot be thought to be in a sealed system; over time, significant amounts of refrigerant would be sure to leak out into the atmosphere. Given the current concern about the effect of chlorine from refrigerants on the earth's ozone layer, widespread use of this system is unacceptable. We nevertheless include the system in this analysis on the chance that a chlorine-free refrigerant can be found.[3]

The technical parameters in Table 3-1 and the economic analysis in Tables 3-2 are based completely on a design for a 10,000 ton district cooling system (Winters91); the numbers are therefore less certain than those for a commercially available system. Since most of the quantities used were scaled up from elementary engineering estimates, little accuracy is lost in scaling them back down again to our 500 ton system. The authors assumed a compressor power demand of 0.91 kW/ton at 20°F evaporator; for consistancy with our design specifications and sources (ASHRAE89), we revise their estimate to a more conservative 0.99 kW/ton. The system was designed with only 20% (by weight) ice in the distribution slurry, although 50% is considered reasonable by others (ASHRAE90); the reason for this design choice was not explained. Since we do not examine the distribution loop in our comparisons, the difference is not important here, and they did assume that a 50% mix could be achieved in the storage tank.

The capital cost factors are taken from Table 4 of Winters91, and are aggregate estimates, including the costs of cooling towers and all other "central plant" installations. Storage is broken out separately, and the range they show, \$15-25/ton-hr or \$5.20-8.60/ft^3, nicely brackets our base storage cost of \$6.33/ft^3 (Landry91). Storage in this system must be a steel tank to ensure maximum retention of the HCFC vapor as it is released from the system water.

Table 3-1: Technical Characteristics of Direct Contact Ice Maker.

Capacity (tons):	116
Compressor Power (kW):	114
Auxiliary Power (kW):	9
Tank Size (ft^3):	16,000

In adding together the capital costs of the central plant and of storage, it is unlikely that they would both be at either extreme of the range of values, and consequently, the range of values for the sum is smaller than what one would calculate by simply summing the extreme values. Although the procedure cannot be justified here as rigorously as it can in the case of physical measurement, we follow the standard theory of errors and treat these ranges as one standard deviation errors around a mean; the standard deviation of the sum is found from

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3) Ammonia, for example, will not work since it will go into solution in water, lowering the freezing point. What is needed is a substance which is not miscible with water, and which can boil at a few degrees below the freezing point of water near atmospheric pressure.

the square root of the sum of the squares of the individual uncertainties and the range in the present value is then calculated by taking the present value based on the average capital cost and adding and subtracting the standard deviation.

The only system considered was based on a water cooled condenser with an evaporative cooling tower. Here we have adjusted their scaling factor, since they assumed a head of 75 feet for the cooling tower pumps; we have taken a head of 25 feet, lowering the cooling tower pump power per ton by a factor of three, resulting in the value shown in Table 3-2. Other components of auxiliary power demand are the chiller feed pumps and the cooling tower fans.

The uncertainties in the operating costs are neglected (both by Winters and Looy and by us); the electricity demand and energy charges indicated in Table 1-2 are calculated for the power levels of Table 3-1 over the 15 week season and totaled over five years at a six percent discount rate. In Table 3-2 the monthly charges are shown and the five year total is incorporated into the present value on the bottom line. The maintenance charge is uncertain since there is no experience with long term operation of these systems; it will cover maintenance for the condenser and compressor and miscellaneous items, but if the refrigerant injector needs care or if the presence of refrigerant causes problems, the bill could be higher.

Table 3-2: Economic Characteristics of Direct Contact Ice Maker.

CAPITAL COSTS	
Ice Maker ($1000):	115-175
Installation:	(Included)
Tank ($1000):	83-140
Tower:	(Included)
OPERATING COSTS:	
Electricity($/mnth):	9850
Maintenance($/mnth):	1000
PRESENT VALUE:	
(Over 5 yrs, $1000):	410 -490

The present value is the sum of the capital outlays (which are not discounted, since they take place in year one) and the annual expenses, discounted at 6% per year over the first five years of the life of the system. These over all costs are compared with other systems in the Summary at the beginning of this report. It is important to remember that, like the polymer film device, this system is hypothetical and has much greater uncertainty in both its operating characteristics and in its costs than do the two following systems, which are currently commercially available.

4. Plate Ice Making System

Ice-making systems based on a flat metal plate heat exchanger using a re-heat release mechanism are currently being widely marketed for cold storage. We base our description directly on one manufacturer's specifications (Mueller92), although this should in no way be taken as an endorsement or evaluation of this particular product. We assert only that these units are exemplary of what is currently available, and use them to estimate operating efficiencies and work up expected capital and operating costs.

