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**ENGINEERING ASSESSMENT OF THE USE OF COMPRESSION
REFRIGERANTS IN THE THIN POLYMER ICEMAKER**

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ABSTRACT

In previous studies we have developed the concept of an ice-making device constructed of thin polymer films, arranged so that when filled with a cold fluid they expand in a set of parallel tubes, on the exterior of which ice forms. When the ice is about one millimeter thick, the cold fluid is removed from the tubes and they collapse, being pulled away from the ice by a vacuum within. The device has been successfully demonstrated in a version where the tubes are filled with a water-ethylene glycol brine. This was followed by an economic assessment which indicated that the device would be even more competitive with existing systems if the compression refrigerant itself were admitted into the polymer film structure on which the ice forms. This report presents an engineering study of such a system, examines the outstanding questions which must still be answered and assesses the potential economics of the system.

In this report we develop preliminary designs for systems which satisfy code requirements using R-134a and R-C318 as refrigerants. We use standard thermodynamic and heat transfer analysis to determine expected performance for the systems. The R-C318 has a somewhat peculiar vapor/liquid equilibrium curve, and we introduce a novel heat exchanger to optimise performance for this refrigerant. We combine these performance assessments with component and assembly cost estimates to prepare life-cycle costs for the two new systems. Comparing them to commercially available systems, we find that these "polymer evaporator" systems seem to be completely feasible technically and that if they are successfully demonstrated, they should have substantial economic advantages over existing ice-making devices.

1. INTRODUCTION AND SUMMARY

A new device for the production of ice has been designed, constructed and tested¹ in a project that is part of the District Heating and Cooling program at Brookhaven National Laboratory. The central feature of this ice maker is a set of tubes made of a thin polymer film. Cold fluid within the tubes causes ice to form on the outside of the tubes, and when the coolant is removed the resulting vacuum causes the tube to collapse and pull away from the ice, which is then free. This system obviates the reheat cycles necessary in existing ice makers, and so offers the potential for higher efficiency.

A subsequent part of this project, an economic assessment of this device² showed that it has the potential to gain a significant share of the market for energy-related ice making systems. Such systems include slush ice production for district cooling and ice production for cold storage systems in buildings.

However, that study also showed that the icemaker's efficiency could be significantly improved. In the tested version, a chiller cools a brine to subfreezing temperatures in a separate evaporator, and the brine is subsequently pumped through the ice making pad. This extra stage of heat exchange leads to higher temperature differences and more work for the compressor, resulting in a higher electric bill and greater capital costs than would otherwise be necessary. If, alternatively, the refrigerant could be allowed to evaporate directly inside the thin polymer ice maker itself, the evaporator temperature would be raised by several degrees, resulting in a substantial increase in system COP and capacity.

In this report we examine the feasibility of this scheme. The choice of refrigerants is increasingly constrained by concerns about the ozone layer; the chemical compatibility of the refrigerants and the polymers must be assured, and most important, the design must ensure that the relatively fragile polymer icemaker is not burst by the pressurized refrigerant. All these questions are addressed in the sections that follow, where a conceptual design is advanced, two candidate refrigerants (R-134a and R-C318) are selected, the systems are analyzed technically and then, using the exact formulation of our earlier study², assessed economically.

The technical analysis includes a thermodynamic cycle analysis based on accurate properties data from the literature and accurate treatment of heat transfer in the evaporator and condenser. We examined a simple single compressor system, without looking at benefits that might accrue from multi-stage compression or other sophisticated improvements. For R-C318, we found that addition of one unorthodox but straightforward heat exchanger improved performance significantly, and included that in our analysis.

The economic analysis is based on a five year life cycle, in which the total cost of meeting a specified load typical of a medium-sized commercial building in the northern part of the United States. During a 15-week cooling season, the system meets a daily peak load of 370 tons; the design day is 500 tons, and storage is sufficient to meet four design days in a row. Electric charges accrue for both demand and energy, and they and estimated maintenance

expenses are discounted back to give a net present value for the five-year cost of system operation. We believe both the design and the economic assessment to be quite conservative.

In Table 1-1, the resulting data on the two "Advanced Polymer" systems are presented in the top rows, while the data for the competitive systems have been imported directly from our economic assessment². "Polymer/Brine" is our design study based directly on our experimental apparatus. The "Flat Plate" and "Tubular Evaporator" represent commercially available systems, and accurately indicate the current costs of ice storage systems (in 1992 dollars). We have omitted one experimental system discussed in our earlier report, a "direct contact" concept in which the refrigerant actually mixes with the system water. This system constantly loses refrigerant to the system water, so it is unlikely to achieve technical viability because of its atmospheric emissions. (Even chlorine-free refrigerants have substantial potential to enhance the greenhouse effect.)

The table makes it clear that the advanced polymer ice maker should be very competitive against existing systems, and that a laboratory test of this concept would be a very worthwhile investment.

Table 1-1: Comparison of System Economics.

System Type	Status	Capital Outlay (\$1000)	Electric Bill (\$1000/mnth)	Net Present Value (\$1000)
Polymer/R-C318	Design	304	7.0	457
Polymer/R-134a	Design	332	6.6	480
Polymer/Brine	Demo	340	9.9	539
Flat Plate	Com.Av.	316	10.0	524
Tubular Evaporator	Com.Av.	372-382	8.7	550-560

2. PHYSICAL/CHEMICAL REFRIGERANT SCREENING

Current refrigeration and ice-making equipment all uses one of three or four common refrigerants, but several concerns, some general, others unique to the thin polymer ice-maker, lead us to examine a much larger set of candidates initially. Then, environmental and other constraints will restrict the choice of refrigerants to a subset of these several dozen initial candidates. In particular, consideration of current threats to the ozone layer will lead us to eliminate all candidates containing chlorine and bromine. Also, operating pressures must be compatible with the tensile strength of the polymer comprising the evaporator and the chemical compatibility of the refrigerants and the polymers must be assured, as well as the impermeability of the polymer to that refrigerant. Finally, levels of toxicity and flammability must be taken into account. In this section we work through these steps in turn.

Several sources of general information on compression refrigerants provide useful data. We start with a reference³ listing 86 standard refrigerants, but do not reproduce the entire table here in the interest of brevity.

In the foreseeable future, the choice of refrigerants will be increasingly constrained by concerns about the ozone layer. CFC production is completely banned by Jan 1, 1996, according to the Montreal Protocol. In addition, 90% of HCFC use is now slated to be phased out over the 1996-2015 period.⁴ Anticipating that these restrictions can only get stricter, we consider only those refrigerants known to be harmless in this regard; this essentially means that they can have no chlorine or bromine in them. The result of applying this filter to the 86 refrigerants mentioned above is Table 2-1, which gives the refrigerant number and chemical formula of the 35 distinct candidates having no chlorine or bromine in them. (Table 2-1 is an abbreviation of Appendix A.)

Second, we are interested only in refrigerants that have reasonable working pressures at the evaporator temperature of 25°F (-4°C). The working pressure at 25°F must be greater than one atmosphere (so the flexible pad can expand) and yet of relatively modest pressure (so that it will not burst). The vapor pressures at 25°F (derived from boiling and critical parameters⁵ in Appendix A) of all sub-critical refrigerants are shown in Table 2-1. Also, any material with a critical temperature below 25°F is unsuitable, and for these substances the critical temperature is shown in Table 2-1, as " $T_c = -150^\circ\text{F}$ ", rather than the pressure at 25°F, which is indeterminate. The candidates with working pressures between one and five atmospheres are presented in Table 2-2, along with various compatibility criteria.

Pressures will be further constrained by the ASME code for pressure vessels,⁶ which requires that refrigerant vessels be capable of withstanding seven times their operating pressure. Exactly how this will be applied will depend on detailed design considerations, which will be developed in later sections.

Third, the chemical compatibility of the refrigerants and the polymers and metals must be assured. Detailed data on the chemical compatibility has been difficult to find. Limiting

Table 2-1: Refrigerants having no chlorine or bromine.

