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COST REDUCTIONS IN ABSORPTION CHILLERS
FINAL REPORT (JUNE 1984 - MAY 1985)

R. W. Leigh, Principal Investigator

Contributors:

D. W. Huszagh, H. S. Isaacs, J. G. Murray*, A. Ravve**,
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May 1986

Prepared for the Gas Research Insitute

DEPARTMENT OF APPLIED SCIENCE

BROOKHAVEN NATIONAL LABORATORY
UPTON, LONG ISLAND, NEW YORK 11973

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ARCHITECTURAL AND BUILDING SYSTEMS DIVISION
DEPARTMENT OF APPLIED SCIENCE
BROOKHAVEN NATIONAL LABORATORY
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Research Summary

Title	Cost Reductions in Absorption Chillers
	GRI Contract Number 5084-243-1002
Contractor	Brookhaven National Laboratory Department of Applied Science Solar and Thermal Systems Division
Principal Investigator	R. W. Leigh
Time Span	September 1984 - June 1985 Final Report
Major Achievements	Absorption chillers have great difficulty competing with the electric-driven compression alternative, due in part to modest operating efficiencies and largely to high first costs. This project is an assessment of the possibility of lowering the costs of absorption chillers dramatically by the use of low material intensity in the design of a new generation of these machines. We established breakeven costs for absorption chillers, their heat exchangers and heat exchanger materials which will allow commercial success. We identified polymeric and metallic materials appropriate to particular components and which meet the cost goals. We tested a subset of these materials and ordered them by success in tolerating conditions and materials found in absorption chiller applications. We developed conceptual designs which indicate the practicality of the low material intensity approach.
Recommendations	The work reported here indicates that there is a high probability that this approach will be successful. RNL recommends a strong continued research effort, incorporating further materials identification and testing, the fabrication and testing of miniature heat exchanger prototypes, studies of corrosion mechanisms in thin metallic structures, studies of control of phase change processes, development of a system modelling capability and initiation of system design studies.
Description of Work Completed	Standard economic techniques were used to establish breakeven costs for components and materials. Both pure and welded samples of polymers were subject to various mechanical and chemical tests before and after exposure to lithium bromide. The resistance of candidate metals to pitting corrosion was tested electrochemically in lithium bromide, and the metals tested mechanically. Thermodynamic and heat transfer analysis was used to ascertain that design criteria could be met by candidate materials.

GRI Comment

The work performed by Brookhaven National Laboratory was to evaluate the possibility of reducing the first costs of absorption chillers sufficiently to improve their competitive posture with respect to compression chillers. The primary mechanism of cost reduction was to be lowered material intensity in chiller components, especially heat exchangers. If successful, this effort would result in enhanced utilization of gas for space cooling not only in the directly fired units examined here but also in applications fired by cogeneration and reject heat. The contractor completed this phase of the work as planned, identifying several routes to significant cost reduction in each of the heat exchangers comprising the absorption chillers.

The Gas Research Institute reviewed this report in conjunction with its cooling systems project advisors prior to publication. Having concurred in principle with the results, GRI and its advisors are supporting the continuation of this effort. The next phase will continue the materials testing and selection process begun here, and will move on to fabricate and test various miniature prototype heat exchangers.

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1. INTRODUCTION AND SUMMARY

1.1 Introduction

The absorption chiller market in the United States is in a state of near total collapse. Recently both Arkla (under new owner Preway) and York have terminated Lithium Bromide absorption operations completely. Japanese competition has attempted to make inroads, but the market success of even these well-engineered machines has been limited. The problem is essentially that the compression alternative is inexpensive, reliable and efficient enough to offer acceptable operating costs. It appears that absorption chillers, and the gas markets that they will create, can only be resuscitated by very substantial improvements in their salient characteristics, whether these be lowered first costs, improvements in efficiency or more reliable operation.

A recent research program at Brookhaven National Laboratory (BNL) has assessed the possibility of lowering the first costs of absorption chillers dramatically through lowered material intensity and the use of lower cost materials, primarily in the heat exchangers which make up the bulk of the operating components of these systems. This must be done while retaining the best performance characteristics available today, a net design point coefficient of performance (COP) of 1.3 and a gross design point COP of 1.0 in a directly fired, double effect unit.

We have investigated several possible routes to these goals, and here report on these findings, focussing on the areas that appear most promising. The candidates include the use of electroplated polymer coatings on inexpensive metal foils to replace heavy steel components or components made of more noble (and more expensive) metals, the use of thin strips of new, corrosion resistant alloys to replace thicker, less impervious metals in applications exposed to gas flames and the use of all-polymer heat exchangers in lower temperature components. The use of such materials is only possible in the context of new heat exchanger and system designs, which are also discussed.

1.2 Background

Our cost goal approach originated in work done for the U.S. Department of Energy (DOE) in pursuit of solar energy systems which could compete economically with conventional energy sources.[1-1] A straightforward assessment demonstrated that appropriate retail costs were far below those then currently available, forcing a significant redesign of solar collectors from the ground up. The key to the new designs was lowered material intensity, since the cost goals dictated that only very small amounts of anything that was not free could be included in the collector. Thin films of material, in many cases polymers, were identified as likely candidates, and the problem at that point was reduced to the engineering problem of finding materials which could perform adequately while withstanding the rigors of the outdoor environment.

Absorption chillers are ripe for a similar analysis and development. While there are many advantages to the underlying design, including quiet, reliable operation, reasonable resource use efficiency and the possibility of cooling through the recovery of cogenerated heat, the systems are currently too expensive to compete with the electric alternative.

We therefore analyzed the economic circumstances of absorption chillers to determine cost goals both for complete systems and on a component by component basis. We then examined available information on possible thin materials to select an initial set of candidates which would meet both technical and cost requirements. Information on the interactions of many materials with the lithium bromide brine used in the chillers was sparse or non-existent, so we report on a set of experiments which allowed us to begin the process of winnowing out the candidates least able to tolerate the environment to which they would be exposed. Finally, to demonstrate that chillers could be constructed using these materials, we developed some preliminary conceptual designs and analyzed several salient questions that arose during this work. The results of this work are summarized here, the details of each area of study are developed in the main body of the report, and various technical details are incorporated into appendices.

1.3 Summary

We first examined existing absorption chillers to determine current engineering practice and system costs; Chapter 2 reviews these areas in detail. Double effect systems directly fired by gas offer distinct efficiency and engineering advantages and were chosen early on as the baseline design for economic assessment and engineering analysis. For the two systems used in most of our economic assessments, of 100 and 400 tons capacity, our survey of current manufacturers revealed retail prices of \$700 and \$325 per ton, respectively. (All cost data are reported in mid-1984 dollars.) Our other significant finding in this area was that contemporary chemical maintenance procedures can prevent failures due to corrosion on the interior of most current systems; water side corrosion is more difficult, since the chemical content of either system water or cooling tower water is harder to control.

We then established baseline designs for assessment purposes. These are systems which could be built with the best of current industry practice; indeed, closely related units have been built. In these baseline designs for systems of 3, 25, 100 and 400 tons we developed tables delineating both the operating conditions which are faced by heat exchangers in these machines and the current practices used in constructing them, including materials choices, flow rates, temperatures, pressure differentials and other information relevant to the assessment of heat exchanger materials. The machines are designed to give gross design point COP's of 1.0, with net COP's (ignoring boiler losses) of 1.3, and this has been achieved in selected production machines.

We then proceeded to use a specified set of assumptions to derive breakeven costs for future absorption chillers. The "breakeven cost"

is the capital cost for an absorption chiller which would make its total lifecycle costs equal to or lower than those of the electric compression alternative. It is thus a price at which absorption chillers could compete effectively with compression systems. Various complexities such as distinctions between factory, retail and installed costs, design point versus seasonal average efficiencies and the selection of suitable fuel price scenarios are detailed in Chapter 4.

The resultant retail breakeven cost estimates are presented in Table 1-1 and the costs are broken down on a heat exchanger by heat exchanger basis in Table 1-2.

Table 1-1
Estimated Breakeven Cost of Absorption Chillers
(1984 dollars per ton)

Capacity (tons)	Current Retail Price	Breakeven Retail Price	Breakeven Factory Price
3*	NA	1340	570
25	950	360	190
100	700	290	165
400	325	175	95

*The 3 ton unit includes air handlers and a cooling tower and is not directly comparable with the larger machines. Such units are currently not available.

Table 1-2
Estimated Breakeven Costs of Heat Exchangers
(Factory Prices, 1984 dollars, 400 ton unit)

Heat Exchanger	Total Breakeven Cost (\$)	Fabrication Breakeven Cost (\$)	Materials Breakeven Cost (\$)	Materials Breakeven Cost (\$/ft ²)
1st Effect Generator	4140	340	3800	6.30
2nd Effect Generator	2070	690	1380	1.40
Condenser	2420	340	2070	3.00
Evaporator	3460	690	2760	1.90
Absorber	3460	690	2760	2.00
Hi Temp Recuperator	1210	690	520	0.90
Lo Temp Recuperator	1210	690	520	0.90

The "Breakeven Retail Price" results for large machines are in reasonable agreement with comparable estimates made by GRI [1-2], as is discussed in Chapter 4. The material costs indicated in Table 1-2 serve as our basis of comparison; if we do not have good reason to think that a material will cost less than the amount shown in Table 1-2, it will not be considered in the first round assessment. At a later date we may consider trading off costs between heat exchangers, the benefits of designs which permit one component to carry out more than one function and the possible increases in allowable material costs which could result from lowering production costs through automation. However, with the use of thin materials, the cost goals of Table 1-2 appear to be easily obtainable.

Chapter 5 then recounts our search for materials that can withstand the conditions delineated in Chapters 2 and 3 and meet the economic criteria of Chapter 4. After elimination of polymers subject to hydrolysis or permeable to water, the remainder of the most complete list of available materials we could assemble was ordered according to the product of thermal conductivity and yield strength. This order is of interest because the thermal conductivity of polymers is sufficiently lower than that of metals to be a matter of concern in heat exchanger design, and because the differential pressures of up to two or three atmospheres put substantial stresses on the heat exchanger materials. Table 1-3 presents the top end of this list. Costs are not indicated in detail because the most expensive materials, particular fluoroplastics, cost \$0.30 per mil-thick square foot, well below the breakeven costs of Table 1-2, and most materials are substantially less expensive. If the polymers can be made to work technically, they will offer dramatic cost advantages.

Chapters 6 and 7 describe the tests performed on these materials at BNL, both on the material properties themselves and on samples welded in simple configurations. Double asterisks in table 1-2 indicate which polymeric materials have been tested at BNL to date. Most of the candidates high on the list which have not been tested have proven to be difficult to obtain in thin (less than ten mil) films. To date ECTFE (Halar) has performed best in our tests, especially in being readily and reliably weldable over long seams, and at \$0.15 per mil-ft² it is very attractive from an economic point of view. The selection process is ongoing, and the strength of all the polymers at the higher temperatures required in some of the heat exchangers has yet to be tested.

At least the first effect generator, facing a gas flame, cannot be made of polymers, and some other heat exchangers operate at temperatures which may cause trouble with thermoplastic materials. For this reason, we also assessed the possibility of using thin metal strips or foils. A list of metals expected to be resistant to corrosion on the basis of tabulated properties was assembled and samples obtained. These were rolled out into thin foils at BNL and tested electrochemically for resistance to pitting corrosion in lithium bromide at 100°C. Table 1-4 lists the tested candidates in order, with the most resistant candidates first. Note that although 3003 aluminum had very little resistance to pitting at 100°C, it is by far the most resistant at 23°C, indicating that it may be a promising material for the evaporator.