These systems are similar in concept to the polymer ice making system described in Chapter 2, but there are several important differences in construction and operation. The metal plates on which the ice is built up are suspended above the storage tank and water is run over them, spread out into sheets by a delivery manifold and irregularities in the surface of the plates. The plates are themselves the evaporator of the compression chiller providing cooling, resulting in the simplicity of eliminating the brine circuit of Chapter 2, but the expense of a substantially increased inventory of refrigerant.

Harvesting of the ice is carried out by automatic valving which directs hot condenser vapor into the plates when the ice has reached a specified thickness (typically five-sixteenths inch); a thin layer of ice in contact with the plates melts, and the rest of the sheet of ice falls into the water below, usually breaking as it does so. The valving reverts to the ice building mode and the plate becomes an evaporator again as the next layer of ice begins to build up. The systems are technically mature, incorporating many engineering details aimed at increasing reliability and efficiency. The main technical characteristics of the system are summarized in Table 4-1; the main economic characteristics in Table 4-2.

One feature affecting the economic performance is the option of running the machine as a chiller with a 40°F evaporator temperature, which substantially increases both COP and capacity. If building cooling is in process, return water is likely to be above 50°F and will not be frozen on the plates, since all capacity will be used cooling it down to 32°F or above. Thus this option can be switched on during on peak hours to cut demand and energy charges at a given capacity level. The process works by pre-cooling the 50°F returning water to 44°F before returning it to the ice tank, so the rate of melting is greatly reduced. Clearly this can only work when the system water is being heated to above 45°F. For our rate structure, peak rates are in effect until 10 PM, well after the cooling load has been turned off at 5 or 6 PM. Therefore, although we can save on-peak <u>energy</u> by running in the chiller mode, the demand charges are not lowered, since we must switch back to ice making at that time. (We did not consider the option of shutting the ice-maker off from 6 to 10 PM, since then we would have needed a larger capacity, minimizing potential savings.)

Table 4-1 describes an actual chiller available (at the time of writing) from a vendor (Mueller92); it comes in two versions, one with an integral, evaporatively cooled condenser, the other with water cooling supplied from a separate cooling tower. The evaporatively cooled condenser provides substantially better compressor performance, since the condenser is held closer to the dew point, but since it requires more fan power,

Table 4-1: Technical Characteristics of Plate Ice Maker.

	Condenser Cooling	
	Evap.	Water
Capacity (tons):	121.6	113.7
Compressor Power (kW):	123	132
Auxiliary Power (kW):	29	11
Tank size (ft^3):	19,500	19,500

the overall electric consumption rates of the two systems are comparable. Both systems are described here to allow comparisons with other devices. The operating parameters describe design point operation, at full capacity and at 78°F wet bulb temperature. Although substantially better performance will occur on average, accurate modelling is time-consuming, so electricity consumption is modelled on the basis of design point characteristics, as with all the systems evaluated in this report.

Table 4-2 presents the economic characteristics of the plate-type ice making system. Because these numbers will be compared directly with the other systems, all quantities have been scaled, where appropriate, to correspond to the 116 ton capacity demanded by the partial storage operating mode. Installation cost is included, based on one man-day to connect the building system to this skid-mounted, turnkey system. The manufacturer offers a site-assembled steel storage tank, but in this size range recommends either a pre-cast (insulated) or poured concrete tank, which we therefore use, in common with the other systems being evaluated. The cooling tower cost is taken from engineering estimates for large towers (Schneider86).

Table 4-2: Economic Characteristics of Plate Ice Maker.

	Condenser Cooling	
	Evap.	Water
Capacity (tons):	116	116
CAPITAL COSTS		
Ice Maker($1000):	162	167
Installation:	(Inc.)	(Inc.)
Tank ($1000):	135	135
Tower ($1000):	$0	14
OPERATING COSTS		
Electricity($/mnth):	10,000	10,000
Maintenance($/mnth):	1000	1500
PRESENT VALUE		
(Over 5 yrs, $1000):	495	524

The monthly electricity bill is calculated as described in Appendix A, with additional accounting for the savings described above from chiller operation for the eight hours per day that demand allows it. The maintenance estimate is the estimate of a contract from a local engineering firm recommended by the manufacturer; note that the cooling tower requires substantially more maintenance, largely because it requires more water. Finally, the "present value" represents the annual costs of electricity and maintenance over the fifteen week heating season, added together over five years after discounting future expenses at six percent per year. These over all costs are compared with other systems in the Summary at the beginning of this report.