Halocarbons:		P @ 25°F	Hydrocarbons:		P @ 25°F
14	CF ₄	T _c =-50	50	CH ₄	T _c =-117
23	CHF ₃	22.7	170	CH ₃ CH ₃	22.1
32	CH ₂ F ₂	7.1	290	CH ₃ CH ₂ CH ₃	4.2
41	CH ₃ F	18.5	600	CH ₃ CH ₂ CH ₂ CH ₃	0.9
116	CF ₃ CF ₃	16.4	600a	CH(CH ₃) ₃	1.3
125	CHF ₂ CF ₃	5.5	610	C ₂ H ₅ OC ₂ H ₅	0.2
134a	CF ₃ CH ₂ F	2.5	611	HCOOCH ₃	0.2
143a	CH ₃ CF ₃	5.4	630	CH ₃ NH ₂	1.1
152a	CH ₃ CHF ₂	2.3	631	C ₂ H ₅ NH ₂	0.4
Cyclic Organic Compounds:			Inorganic Compounds:		
C318	C ₄ F ₈	1.1	702	H ₂	T _c =-400
Unsaturated Organic Compounds:			704	He	T _c =-450
1114	CF ₂ =CF ₂	15.7	717	NH ₃	3.6
1132a	CH ₂ =CF ₂	20.7	718	H ₂ O	T _f =32
1141	CH ₂ =CHF	13.4	720	Ne	T _c =-380
1150	CH ₂ =CH ₂	38.0	728	N ₂	T _c =-232
1270	CH ₃ CH=CH ₂	5.2	732	O ₂	T _c =-181
			740	A	T _c =-188
			744	CO ₂	27.3
Temperature in °F			744a	N ₂ O	28.4
Absolute Pressure in Atmospheres			764	SO ₂	1.3

ourselves to the PFA teflon from which the demonstration ice makers have been constructed, we will rely on the widely-known resistance of this material to acids, alkalies, oxidizers, oils and fuels and hydrocarbons⁷ and organic solvents⁸ to assume that chemical compatibility will not be a problem. Immersion creep testing⁹ would be an early step in an experimental program.

Table 2-2: Chemical and Hazard Characteristics of Candidate Refrigerants.

Refrigerant #	Incompatible with:	Explosive Limits (vol. %)	Q _c /Q _t (1)	UL	ANSI/ASHRAE	OSHA PEL(2) ppm	
134a	Mg, Zn	none	7			A	
152a	Mg, Zn	3.7-20	37			A	
C318	Mg, Zn	none	2			A	
1270		2-11	120		1	A	
290		2.1-9.6	122	5b	1	1000	
600a		1.7-9.7	110	5b	1	-	
630		4.9-20.7	38		3	10	
717	Cu, Brass,..	15-33	15	2	3	50	
764	Water (->H ₂ SO ₄)	Non-flammable	0	1	3	5	
Toxicity scales:				Maximum hazard:	1	3	->0
				Miimumum Hazard:	6	1	10 ⁶
1) Q _c /Q _t = Heat of combustion/Latent heat of vaporization.							
2) "PEL" = "Permanent Exposure Limit" set by OSHA; "A" = Simple Asphyxiant							

Several incompatibilities exist between refrigerants and metals¹⁰, and these are listed in Table 2-2. The first three refrigerants are halogenated and as such, should not be used with magnesium or zinc-bearing alloys here, where water could be present. A more serious problem is the tendency of ammonia (R717) to attack copper and copper-bearing alloys. This is not a problem within the compression chiller system itself, where copper can be avoided, but when an ice-making pad fails, refrigerant could be released into the building system water. If ammonia is used, it might be necessary to flush the system often to keep concentrations acceptably low. Even more serious is the conversion of sulfur dioxide (R764) into sulfuric acid on contact with water, which would occur whenever a pad failed. We have not identified any compatibility problems for the hydrocarbons (R1270, R290 and R600a) or for methyl amine (R630).

Fifth, the impermeability of the polymer to the refrigerant must be ensured. Here we have no data for any candidate refrigerants or for PFA teflon. We do know that related FEP Teflon will transmit 2.4 gram-moles (1.5 ounces) of water per square foot per year through a typical 5-mil film, and 5-10 gram-moles of gasses such as O₂, N₂ or CO₂⁸ at a differential

pressure of one atmosphere⁸. It would not be surprising if similar rates occurred for ammonia and sulfur dioxide, and lower rates for the larger molecules of the other candidates, but the only way to be sure is testing.

Sixth, safety issues must be considered; various attributes are shown in Table 2-2 for the candidate refrigerants. Arguing that these ice-makers will be used in large industrial applications where special measures can be taken, and that the whole refrigeration industry may be making similar difficult choices in the near future, we will not use either toxicity or flammability to eliminate candidates automatically. Table 2-2 gives a comparison of the hazards to life and the explosive limits for the same nine candidates.

The explosive limits^{5,11} refer to volumetric percentages; for a given leak, the wider the range, and the lower the lower limit, the more easily an explosion can be triggered. In the next column we compare the possible damage from such an accident across refrigerants by dividing the heat of combustion, Q_c ,⁵ (representing the energy released in a fire or explosion) by the latent heat of vaporization, Q_l , (representing the quantity of refrigerant needed for a given refrigeration capacity).

Three measures of toxicity are presented, with the scales explained on the bottom line. The ANSI/ASHRAE Standard 15-78 is an arbitrary measure based on "hazards involved in use".¹² The Underwriters Laboratories classification¹³ is based on toxicity experiments with guinea pigs; "Group 1" materials produce lethal or serious injury after five minutes exposure at 0.5-1.0% concentrations, while "Group 6" materials appear to be benign, other than the possibility of suffocation. Finally, the U.S. Occupational Safety and Health Authority (OSHA) Permanent Exposure Limit (PEL) is given¹⁴ in parts per million, where such a limit has been defined. (These numbers refer to "time weighted averages" or "TWA's.") An "A" in this column means the substance is classified as a simple asphyxiant.

Screening based on these data, we discard R764 (sulfur dioxide) because of its tendency to form sulfuric acid and its high toxicity. R1270 (propylene), R290 (propane) and R600a (Isobutane) are dropped due to their high heats of combustion. The remaining five compounds fall into two groups: R-C318 (Octofluorocyclobutene) and R630 (methyl amine) with vapor pressures just above one atmosphere at 25°F and the remainder, R134a (tetrafluoroethane), R152a (difluoroethane) and R717 (ammonia) with higher vapor pressures which would require careful engineering of the pad to meet code requirements and avoid pressure-induced failures.

We will choose one from each group for further analysis: R-C318 because of its non-explosive nature, low heat of combustion and lower toxicity, all when compared to R630, and R134a from the second group for essentially the same reasons.

3: THERMODYNAMIC PROPERTIES OF SELECTED REFRIGERANTS

The pressure and temperature relations and other thermodynamic properties of the refrigerants are basic to the design of an ice making system. In this section we discuss data on the vapor pressure, enthalpy, entropy and other relevant properties of R134a and R-C318.

For R134a, a detailed program of measurement and analytic characterization has recently been carried out by Wilson and Basu¹⁵. The authors have made a Fortran program available which allows straightforward calculation of tables of all quantities needed for cycle analysis. The analytic representations include an expression for vapor pressure as a function of temperature with six adjustable constants, a four-constant expression for liquid density, a 13-constant Martin-Hou equation of state and a five-constant expression for the ideal gas heat capacity. The latter agrees with the expression in Daubert and Danner⁵ to better than one per cent. Values for pressure and changes in enthalpy and entropy derived from Wilson and Basu's equations also agreed with tabular data from ASHRAE¹⁶ to better than one per cent in the regimes relevant to our calculations.

Figure 3-1: Temperature-Entropy Equilibrium Curve of R-C318.

Refrigerant C-318 (perfluorocyclobutane, C_4F_8) has not been used commercially to our knowledge, and less technical data is available. Basic property data from Daubert and Danner⁵

were useful, but the virial equation of state was not sufficiently precise to generate values of enthalpy and entropy. One recent paper¹⁷ was rendered useless by typographical errors and an incomplete presentation. Older work by Tawfik Morsey¹⁸ proved far more useful. His extensions of the B-W-R equation of state explicitly guarantee a set of important thermodynamic constraints at the critical point. Programmed in a spreadsheet, Morsey's equation of state agreed to within one percent with that of Dauberg and Danner and with the extensive empirical data of Martin¹⁹ (from which Morsey had derived his constants). Temperature-entropy and temperature-enthalpy diagrams derived from Morsey's equations agreed to within graphical accuracy with figures in Cipollone's¹⁷ paper.

This last agreement is welcome confirmation of the somewhat peculiar behavior of C-318 in the temperature-entropy plane, shown in Figure 3-1: the saturated vapor side of the curve slopes sharply out to the right. Since the compression part of the refrigeration cycle is (approximately) isentropic, this means that compression of saturated vapor will result in the formation of small amounts of liquid, probably in the form of a mist. In the cycle analysis to follow we will deal with this unusual behavior in two different ways.

4. PRELIMINARY SYSTEM DESIGN

In this section we develop preliminary system designs which will serve as models in which to assess system performance and for which we develop cost estimates. Many details which would be included in an actual design are combined or included only in summary, but we have attempted to include enough detail to permit meaningful comparisons with the existing systems examined earlier and summarized in Chapter 1. Because there are significant differences between systems designed to operate on R-134a and on R-C318, we present them in separate sections.