Table 1-3
Selected Candidate Materials

PET (Polyethylene terephthalate)
Polyimide (Thermoset)
Polysulphone
Acrylonitrile methyl acrylate
Acetal Copolymer
Polymethyl Methacrylate-Stndrd (Acrylic)
ETFE (Ethylene tetrafluoroethylene)
PVF (Polyvinyl Fluoride; Tedlar)**
Polyethylene (High Molecular Weight
Olefin-modified styrene-acrylonitrile
SAN (Styrene-acrylonitrile)
ABS (Acrylonitrile-butadiene-styrene
ECTFE (Ethylene Chlorotrifluoroethylene; Halar) [Best tested to date]
Polyethylene (High Density) **
Polyethylene (Ultra High Molecular Weight)
PCTFE(Polychlorotrifluoroethylene; Aclar)**
PFA (Perfluoroalkyl; Teflon)
Polystyrene
PVC (Nonplasticized Extrusion)
Polypropylene - Oriented
PVC (Nonplasticized Solvent Cast)
PVDF (Polyvinylidene Fluoride)
Polyethylene (Linear Low Density)
Polyethylene (Medium Density)
Ethylene-Methyl Acrylate
FEP (Fluorinated Ethylene-Propylene; Teflon) **
PVDC (Polyvinylidene Fluoride; Saran)

** Polymers tested at BNL to date.

Table 1-4 also indicates the current prices of strips and foils. Again, if the materials can be made to work technically, there should be no trouble demonstrating strong economic advantages.

Key to making heat exchangers of thin metals are welding techniques, and Chapter 7 discusses our efforts in this regard. We have developed an ultrasonic welding facility which has been used very successfully on coppers in other applications. Consequently, we were able to weld the aluminum and Monel samples easily and reliably. Steels have historically been a problem, since the weld head is steel and tends to stick to the material. We had hoped that the use of a tungsten carbide coating on the weld head would overcome this difficulty, but to date our industrial partner's attempts to make the coating stick to the weld head under welding conditions have not borne fruit. Several new ideas are now being tried, and we expect to be able to report progress

Table 1-4 Corrosion Resistance and Cost of Metal Candidates	
Material	Cost of 20 mil Strip (\$/ft ²)
AL-6X SS*	3.27
AL-6XN SS	3.27
S31803 (Alloy 2205) SS	2.04
AL 29-4C SS	2.21
316 SS	1.29
SEA-CURE SS	2.40
M-400 Monel	high
304 SS	1.23
CDA-614 (Bronze)	moderate
3003 Aluminum (Best at 23oC)	0.65
1010 Carbon Steel (Not tested)	0.75
* SS = Stainless steel.	

soon, but to date we have not been able to weld the stainless steels reliably. The other options for thin materials, laser welding and electron beam welding, are undesirable because they are quite expensive.

Finally, in Chapter 8, we discuss the allocation of materials to the different heat exchangers on the basis of heat exchanger requirements and material properties. We also present a conceptual design of a double effect absorption chiller to demonstrate that the lack of rigidity which characterizes all thin materials does not preclude making almost the entire machine from them. This demonstration of feasibility constitutes the final necessary link, and we conclude that although very substantial amounts of research, testing, design and development remain to be done, the possibility of dramatically lowering costs in absorption chillers through the use of thin materials and innovative design is very real and that the returns may be very large.

2. STATUS OF THE ABSORPTION COOLING INDUSTRY AND ABSORPTION COOLING TECHNOLOGIES

2.1 Introduction

The absorption refrigeration cycle was invented by Ferdinand Carre, who obtained a U.S. Patent in 1860. His first machine used sulfuric acid and water, and in 1859 he built an ammonia-water system. One of the earliest uses of absorption cooling was by the Confederacy in 1863, during the Civil War, after their ice supply had been halted by the North. After the war, Daniel Holden made several improvements, including placing steam coils in the generator, which resulted in a number of units being installed in the south to produce ice.[2-1]

Commercialization of lithium-bromide and water systems occurred in the 1940's for building air conditioning systems with steam or hot water from oil or gas boilers used as the heat source. Because of falling electricity prices in the 1960's and the uncertainties in oil and gas during the 1970's, directly fired absorption units declined in popularity. However, the rise in energy prices has resulted in a new demand for waste heat driven applications.[2-2]

A simple single stage absorption chiller is shown in Figure 2-1. Cooling is provided to the load as refrigerant (pure water) evaporates off of a heat exchanger containing circulating cooling system water and known as the "evaporator". Evaporation continues because the heat exchanger known as the "absorber" maintains a vacuum around the evaporator by absorbing the water vapor into a film of dense brine, a mixture of water and lithium bromide salt. The water is absorbed because the vapor pressure of the dense brine is much lower than that of pure water; heat released as the water vapor condenses on the absorber is carried to a cooling tower by circulating water.

This process continually reduces dense brine to a weaker solution by mixing it with refrigerant (water), and a method is needed to separate out pure water (to serve again as refrigerant) and dense brine (to serve again as absorbent). This is accomplished by the generator and condenser, which constitute a still: weak solution is boiled in the generator, the vapor is condensed into refrigerant with the release of heat to the cooling tower through the condenser, and the dense brine solution exiting from the generator serves as the new absorbent. Appropriate pressure reduction devices and pumps keep all fluids flowing in the correct directions while a recuperator aids system efficiency.

A double effect absorption system is shown below in Figure 3-1. It uses a second generator as well as an additional recuperator to achieve a higher COP. Vapor from the first generator is condensed in the second generator, acting as its heat source. The condensed liquid is then injected into the condenser.

Figure 2-1. Single effect LiBr-H₂O absorption chiller cycle

2.2 Characteristics of Refrigerant-Absorbent Pairs

The ASHRAE 1977 Fundamentals [2-3] discusses nine desirable characteristics of the refrigerant-absorbent pair. These include: no solid phase formed during operation, high refrigerant volatility compared to absorbent volatility under generator conditions, high absorbent affinity for refrigerant during absorption, reasonable operating pressures, chemical stability, noncorrosiveness, nontoxicity, nonflammable, low viscosity and high refrigerant latent heat.

No fluid pair meets all of these criteria, but the most commonly used pair is LiBr and H₂O. Ammonia-water is also used and additional fluids are being actively explored in various research programs.

Ammonia is toxic, and this restricts its use in residences and commercial buildings. The Underwriters' Laboratory classifies ammonia in Group 2, which is defined in the ASHRAE Fundamentals [2-3] as "Gases or vapors which in concentrations of about 1/2 to 1 percent for durations of exposure of about 1/2 hour are lethal or produce serious injury." In addition, the operating pressures are somewhat high and the volatility ratio requires that water must be separated from the ammonia using additional components, adding to the cost.

The operating range for LiBr-H₂O is limited by both the freezing of water and the crystallization of LiBr. Although the solution pressures are low, and the viscosity can be high, proper design can minimize these problems.[2-3] Consequently, this is the most common refrigerant pair today.

2.3 Current LiBr-H₂O Technology

Since the generator and condenser are at the same pressure, they are confined in one pressure vessel for typical single effect LiBr-H₂O systems. Similarly, the absorber and evaporator are in one low pressure container. Usually refrigerant is sprayed or dripped over the evaporator, while the system water circulates within it. The absorber and condenser are generally water cooled using a cooling tower for heat rejection.

For part load operation, three main methods are used for capacity control: (1) reducing the flow rate of the various pumps, (2) reducing the generator temperature, and (3) increasing the condensing temperature. All these methods have different advantages and problems; further details may be found in reference 2-2.

At sufficiently high mass fractions of LiBr and low temperatures, crystallization of the solution occurs, forming a slush which may block the pipe and interrupt operation. This is most likely to happen at the exit of the recuperating heat exchanger after leaving the generator. Control systems are available to sense impending crystallization and shut the unit down, and various flow bypass systems are also in use to prevent crystallization.

2.4 General Operating and Maintenance Requirements

There are several operating and maintenance requirements common to all LiBr machines.[2-4] The system must be kept clean internally and have minimal leakage. Since the refrigerant operates below atmospheric pressure, noncondensable gases must be periodically purged, either automatically or manually. One common air leakage path is through the seals on the circulating pumps, and seal replacement is time consuming, typically requiring about three man days of labor. In many modern systems, however, these pumps are hermetically sealed, with magnetic drives, thus alleviating this problem. In addition, tube bundles may have to be removed, cleaned and serviced, although proper water treatment prolongs the service interval. Maximum and minimum temperatures to the unit must not be exceeded.

A study by The Hartford Steam Boiler Inspection and Insurance Company [2-5], analyzed tube failures in absorption machines occurring between 1971 and 1983. These tube failures usually resulted in chemical contamination of the LiBr, with the necessary fluid replacement being quite expensive. Most often corrosion was the cause, with stress corrosion cracking predominating. Ammonia or amines are a key ingredient in stress corrosion in these machines. They are present because lithium nitrate (or a comparable compound) is sometimes added as an inhibitor. The problem can be surmounted by using cupronickel rather than the phosphor-containing copper in common use or by using lithium chromate as alternative inhibitor, but the best solution is simply annual testing of the working fluid and adjustment of the pH and other chemical characteristics. Hydro-bromic acid or octal alcohol are other useful inhibitors. It is the opinion of practitioners that through such annual chemical checks, one can reduce internal corrosion to the point where all problems will arise on the system water side.[2-6]

Corrosion can also be detected at a very early stage using eddy current analysis. This is a well known (in the field), non-destructive testing technique involving threading a driven (radiating) coil and a pickup coil through the tubes of the heat exchanger. The induced current depends on the eddy currents which in turn depend on the conductivity, especially the surface conductivity, so the signal, displayed on an oscilloscope, indicates the condition of the tubes in a manner which is readily interpreted by a technician. Thus tubes which are corroding can be detected and replaced before they break, preventing uncontrolled loss of vacuum and contamination of the solution.

Thus up to date maintenance procedures can make contemporary absorption machines quite reliable; questions to be addressed include whether newer materials will permit some lessening of this vigilance, whether these advanced techniques will allow the use of thinner materials and whether the use of new materials will lessen the effectiveness of these procedures.

2.5 Advanced Concepts in Absorption Machinery

A number of research projects in the U.S. are exploring advanced absorption machines.[2-7] One of these, the Battelle Dual-Cycle Absorption

Heat Pump, sponsored by the Gas Research Institute, incorporates novel heat exchanger concepts into the design. Their machine links a LiBr-H₂O cycle to a NH₃-H₂O cycle with the LiBr-H₂O condenser supplying the heat to the NH₃ generator. Four concentric vertical tubes with enhanced surfaces are used for the generators and condensers. Flue gas flows up the center tube, while LiBr-H₂O solution flows down the tube outer diameter, which is covered with spiral fins. The water vapor that is generated condenses on the inner surface of a surrounding concentric tube, and the latent heat is transferred through the tube to the ammonia-water solution on the outer diameter, driving off ammonia vapor. The outer diameter of this tube is also covered with spiral fins. The ammonia vapor then condenses on the inner diameter of another surrounding tube, giving up its latent heat to cooling water flowing in the annulus between the third and fourth tubes. These concentric tubes can be manufactured and configured in modular form giving the ability to heat the tubes individually and resulting in a capacity control technique. The use of two fluid pairs allows some optimization. Other fluids are being explored.