5. Tubular Ice Making System

Ice making systems based on vertical tubular metal heat exchangers are also currently widely marketed. Our description is based on one manufacturer's specifications (Morris92), although this should in no way be taken as an endorsement or evaluation of this particular product. We assert only that these units are exemplary of what is currently available, and use them to estimate operating efficiencies, capital outlays and operating costs.

These systems are similar in many respects to the metal plate ice makers described in Chapter 4, although the shape of the evaporator is quite different. Here the ice is made on a tubular metal shell which is suspended vertically over the storage tank; one shell is shown in outline in Fig. 5-1. During freezing, refrigerant floods the interior of the shell and ice forms on both the outer and inner surfaces of the shell. As with the plate ice maker, harvesting is carried out by momentarily valving hot condenser gas into the shell, melting a thin layer of ice nearest the metal surface so the ice is free to fall. The shell is tapered, both internally and externally, so that as the ice begins to fall it separates further from the surface and is not impeded by the viscosity of the layer of water.

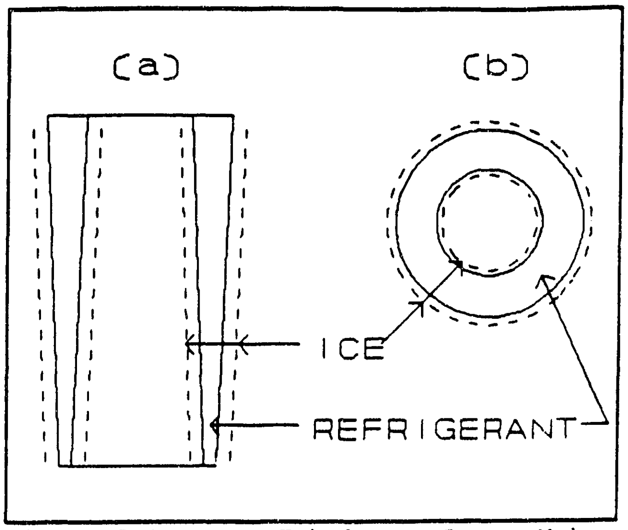

Figure 5-1: Tubular Ice Maker Evaporator Element; (a) Vertical Section; (b) Horizontal Section.

Standard operation is quite similar to the plate chiller, with the various options for full, intermediate or partial storage available, as well as the option of running the machine as a chiller with a 40°F evaporator temperature substantially increasing both COP and capacity. As with the plate-type chiller, this option is only useful while there is a cooling load, so while it allows some energy savings, it has no effect on demand charges under our rate schedule.

A significant difference from the system design of plate type ice makers is the use of levelling screws in the tank. The tubes of ice falling into the tank may be somewhat stronger and more prone to build up in piles than the sheets from the plate- type machine, but whatever the reason, the manufacturer has found it advantageous to add coarse screws running the length of the storage tank. These screws rotate during ice making operations, pulling the tops of any piles that form off to less full areas of the tank and leveling and packing the mass of ice. This allows a more compact design for the ice maker itself, since it need not cover the entire tank, as it must if the ice is simply left to fall. More importantly, the addition of the ice leveling screws allows the ice to be packed more tightly, so their specifications call for 3.0 ft^3/ton-hr (ice comprising 48% of the volume) rather than Mueller's 3.3 ft^3/ton hour (ice comprising 44% of the volume). Although the screws cost more to install and operate, this allows the tank to be downsized by 23%, a substantial saving.

Table 5-1 describes actual chillers available (at the time of writing) from a vendor; two models are described, with larger and smaller compressor motors. Both have water cooling supplied from a separate cooling tower. (Models with evaporatively cooled and air cooled condensers are also available, but are omitted for brevity.) The operating parameters describe design point operation, at full capacity and at 78°F wet bulb temperature. Although substantially better performance will occur on average, accurate modelling is time-consuming, so electricity consumption is modelled on the basis of design point operation (as with all the systems evaluated in this report). Most auxiliary power is consumed in the cooling tower (Winters91); in the ice making mode, about 2 kW is also needed to recirculate water over the evaporator tubes.

Table 5-1: Technical Characteristics of Tubular Evaporator Ice Maker.