4.1 Preliminary Design Parameters

The system design will follow the one developed for the New York climate during our economic assessment of the original (brine-filled) polymer ice maker². The peak design-day load was taken as 500 tons, with a design-day load factor of 0.75 for the ten hours during which the system supplies cooling to a commercial building. The optimal system incorporated a 16,300 ft³ ice storage tank capable of storing 5520 ton-hours* of ice. With this storage, an ice maker with a steady state capacity of 116 tons (1.4 mmBtu/hr or 408 thermal kilowatts) was sufficient to operate the system through a fifteen week cooling season.

As discussed in Ref. 2, optimal operation for the system involved running the ice maker at all times, but using it in "cooling mode" when warm (55°F) water was returning from the building distribution system, which we assumed occurred eight hours per day, during the peak electric rate period. With the pads suspended above the water in the tank, returning water is sprayed onto them and is cooled to about 45 °F before dripping onto the tank. This allows the evaporator (the pads) to run at a considerably higher temperature (above 40°F), and so lowers the power requirements of the compressor and the electric bills. When no warm water is returning, the pads are cooled below freezing and cold water is sprayed onto them. When ice has built up to perhaps a millimeter thick, the pads are deflated and the ice falls off and into the bath. This combined operating mode is commonly used in current commercial (rigid evaporator) ice making systems, and results in substantial savings in operating costs.

An apparent design obstacle with the thin polymer pad design is the limited strength of polymers, including the PFA used in our experiments. The internal pressure indicated in Table 2-1 for R-134a, 36 psia at 25°F, is substantially higher than pressures exerted on the polymer tubes by the brine during previous testing of the icemaker. It is necessary to relate the differential pressure between the inside and the outside of the tubes making up the pad to the

*. 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, and a "ton-hour" is 12,000 Btu or 2000/24=83.3 lb of ice. Also, 1.0 mBtu = 1.0x10³Btu and 1.0 mmBtu = 1.0x10⁶Btu

internal stresses in the polymer, and to compare these to the various characterizations of the strength of the material.

First, we consider the material characteristics. PFA has a listed "tensile strength" of 4000 psi⁸. However, this is not a good basis for design since it does not incorporate possible weakening at the welds and does not account for long term "creep" of the polymer (an ongoing stretching of the material) in response to stress, which is the primary strength consideration for any long life polymer application.

Previous tests performed in an investigation of possible absorption chiller applications⁹ showed that welded PFA tubes could withstand internal stress of up to 2180 psi (in the polymer) at 70°F and underwent creep of 1.5% in the first year under an internal stress of 810 psi at 220°F. Creep slows logarithmically with aging, so the next 1.5% should take approximately ten years. Since creep also slows dramatically with decreasing temperature, a design point internal stress of 1000 psi will be quite conservative at 32°F. Creep tests should be carried out at these temperatures before designs are made final.

Figure 4-1 shows the relationship between internal stress in the polymer, S , and the differential pressure between the inside and the outside of the tube, P . The diameter of the tube is d , the thickness of the material is t and we are looking at a cross-section of a length ℓ of tube. The top half of the tube is in equilibrium between the upward force $F = Pd\ell$ due to the differential pressure and the downward force $F' = 2(S t \ell)$ due to the stress where the lower half of the tube is pulling on the upper half. Since $F = F'$, $Pd\ell = 2St\ell$ and to withstand creep, the tubes should be subjected only to a differential pressure of $P < 2tS/d$. Taking "t" as 0.003 inches (as in our experiments) and "S" as 1000 psi, we find that the differential pressure should be constrained to $P < 6.00/d$ psi for d in inches.

Figure 4-1: Stress-pressure relations.

4.2 Preliminary Design for R-134a

R-134a would appear to be a more desirable choice for refrigerant than R-C318, since it is already in wide use in automotive air conditioning and is expected to gain more and more market share as chlorine-based compounds are phased out. However, for a polymer-based design, it presents significant design challenges. We will see in Chapter 5 that a reasonable maximum working temperature for the R-134 in the tubes will be 40°F (when the system is

operating in cooling mode), and the data described in Section 3 indicate that the corresponding pressure in the evaporating refrigerant is 50 psia. This pressure must be contained in accord with two distinct criteria: the design criteria that "creep" should be minimal and the ASME code requirement that these tubes be treated as a pressure vessel and be capable of withstanding seven times the design differential pressure. We deal with these two criteria in turn.

For design purposes, we will allow a steady differential pressure of 14.7 psi, for which the creep limitations derived just above imply a tube diameter of 0.404 inches, smaller than the 0.45 inches used in the experimental ice maker tested in earlier experiments¹. To ensure differential pressures less than 14.7 psi, the volume in which the pads reside, including the ice water and storage tank, must be pressurized to at least $50 - 14.7 = 35$ psia, or 21 psig.

The ASME code requirements can be met on the basis of tensile strength rather than creep criteria. We take the tensile strength of PFA teflon from earlier tests²⁰ as 2180 psi. Using the formula from the previous paragraph divided by seven, we find the maximum differential pressure across the tube consistent with the ASME requirement to be 4.6 psi. Consequently, to keep the differential pressures within code limits, the ice water in the storage tank must be pressurized to 45 psia, or 31 psig (a more stringent requirement than that derived in the previous paragraph).

This design, with the entire ice tank pressurized to about two atmospheres, will not cost anything energetically, since pumping is needed to push the cold water or slush ice around the building distribution system anyway; we are simply putting the pump before the tank rather than after it. Further, the tank offers protection to the local environment against pad breakage and refrigerant release. We are not sure whether the pressure vessel code would require the tank to adhere to the "7X" standard as well, but will assume below that it does. If the standard can be relaxed, either for the pads or for the tank, the costs of the system can be lowered.

The resulting assembly is shown in Figure 4-2. The tanks are capped cylindrical steel with substantial insulation. Appendix B details and further illustrates the sizing calculations based on the 16,300 ft³ storage volume²; sizing is more practical with four equal tanks, and this offers advantages in operational security and maintenance flexibility as well. The pads occupy a rectangular volume at the top of each tank, and can be extracted for maintenance, along with refrigerant piping and supports, from one end.

Appendix B also details the wall thickness calculations, based on a value for the tensile strength of ASTM A284 steel, 55,000 psi²¹, which when coupled with a calculation similar to that performed just above for the tubes in each pad indicates that 0.33 inch thick steel is adequate for the tanks, and that they will require 90,500 lbs of steel, plus supports, ribbing, flanges and so forth. It might be possible to use less material through clever ribbing, but we will use these values to be conservative.

The total ice-making area and the number of pads required will depend on the speed with which ice forms on the surfaces, and this will depend on the thermodynamic calculations to be

Figure 4-2: Thin Polymer Ice Maker and Tank: Conceptual Design.

developed in Section 5, below. As shown in Figure 4-2 and Appendix B, the pads must fit into a rectangular space two feet high, seven feet wide and extending the length of the tank. Leaving space for individual support structures, the active height of a pad will be 18 inches, consisting of 34 tubes each 0.404 inches in diameter. Appendix C details and illustrates various pad dimensions and quantities as outlined here and developed below.

For optimal heat transfer, each evaporator tube should remain flooded over its entire length. This can be assured by inserting a narrow (1/16-inch ID) metal tube equipped with regularly spaced orifices down the center of each polymer evaporator tube and injecting the fluid through it, so that the fluid sprays out radially within the evaporator tube. In Appendix D we calculate pressure drops in a system of distribution tubing between the pads and in pads composed of "Headers" at each end and "Tubes" on which the ice actually forms. The calculation is presented for the more demanding water chilling mode of operation; all pressures are lower when ice is being made. The "System Totals" show that pressure drops are acceptable if the tubes are three feet long, and we will assume this is the case in subsequent modelling. Power consumption is also calculated in Appendix D for use in Chapter 6.

We believe the design outlined here presents a conservative basis for calculating performance and estimating system costs. Some trade-offs should be kept in mind: although operating in cooling mode offers substantial operating savings, it is forcing us to design a system based on the vapor pressure of R-134a at 40°F. If the system were restricted to operating solely to make ice, the maximum R-134a temperature would be about 28°F, the pressure under 40 psia and the required steel reduced to 59,000 pounds. This option should be examined if real designs are to be developed along the lines sketched here.