The U.S. Department of Energy (DOE), through Oak Ridge National Laboratory (ORNL) is sponsoring absorption heat pump research at Carrier, Trane and Phillips Engineering.[2-8] Each contractor is selecting and evaluating an advanced cycle using a well characterized fluid pair, and is identifying and evaluating an advanced cycle using ternary fluids. Target COP's for near term are 0.7 for cooling (95°F outdoor air) and 1.6 for heating (47°F outdoor air), while far term COP's are 1.6 for cooling and 2.2 for heating. Heating COP's assume 85% waste heat utilization, while parasitics are not considered in any of the COP's. Additional DOE projects are in base technology research, such as developing analytical tools for cycle analysis, and developing binary fluid models. The remaining work is in absorption fluids: sensitivity analysis of cycle efficiency calculations (due to uncertainty in fluid properties, for instance), and surveying and compiling fluids property data.

2.6 Current Machines

In this section, the characteristics of commercially available LiBr machines from 6 manufacturers are presented. All units use water cooled condensers with cooling towers for heat rejection.

Carrier makes 15 single effect units ranging in capacity from 70 to 815 tons. Low pressure steam or hot water is used as the heat source. Available capacities are 100, 116, 132, 174, 201, 233, 270, 312, 349, 397, 455, 490, 522, 562 and 618 tons.

Hitachi makes double effect chillers of 100-1500 tons using gas, propane, oil, steam or waste heat for the heat source. COP's of 1.02, 1.14, 1.2 are reported for direct firing, heat recovery and steam sources, respectively. Available capacities are 100, 125, 140, 150, 170, 200, 250, 270, 320, 345, 400, 430, 500, 600, 700, 1000, 1250 and 1500 tons.

Sanyo's machines are marketed in the U.S. through an agreement with Bohn. They have double effect units in a 10-120 ton modular series and in a

100-950 ton large-scale series, with typical COP's of 0.95. These are all combination chillers/hot water heaters which are directly fired with natural gas. Capacities of units in the modular series are 10, 20, 30 and 40 tons. These may be combined to give systems with a range of capacities up to $3 \times 40 = 120$ tons. The large-scale series includes units of 100, 120, 150, 180, 210, 240, 280, 320, 360, 400, 450, 500, 550, 620, 690, 770, 800 and 950 tons.

Trane makes 23 single effect units ranging from 101 to 1660 tons, which use hot water or steam as the heat source. In addition, they make 8 double effect units with capacities from 385 to 1060 tons. Single effect machines are available with capacities of 101, 112, 129, 148, 174, 200, 228, 256, 294, 354, 385, 420, 465, 520, 590, 665, 750, 852, 955, 1125 and 1250 tons. Double effect machines include capacities of 385, 465, 527, 656, 750, 852, 935 and 1060 tons.

Yazaki makes single and double effect chiller/hot water heaters that are gas and/or water fired. Twenty and 30 ton modules can be combined to give capacities up to 150 tons. Models are listed in Table 2-1.

York has 21 single effect units ranging from 120 tons to 1377 tons. They are hot water or steam fired. Available capacities are 120, 155, 172, 205, 235, 273, 311, 334, 363, 410, 446, 518, 565, 617, 704, 794, 908, 960, 1148, 1235 and 1377 tons.

Cooling towers are a significant part of the bulk and the expense of an absorption chiller system. Sizes for representative wet cooling towers are listed in Table 2-2, with the data supplied by Yazaki.

Table 2-1
Yazaki Absorption Chiller Systems

Model No.	Capacity (tons)	COP	Description
CH-SD20 & 30	20 - 30 mod (150 max)	0.75 0.95	Single effect hot water heat source or double effect gas fired
CH-D7.5 to CH-D30	7.5 - 30 mod (150 max)	0.95	Double effect gas fired
WFC- 1.3 to WFC-10	1.3 - 10 mod (50 max)		Single effect hot water heat source
CH-L100	1120	1.0	Double effect gas fired
CH-G30 to 50	30 - 50 mod	0.95	Double effect gas fired
CH-E7.5P	7.5 mod (150 max)	0.92	Double effect gas fired

Table 2-2
Cooling Tower Dimensions

Capacity (tons)	Length (in.)	Width (in.)	Height (in.)
7.5	40.9	40.9	71.8
10	47.6	47.6	72.1
15	56.7	56.7	81.2
20	64.6	64.6	87.5
25	72.4	72.4	98.7
30	78.7	78.7	100.2
40	123.6	45.7	91.1
50	139.0	72.4	106.3
60	151.6	78.7	105.5
80	204.3	72.4	106.3
90	223.2	78.7	105.5
100	269.7	72.4	106.3
125	294.9	78.7	105.5
150	201.6	136.2	111.1
175	220.5	136.2	110.3

2.7 Costs of Absorption Machinery

Equipment costs were obtained from four manufacturers for single and double effect units for capacities ranging from 25 tons to 1000 tons. Both direct fired and steam source units were considered. Figure 2-2 shows that the cost of direct fired double effect machines is very close to several of the double effect steam source machines. This is somewhat surprising, since double effect steam driven machines require high pressure steam which demands expensive piping for safe containment. We were not able to get detailed cost breakdowns of the various machines however (except for the prototype direct fired units in Chapter 3), so we were unable to determine the factors tending to equilibrate the costs of the units. It is always possible that differences in markups and profits were significant.

The curves in Figure 2-2 refer to double and single effect steam-fired machines; they were taken from the equations of reference [2-4], corrected to mid 1984 dollars. They indicate that the constant dollar price has not changed significantly since 1977, when the curves were published. These curves also indicate that the cost is proportional to capacity to nearly the $2/3$ power, which is the approximate way that heat exchanger prices scale. This is not surprising, of course, since the primary components in absorption units are the heat exchangers and their cost will dominate the overall cost.

A detailed review of absorption machinery, including costs, operating principles, maintenance and adaptability to off-design conditions is included in reference [2-9].

Figure 2-2. Current retail equipment costs of absorption chillers. Key: "+" → direct fired double effect, "O" → steam fired double effect, "X" → steam fired single effect, and solid lines are single and double effect steam fired units (see text).

3. CURRENT CHARACTERISTICS OF ABSORPTION COOLING EQUIPMENT

3.1 Introduction

To assess the suitability of various new materials for use in the heat exchangers of absorption chillers, a statement of the technical requirements to be faced by these materials is needed. These requirements include performance characteristics, such as heat transfer capability or high wettability, ability to withstand the stresses of the system environment, such as tensile strength and corrosion resistance and finally economic criteria, such as cost per unit area or cost per unit heat transfer capability. To serve as a basis for establishing these requirements quantitatively, we have defined four systems which we take to represent the best that could be produced using current industrial practice.

Our suggested improved heat exchangers must then appear likely to exceed these components in some significant performance parameter. For each system we present two detailed matrices. The first gives the functional requirements that heat exchangers must meet in that system, and the second summarizes current design practices in a way which allows us to derive the relative costs of the different components. In this way we can focus our efforts on those components likely to afford the largest cost reductions. In the following sections we delineate the system design assumptions, present and describe the four systems and summarize the results. In Appendix 1, we discuss the design procedures used to derive the system characteristics presented in the functional characteristics matrices.

3.2 System Design Assumptions

The systems were designed in accordance with current industry standards. For example, the chilled water enters at 54°F and leaves at 44°F, with a resultant flow rate of 2.4 gpm per ton. Cooling water entered the machines at 85°F and 3.6 gpm per ton; we did not calculate cooling tower dimensions or requirements, but assumed that the tower was capable of reducing the resultant exit water temperature back down to 85°F, and that make up water was provided as required. Lithium Bromide and water were chosen as the working fluid on the basis of safety, potential efficiency and widespread experience and acceptance. Standard tables were used for the properties of LiBr-water solutions.[3-1]

To achieve efficiencies competitive with compression equipment, a double effect system was necessary. Since a double effect system requires thermal energy at about 300°F, a choice must then be made between a directly fired system directly incorporating a gas burner and a high pressure steam system using a separate boiler. Safety requirements for high pressure steam include painstaking construction and the presence of an operating engineer, so we chose directly fired systems for simplicity and applicability to all size ranges, rather than only to systems measured in hundreds of tons. By designing around high concentrations of LiBr we maintained sub-atmospheric operation under all conditions, minimizing code problems and the stresses a system

subject to variations between sub- and super-atmospheric internal pressures would experience.

Gas is assumed supplied in suitable pressures for the burner head in each installation. The overall efficiency of the first effect generator is taken as 75%, including both flue and jacket losses, for all sizes. In these base-line systems, in which a co-flowing froth is created in the horizontal annular spaces of the boiler, it would be difficult to exceed this efficiency by much. It may be possible to improve on this somewhat by considering counter-flow designs, but bringing the hottest part of the flame into contact with the froth, rather than with a completely liquid part of the solution, increases the danger of hot spots and burnout in the heat exchanger.

A schematic drawing of a double effect chiller system is shown in Figure 3-1. Q (source) is of course the gas flame, Q (load) the system water and Q (sink) the water carrying thermal energy off to the cooling tower. Some form of expansion valve, which may in fact be a manometer arm, must reduce the (relatively) high pressure of the refrigerant and strong solution before dripping or spraying them onto the evaporator and absorber, respectively. Other elements are keyed to lines in the tables which follow.

It should be emphasized that the designs which follow are not hypothetical. The designs for the 25, 100 and 400 ton units are based on comparable units in a 15 ton size, of which some 200 were built and sold by one of the authors (J. Murray); some of these had measured gross COP's of 1.1 and net COP's of 1.4. Some six prototypes at 25 tons were also constructed and tested at that time. We are therefore confident that these designs represent a reasonable representation of the best to be realistically expected from current engineering practice.

3.3 3 Ton Evaporatively Cooled Absorption Chiller

This small residential unit is included for completeness, although it is not used extensively in the analysis which follows. The residential market is large and important, but there are economies of scale which indicate that absorption units can more easily fight their way back to popularity in larger sizes first, and focus on residential markets at a later point.

This unit differs from the larger units in having an integral, evaporatively cooled condenser and absorber. It is quite similar to a solar design built and tested for DOE [3-2, 3-3] with the solar components omitted. The primary problem with the prototype was the potential for corrosion of the surfaces of the integral condenser and absorber, which required that the whole system be dip galvanized; this was a very expensive process, and is as yet of uncertain reliability in production, so more corrosion resistant heat exchange surfaces would give similar units a major push towards commercial success, and they are perhaps worth examining for this reason alone.

Figure 3-1. Double effect LiBr-H₂O absorption chiller cycle.

Table 3-1 presents the operating conditions of the unit, which constitute the functional requirements for new heat exchangers and heat exchange materials. In the first column the various fluid flows in each component are listed; they may be correlated with the flows indicated in Figure 3-1. The second column gives the total heat being transferred through each heat exchanger; the net COP may be calculated by dividing the heat transfer of the first effect generator by the heat transfer of the evaporator, giving 1.33; the gross COP of 1.0 is obtained by dividing the evaporator heat transfer by the 36 mBtu/hour gas input from footnote 4 of Table 3-1.

The next column shows design pressure for the various fluids, indicating the differential pressures that the heat exchange surfaces must be able to withstand. "Atmos." of course means atmospheric pressure, while "system" refers to the pressure of the supply or circulating water. Since the entire chamber will draw down to a few millimeters when it is shut off, all surfaces must be able to withstand pressures of one atmosphere, plus a margin of safety, plus (for the evaporator) net system pressure.