ICE MAKING OPERATION		
Capacity (tons):	110	130
Compressor Power(kW):	109	129
Auxiliary Power (kW):	11	11
Tank Size (ft^3):	16,300	16,300
CHILLER OPERATION		
Capacity (tons):	143	169
Compressor Power(kW):	110	132
Auxiliary Power (kW):	9	9

Table 5-2 presents the economic characteristics of the tubular evaporator ice making system. Because these numbers will be compared directly with the other systems, all quantities have been scaled, where appropriate, to correspond to the 116 ton capacity demanded by the partial storage operating mode. Installation cost consists of one man-day to connect the building system to this skid-mounted, turnkey system. For comparison with other systems, we examine only a poured concrete tank. The monthly electricity bill is calculated as described in Appendix A, assuming design point operation at all times. Demand charges are lowered during on-peak operation by following the recommended schedule: during on-peak operation, the evaporator gas temperature is allowed to rise to 40°F, so that the tubes are acting as chillers rather than ice makers. The maintenance estimate was recommended by the manufacturer. Finally, the "present value" represents the annual costs of electricity over the fifteen week heating season, added together over five years after discounting future expenses at six percent per year. These over all costs are compared with other systems in the Summary at the beginning of this report.

Table 5-2: Economic Characteristics of Tubular Evaporator Ice Maker.

Capacity (tons):	116
CAPITAL COSTS	
Ice Maker ($1000):	245-255
Installation ($1000):	(Inc.)
Tank ($1000):	113
Tower ($1000):	$14
OPERATING COSTS	
Electricity ($/mnth):	8670
Maintenance ($/mnth):	1000
PRESENT VALUE	
(Over 5 yrs, $1000):	550-560

Appendix A: Comparison of Full, Intermediate and Partial Storage Options.

As discussed in Section 1, there are then three possible ways of designing and operating a cold storage system under a three-tired rate structure like that shown in Table 1-2. In the first, called "full storage", components are sized and operations arranged so that ice making occurs only during off peak hours when electric power is least expensive. Since there are only a limited number of hours per week when the system can be run, the ice making capacity must relatively large. In the second operating mode, called "intermediate storage", the ice maker runs during off-peak and intermediate rate periods, and is turned off during the on-peak period. Since it runs more hours per day, the ice maker itself can be smaller and still meet the same average load. The third operating mode, called "partial storage", allows the ice maker to operate at all times, but uses stored ice to follow the peak load in order to minimize demand and energy charges during the day. Although more expensive to operate, this mode allows an even smaller ice making capacity, saving on capital costs. The component sizes and capacities which result for each of these options are derived and displayed in Tables A-2 through A-3.

Table A-1: Component Capacities and Economics - Full Storage.

Tank Capacity:	
Ice =	6675 ton-hr
Volume=	19,300 ft^3,50%ice
Ice Maker Capacity:	
Production=	116 tons(ice)/day
Operation=	7 hrs/day
Capacity=	396 tons
Economics:	
Tank Cost=	$122,500
Ice Maker=	$423,000
Electric=	$6300/month
Present Value=	$637,000 (5 yrs, 6%/yr)

Table A-2: Component Capacities and Economics - Intermediate Storage.

Tank Capacity:	
Ice =	5980 ton-hr
Volume=	17,300 ft^3,50%ice
Ice Maker Capacity:	
Production=	116 tons(ice)/day
Operation=	12 hrs/day
Capacity=	230 tons
Economics:	
Tank Cost=	$110,000
Ice Maker=	$247,000
Electric=	$8200/month
Present Value=	$477,000 (5 yrs, 6%/yr)

Each of these systems implies a different set of capital and operating costs, and these are also indicated in Tables A-1 through A-3. The capital costs for the tank ($6.33/cubic foot) and for the ice maker itself ($1070/ton) are taken from a real, well documented 300 ton installation (Landry91). The cost estimate for the tank is consistant with $3.75 - $8.75, derived from an ASHRAE nomogram (ASHRAE87). The electricity outlays are based on the rates reported in Table 1-1. They are broken out into monthly energy and demand charges, and summed over two to ten years of operation using a six per cent discount rate. (Of course the expenses only accrue during the fifteen weeks per year of operation.) The capital costs and discounted future costs are then added to give the total present value of each system. Since many charges which were assumed to be roughly equal among the systems, such as maintenance, circulation pump expenses and so on were omitted, this is not really a "total" present value, but it will represent the cost variations between the three systems adequately for optimization purposes. (The only significant difficulty with this method is that it exaggerates the importance of optimization; since true total system costs will be about twice those reported here, an apparent difference of ten percent between systems will in reality turn out to be only a five per cent difference.)