4.3 Preliminary Design for R-C318

An even more optimal solution would result from the use of C-318, since its lower operating pressures would allow the use of less expensive containment. The analysis of Section 4.2 is readily adapted to this case. With respect to our long-term design goal of containing creep, assuming the same geometry and material, we still need to keep the differential pressure across the tubes under 14.8 psi. However, this is now easy, since the analysis of Chapter 5 will show that during water chilling the pressure of the evaporating R-C318 will be 145.6 kPa, equal to 21.2 psia or 6.5 psig, well below the 14.8 psig limit.

The requirements of the ASME pressure vessel code are again somewhat more exacting. For the same geometry and material, the maximum differential pressure is again 4.6 psi. Since this is less than the 6.5 psig of the evaporating R-C318, the structure would have to be pressurized, but only at 1.9 psig. Following the approach of Section 4.2 and enclosing the pads and ice water in steel pressure vessels, it could be constructed of 20-mil strip steel.

This would not be strong enough to retain the ice water, and for R-C318, our conceptual design will assume that the ice water storage is in insulated, rectangular concrete tanks, as is often done with commercial systems today, and as was assumed in our earlier study². The modest pressurization would be achieved by installing a sealed semi-cylindrical steel cover which would house the pads and distribution piping. Except for these greatly reduced housing requirements, the system is identical to that developed for R-134a, and the cost estimates to be developed in Chapter 6 will reflect this. Appendix E shows that because more refrigerant must be pumped, the pressure losses and power consumption are higher than for R-134a. Although pipe sizing should be optimized before such a system is built, we have left all sizes the same as for R-134a; the result is a conservatively over-priced, un-optimized system.

We have confidence in our system design and cost estimates, but the practicality of using R-C318 is considerably less certain than was the case with R-134a. The properties of the pure chemical are well known, but we have not found a case where it has been used commercially. Its interactions with lubricants and seals are unknown, as is its long-term stability in a continuously cycling application. Its price in quantity at commercial purity is not clear either; that of highly purified laboratory material is prohibitive. However, the promise our systems analysis reveals indicates that investigation of these concerns would be a worthwhile endeavor.

5. SYSTEM MODELLING AND EXPECTED EFFICIENCY

In this section we develop a thermodynamic model of the thin polymer ice maker suitable to estimate the overall efficiency of the system and to determine typical operating parameters and conditions. We apply this model to a system based on R-134a and to two systems based on R-C318.

5.1 Description of Model

The model includes standard characterization of heat transfer in the thin polymer evaporator and in a water-cooled condenser, which is assumed connected to a wet cooling tower, as shown in Figure 5-1. The chilling cycle itself is represented by the ASHRAE "Actual Basic Vapor Compression Cycle"²², which includes the major irreversible losses associated with the vapor compression cycle. The model is constructed so that when the evaporator temperature changes as ice-making and water-cooling operation are compared, the compressor operates at a constant displacement (rather than constant mass flow or constant evaporator capacity), as a standard, single-speed compressor would.

Figure 5-1: Ice Maker System Diagram.

The model is displayed for the case where R-134a is used in ice making in Figure 5-3, in chilling 45°F water in Figure 5-4 and in similar calculations for R-C318 in following figures. In these figures, the numbers in gray boxes are determined externally and typed in, while those in white areas are calculated by algorithms within the spreadsheet. To the right in each figure, temperature-entropy and pressure-enthalpy diagrams are sketched in. The (approximately) reversible portions are shown as solid lines, the irreversible parts as heavy dotted lines and the approximate position of the refrigerant's saturation curves is indicated by light dashed lines.

The evaporator is described in the upper left-hand box of the figures. For ice making, the key temperature is that at the ice-water interface, where freezing is actually occurring. As shown in Figure 5-2, this is the source of latent heat of fusion which must be transported through the ice and the polymer wall to the refrigerant vapor. If the water is above the freezing point, it must also be cooled down, but a simple calculation shows that for water a few degrees above freezing, this will produce heat amounting to only a few per cent of the latent heat of fusion²³, which we ignore in these calculations.

The ice building process takes place in intervals such that the maximum thickness is about one millimeter (0.04 in.), so on average the thickness is one-half millimeter which combines with the thermal conductivity of ice to give a heat transfer coefficient of $3.2 \text{ kW/m}^2\text{-}^\circ\text{C}$ (all internal calculations in the model are in SI units; results are given in both SI and conventional American units.) For 3-mil (75 micron) PFA film, the thermal conductivity⁸ gives a heat transfer coefficient of $0.99 \text{ kW/m}^2\text{-}^\circ\text{C}$.

Finally, ASHRAE gives an estimate of $2.0 \text{ kW/m}^2\text{-}^\circ\text{C}$ for R-12 (Ref. 3, p. 4.5, Fig. 8) as the refrigerant boils in the evaporator tube. R-134a should show similar heat transfer performance²⁴, and experimental data for R-134a found after our calculations were completed²⁵ indicates that this is a reasonable and perhaps conservative assumption. The flow configuration used in these recent experiments differed from ours, and without experimental data for the polymer tubes in our arrangement it is doubtful that more detailed predictions would yield greater confidence. Also, the relatively high thermal resistance of the polymer wall insures that variations of $1 \text{ kW/m}^2\text{-}^\circ\text{C}$ up or down in the boiling heat transfer coefficient of the refrigerant result in only a 10-20% variation in the overall coefficient. Consequently, we simply use this R-12 estimate for R-134a. We have found no phase-change heat transfer data for R-C318, and use the R-12 estimate in those calculations as well.

Pressure loss down the evaporator tubes is ignored in the cycle calculations (although included in Appendices D and E). The total heat exchange area of 185 m^2 is found by an iterative process (described below). This area is multiplied by the inverse of the sum of the inverses of these three series-connected elements to give the overall evaporator heat transfer coefficient of $101 \text{ kW/}^\circ\text{C}$. The evaporator load of 407 kW (calculated below) is then used to find the consistent refrigerant temperature of 28°F (-2.2°C), which is used in the cycle calculations below.

For the water chiller analysis of Figure 5-4, the evaporator is analyzed in the same way except that the heat source is 45°F (7.2°C) water, and the corresponding heat transfer coefficient for cold water flowing over the outside of horizontal tubes is $0.9 \text{ kW/m}^2\text{-}^\circ\text{C}$ ²⁶. The results are discussed below.

The condenser is assumed to have refrigerant condensing on tubes containing water from a cooling tower, taken to have a temperature of 95°F (35°C) over the entire length of the condenser, a simpler but more conservative assumption than the standard of 85°F entering and 95°F leaving water temperatures. The heat transfer coefficient from this in-tube water to the

Figure 5-2: Heat Flow in Ice Building.

Ice Maker		R134a Tev=32F			
EVAPORATOR:		e-a			
Tf:	0.0 oC =	32.0 oF =	273 oK		
Uice:	3.20	Uw:	0.99	Ur:	2.00 kW/m2-oK
A:	155 m2	UAtot=	101.5 kW/oK		
Te=	-2.23 oC =	27.99 oF =	270.9 oK		
Load=	407 kW =	116 tons	1390 mBtu/hr		
CONDENSER:		b-c-d			
Tf:	35.0 oC =	95.0 oF =	308 oK		
Uwtr:	1.0	Uw:	7880	Ur:	0.30 kW/m2-oK
A:	350 m2	UAtot=	81 kW/oK		
Tc=	38.32 oC =	101.0 oF =	311.5 oK		
Load=	-483 kW =	-137 tons	-1648 mBtu/hr		
Superheat:	4.2 oC =	7.6 oF			
COMPRESSOR:		a-b			
Mass flow:	2.86 kg/s=	213.3 ltr/s=	452 cfm		
Powr=	75.7 kW(m)	101.5 hp =>	0.876 hp/ton		
COPe	4.84 =	0.73 kW(e)/ton with motor @			
	7.80 (Carnot)	90% eff.			
	T	T	P	h	s
	oK	oC	kPa	kJ/kg	kJ/kgK
a:	271	-2.2	269.9	246.9	0.920
b:	316	42.6	971	272.4	0.9202
c:	311	38.3	971	267.5	0.905
d:	311	38.3	971.5	103.7	0.379
e:	271	-2.2	270	104	0.395
					0.28
a:	271	-2.2	270	246	0.920
e':	271	-2.2	270	47.1	0.186
					0.00
Based largely on "Actual Basic Vapor Compression Ref. Cy ASHRAE Fundamentals - SI - 1985 (p. 1.11)					
Constants:					
R= 8.314 J/mol-K					
To= 273.2

Figure 5-3: Analysis of Ice-Building Operation with R-134a.