The remaining eight columns indicate the condition of each fluid as it enters and leaves each heat exchanger. Of course refrigerant and water have no salt concentration, but the concentration of the brine solution varies as shown. The differences do not appear to be significant with respect to corrosion, but temperature differences certainly are; they will play a key role in the selection of candidate materials for each component. Also, examination of the entries in the two "State" columns will indicate where phase changes are taking place, and this, too, will be significant later.

Table 3-2 details the characteristics of heat exchangers and selected other components typically used to meet these requirements. Current materials and geometry are listed in the second and third columns, pressure drops under current design criteria in the next. The assumed surface area, pounds of material and material costs at mid 1984 prices are shown next. These heat exchange areas are the result of current design assumptions and criteria, and variations from this baseline will be considered later.

In the next to the last column the fabrication cost is shown, including both direct labor and labor burden, or overhead, for each component. Labor burden, calculated from recent experience at 400% of direct labor, covers the capital cost of the production plant, administration, electricity and so on. Although clever design may cut the direct labor required, we must find ways to do it which do not impose a significant capital cost burden on the factory if we are to lower the entries in the last column, a tabulation of component by component total factory costs.

Examination of these cost figures indicates where material or component cost reductions can have the greatest effect, and this will be considered in Chapter 4. For now we note only that the distributor's and dealer's markups form a significant part of the final cost to the customer of \$1600 to \$2000 per ton, and that costs seem to be spread among most of the components, with no single item dominating the economics.

3.4 The 25, 100 and 400 Ton Absorption Chiller Systems

The 25 ton and larger systems are more traditional with respect to heat removal, relying on separate, presumably wet cooling towers, which are not analyzed in this chapter. The functional requirements of the 25 ton system are shown in Table 3-3. The condenser and absorber must now be able to withstand "cooling system" pressures rather than just atmospheric pressures, which was the case for the three ton unit. In most other respects the system is similar to the three ton unit, although the larger size permits higher temperatures and the insulation from ambient conditions permits higher brine concentrations without fear of crystallization. The serial flow of cooling water from absorber to condenser means that the latter runs hotter than it did for the small unit.

Table 3-4 displays the typical characteristics of the 25 ton units. The coated mild steel absorber and condenser have been replaced by copper tubes and we now have two separate cylindrical vessels, the smaller containing the second effect generator and condenser and the larger containing the evaporator and absorber. The first effect generator is in a third, smaller cylindrical vessel.

Tables 3-5 through 3-8 display the functional requirements and typical characteristics of the 100 and 400 ton units in the same manner. In many respects the machines are simply larger scale models of each other, but there are differences. In the 400 ton machines, for instance, water is sprayed onto the evaporator, rather than dripped, leading to higher parasitic losses from the pumps. Other differences do not show up in the tables: larger systems, with a smaller surface to volume ratio, will experience lower thermal losses and greater temperature stability than the smaller systems do.

3.5 Cost Structure of Absorption Chillers

The cost figures of Tables 3-2, 3-4, 3-6 and 3-8 indicate a cost structure for absorption chillers which is summarized in Figure 3-2 for the three larger systems. We see that the heat exchanger materials cost roughly the same as the materials for the balance of the system, that the total materials costs constitute about one half of the total factory costs, and that current distributor's and dealer's markups of 60 to 100% add significantly to the customer's cost. How these costs will scale if factory costs are reduced dramatically will be very significant in the potential commercial success of new systems.

Figure 3-2 also shows the cost of current systems, taken from Figure 2-3, for the 100 and 400 ton systems. The excess of current costs above our estimates is labelled "current cost increment". While current costs are only slightly in excess of our estimates for the 400 ton system, current costs for the 100 ton unit are approximately twice our estimates. Not having access to cost breakdowns for the units currently available, we have not been able to determine the cause for this disagreement. Indeed, the causes could differ from company to company - Japanese firms with recent research to amortise may

Figure 3-2. Cost structure of absorption chillers.

have high overhead as a result, while the shrinking market could lead to very large factory overhead or distribution markups in the case of the American companies. In any event, the effect on our cost goal analysis in Chapter 4 will not be severe, as we will use the 400 ton estimates and will use them only to determine the fraction of system costs to be allocated to a particular heat exchanger or other component.

Table 3-1. Typical Functional Requirements for 3 Ton Evaporatively Cooled Absorption Chiller Heat Exchangers

Component	Heat Transfer (mBtu/hr)	Design Pressure (mm)	Entering Conditions				Leaving Conditions			
			Flow Rate ¹	Temp. (°F)	Concen- tration ²	State ³	Flow Rate ¹	Temp. (°F)	Concen- tration ²	State ³
1st Effect Generator	27 ⁴									
Solution		485	6.75	247	0.52	l	6.42	277	0.55	l
Refrigerant		-	-	-	-	-	0.32	277	-	v
2nd Effect Generator	13									
Solution		47 ⁵	6.42	163	0.55	l	6.16	177	0.57	l
Refrigerant (HP)		-	0.33	270	-	v	0.32	190	-	l
Refrigerant (LP)		-	-	-	-	-	0.27	172 ⁸	-	v
Condenser	20									
Refrigerant		47 ⁵	0.59	172 ⁷	-	v	0.59	98	-	l
Water		atmos.	5.0	85	-	l	5.0	93	-	l, v ⁸
Evaporator	36									
Refrigerant		6.0 ⁵	0.59	98 ⁷	-	l	0	41	-	v ⁶
Water (load)		building system	7.2	55	-	l	7.2	45	-	l
Absorber	43									
Solution		6.0 ⁵	6.16	108	0.57	l	6.75	94	0.52	l
Refrigerant		6.0	0.59	41	-	v	0	-	-	-
Water		atmos.	5.0	85	-	l	5.0	93	-	l, v ⁹
Low Temp Recuperator	12									
St Solution		10	6.16	172	0.57	l	6.16	108	0.57	l
Wk Solution		600	6.75	94	0.52	l	6.75	151	0.52	l
High Temp Recuperator	20.5									
Intermediate Solution		485	6.42	277	0.55	l	6.42	163	0.55	l
Wk Solution		550	6.75	151	0.52	l	6.75	247	0.52	l

1) Gpm for water, lbs/min for brines and refrigerant.

2) Brine concentration, pounds salt per pound brine.

3) 'v' is vapor, 'l' is liquid.

4) 36 mBtu/hr gas input = 27,000 to solution
+ 1100 jacket loss + 7900 flue loss.

5) Vapor pressure.

6) 5% is flash loss, 95% to load.

7) Superheated.

8) 20 lb/hr vaporized.

9) 43 lb/hr vaporized.

Table 3-2. Typical Characteristics of 3 Ton Absorption Chiller Heat Exchangers
Evaporatively Cooled Unit

Component	Material ¹	Geometry	Pressure Drop (psi)	Parasitic Power (hp)	Surface Area (ft ²)	Quantity (lbs)	Material Cost (\$)	Fabrication Cost Labor & Burden (\$)	Total Cost (\$)
1st Effect Generator	SS	Shell	0.5	--	5.5	15	80	75	155
2nd Effect Generator	MS	Tube	10/0rif	--	12	30	15	75	90
Condenser	MS/C	Shell/C	0.1	--	55	110	30	120	150
Evaporator	COP	Tube	3	1/4	15	30	65	25	90
Absorber	MS/C	Shell/C	0.1	--	90	175	65	175	240
Hi Temp Recuperator	MS	Flatplate	2.0	--	33	60	25	100	125
Lo Temp Recuperator	MS	Flatplate	2.0	--	33	60	25	100	125
Cooling Fan/Mo.	--	--	--	1/3	--	--	35	--	35
Generator Vessel	MS	--	--	--	--	--	10	50	60
Absorber Vessel	N/A	--	--	--	--	--	--	--	--
Sol/H ₂ O Pumps	SS/MS	Semi-Hermetic	--	1/4	--	--	300	30	330
Coating of Absorber and Condenser Mild Steel Outer Surfaces							200	200	400
Leak Tests & LiBr Charge							150	500	650
Wiring and Controls and Operation Test							75	50	125
Air Shield, Cabinet, Frame, and Detail Piping of Components							25	400	425
							<u>\$1100</u>	<u>\$1900</u>	<u>\$3000</u>
Estimated Cost to Customer; includes Distributor's markup									\$5200 - 6000

¹SS = Stainless Steel, MS = Mild Steel, N/A = Not Applicable, COP = Copper, C = Coated.

Table 3-3. Typical Functional Requirements for 25 Ton Absorption Chiller Heat Exchangers

Component	Heat Transfer (mBtu/hr)	Design Pressure (mm)	Entering Conditions				Leaving Conditions			
			Flow Rate ¹	Temp. (°F)	Concen- tration ²	State ³	Flow Rate ¹	Temp. (°F)	Concen- tration ²	State ³
1st Effect Generator	225 ⁴									
Solution		725 ⁵	67.6	277	0.57	l	64.9	305	0.60	l
Refrigerant		-	-	-	-	-	2.7	305	-	v
2nd Effect Generator	170									
Solution		60 ⁵	64.9	177	0.60	l	62.7	194	0.62	l
Refrigerant (HP)		725	2.7	300	-	v	2.7	240 ⁶	-	l
Refrigerant (LP)		0	-	-	-	-	2.2	186 ⁷	-	v
Condenser	165									
Refrigerant		60 ⁵	4.9	186 ⁷	-	v	4.9	101	-	l
Water		cooling system	90	94	-	l	90	98	-	l
Evaporator	300									
Refrigerant		6.5 ⁵	4.9	101 ⁷	-	l	4.9	41	-	v ⁸
Water (load)		building system	60	55	-	l	60	45	-	l
Absorber	360									
Solution		6.0 ⁵	62.7	110	0.62	l	67.6	97	0.57	l
Refrigerant		6.0	4.9	41	-	v	90	abs'd.	-	-
Water		cooling system	90	85	-	l	-	94	-	l
Low Temp Recuperator	116									
St Solution		10	62.7	194	0.62	l	62.7	110	0.62	l
Wk Solution		850	67.6	97	0.57	l	67.6	167	0.57	l
High Temp Recuperator	185									
Intermediate Solution		725	64.9	298 ⁹	0.60	l	64.9	177	0.60	l
Wk Solution		800	67.6	167	0.57	l	67.6	277	0.57	l

1) Gpm for water, pounds per min for brines and refrigerant.

2) Brine concentration, pounds salt per pound brine.

3) 'v' is vapor, 'l' is liquid.

4) 300 mBtu/hr gas input = 225 to solution + 9 jacket loss + 66 flue loss.

5) Vapor pressure.

6) Sub-cooled.

7) Superheated.

8) Refr. Flow is combined flash loss and load vaporized.

9) 7° drop due to flash in separator.