The results shown in Tables A-1 through A-3 are based on a five year "planning horizon"; however, different people and institutions will be interested in discounting future expenses over different time intervals. We therefore calculated the present value of each of the three systems over six different planning horizons. The results, in Table A-4, indicate that over the full range of possible planning horizons, intermediate storage is more expensive than partial storage by a margin ranging from about 10% to 50%, depending on the period over which costs are discounted.

Table A-3: Component Capacities and Economics - Partial Storage.

Tank Capacity:	
Ice =	5520 ton-hr
Volume=	16,000 ft^3,50%ice
Ice Maker Capacity:	
Production=	116 tons(ice)/day
Operation=	24 hrs/day
Capacity=	116 tons
Economics:	
Tank Cost=	$101,000
Ice Maker=	$123,000
Electric=	$11,400/month
Present Value=	$392,000 (5 yrs, 6%/yr)

This study was carried out assuming a 15-week cooling season. In many circumstances, either because of a warmer climate or because of greater internal loads in the buildings being cooled, the heating season may be substantially longer, up to an entire year. Table A-5 shows the results of applying the same analysis to a longer cooling season, and we see that with a 52 week cooling season, intermediate storage is preferable to partial storage if one's planning horizon is four years or more. Similar calculations show that intermediate storage is preferable if 26 weeks of cooling is required, but only if the planning horizon is ten years or more. Similar variations would follow even for a short cooling season if cheaper off-peak power were available, but since off-peak power is becoming more and more expensive, this possibility was not explored in detail.

Table A-4: Net Present Value v. Planning Horizon, 15 weeks/year Operation.

Planning Horizon (years)	--- Storage Mode ---		
	Full	Intmdt	Partial
	(PV in $1000's)		
2	544	372	263
3	563	396	297
4	580	419	328
5	596	440	358
7	626	480	412
10	665	531	483

Table A-5: Present Value v. Planning Horizon at 52 weeks/year Operation.

Planning Horizon (years)	--- Storage Mode ---		
	Full	Intmdt	Partial
	(PV in $1000's)		
2	643	502	443
3	707	585	559
4	766	664	668
5	823	738	770
7	926	875	959
10	1061	1051	1204

Appendix B: System Comparison Spreadsheets

Table B-1: System Operating Conditions

Design Day:						
Peak Load:	500	tons	Daily			
Load Factor:	0.75		Total=	3750	ton-hrs	
Length:	10	hours				
Seasonal Quantities:						
Length:	15	weeks				
	6	days/week				
Daily Peak:	370	tons	Daily			
Load Factor:	0.75		Total=	2775	ton-hrs	
Max. Succesive			Heat Wave			
Design Days:	4		Capacity	3900	ton-hrs	
			Total=	6675	ton-hrs	
Economic Quantities:			Electric Rates:			
Discount Rate:	6.00%			hrs/day	Demand	Energy
Plnng Hrzn:	5	yrs	Off-pk	7	0	0.07
			Intrmdt	5	4.83	0.0861
			On-Peak	12	25.08	0.0984

Table B-2: Polymer Ice Making System

Tank Capacity:							
Ice=	5519	ton-hrs=	7998	ft3 of ice			
Volume=	142656	gallons =	19123	ft3 in tank @		49%	ice max
						3.0	ft3/tn-hr
Ice Maker Capacity:							
Production=	116	tons/day					
Capacity =	9635	lbs/hr,	24	hrs/day			
=	116	tons					
Comp. Power:	0.99	kW/ton =	114	kW total			
Aux Power:	0.086	" =	10	kW			
Total=	1.076	"					
Economics:							
Tank @	$6.90	/ft3, Icemaker @	$1,828	/ton		$197,000	Machine
Tank Cost= $	131948						
Ice Maker=	225739					$14,369	CoolTow
5 -yr electric=	145252	@ mnthly	$3,122	dmnd	$6,800	energy	
+ Maintenance	50548		$1,000	/month			
Discounted @	6%	/year					
TotalPV($1000)=	553						

Table B-3: Advanced Polymer Ice Making System

Tank Capacity:							
Ice=	5519	ton-hrs=	7998	ft3 of ice			
Volume=	142656	gallons =	19123	ft3 in tank @		49%	ice max
						3.0	ft3/tn-hr
Ice Maker Capacity:							
Production=	116	tons/day					
Capacity =	9635	lbs/hr,	24	hrs/day			
=	116	tons					
Comp. Power:	0.913	kW/ton or	0.68	kW/ton	as chllr	8	hr/day
Aux Power:	0.086	"	0.09	"			
Total=	0.999	"	0.77	"			
Economics:				300	pads @	$140	/pad =>
Tank @	$6.90	/ft3, Icemaker @		$1,439	/ton	$42,000	Pads
Tank Cost= $	131948					$110,000	Chiller
Ice Maker=	180739					$14,369	CoolTow
5 -yr electric=	118701	@ mnthly	$2,898	dmnd	$5,205	energy	
+ Maintenance	50548		$1,000	/month			
Discounted @	6%	/year					
TotalPV($1000)=	482						