Water Chiller R134a $T_{ev}=45F$									
EVAPORATOR: e-a									
TT:	7.2	OC	=	45.0	OF	=	280	OK	
Uice:	0.99	M2		0.99	UR:	2.00	KW/M2-OK		
A:	185	M2		70.58	UAtot:		KW/OK		
Te:	3.28	OC	=	37.90	OF	=	276.4	OK	
Load=	501	KW	=	142	tons		1710	MBtu/hr	
CONDENSER: b-c-d									
TT:	35.0	OC	=	95.0	OF	=	308	OK	
Uw:	1.0	M2		7960	UR:	0.30	KW/M2-OK		
A:	350	M2		81	UAtot:		KW/OK		
Tc:	38.99	OC	=	102.2	OF	=	312.1	OK	
Load=	-580	KW	=	-165	tons		-1978	MBtu/hr	
COMPRESSOR: a-b									
Superheat:	3.5	OC	=	6.2	OF				
Mass flow:	3.47	KG/S	=	213.3	LT/S	=	452	CFM	
Power=	78.7	KW(M)		105.5	HP	>	0.741	HP/TON	
COPe	5.73		=	0.61	KW(E)/ION	WITH MOTOR @			
10.09 (Carnot) 90% eff.									
T	P	h	s	x	v	OK	KJ/KG	%VAP	M3/KG
a:	276	3.3	329.2	249.1	0.917	1.00	0.062		
b:	316	42.4	989	271.8	0.9172	1.00			
c:	312	39.0	989	267.8	0.904	1.00			
d:	312	39.0	989.1	104.7	0.382	0.00			
e:	276	3.3	329	105	0.395	0.26			
a:	276	3.3	329	249	0.917	1.00			
e:	276	3.3	329	54.4	0.213	0.00			
Based largely on "Actual Basic Vapor Compression Ref. Cy									
ASHRAE Fundamentals - SI - 1985 (p. 1.11)									
Constants:									
R = 8.314 J/mol-K									
To = 273.2									

Figure 5-4: Analysis of Water-Chilling Operation with R-134a.

metal of the tubes is $1.0 \text{ kW/m}^2\text{-}^\circ\text{C}^{27}$, the metal is an essentially infinite $7860 \text{ kW/m}^2\text{-}^\circ\text{C}$ and the film of condensing, nearly static refrigerant has a limiting heat transfer coefficient of $0.3 \text{ kW/m}^2\text{-}^\circ\text{C}^{28}$. As before, these are combined with an exogenously-derived heat exchange area of 350 m^2 to give an over-all condenser heat transfer coefficient of $81 \text{ kW/}^\circ\text{C}$. The condenser load, calculated below as 483 kW , then determines the delta-T and the refrigerant temperature of 101°F (38.3°C). The superheat, calculated below, is also reported in this box.

The compressor model is governed by the volumetric displacement, entered by the user. The vapor density (calculated below) then gives the mass flow, and the pressure difference between points "a" (compressor input port) and "b" (output port) multiplied by the volumetric displacement gives the mechanical power requirement reported here. Overall motor and compressor efficiency, taken as 90%, results in the indicated electric power requirements and the COP, which is compared to the efficiency of a Carnot refrigerator operating between the temperatures of the ice (or chilled water) and the 95°F cooling water.

The next box displays temperature, pressure, enthalpy, entropy and vapor fraction at the five state points and vapor density at the compressor input. Points "a" and "e" in the evaporator have the same pressure and temperature, differing in vapor fraction. (We have assumed a perfectly sized evaporator, so that the vapor fraction becomes unity just as the vapor enters the compressor.)

Points "b", "c" and "d" in the condenser are all at the same pressure, and "c" and "d" are at the same (liquid-vapor equilibrium) temperature. Similarly to our treatment of the evaporator, we assume the condenser is sized so as to just avoid sub-cooling the liquid after it has all condensed. The temperature at "b" is determined by demanding both $P_b = P_c$ and $S_b = S_a$ as a result of isentropic compression.

The other enthalpies and entropies are determined using the formulas for R-134a discussed in Section 3 above. The enthalpy at the hypothetical point "e'" (which is at the evaporator pressure and equilibrium temperature, but is 100% liquid) is combined with the enthalpies at "e" and "a" to determine the vapor fraction at e, although it is not required for the cycle analysis.

Three exogenously determined parameters define the mechanical system: the heat exchange areas of the evaporator and condenser and the displacement of the compressor. Also, two temperatures, that of the condenser cooling water and of the ice or chilled water in contact with the evaporator, define the operating conditions. With these five quantities specified, it is necessary to iterate the model two or three times to find a set of internally consistent temperatures and pressures, since changing the heat flow through the condenser and evaporator will change the temperature drop across them. Repeating these iterations for each configuration, variation of the compressor displacement and heat exchange areas allows determination of a system having the desired capacity and reasonable temperature drops across the condenser and evaporator.

The model was first validated by using R-22 data and duplicating ASHRAE performance tables²⁹ with respect to evaporator and condenser temperatures; in these cases the heat exchange areas were set very large, since ASHRAE dealt only with thermodynamic calculations. Then the heat exchange areas of the evaporator and condenser were decreased to physically reasonable values, and capacity and COP both dropped, as expected. In a detailed design study, optimal sizes would be found considering the trade-off between the cost of more heat exchanger and the lowered operating costs that more heat exchange area makes possible. For our purposes, such a detailed study would be premature, and we simply chose heat exchange areas and a compressor displacement that gave modest temperature drops across the evaporator and condenser.

Clearly there are ways this model could be over-estimating performance. We have mentioned ignoring pressure drops down the condenser and evaporator tubes and assuming perfect sizing for these components, so that vapor fractions always turn out exactly as they should. Also, our heat transfer coefficients are theoretical or rule-of-thumb estimates, a situation which can only be convincingly improved by building a prototype machine. However, there are also steps which can be taken in large machinery to improve efficiency, such as economizers, inter-cooled two stage compression and variable speed compressors, and we have not incorporated any of these gains into the model. We are confident that the level of accuracy employed here is appropriate for a demonstration of viability.

5.2 System Modelling with R-134a

The refrigerant data were then switched from R-22 to R-134a, with no other changes to the model. The result was a machine with lower capacity, higher COP and less superheat.

We then re-sized the components to model a machine with 116 tons of cooling power. Again, we did not truly optimize, but selected an evaporator temperature drop of 4°F (2.2°C) and a condenser drop of 6°F (3.4°C). The result is the system characterized in Figure 5-3. It has an over-all COP of 4.8, corresponding to an efficiency of 0.81 kW_e/ton. With an evaporator area of 185 m² (1990 ft²) it meets our design point cooling load of 116 tons (407 kW_{th}). This is less than one half the pad area required for our earlier design², which will result in substantial cost savings, so we did not attempt to lower pad area further by allowing larger temperature drops between ice and refrigerant.

Figure 5-4 then shows that this same system can provide 142 tons (501 kW_{th}) of cooling to 45°F water at a COP of 5.7 (0.68 kW_e/ton). Operating in this mode when water that has been warmed in the building's cooling system is available offers substantial savings. The efficiency figures derived for both ice-building and water chilling are used in Chapter 6 to estimate annual operating costs.

5.3 System Modelling with R-C318

Operating the model shown in Figure 5-1 with the refrigerant characteristics for R-C318 described in Chapter 3, we found that the machine described in Figure 5-1 and Figure 5-5 would produce ice with a design point capacity of 116 tons. Because the heat transfer properties are the same, the heat exchanger sizes are unchanged, but R-C318 requires a compressor with more than twice the volumetric capacity of that in the R-134a system. The system operates with an acceptable COP of 4.4, corresponding to 0.89 kW_e per ton. Water chilling operation is described in Figure 5-6. Again, the COP is about 9% lower than for the comparable R-134a machine, but otherwise operation is comparable.

For both ice-making and water chilling, the temperature-entropy diagrams reveal the unusual behavior described in Chapter 3: because the vapor equilibrium curve bends out to the right as entropy increases, the refrigerant will partially condense during compression. Calculating this effect required some straightforward re-arrangement of the spreadsheet, which now finds the vapor fraction at "b" (the compressor output/condenser input) as well as at "e". About 12% of the vapor is condensed during the compression stroke. This is probably not analogous to the "slugging" that occurs when un-evaporated liquid is pulled into the compressor, since the condensation will take place throughout the well-mixed vapor, resulting in a heavy mist which should not cause undue stress on the compressor.

However, this situation is one that is unfamiliar and therefore troubling, and it also results from a cycle that is less than optimal. If the vapor coming from the evaporator were suitably superheated before compression, it would enter the condenser with a vapor fraction of 100%, avoiding the uncertainties of "misting" in the compressor and the attendant inefficiencies. This can be achieved by adding a heat exchanger between fluid exiting the condenser and that which is entering the compressor, as shown in Figure 5-7.