Table 3-4. Typical Characteristics of 25 Ton Absorption Chiller Heat Exchangers

Component	Material ¹	Geometry	Pressure Drop (psi)	Parasitic Power (hp)	Surface Area (ft ²)	Quantity (lbs)	Material Cost (\$)	Fabrication Cost Labor & Burden (\$)	Total Cost (\$)
1st Effect Generator	SS	Shell	0.5	--	50	75	650	300	950
2nd Effect Generator	MS	Tubes or Flatplate	10/0rif	--	65	120	175	350	525
Condenser	COP	Tubes	3 ²	See Abs'r	75	130	290	190	480
Evaporator	COP	Tubes	4 ²	1/2-3/4	100	200	375	275	650
Absorber	COP	Tubes	4 ²	3/4 - 1	150	300	560	450	1010
Hi Temp Recuperator	MS	Flatplate	2.0	--	100	180	75	350	425
Lo Temp Recuperator	MS	Flatplate	2.0	--	100	180	75	350	425
Generator Vessel	MS	Round	--	--	--	200	60	100	160
Absorber Vessel	MS	Round	--	--	--	300	90	150	240
Pumps, Sol'n & Ref	SS/MS	Semi-Hermetic	--	1/4+1/10	--	--	750	--	750
Controls, wiring, etc.						--	800	300	1100
Purge, piping, structural						500	500	700	1200
LiBr charge, leak test, final assembly:							600	1100	1700
							<u>\$5000</u>	<u>\$4615</u>	<u>\$9615</u>
Estimated Cost to Customer; includes Distributor's markup								\$16,000 - 20,000	

¹SS = Stainless Steel, MS = Mild Steel, COP = Copper.²Water Side.

Table 3-5. Typical Functional Requirements for 100 Ton Absorption Chiller Heat Exchangers

Component	Heat Transfer (mBtu/hr)	Design Pressure (mm)	Entering Conditions				Leaving Conditions			
			Flow Rate ¹	Temp. (°F)	Concen- tration ²	State ³	Flow Rate ¹	Temp. (°F)	Concen- tration ²	State ³
1st Effect Generator	900 ⁴									
Solution		725 ⁵	270	277	0.57	l	259	305	0.60	l
Refrigerant		-	-	-	-	-	11	305	-	v
2nd Effect Generator	680									
Solution		60 ⁵	259	177	0.60	l	250	194	0.62	l
Refrigerant (HP)		725	11	300	-	v	11	210 ⁶	-	l
Refrigerant (LP)		-	-	-	-	-	9	186 ⁷	-	v
Condenser	660									
Refrigerant		60 ⁵	20	186 ⁷	-	v	20	101	-	l
Water		cooling system	360	93	-	l	360	97	-	l
Evaporator	1200									
Refrigerant		6.5 ⁵	20	101	-	l	0	41	-	v ⁸
Water (load)		building system	240	55	-	l	240	45	-	l
Absorber	1440									
Solution		6.0 ⁵	250	110	0.62	l	270	97	0.57	l
Refrigerant		6.0	20	41	-	v	-	-	-	-
Water		cooling system	360	85	-	l	360	93	-	l
Low Temp Recuperator	464									
St Solution		10	250	194	0.62	l	250	110	0.62	l
Wk Solution		850	270	97	0.57	l	270	167	0.57	l
High Temp Recuperator	740									
Intermediate Solution		725	260	298 ⁹	0.60	l	260	177	0.60	l
Wk Solution		800	270	167	0.57	l	270	277	0.57	l

1) Gpm for water, pounds per min for brines and refrigerant.

2) Brine concentration, pounds salt per pound brine.

3) 'v' is vapor, 'l' is liquid.

4) 1200 Btu/hr gas input = 900 to solution + 264 flue loss + 36 jacket loss.

5) Vapor pressure.

6) Subcooled.

7) Superheated.

8) Refrigerant flow is combined flash loss and load vaporized.

9) Down from 305°F due to flash in separator.

Table 3-6. Typical Characteristics of 100 Ton Absorption Chiller Heat Exchangers

Component	Material ¹	Geometry	Pressure Drop (psi)	Parasitic Power (hp)	Surface Area (ft ²)	Quantity ³ (lbs)	Material Cost (\$)	Fabrication Cost Labor & Burden (\$)	Total Cost (\$)
1st Effect Generator	SS	Shells	1.0	--	200	300	2600	300	2900
2nd Effect Generator	MS	Tubes	10/0rif	--	260	480	700	750	1450
Condenser	COP	Tubes	3 ²	See Abs'r	300	525	1200	250	1450
Evaporator	COP	Tubes	4 ²	2	425	850	1600	500	2100
Absorber	COP	Tubes	4 ²	3	650	1300	2500	700	3200
Hi Temp Recuperator	MS	Tubes	2.0	--	400	650	260	600	860
Lo Temp Recuperator	MS	Tubes	2.0	--	400	650	260	600	860
Generator Vessel	MS	Round	--	--	--	400 shell	120	150	270
Absorber Vessel	MS	Round	--	--	--	600 shell	180	250	430
Sol/Refr Pumps	SS/MS	Semi-Hermetic	20 lift	1 + 1/4	--	--	1500	--	1500
Controls, wiring, etc.							1200	350	1550
Purge, piping, structural							1500	1200	2700
LiBr charge and inhibitors							1200	--	1200
Leak testing: Individual components and complete unit, sub-tests through mass spectrometer.							--	1800	1800
							<u>\$14,820</u>	<u>\$7450</u>	<u>\$22,270</u>
Estimated Cost to Customer; includes Distributor's markup								\$38,500 - 45,000	
¹ SS = Stainless Steel, MS = Mild Steel, COP = Copper.				² Water Side.		³ Material in heat exchanger surface.			

Table 3-7. Typical Functional Requirements for 400 Ton Absorption Chiller Heat Exchangers

Component	Heat Transfer (mBtu/hr)	Design Pressure (mm)	Entering Conditions				Leaving Conditions			
			Flow Rate ¹	Temp. (°F)	Concen- tration ²	State ³	Flow Rate ¹	Temp. (°F)	Concen- tration ²	State ³
1st Effect Generator	3600 ⁴									
Solution		725 ⁵	1082	277	0.57	l	1039	305	0.60	l
Refrigerant		-	-	-	-	l	43	305	-	v
2nd Effect Generator	2720									
Solution		60 ⁵	1039	177	0.60	l	1003	194	0.62	l
Refrigerant (HP)		725	43	300	-	v	43	210 ⁶	-	l
Refrigerant (LP)		-	-	-	-	-	36	186 ⁷	-	v
Condenser	2640									
Refrigerant		60 ⁵		186	-	v	79	101	-	l
Water - Tower Side		cooling system	1440	93	-	l	1440	97	-	l
Evaporator	4800									
Refrigerant		6.5 ⁵	79	101	-	l	79	41	-	v
Water (load)		building system	960	54	-	l	960	44	-	l
Absorber	5800									
Solution		6.0 ⁵	1003	110	0.62	l	1082	-	0.57	l
Refrigerant		6.0 ⁵	79	41	-	v	-	Abs'd.	-	-
Water		cooling system	1440	85	-	l	1440	93	-	l
Low Temp Recuperator	1860									
St Solution		10	1003	194	0.62	l	1003	110	0.62	l
Wk Solution		850	1082	97	0.57	l	1082	167	0.57	l
High Temp Recuperator	2960									
Intermediate Solution		725	1039	298	0.60	l	1039	177	0.60	l
Wk Solution		800	1082	167	0.57	l	1082	277	0.57	l

1) Gpm for water, pounds per min for brines and refrigerant.

2) Brine concentration, pounds salt per pound brine.

3) 'v' is vapor, 'l' is liquid.

4) 4800 mBtu/hr gas input = 3600 to solution + 1056 flue loss + 144 jacket loss.

5) Vapor pressure.

6) Subcooled.

7) Superheated.

Table 3-8. Typical Characteristics of 400 Ton Absorption Chiller Heat Exchangers - Double Effect LiBr Unit

Component	Material ¹	Geometry	Pressure Drop (psi)	Parasitic Power (hp)	Surface Area (ft ²)	Quantity ³ (lbs)	Material Cost (\$)	Fabrication Cost Labor & Burden (\$)	Total Cost (\$)
1st Effect Generator	SS	Shells	2	--	600	900	5500	500	6000
2nd Effect Generator	MS	19 fin/in. 10/0rif		--	1000	1500	2000	1000	3000
Condenser	COP	Tubes	5 ²	5	700	1200	3000	500	3500
Evaporator	COP	Tubes	15 ²	20	1450	2900	4000	1000	5000
Absorber	COP	Tubes	5 ²	10	1400	2800	4000	1000	5000
Hi Temp Recuperator	MS	Tubes	2.0	--	600	1000	750	1000	1750
Lo Temp Recuperator	MS	Tubes	2.0	--	600	1000	750	1000	1750
Generator Vessel	MS	Round	--	--	Includes water headers for condenser		500	500	1000
Absorber Vessel	MS	Round	--	--	Includes water headers for evap and abs.		1750	750	2500
Sol/Refr Pumps	SS/MS	Semi-Hermetic	20 lift	10	--	--	6000	--	6000
Controls, wiring, etc.							2500	600	3100
Purge, piping, structural							3000	3000	6000
LiBr charge and inhibitors							4000	500	4500
Leak testing labor							--	2500	2500
Startup services including one week travel							--	3400	3400
							<u>\$37,750</u>	<u>\$17,250</u>	<u>\$55,000</u>
Estimated Cost to Customer; includes Distributor's markup								\$95,000 - \$110,000	

¹SS = Stainless Steel, MS = Mild Steel, COP = Copper.²Water Side.³Material in heat exchanger surface.

4. PERFORMANCE REQUIREMENTS AND ESTIMATED COST GOALS FOR COMPETITIVE ABSORPTION CHILLERS

4.1 Introduction

Current absorption chillers have difficulty competing with electric chillers in today's markets. To estimate the conditions under which absorption equipment could compete more successfully with electric equipment requires detailed analysis. In this section we compare the life cycle costs associated with owning and operating absorption and electric equipment. Assuming that the electric equipment available today will undergo only evolutionary improvements, we use this comparison to estimate the minimum efficiency and maximum capital cost an absorption unit must have to ensure that its total life cycle costs will be less than or equal to those associated with comparable electric equipment.

This maximum allowable capital cost will then serve as our estimate of the "breakeven cost", the target to which the cost of chillers must be reduced. All economic estimates of this sort are subject to large uncertainties, due in part to the uncertain future prices of gas and electricity or to varying usage patterns. Therefore, such estimates must be used as guides rather than rigid conclusions, but we will see in Section 4.5 that within these constraints, our estimates of breakeven costs are in reasonable agreement with estimates developed by GRI in a somewhat different context.

We then convert these breakeven capital costs to a breakeven factory cost by subtracting out installer's and dealer's markups. Using factors characteristic of the fabrication of today's absorption chillers (as developed in Chapter 3) we will then distribute these breakeven production costs among materials, labor and labor burden components. We will thus arrive at a maximum, or breakeven, cost per square foot for heat exchanger materials and will use this as a basis for selecting candidate materials and establishing an upper limit on the thickness of the heat exchanger walls. Alterations in the detailed breakdown of production costs used here might result from new fabrication techniques or other design improvements; as these are identified in later phases of our work we will assess their impact on these estimates.

4.2 The Four Model Systems

Our results should be as general as possible, covering, in particular, a wide range of system sizes. However, because cooling systems of different sizes will normally involve quite different designs, the assumptions and results do not scale well. We will therefore look at four particular sizes; 3, 25, 100 and 400 tons, as has been discussed in Chapter 3. Some results will be expressed in units such as \$/ton, but they must be understood to refer only to modest adjustments in system capacity, of the order of ± 20 or 30%.