Table B-4: Direct Contact System

Tank Capacity:							
Ice=		5519	ton-hrs=	7998	ft3 of ice		
Volume=		119333	gallons =	15996	ft3 in tank @		50% ice max
Ice Maker Capacity:							
Production=		116	tons/day				
Capacity =		9635	lbs/hr,	24	hrs/day		
=		116	tons				
Comp. Power:		0.99	kW/ton =	114	kW Total		
Aux Power:		0.078	" =	9	kW Total		
Total=		1.0678	"				
Economics:							
Tank @		$6.90	/ft3, Icemaker @	$1,250	/ton		
Tank Cost=	$	110375					
Ice Maker=		144531					
5 -yr electric=		144088	@ mnthly	$3,097	dmnd	$6,746	energy
+ Maintenance		50548		$1,000	/month		
Discounted @		6%	/year				
TotalPV($1000)=		450					

Table B-5: Piate Ice Making System - Evaporative Condenser

Tank Capacity:						
Ice=	5519	ton-hrs=	7998	ft3 of ice		
Volume=	145528	gallons =	19508	ft3 in tank @	41%	ice max
					3.5	ft3/ton-hr
Ice Maker Capacity:						
Production=	116	tons/day				
Capacity =	9635	lbs/hr,	24	hrs/day		
=	116	tons				
Comp. Power:	1.01	kW/ton or	0.68	kW/ton as chllr	8	hr/day
Aux Power:	0.24	"	0.24	"		
Total=	1.25	"	0.9167	"		
Economics:						
Tank @	$6.90	/ft3, Icemaker @	$1,398	/ton	$170,000	
Tank Cost=	$ 134604				121.6	tons
Ice Maker=	161647					
5 -yr electric=	148325	@ mnthly	$3,625	dmnd	$6,508	energy
+ Maintenance	50548		$1,000	/month		
Discounted @	6%	/year				
TotalPV($1000)=	495					

Table B-6: Plate Ice Making System - Water Cooled

Tank Capacity:								
Ice=		5519	ton-hrs=	7998	ft3 of ice			
Volume=		145528	gallons =	19508	ft3 in tank @		41%	ice max
							3.5	ft3/ton-hr
Ice Maker Capacity:								
Production=		116	tons/day					
Capacity =		9635	lbs/hr,	24	hrs/day			
=		116	tons					
Comp. Power:		1.16	kW/ton or	0.78	kW/ton as chllr		8	hr/day
Aux Power:		0.079	"	0.079	"			
Total=		1.239	"	0.8562	"			
Economics:								
Tank @		$6.90	/ft3, Icemaker @	$1,445	/ton		$150,000	
Tank Cost=	$	134604					113.7	tons
Ice Maker=		167045					$14,264	CoolTow
5 -yr electric=		146949	@ mnthly	$3,593	dmnd	$6,445	energy	
+ Maintenance		75823		$1,500	/month			
Discounted @		6%	/year					
TotalPV($1000)=		524						

Table B-7: Tubular Ice Making System

Tank Capacity:								
Ice=		5519	ton-hrs=	7998	ft3 of ice			
Volume=		121768	gallons =	16323	ft3 in tank @		49%	ice max
							3.0	ft3/tn-hr
Ice Maker Capacity:								
Production=		116	tons/day					
Capacity =		9635	lbs/hr,	24	hrs/day			
=		116	tons					
Comp. Power:		0.99	kW/ton or	0.78	kW/ton	as chllr,	8	hr/day
Aux Power:		0.095	"	0.095	"			
Total=		1.085	"	0.875	"			
Economics:								
Tank @		$6.90	/ft3, Icemaker @	$2,050	/ton		$237,031	Machine
Tank Cost=	$	112628					12500	Headers
Ice Maker=		263901					$14,369	CoolTow
5 -yr electric=		128918	@ mnthly	$3,146	dmnd	$5,656	energy	
+ Maintenance		50548		$1,000	/month			
Discounted @		6%	/year					
TotalPV($1000)=		556						

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