Figure 5-7: Ice Maker Superheated System Diagram.

Operation of this system is shown in Figure 5-8 and Figure 5-9. The vapor exiting the evaporator at "a" is superheated until its entropy at "b" equals that at "c", the entrance to the condenser. Knowing the entropy and temperature allows us to determine the temperature and enthalpy at "b" from the refrigerant properties, and this determines the quantity of heat needed to produce this superheat. The hot liquid exiting the condenser at "d" is sub-cooled until it enters the expansion valve at "d", with its enthalpy, entropy and temperature all reduced by the removal of the required quantity of heat.

Ice Maker		R-C318		Tev=32F			
EVAPORATOR:		e-a		(No superheat)			
Tf:	0.0 oC =	32.0	oF =	273	oK		
Uice:	3.20	Uw:	0.99	Ur:	2.00 kW/m2-oK		
A:	185 m2	UAtot=	101.5	kW/oK			
Te =	-2.23 oC =	27.98	oF =	270.9	oK		
Load=	408 kW =	116	tons=	1392	mBtu/hr		
CONDENSER:		b-c-d					
Tf:	35.0 oC =	95.0	oF =	308	oK		
Uwtr:	1.0	Uw:	7860	Ur:	0.30 kW/m2-oK		
A:	350 m2	UAtot=	81	kW/oK			
Tc =	38.38 oC =	101.1	oF =	311.5	oK		
Load=	-491 kW =	-140	tons=	-1677	mBtu/hr		
Superheat:	0.0 oC =	0.0	oF				
COMPRESSOR:		a-b					
Mass flow:	5.96	kg/s=	528	l/s=	1118 cfm		
Powr=	83.5 kW(m)	111.9	hp =>	0.965 hp/ton			
COPe	4.40 =	0.80	kW(e)/ton with motor @				
	7.80 (Carnot)		90% eff.				
	T	T	P	h	s	x	v
	oK	oC	kPa	kJ/kg	kJ/kgK	%vap	m3/kg
a:	271	-2.2	118.0	614.2	1.405	1.00	0.089
b:	312	38.4	470	628.2	1.405	0.88	
c:	312	38.4	470	639.8	1.442	1.00	
d:	312	38.4	469.6	545.7	1.140	0.00	
e:	271	-2.2	118	545.7	1.152	0.39	
a:	271	-2.2	118	614.2	1.405	1.00	
e':	271	-2.2	118	501.6	0.989	0.00	
Based largely on "Actual Basic Vapor Compression Ref. Cycle", ASHRAE Fundamentals - SI - 1985 (p. 1.11)							
Constants:							
R= 8.314 J/mol-K							
To= 273.2

Figure 5-5: Analysis of Ice-Building Operation with C318.

Water Chiller			R-C318		Tev=45F		
EVAPORATOR:			e-a (No superheat)				
Tf:	7.2 oC =	45.0	oF=	280	oK		
Uwtr:	0.90	Uw:	0.99	Ur:	2.00	kW/m2-oK	
A:	185	m2		UAtot=	70.58	kW/oK	
Te=	3.15 oC =	37.67	oF=	276.3	oK		
Load=	518 kW =	147	tons=	1767	mBtu/hr		
CONDENSER:			b-c-d				
Tf:	35.0 oC =	95.0	oF=	308	oK		
Uwtr:	1.0	Uw:	7880	Ur:	0.30	kW/m2-oK	
A:	350	m2		UAtot=	81	kW/oK	
Tc=	39.17 oC =	102.5	oF=	312.3	oK		
Load=	-606 kW =	-172	tons=	-2070	mBtu/hr		
Superheat:	0.0 oC =	0.0	oF				
COMPRESSOR:			a-b				
Mass flow:	7.29	kg/s=	528	ltr/s=	1118	cfm	
Powr=	88.8 kW(m)	119.1	hp =>	0.809	hp/ton		
COPe	5.24 =	0.67	kW(e)/ton with motor @				
	10.09 (Carnot)		90% eff.				
	T	T	P	h	s	x	v
	oK	oC	kPa	kJ/kg	kJ/kgK	%vap	m3/kg
a:	276	3.1	145.6	617.6	1.410	1.00	0.072
b:	312	39.2	480	629.8	1.410	0.89	
c:	312	39.2	480	640.3	1.443	1.00	
d:	312	39.2	480.5	546.6	1.143	0.00	
e:	276	3.1	146	546.6	1.153	0.36	
a:	276	3.1	146	617.6	1.410	1.00	
e':	276	3.1	146	507.2	1.010	0.00	
Based largely on "Actual Basic Vapor Compression Ref. Cycle", ASHRAE Fundamentals - SI - 1985 (p. 1.11)							
Constants:							
R= 8.314 J/mol-K							
To= 273.2

Figure 5-6: Analysis of Water-Chilling Operation with C318.

Ice Maker		R-C318		T _{ev} =32F	
EVAPORATOR: e-a (With superheat)					
T _f :	0.0	oC =	32.0	oF =	273 oK
U _{ice} :	3.20	U _w :	0.99	U _r :	2.00 kW/m ² -oK
A:	185	m ²	U _A tot=	101.5	kW/oK
T _e :	-2.23	oC =	27.98	oF =	270.9 oK
Load=	408	kW =	116	tons=	1393 mBtu/hr
CONDENSER: c-d					
T _f :	35.0	oC =	95.0	oF =	308 oK
U _{wtr} :	1.0	U _w :	7850	U _r :	0.30 kW/m ² -oK
A:	350	m ²	U _A tot=	81	kW/oK
T _c :	38.35	oC =	101.0	oF =	311.5 oK
Load=	-488	kW =	-139	tons=	-1664 mBtu/hr
COMPRESSOR: b-c					
Mass flow:	5.18	kg/s=	488	ltr/s=	1034 cfm
Powr=	79.3	kW(m)	106.4	hp =>	0.916 hp/ton
COPe	4.63	=	0.76	kW(e)/ton with motor @	
	7.80	(Carnot)		90% eff.	
	T	T	P	h	s
	oK	oC	kPa	kJ/kg	kJ/kgK
a:	271	-2.2	118.0	614.2	1.405
b:	284	10.7	118	624.5	1.442
c:	312	38.4	469	639.8	1.442
d:	312	38.4	469.3	645.7	1.140
d'	302	29.33	469	535.4	1.107
e:	271	-2.2	118	535.4	1.114
a:	271	-2.2	118	614.2	1.405
e':	271	-2.2	118	501.6	0.989
Based largely on "Actual Basic Vapor Compression Ref. Cycle", ASHRAE Fundamentals - SI - 1985 (p. 1.11)					
Constants:					
R= 8.314 J/mol-K					
T _o = 273.2

Figure 5-8: Analysis of Ice-Building with Superheated C318.

Water Chiller			R-C318		Tev=45F	
EVAPORATOR:			e-a (With superheat)			
Tf:	7.2 oC =	45.0	oF=	280	oK	
Uice:	3.20	Uw:	0.99	Ur:	2.00	kW/m2-oK
A:	185	m2	UAtot=	101.5	kW/oK	
Te=	4.28 oC =	39.70	oF=	277.4	oK	
Load=	538 kW =	153	tons=	1835	mBtu/hr	
CONDENSER:			c-d			
Tf:	35.0 oC =	95.0	oF=	308	oK	
Uwtr:	1.0	Uw:	7860	Ur:	0.30	kW/m2-oK
A:	350	m2	UAtot=	81	kW/oK	
Tc=	39.28 oC =	102.7	oF=	312.4	oK	
Load=	-623 kW =	-177	tons=	-2125	mBtu/hr	
COMPRESSOR:			b-c			
Mass flow:	6.65	kg/s=	488	l/s=	1034	cfm
Powr=	85.0 kW(m)	114.0	hp =>	0.745	hp/ton	
COPe	5.69	=	0.62	kW(e)/ton with motor @		
	10.09 (Carnot)			90% eff.		
	T	T	P	h	s	x
	oK	oC	kPa	kJ/kg	kJ/kgK	%vap
a:	277	4.3	152.1	618.4	1.411	1.00
b:	289	15.6	152	627.6	1.443	1.00
c:	312	39.3	482	640.4	1.443	1.00
d:	312	39.3	482.0	646.8	1.144	0.00
d'	304	31.20	482	537.5	1.114	0.00
e:	277	4.3	152	537.5	1.119	0.26
a:	277	4.3	152	618.4	1.411	1.00
e':	277	4.3	152	508.5	1.014	0.00
Based largely on "Actual Basic Vapor Compression Ref. Cycle", ASHRAE Fundamentals - SI - 1985 (p. 1.11)						
Constants:						
R= 8.314 J/mol-K						
To= 273.2

Figure 5-9: Analysis of Water-Chilling with Superheated C318.