Systems are more easily and rigorously compared when they have many elements in common, and we have employed as many common elements in both electric and gas-fired systems as possible without distorting engineering

practice. The costs of these common elements can then be ignored, as they will weigh equally on both electric and gas-fired systems. Building HVAC elements such as air ducts or chilled water circulation are assumed the same for both systems. Heating systems are assumed the same and ignored; the economics may work out quite differently if the electric chiller can double as a heat pump to stand in place of a gas furnace or even a gas-fired heat pump, but this question is beyond the scope of the current study. On the other hand, cooling towers will be substantially larger for absorption systems, so they must be specifically accounted for.

The 3 ton system, suitable for a 2000 square foot single family house or similar load, will be taken as a stationary, central unit, fitted to provide cool air to a single zone duct system. The absorption system will have an integral wet cooling tower, as described in Chapter 3, and will be a self contained unit with a first effect generator directly fired by gas. Its cost will include an evaporator water to duct air heat exchanger and plumbing to provide make up water to the cooling tower. This absorption chiller will be compared with a 3 ton reciprocating electric chiller, designed with an integral ambient air-cooled condenser and a direct duct air-heated evaporator. This capability of the electric chiller to produce cool air directly is what forces us to include an evaporator water-to-duct air heat exchanger in the absorption unit. For systems of this size, chilled water circulation in the building is not normally practiced. One consequence of these assumptions is that system costs per ton for the 3 ton unit are not at all comparable with those for the larger systems.

The absorption and compression twenty five ton systems are more directly comparable, since both provide chilled water to a central distribution system. The absorption system again has a directly gas fired first effect generator. The absorption chiller has a separate wet cooling tower, while the reciprocating electric chiller, in keeping with current practice for units of this size, again has an air cooled condenser, either integral or plumbed in at some modest distance from the central unit.

Finally, the 100 and 400 ton units are functionally identical, both providing chilled water to a central distribution system and both using wet cooling towers located at some distance. Again, the first effect generator will be directly fired in an integral boiler. The electric chillers are taken to be centrifugal, giving substantially higher COP's than were used for the smaller systems. Hermetic design is almost universal among machines of 100 tons capacity or less and is assumed here. Open designs (with the motor mounted externally and cooled independently) are more efficient, but are not common below 1000 tons. They are available, however, so we use one as our model in the 400 ton example.

The characteristics of the various systems are summarized in Table 4-1. The cooling season varies widely in length across the US, leading to uncertainty in the hours per year of operation. We have assumed the systems will be sized to have load factors of 60 to 80% during the cooling season and that the two smaller systems will be shut down 30% of the time during absences due

Table 4-1
Cooling System Characteristics

Nominal System				
Capacity:	3 tons	25 tons	100 tons	400 tons
Provides chilled:	air	water	water	water
Duty Cycle (hr/yr):	300-1200	300-1200	420-2000	420-2000
Absorption Chiller				
Cooling tower:	integral	wet	wet	wet
Boiler efficiency (%):	75	75	75	75
Parasitics (kW):	0.62	1.6	6.2	25
Cooling Tower (kW):	0	2.0	16.3	65
Electric Chiller				
Type:	reciprocating hermetic	reciprocating hermetic	centrifugal hermetic	centrifugal open
Cooling Tower:	integral, air	integral, air	tower, wet	tower, wet
Design Efficiency (kW/ton)*:	0.77	0.77	0.76	0.62
Seasonal COP*:	4.1	4.1	4.3	5.3
Cooling Tower and Parasitic Power (kW):	-	-	13	51

*Systems chosen are the most efficient that are currently widely available, and the analysis is structured around this assumption.

to working hours for the 3 ton unit or to nighttime for the 25 ton commercial unit. Since there are 600 cooling hours per year in New York City and 2800 in south Florida we chose the range of 300 to 1400 actual operating hours per year for the two smaller units and 420 to 2000 hours per year for the 100 and 400 ton units, which have no night shutdown. Clearly fuel and operating expenses will carry more economic weight at the upper end of this scale and we will determine the range of breakeven costs it implies.

Note that as the (unspecified) COP of the absorption units increases, the cooling towers can become smaller. In particular, the cooling tower dissipative power $P_{ct} = \text{Load}(1+1/\text{COP})$, where "COP" refers to the net coefficient of performance and, does not include the boiler efficiency η_b , since these losses go up the stack. This factor will have a significant effect on the comparative economics.

The boiler efficiencies are based on Tables 3-2 through 3-8. They are seasonal efficiencies for non-condensing systems since the first effect generator operates above 250°F. Parasitics which are common to both systems, such as chilled air or water circulation are simply ignored; the values of parasitics which are not common to both systems are taken from those of current

commercial units [4-2, 4-3, 4-4], as are the estimates of cooling tower power, or from Tables 3-2 and 3-4 for the 3 and 25 ton systems.

The efficiencies assumed for the compression units will have a significant effect on the analysis, and a number of variables must be considered. First, there are differences between the most efficient units available today and the average unit sold. In 1982 the average central residential air conditioner had a seasonal energy efficiency (SEER, in Btu/hr-W) of a little over 8, while units with rated SEER's of 14 were available.[4-4] We will use the efficiency of the best widely available units in each category, charging an appropriate cost premium in the following section. Although the performance of available units will certainly continue to improve, this improvement will be accompanied by further cost increments, and we will assume that these two effects cancel each other to some extent.

There is also a difference between design point COP and seasonal performance. For residential central compression systems there are well established correlations between seasonal and design point energy efficiency, and the quoted numbers are in fact seasonal averages. Small, typical air-to-air systems require up to 20% more energy than a design point analysis would indicate [4-5]; for our high efficiency systems we assume this is reduced to 10% and apply this factor to the 3 and 25 ton absorption systems to get the design point ratings. For the larger systems we have found very little information, and will assume more careful system design and lower cycling losses lead to 7% seasonal losses for 100 and 400 tons. The resulting design point efficiencies and seasonal COP's are shown in Table 4-1. Lacking any other data, these same ratios were applied to the design point COP's of the absorption machines to determine annual gas consumption.

4.3 The Breakeven Cost Model

In this section the economic relationships used in the breakeven cost model are developed and justified; the model itself is included (in Basic) as Appendix 5. The primary purpose of the model is to determine the maximum installed capital cost that an absorption chiller can have and still be competitive with a comparable compression unit. We find this "breakeven capital cost" as a function of absorption chiller COP and use this result to establish threshold efficiencies for absorption machinery. Accordingly, let:

$$\begin{aligned} \text{Breakeven Absorption Capital Cost} &= \text{Compression Capital Cost} \\ &+ \text{Compression Cooling Tower Cost} \\ &+ \text{Compression Installation Cost} \\ &+ \text{Lifecycle Compression Electricity Cost} \\ &+ \text{Lifecycle Compression O\&M Costs} \\ &- \text{Absorption Cooling Tower Cost} \\ &- \text{Absorption Installation Cost} \\ &- \text{Lifecycle Absorption Gas Cost} \\ &- \text{Lifecycle Absorption Electricity Cost} \\ &- \text{Lifecycle Absorption O\&M Costs.} \end{aligned} \quad (4-1)$$

We discuss the various terms in the order shown in Table 4-2, where the actual values used are presented. It is worth noting that prices and cost estimates are derived from a number of sources which presented data in terms of dollars for several different years. All figures quoted here have been converted to mid 1984 dollars, using the producer price index for overall prices and the individual price indices for fuel and machinery where appropriate.[4-8]

Table 4-2
Cost Factors in Breakeven Cost Model
(mid 1984 dollars)

Nominal System Capacity (tons)	3	25	100	400
<u>Absorption Systems</u>				
Cooling Tower Cost, COP=1 (\$)	(included)	5200	14,900	46,300
Variable C.T.C. (\$/mBtuh)		4.6	5.3	2.8
Installation Cost (\$)	1000	7000	23,000	43,400
1984 Gas Cost (\$/mmBtu)	6.15	5.70	5.70	5.70
Maximum 1984 Electricity Cost (\$/kWh)	0.15	0.17	0.16	0.16
O&M (\$/year)	230	1200	2350	3600
<u>Compression System</u>				
<u>Capital Cost:</u>				
Equipment (\$)	3900	12,750	38,000	87,000
Installation (\$)	1000	6,150	15,000	35,000
Cooling Tower Cost (\$)	(included)	(included)	10,000	32,000
Maximum 1984 Electricity Cost (\$/kWh)	0.14	0.16	0.13	0.13
O&M (\$/year)	230	1200	1900	3700
<u>Common Elements</u>				
Discount Rate (%/year)	6	6	6	6
Economic System Lifetime (Years)	7	7	7	7
Gas Price Escalation (%/year)	0,3	0,3	0,3	0,3
Electricity Price Escalation (%/year)	3,0	3,0	3,0	3,0
Minimum 1984 Electricity Cost (\$/kWh)	0.08	0.08	0.08	0.08

The capital costs of cooling towers were taken from Means' Mechanical and Electrical Cost Data Guide [4-9] and include materials and installation labor, subcontractor's markups and a 10% general contractor's overhead and profit. The results compare favorably (within 5%) with costs for comparable systems in [4-3]. The variable cooling tower cost indicates the amount the total cost of the cooling tower changes as variations in COP above or below 1.0 produce variations in the thermal power it must dissipate. It is calculated from the change in installed cost for two systems somewhat above and somewhat below the nominal system capacity in each case.

We seek a breakeven capital equipment cost, so the cost of installing the absorption systems must be subtracted out, as shown in Eq. 4-1. For the 400 ton unit, we interpolate between systems of 320 and 800 tons in [4-3], where installation costs are broken out in detail, to obtain \$38,800 for a direct fired system and an increment of 13% for installation of a steam fired unit, due largely to the steam piping. Interpolating between systems in [4-9] we find installation costs for similar steam fired units of \$54,300; lowering this by 13% and averaging with the value from [4-3], we find the value in Table 4-2, which is probably certain to plus or minus 10%, as well as one can do in estimates of this sort. Similarly, for the smaller units, we have have folded together data from [4-9] and [4-10], using geometric averages where the values differed widely; the resulting entries should only be regarded as accurate to plus 100 or minus 50 percent.

The breakeven cost analysis is carried out for a range of possible electricity and gas prices, expressed in terms of two scenarios which we assert bracket reasonable expectations for future circumstances. One of these assumes price structures favorable to gas consumption, while the other examines the case where electricity develops a price advantage. Gas prices are currently reasonably uniform across the country [4-12], so we use current national average prices [4-13] for residential (3 ton system) and commercial (25 and 100 ton systems) customers. The scenario which is more favorable to gas will assume that prices only increase with inflation (have zero escalation) while the scenario assuming an advantage for electricity will use a 3% per year escalation in the price of gas above inflation.

Electricity prices are far more variable than gas prices. Figure 4-1 attempts to capture this variability by displaying the fraction of customers who are paying more than a given price as a function of that price.[4-14] (The data represent prices in 40 large metropolitan areas.) Thus almost all customers pay more than 6.1 cents per kilowatt hour, 50% pay more than 8.1 cents and an astonishing 20% pay almost 16 cents. It is noteworthy that only a small number of customers pay between 10 and 15 cents, so a breakeven cost analysis based on 10 cents might as well be based on the upper limit prices.

Accordingly, for the scenario in which electricity is favored, we will use a 1984 electricity price of 8 cents per kilowatt hour, ensuring that our product will be competitive in one half of the national market, and will couple this price with zero constant dollar escalation. For the scenario in which gas is favored, electricity prices are set equal to the highest current price in a U.S. service area for the appropriate customer class [4-15], coupled with a 3%/yr escalation rate. In all cases the system owner was assumed to have a total non-air conditioning electric load equal to the compression air conditioning load. This determined an average monthly purchase level which determined customer class for residential (3 ton) and commercial (25 and 100 ton) systems. The same approach was used to establish electric rates for absorption systems, but here the lower electricity consumption of the cooling systems led to higher unit costs.