This subcooling benefits the performance by moving the state point "e" to the left in the pressure-enthalpy diagram, providing more cooling on the way from "e" to "a". This benefit is partly offset by increased enthalpy of compression in the superheat region. There is a net benefit, however, and the result is a system that compresses without producing condensation and which has an ice-making COP 5% higher than without the extra heat exchanger providing subcooling. In the water chilling mode the COP is 8.6% higher with subcooling. Electric loads are correspondingly lower in both cases.

However, to be sure this is a reasonable system, the required heat exchanger size should be determined. The enthalpy differences between points "a" and "b" and the rate of mass flow imply that the heat exchanger must transfer 53 kW of thermal energy between the two streams in ice-building operation, and 61 kW in water-chilling operation. The latter is the more demanding case for the heat exchanger, with a log mean temperature difference of 25°C requiring a heat exchanger with "UA" equal to 2.5 kW/°C. The ice building mode has an LMTD of 30°C and requires "UA" only equal to 1.8 kW/°C. In either case, the heat exchanger will have less than one square meter of heat exchange area, quite modest for a system of this size, and will have a small effect on total cost.

The result is a system which appears practical and which, we will see in the next chapter, will be much less expensive to construct due to the reduced pressure requirements. Clearly many questions about R-C318 remain to be answered, but the technical promise of the refrigerant seems substantial.

6. SYSTEM COST ESTIMATES

Here we integrate the operational data of the previous sections to derive life-cycle costs for a thin polymer ice making system based on the use of compression refrigerants in the polymer evaporator. The approach and economic parameters will be taken from our previous economic assessment², and we will continue to work in the early 1992 dollars employed there to permit direct comparisons. The reader can estimate costs in current (1995) dollars by increasing the numbers presented here by 6%³⁰.

In Section 4 we developed component sizes based on the requirements of our economic assessment. We also continue to use the electric rate structure presented in the economic assessment, which we repeat in Table 6-1 for reference. The rates are high compared to U.S. national averages, but it is in just such locations that cold storage is most attractive.

6.1 Cost Estimates with R-134a

We then prepared a list of materials for the R-134a ice maker itself, presented in Table 6-2. The costs are largely from construction estimating handbooks³¹, updated to 1992 using the capital goods price inflator³⁰.

Table 6-1 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	

Since we need a total area of 185 m² (1991 ft²) (from Table 5-1) and since each pad has a heat exchange area of 10.8 ft², 184 pads are needed. The PFA film is seen to be a relatively small part of the total cost, indicating that there is little to be gained by trying to use some less expensive polymer, which would risk lowering system reliability. The header/support structure is complex, involving 68 tee's for each pad as well as the narrow tube which distributes refrigerant down the length of the evaporator tube, and for costing purposes is assumed to be made of copper. It should be possible to mass-produce these headers (one for each side) from some less expensive material, which could lower the cost of the pads substantially, but we have not assumed that here. Rather, we have treated the entire job as a custom plumbing installation, taking the costs from the estimating handbooks³¹. Since the pads would actually be assembled in a factory and shipped to the job site, we have compensated for this over-estimate by not charging mark-ups ("general and administrative" or "overhead and profit") on these components.

The distribution piping is also complex, and again, we have used prices appropriate to copper. It should be possible to use aluminum, since it is quite corrosion-resistant at low

Table 6-2: Component Costs for Polymer Film Ice Maker.

Item	Quantity /Size	Units	Unit Cost (\$/unit)	Item Cost(\$)	System Cost(\$)
Pads:					
	184				
PFA Film	12.9	sf	1.16	15	
Header/Support	1		56.00	56	
Check Valve	1		11.00	11	
Insulation	4	ft	0.33	1	
Assembly				40	
Pad Total=				123	22,632
Piping:					
1-1/4" Copper	4x40	ft	12	1920	
3" Copper	4x40	ft	28	4480	
1-1/4" Tees	184		35	6440	
3" Tees	184		63	11,592	
Total:				24,432	24,432
Compressor & Condenser:	142 (nominal)	tons	84,870	+15%O&P	97,600
ICE MAKER TOTAL:					144,664

temperatures, and this would lower the cost for materials by two-thirds. Again, these manifolds would be factory-assembled, so we have not charged mark-ups.

The cost of the compressor/water cooled condenser combination was obtained from the cost of an efficient water-cooled water chiller³² by taking a \$10,000 credit for the unneeded evaporator; here the recommended 15% mark-up was added. The four tanks, described in Appendix B, are quite large and it may be advisable to bury them, leaving access space at one end. Means reports buried 30,000 gallon tank costs (with three-eighths inch walls and galvanic protection) which scale to ⁹²\$36,800 for 35,692 gallons. Including a 17% recommended mark-up, the four tanks will then cost \$173,064, more than was estimated in our previous study. We use a previously-derived² \$14,000 capital cost for the cooling tower, and assume (as there) that 10 kW of auxiliary power is required for the cooling tower itself.

These costs are combined with the electric rates above to obtain life-cycle costs and the net present value of operating this system. The details are in Appendix F, where ongoing operating costs are converted to a present value using a 6% per year discount rate. The calculations in Appendix F assume twenty-four hour per day operation of the system with eight hours per day in water chilling mode, which occurs during the time of peak electric rates. (The electric rate structure is embedded in the spreadsheet.) The power needed to distribute the refrigerant around the system, calculated in Appendix D as 259 watts for water chilling and 199 watts for ice making is added to the 10 kW needed for the cooling tower. Maintenance is charged out at \$1000 per month, including repairs and replacements.

Table 6-3: Economic Characteristics of the Polymer Ice Maker, R-134a.

CAPITAL COSTS	
Ice Maker (\$1000):	145
Installation (\$1000):	(Inc.)
Tank (\$1000):	173
Tower (\$1000):	14
OPERATING COSTS	
Electricity (\$/mnth):	6642
Maintenance (\$/mnth):	1000
PRESENT VALUE	
(Over 5 yrs, \$1000):	480

These results are summarized in Table 6-3 and compared to the competition in the Summary. The thin polymer ice maker used directly as the evaporator of the R-134a based compression chiller certainly seems to offer substantial technical and economic advantages over existing systems.

6.2 Cost Estimates with R-C318

Our treatment of the R-C318 based system parallels that of the R134a system closely. Table 6-2 still describes the ice maker itself and the compressor and condenser and is still, we believe, quite conservative in assuming the entire apparatus is constructed as a piecework copper plumbing job, rather than in a factory from less expensive materials such as aluminum. One might question whether the more than doubled displacement required by the R-C318 would not lead to higher costs for the compressor, but the power requirements have only increased by 10%. Further, although a greater volume of refrigerant must be moved, the peak pressures are half those in the R-134a system, so many construction constraints can be relaxed.

The remaining economic characteristics are detailed in Appendix G and summarized in Table 6-4. The major item is the tank cost. We used an estimate of \$132k in our previous study², taken from a well-documented real system³³. We here assume that the minor pressurization (comparable to that in an inflatable tennis arena) can be accomplished by strengthening and sealing that structure, and simply increase the cost by 10% to represent this modest additional work. The result, \$145k, is substantially below the cost of the pressurized system required by the use of R-134a.

Other characteristics of the R-C318 system are transferred into Appendix G as they were in the previous case, and the resulting net present value, \$457k, is sufficiently less than that of R-134a and of the available commercial units to merit considerable continued investigation.

Table 6-4: Economic Characteristics of the Polymer Ice Maker, R-C318.

CAPITAL COSTS	
Ice Maker (\$1000):	145
Installation (\$1000):	(Inc.)
Tank (\$1000):	145
Tower (\$1000):	14
OPERATING COSTS	
Electricity (\$/mnth):	6964
Maintenance (\$/mnth):	1000
PRESENT VALUE	
(Over 5 yrs, \$1000):	457

31. "Means Mechanical Cost Data", Robert Snow Means Co., Kingston MA, 02364 (1985).
32. Means, op. cit., 15.7, line 158.
33. C. M. Landry and C. D. Noble, "Making Ice Thermal Storage First-Cost Competitive", ASHRAE Journal 33, No. 5, p. 19 (May 1991).