Figure 4-1. The fraction of electricity customers paying a given price or more, as a function of that price.

Maintenance estimates for the 400 ton absorption machine [4-3] do not include condenser tube cleaning, and this is also omitted from the estimates for the 400 ton compression machine. After allowing for this, they are consistent with the estimates of [4-10], used for the 100 ton unit. The estimates of [4-10] for maintenance of the two smaller absorption machines seem untrustworthy, since they turn out to be less than estimates for compression machines. We therefore use the maintenance costs derived below for 3 and 25 ton compression machines for the small absorption machines as well. All maintenance charges are assumed constant except for inflation.

The cost factors for the 400 ton compression unit are taken from [4-3], interpolated between those for 300 and 800 ton units. Again, the maintenance does not include condenser tube cleaning. References [4-7] and [4-11] give slightly lower capital costs, perhaps reflecting somewhat lower quality equipment, while their installation costs are dramatically above and below that used here, from [4-3], which is quite close to the average over all three. Capital and installation costs for the three smaller machines are all taken from Means' Mechanical Cost Data [4-11], and are less certain than the 400 ton figures. The installation cost for the 3 ton unit from [4-11] (\$1990) seems ridiculously high, and we use the estimate obtained for the 3 ton absorption machine instead. Maintenance costs are taken from [4-10] (3 and 25 tons) and from [4-3] (100 and 400 tons).

Future payments for gas, electricity and maintenance must be discounted to allow for the returns that could be gained on the funds that will be used to make those payments in the meantime.[4-16] Consistent with the fuel escalation rates, we use a constant dollar discount rate of 6%/yr in all cases, corresponding to a crude estimate of alternative investments. The net present value (NPV) of any current annual payment (P) is then:

$$NPV = \sum_{k=1}^7 P[(1+e)/(1+d)]^k, \quad (4-2)$$

where d is the discount rate, e the escalation rate and k the year index.

4.4 Operating Expenses

Before evaluating breakeven costs we consider the first year operating expenses of the four model systems, displayed in Table 4-3. They include only the variable operating expenses of Tables 4-1 and 4-2, not the capital outlays or fixed maintenance expenses. Results are displayed for the two cases developed above, in one of which gas is less expensive than electricity and in one in which it is more expensive. Since the calculation is for first year operating expenses, it includes no escalation of either gas or electric prices. For the absorption chillers the total operating expenses are further broken down into their electric and gas components, and all calculations are done assuming a net seasonal COP of 1.0 for absorption units.

Table 4-3
Operating Cost Comparisons
(\$/hour in first year)

System (tons):	3	25	100	400
Inexpensive Electricity Scenario:				
Electric Chiller:	0.21	1.70	7.10	25.30
Absorption Chiller:	0.27	2.00	8.60	34.60
(Electricity:	0.05	0.30	1.80	7.20)
(Gas:	0.22	1.70	6.80	27.40)
Expensive Electricity Scenario:				
Electric Chiller:	0.36	3.40	11.60	41.10
Absorption Chiller:	0.31	2.30	10.40	41.80
(Electricity:	0.09	0.60	3.60	14.40)
(Gas:	0.22	1.70	6.80	27.40)

First note that in the case of relatively inexpensive electricity, i.e. the median electricity price versus the median gas price, the operating expenses of the absorption chillers are higher than those of compression systems in all sizes. This is disturbing because the primary selling point of absorption chillers has historically been low operating expenses, while these results indicate that to reach the "average" American market we will have to lower the capital cost sufficiently to make the higher operating costs acceptable to the purchaser. Even if this can be done, it will entail a complete reversal of sales philosophy which the industry may find hard to carry out.

The scenario in which electricity is relatively expensive refers to operation in localities like New York and San Diego where gas prices are average and electricity prices quite high. Here the absorption chiller's operating expenses are indeed lower than those of compression systems, except for the largest size, where the very high efficiency of the open centrifugal electric chiller produced a dead heat, to the accuracy of this sort of calculation.

These results lead us to two important conclusions. First, we must recognize that a revitalized absorption chiller industry selling machinery at these levels of efficiency, will most likely be selling systems only to the 20% of customers who pay premium prices for electricity. This target is less than the entire country, but it is a substantial and stable market, and one well worth pursuing.

This conclusion is fairly robust: it is easy to calculate that moving from eight cents to ten cents for the low priced electricity has little effect

on the balances of Table 4-3; absorption still has higher operating costs for all but the 25 ton case. Furthermore, as pointed out above, Figure 4-1 shows that almost everyone paying 10 cents per kilowatt hour or more actually pays 16 cents, so there is no point in considering 10 cent electricity as a viable scenario; it gains very few customers over the much more advantageous 16 cent regime.

Second, we may do well to aim initially at systems in the 25 - 100 ton class. Larger compression systems can afford the maintenance that goes with open systems and higher efficiency; smaller systems find it harder to justify these maintenance costs, which are relatively independent of system size, and are much more likely to choose hermetically sealed systems with attendant lower efficiencies. This niche appears to be the best place to start, but in order not to limit ourselves to too small a market in the process of selecting cost goals, we will work largely from the 400 ton model, where the operating expenses of the competing systems are almost equal. Systems which meet viable breakeven cost goals at 400 tons will enjoy clear advantages at somewhat smaller sizes.

4.5 Estimates of Breakeven Costs of Absorption Chillers

The above parameters and relationships were combined in a program which is included as Appendix 5. Execution of the program resulted in the estimates of breakeven cost displayed in Figures 4-2 through 4-6, which are discussed in this section. For each system, the figure shows the price a "rational" consumer would pay for an absorption system of the indicated COP, based on the cost of the compression alternative.

Figures 4-2 through 4-5 show the components of this breakeven cost for chillers of our baseline design with net COP's of 1.0. Starting on the left of each figure, the first costs associated with the compression system are added in and the first costs of absorption equipment (other than equipment costs) are subtracted out. This leaves the "first cost component" of the breakeven costs. Adding the net present value of maintenance costs for compression equipment and subtracting the corresponding charges for absorption equipment leaves the non-energy component of each breakeven cost estimate stretching across the middle of each figure.

Issuing from this baseline are the components of breakeven cost associated with the net present value of the operating costs for each of four scenarios for each machine. In all scenarios involving high electricity prices the net effect of the operating cost components is to increase the breakeven cost; the lowered fuel expenses are translated into higher allowable first costs for the absorption equipment. Similarly, in the scenarios involving more moderate electricity prices, the higher operating costs of the absorption equipment lower the allowable first costs. In the cases of higher load factors these differences become strikingly more pronounced.

On the right side of Figures 4-3 through 4-5 the current price of absorption equipment is shown, taken from Figure 2-2. (No 3 ton machine is

Figure 4-2. Components of the breakeven cost of a 3 ton absorption chiller.

Figure 4-3. Components of the breakeven cost of a 25 ton absorption chiller.

Figure 4-4. Components of the breakeven cost of a 100 ton absorption chiller.

Figure 4-5. Components of the breakeven cost of a 400 ton absorption chiller.

Figure 4-6. Breakeven cost of 400 ton chiller as a function of coefficient of performance.

currently available.) It is clear that within the limits of the scenarios we have used, current gas-fired absorption machinery is not broadly competitive. A few market niches where absorption chillers are currently competitive do exist, based on low cost local gas or similar anomalies. For example, systems in which the chiller is used as a heat source in the winter look considerably more acceptable. The capital savings on a boiler can add up to several hundred dollars per ton and Figures 4-2 through 4-5 show clearly that this could tip the balance in favor of the absorption system, but only in an application where such a system would be able to meet all demands, including hot water during the cooling season. Because of the application-specific nature of these systems, we have not used them to estimate breakeven costs.

We base our cost goals on the estimates of breakeven costs indicated in the figures. For each system, we will consider the cost goal to be the breakeven cost estimate corresponding to the low duty cycle and the high set of electricity prices, and these values are presented in Table 4-4. This scenario was chosen to be representative by elimination: at this time, it appears that meeting the goals corresponding to the moderate electricity price scenario would be very difficult for the low duty cycle case and obviously impossible for the high duty cycle case. On the other hand, meeting only the less restrictive requirements for the high electricity price, high duty cycle scenario would leave us with an unacceptably small potential market share.

These two bracketing scenarios are used in Table 4-4 to indicate the range of uncertainty in these estimates.

Table 4-4
Estimated Retail Breakeven Equipment Costs (BEC) for Absorption Chillers
(1984 \$ per ton; COP = 1.0)

Capacity	Absorption Equipment Current Price	BEC Range	Cost Goal	Cost Goal Minus Compression Equipment Costs
3	NA	1240 - 1445	1340	
25	950	230 - 630	360	-150
100	700	180 - 535	290	-90
400	325	85 - 220	175	-45

The rigorous nature of the challenge here is indicated in the last column, which shows that for the three larger sizes, the cost goal can only be met if the absorption chiller equipment costs are below comparable compression costs by 20-25%. In the following section we show how these requirements can be met.

These results are consistent with, although not identical to, comparable estimates obtained by GRI in a recent program document.[1-2] Because of modest differences in assumptions with respect to duty cycles, because of differing numbers of electricity price markets, and because the scope of the GRI effort did not include sizing of cooling towers, seasonal versus design point efficiencies and several comparable details, strict numerical agreement would be a very unlikely occurrence. However, Figure 5 of reference 1-2 indicates that over the range of duty cycles considered here, the GRI estimate for absorption chiller breakeven costs ranges from \$50 above the cost of compression equipment to \$50 below it. Using their estimates for the cost of compression equipment results in breakeven cost estimates from \$100 to \$200 per ton of absorption equipment, a final result which is satisfyingly close to ours, shown in Table 4-4 above.

The model is also capable of projecting breakeven cost estimates as a function of COP, and the results for the 400 ton chiller operating under the four scenarios described above are shown in Figure 4-6. It is clear that while efficiency improvements could have a dramatic effect on marketability in the southern tier (high duty cycle) market, the effect would be more modest in low duty cycle markets, with a 35-70% improvement in breakeven cost following from a very optimistic increase in COP up to 1.5. The clear implication is that while increases in the efficiency of absorption chillers can only help marketability, they cannot turn the situation around; only dramatic decreases in equipment costs can accomplish this, and in the next section we discuss in detail how this may be accomplished.

4.6 Breakeven Costs of Heat Exchangers and Components

In this section we disaggregate the retail breakeven costs derived above for the 400 ton chiller to determine breakeven costs for the various heat exchangers and for the materials of which they are made. Several assumptions are necessary which are more matters of judgement than of fact, and we will make these explicit as we use them.

Since we are working largely with the 400 ton system, we will simply use cost fractions derived directly from Table 3-8 without concern for the discrepancies between our estimates of appropriate retail prices and the prices for which these systems are actually available, as shown in Figure 3-1. We will assume that the errors and divergences which produced this discrepancy were spread over the entire process of manufacturing and distributing the units, and will use the quantities in Table 3-8 to determine the fraction of total cost ascribable to the materials in a particular heat exchanger. These fractions will then be applied to the breakeven cost to determine component breakeven costs. In this way the effects of the uncertainties associated with the current cost increment will be minimized.