APPENDICES A - G

Appendix C: Pad Design Calculations

Pad Characteristics:

Pad Length: 3.0 ft
Tubes: 34.0
WeldWidth: 0.125 in
Thickness: 3 mils

Expanded:

TubeDiam: 0.404 in
Total Height= 18.0 in
HX Area= 10.8 ft²/pad
= 83.6% of total

Deflated:

WeldSep= 0.760 in
ShtHght= 25.8 in
PolArea= 12.9 ft²/pad

Long term design limits:

Stress < 1000 psi (in polymer)
=> Diff P < 14.8 psi (across wall)

ASME Code calculation:

Burst stress: 2180 psia at 70oF (Ref. 9, Table 1-3)
Diff.P @ 1/7: 4.6 psi, << long-term design limit above.
R134a @ 40F: 50 psia
=> P(water) 45.4 psia

Appendix D: Pipe Sizing Calculations for R-134a

System Mass flow:	3.47 kg/s	Water Chilling at T =	3.3 °C
Volume Flow with	26% flash not circulated.	Density	Viscosity
Liquid:	2.00 l/s = 31.7 gal/min @	1.284 kg/liter	1.40E-04
Vapor:	158 l/s = 334 cfm @	0.016 kg/liter	1.25E-05
Distribution			Pa-s
# of sub-systems:	4 tanks		
Pipe Length:	12.2 m = 40.0 ft		
Liquid pipe ID:	3.175 cm = 1.25 in		
Liquid Fluid Vel=	0.63 m/s = 2.07 ft/s		
Liq. Reynold's #=	183,570 f= 0.0157		
Liq.Press. Drop =	1.54 kPa = 0.224 psi		
Liq. Pump Power=	3.09 W = 0.0041 hp, total		
Vapor pipe ID:	7.62 cm = 3.0 in		
Vapor Fluid Vel=	8.64 m/s = 28.33 ft/s		
Vap. Reynold's #=	859,192 f= 0.0119		
Vap.Press. Drop =	1.15 kPa = 0.167 psi		
Vap. Pump Power=	181.85 W = 0.2439 hp, total		
Headers			
# of sub-systems:	184 pads => 46 pads/tank		
Pipe Length:	0.6 m = 2.0 ft		
Liquid pipe ID:	0.3175 cm = 0.125 in		
Liquid Fluid Vel=	1.37 m/s = 4.50 ft/s		
Liq. Reynold's #=	39,907 f= 0.0224		
Liq.Press. Drop =	5.20 kPa = 0.754 psi		
Liq. Pump Power=	10.40 W = 0.013945 hp, total		
Vapor pipe ID:	1.27 cm = 0.5 in		
Vapor Fluid Vel=	6.76 m/s = 22.18 ft/s		
Vap. Reynold's #=	112,069 f= 0.0172		
Vap.Press. Drop =	0.31 kPa = 0.045 psi		
Vap. Pump Power=	48.56 W = 0.0651 hp, total		
Tubes			
# of sub-systems:	1564 tubes <= 34 tubes/pad		
Pipe Length:	0.9 m = 3.0 ft		
Liquid pipe ID:	0.16 cm = 0.063 in		
Liquid Fluid Vel=	0.65 m/s = 2.12 ft/s		
Liq. Reynold's #=	9,390 f= 0.0321		
Liq.Press. Drop=	4.96 kPa = 0.719 psi		
Liq. Pump Power=	9.92 W = 0.0133 hp, total		
Vapor pipe ID:	1.03 cm = 0.404 in		
Vapor Fluid Vel=	1.22 m/s = 4.00 ft/s		
Vap. Reynold's #=	16,317 f= 0.0280		
Vap.Press. Drop=	0.03 kPa = 0.004 psi		
Vap. Pump Power=	4.75 W = 0.0064 hp, total		
System totals:			
Liq.Press. Drop=	11.70 kPa = 1.697 psi		
Liq. Pump Power=	23 W = 0.0314 hp		
Vap.Press. Drop=	1.49 kPa = 0.216 psi		
Vap. Pump Power=	235 W = 0.3153 hp		
Total Press.Drop=	13.20 kPa = 1.914 psi		
Total Pump Power=	259 W = 0.3467 hp		

Appendix E: Pipe Sizing Calculations for R-C318

System Mass flow:	5.85 kg/s	Water Chilling at T =	3.3 °C
Volume Flow with	26% flash not circulated.	Density	Viscosity
Liquid:	3.11 l/s = 49.3 gal/min @	1.581 kg/liter	5.37E-04
Vapor:	354 l/s = 750 cfm @	0.014 kg/liter	1.08E-05
Distribution			Pa-s
# of sub-systems:	4 tanks		
Pipe Length:	12.2 m = 40.0 ft		
Liquid pipe ID:	3.175 cm = 1.25 in		
Liquid Fluid Vel=	0.98 m/s = 3.22 ft/s		
Liq. Reynold's #=	91,955 f= 0.0182		
Liq.Press. Drop =	5.33 kPa = 0.773 psi		
Liq. Pump Power=	16.58 W = 0.0222 hp, total		
Vapor pipe ID:	7.62 cm = 3.0 in		
Vapor Fluid Vel=	19.41 m/s = 63.68 ft/s		
Vap. Reynold's #=	1897352 f= 0.0104		
Vap.Press. Drop =	4.35 kPa = 0.631 psi		
Vap. Pump Power=	1539.99 W = 2.0651 hp, total		
Headers			
# of sub-systems:	184 pads => 46 pads/tank		
Pipe Length:	0.6 m = 2.0 ft		
Liquid pipe ID:	0.3175 cm = 0.125 in		
Liquid Fluid Vel=	2.14 m/s = 7.01 ft/s		
Liq. Reynold's #=	19,990 f= 0.0266		
Liq.Press. Drop =	18.44 kPa = 2.674 psi		
Liq. Pump Power=	57.38 W = 0.076951 hp, total		
Vapor pipe ID:	1.27 cm = 0.5 in		
Vapor Fluid Vel=	15.19 m/s = 49.83 ft/s		
Vap. Reynold's #=	247,481 f= 0.0148		
Vap.Press. Drop =	1.14 kPa = 0.166 psi		
Vap. Pump Power=	404.44 W = 0.5424 hp, total		
Tubes			
# of sub-systems:	1564 tubes <= 34 tubes/pad		
Pipe Length:	0.9 m = 3.0 ft		
Liquid pipe ID:	0.16 cm = 0.063 in		
Liquid Fluid Vel=	1.01 m/s = 3.30 ft/s		
Liq. Reynold's #=	4,704 f= 0.0382		
Liq.Press. Drop=	17.59 kPa = 2.551 psi		
Liq. Pump Power=	54.74 W = 0.0734 hp, total		
Vapor pipe ID:	1.03 cm = 0.404 in		
Vapor Fluid Vel=	2.74 m/s = 8.98 ft/s		
Vap. Reynold's #=	36,034 f= 0.0230		
Vap.Press. Drop=	0.11 kPa = 0.015 psi		
Vap. Pump Power=	37.72 W = 0.0506 hp, total		
System totals:			
Liq.Press. Drop=	41.35 kPa = 5.997 psi		
Liq. Pump Power=	129 W = 0.1726 hp		
Vap.Press. Drop=	5.60 kPa = 0.812 psi		
Vap. Pump Power=	1982 W = 2.6581 hp		
Total Press.Drop=	46.95 kPa = 6.809 psi		
Total Pump Power=	2111 W = 2.8307 hp		

Appendix F: Economic Analysis for R-134a

Tank Capacity:

Ice=	5519 ton-hrs=	7998 ft3 of ice	
Volume=	142,656 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		
Power: Compressor:	0.73 kW/ton or	0.61 kW/ton as chiller for	8 hr/day
Auxiliary:	10.199 kW or	10.259 kW as chiller	
Total=	0.818 kW/ton or	0.699 kW/ton as chiller	

Economics:

Ice Maker=	\$144,664 or	\$1,251 /ton	
Tank Cost=	173,064 or	\$9.05 /ft3	
Cool Tower=	14,369		
5 -yr electric=	97,381 based on	\$2,373 demand	\$4,269 energy monthly
+ Maintenance	50,548	\$1,000 /month	
Discounted @	6% /year		
TotalPV(\$1000)	480		

Appendix G: Economic Analysis for R-C318

Tank Capacity:

Ice=	5519	ton-hrs=	7998	ft3 of ice	
Volume=	142,656	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			
Power: Compressor:	0.76	kW/ton or	0.62	kW/ton as chiller for	8 hr/day
Auxiliary:	11.315	kW or	12.111	kW as chiller	
Total=	0.858	kW/ton or	0.725	kW/ton as chiller	

Economics:

Ice Maker=	\$144,664	or	\$1,251	/ton	
Tank Cost=	145,000	or	\$7.58	/ft3	
Cool Tower=	14,369				
5 -yr electric=	102,068	based on	\$2,488	demand	\$4,476 energy monthly
+ Maintenance	50,548		\$1,000	/month	
Discounted @	6%	/year			
TotalPV(1000)=	457				