Our retail breakeven cost is \$175 per ton, or \$70,000 for a 400 ton unit. Table 3-8 indicates that a distributor's markup of 86% is common in this size range. It can be argued that distributors and dealers will require the same absolute amount per unit sold, whether retail prices drop or not. It can also be argued that increased sales will allow dealers to lower the markup more than in proportion to wholesale price cuts. We choose the simplest path, leaving the markup constant, and derive a factory price of \$38,000.

Then the "Total Cost" column of Table 3-8 can be used to show that 47% of the factory price is used for materials and fabrication of heat exchangers, the remainder being used for vessels, pumps, and other balance of system components. We will show in Chapter 8 that significant cost reductions may be possible through advanced designs which allow some components to fill more than one function. Here, we ignore these possibilities and the increased breakeven costs they may allow for individual components and use the figure from Table 3-8 (47%) to derive a total breakeven cost for the heat exchangers of \$17,800.

Then, as a crude estimate, the 6350 square feet of heat exchanger in our design results in a total (materials and labor) breakeven cost of \$2.80 per square foot. For comparison, the same calculations for the three smaller systems result in total breakeven costs of \$3.00, \$3.50 and \$3.60 per square foot for the 3, 25 and 100 ton units respectively.

We can disaggregate these totals into materials and fabrication components, averaged over the whole system. Using the ratios obtained by totaling the heat exchanger materials costs and the heat exchanger fabrication costs in Table 3-8 and dividing by total heat exchanger costs, we find that the 400 ton unit has an average heat exchanger materials breakeven cost of \$2.20 and a fabrication breakeven cost of \$0.60 per square foot. For the

smaller units, fabrication takes up more and more of the allowed costs as subcomponents such as pipes become smaller and the work of joining them more significant. Thus the average materials breakeven costs for the smaller units, obtained in the same way, is \$0.95, \$1.70 and \$2.60 per square foot for the 3, 25 and 100 ton units.

However, these average values will not be very valuable in the selection of materials for particular heat exchangers, since the conditions they face are quite different. The first effect generator, facing gas flames, is made of considerably more corrosion resistant and expensive material than are the recuperators, in their completely controlled environment. To determine in first approximation what we can afford to pay for the materials and fabrication of individual heat exchangers, we take that fraction of the factory breakeven cost allocated to heat exchangers ($6350 \text{ ft}^2 * \$2.80/\text{ft}^2 = \$17,800$) and disaggregate it in proportion to the fraction of total heat exchanger cost each heat exchanger's materials or fabrication represents in Table 3-8. The results are presented in Table 4-5.

Table 4-5
Breakeven Costs of Absorption Chiller Heat Exchangers
400 ton unit

Heat Exchanger	Total Breakeven Cost (\$)	Materials Breakeven Cost (\$/ft ²)	Fabrication Breakeven Cost (\$)
1st Effect Generator	4140	6.30	340
2nd Effect Generator	2070	1.40	690
Condenser	2420	3.00	340
Evaporator	3460	1.90	690
Absorber	3460	2.00	690
Hi Temp Recuperator	1210	0.90	690
Lo Temp Recuperator	1210	0.90	690

These results, as should be clear from the preceeding paragraphs, also apply reasonably accurately to the 100 ton unit, and they will serve as the basis by which we judge candidate materials economically. To anticipate, there are many materials which cost less than this which appear suitable for each of the heat exchangers, and it will probably prove possible to make absorption units with heat exchanger costs well below these breakeven limits if the materials can withstand the technical rigors of the chiller environment. Then, if systems can be constructed using these materials, we will be in a very strong position to challenge compression systems in areas with high electricity prices. Only after this appears likely will it be a suitable time to assess the chances of making inroads in the rest of the market, where absorption chillers would be attractive only because of low first costs, and not at all because of low operating costs.

5. PROPERTIES OF CANDIDATE MATERIALS FOR HEAT EXCHANGERS AND OTHER COMPONENTS

5.1 Introduction

In this chapter we screen a large number of candidate materials for their suitability to the technically and economically demanding environment characteristic of absorption chiller heat exchangers. The first stage of the assessment relies on known characteristics of the materials, culled from the technical literature, from manufacturer's specifications and from personal contact with experts in the field. As a result of this initial screening we selected a small number of materials for detailed testing, and this testing is described in Chapters 6 and 7.

5.2 Specification of the Required Data on Candidate Materials

We describe the materials using a modest number of characteristics. Physically, we enumerate each material's chemical composition or type number, specifying different heat treatments or other manufacturing variations where appropriate, typical thickness, density in pounds per square foot and tensile strength against rupture, to assure that the foils can withstand the one to three atmospheres of static differential pressure encountered in the heat exchangers. The metals are thin enough to ensure that heat transfer will be dominated by boundary layer effects in the fluids rather than heat transfer through the materials, so the thermal conductivity of the metals is not included; for the polymers and polymer coatings thermal conductivity is a matter of concern and is discussed in detail. Surface characteristics that may affect the heat transfer properties on the water side or in the films of evaporating or condensing fluids (all discussed in Chapter 3) are also discussed, especially in the section on polymer coatings.

The heat exchangers will have to operate in various thermal environments, ranging from LiBr solution or pure water from 40 to 212°F to the 1000 to 1500°F external environment of the directly fired first effect generator. (The temperature of even the external surface of the generator will of course be substantially lower.) For each material the temperature at which it begins to weaken or at which corrosion is significantly enhanced is indicated. In particular, the directly fired first effect generator requires materials and approaches distinctly different from those of the internal heat exchangers.

Also, the susceptibility of each material to corrosion in the harsh environment of the strong LiBr solution must be evaluated. Determining the potential effect of LiBr is a bit involved, since the corrosive processes take place only in presence of oxygen, and can also be prevented through the use of regular chemical analysis and maintenance of the working solution.[5-1] However, the expense involved in ensuring a sufficiently oxygen-free environment or in such chemical management will be inversely related to the susceptibility of the materials to attack by these brines. Thus we present data as available on the corrosion susceptibility of the various materials to serve as a guide in this selection process.

In addition, the heat exchangers by which heat is removed from the chiller must face various pollutants picked up by the cooling water in the wet cooling tower, and since this cooling process involves evaporation, these pollutants can be concentrated, as can trace contaminants in the makeup water. We therefore also enumerate the susceptibility of these materials to corrosive agents found as aerosols or in water, primarily weak acids and sulfur compounds. This same data is important in evaluating the potential of these materials for use in the directly fired first effect generator.

Finally, the basic economic parameter is cost per square foot. This will depend on the thickness of the material and the amount of processing involved in complex ways, and data is included for a number of the most promising candidates.

5.3 Properties and Selection of Candidate Polymers

Our approach to potential polymeric materials was as exhaustive as possible given the resources available. First, a list was compiled of all polymers currently available as film or as calendered sheets (which could presumably be made quite thin) or which could be extruded into specific shapes if the heat transfer characteristics of the material permitted relatively thick structures. References 5-2 and 5-3 were the primary sources for this list, augmented by individual suggestions by manufacturer's representatives, articles in trade journals and other sources.

Then all materials known to be hydrophilic or water permeable [5-1], as listed in Table 5-1, were dropped. It might be possible, at some later date, to consider the use of certain hydrophilic materials, arguing that as long as the salts and conditioners do not pass through the barrier, the passage of pure water could easily be made up. The complexity of testing this possibility, especially for the passage of oxygen, and the small likelihood of success, precluded its consideration here.

The remaining materials are listed in Table 5-2; thus Tables 5-1 and 5-2 comprise a complete list of all polymers examined. The next major selection process was based on thermal conductivity and strength. In short, we must have films which are thin enough to ensure that the thermal resistance of the material does not significantly degrade the performance of the chiller, while the material must be strong enough to withstand the differential pressure between system water and the near vacuum within the chiller shell.

The thermal resistance of the metal comprising the heat exchange surface in current designs is negligible in comparison to the thermal resistance of the water (on the water side) or of the film of solution or refrigerant (on the solution side); the entire temperature differential is due to these two factors. Adding a thermally resistant layer of material will increase this temperature differential by an amount equal to the thermal flux divided by the thermal conductivity of the new layer. We therefore established the maximum polymer film thickness by finding, for each heat exchanger, the polymer film thickness which would lead to a twenty percent increase in this temperature

Table 5-1
Polymers Eliminated due to Interactions with Water

Polymer	Problem ¹	References ²
Cellulose Acetate	T,A	22, 502
Cellulose Triacetate	T,A	22, 502
Cellulose Propionate	T,A	22, 502
Ethyl Cellulose	T,A	21, 502
Cellulose Acetate Butyrate	T,A	22, 502
Cellulose Nitrate	T,A	21, 474
Polyamide-Nylon 6-Extruded	A,S	28, 503
Polyamide-Nylon 6-Oriented	A,S	28, 503
Polyamide-Nylon 11,12	A,S	28, 483, 504
Polycarbonate	T	41, 504
Polyimide (Thermoplastic)	T	65, 505
Polymethyl Methacrylate-Type A	A	14, 505
Polysulphone	T	94, 505
Polyethersulphone	A,T	95, 505
Polyurethane Elastomer	A,T,S	80, 506
PVC (Plasticized Extrusion)	T	84, 506
PVC (Plasticized Solvent Cast)	T	84, 506
Vinyl chloride-acetate (Nonplasticized)	T	90, 506
Vinyl Nitrile Rubber	T	27, 506

¹A = Absorption; T = Transmission; S = Stability

²Page reference in Modern Plastics Encyclopedia, Ref. 5-2.

Table 5-2
Initial Candidate Polymer Materials

Polymer	Thermal Conductivity (Btu-mil/hr-°F-ft ²)	Tensile or Yield Strength (psi)*	References
Films or Extrusions:			
Acrylonitrile methyl acrylate	1620	9000 T	473, 502
Acetal Homopolymer	1600	1100 Y	8, 471
Acetal Copolymer	1600	8800 Y	9, 471
E-CTFE (Ethylene Chlorotrifluoroethylene)	1100	8000 T	25, 478, 503
ETFE (Ethylene tetrafluoroethylene)	1650	7000 T	26, 477, 503
FEP (Fluorinated Ethylene-Propylene)	1750	2500 T	25, 477, 503
PFA (Perfluoroalkoxy)	1750	4000 T	25, 477, 503
PCTFE (Polychloro-trifluoroethylene)	1450	5000 T	25, 477, 503

Table 5-2 (continued)
Initial Candidate Polymer Materials

Polymer	Thermal Conductivity (Btu-mil /hr-°F-ft ²)	Tensile or Yield Strength (psi)*	References
<u>Films or Extrusions:</u>			
PTFE (Polytetrafluoroethylene)	1750	1500 T	24, 477, 503
PVF (Polyvinyl Fluoride)	1650	7000 T	25, 503
PVDF (Polyvinylidene Fluoride)	930	5500 T	26, 477, 503
Polybutylene	1510	2100 Y	40, 484
PET (Polyethylene terephthalate)	1054	27,500 T	46, 486, 504
Polyetherimide	460	15,200 Y	54, 489
Polyethylene (Low Density)	2320	1500 T	55, 489, 504
Polyethylene (Linear Low Density)	2320	2100 Y	60, 489
Polyethylene (Medium Density)	2320	2000 T	55, 489, 504
Polyethylene (High Density)	3340	2400 T	55, 490, 504
Polyethylene (Hi Mol Wt)	2941	3350 Y	60, 490
Polyethylene (Ultra Hi Mol Wt)	2561	3000 T	62, 491, 504
Ethylene-methyl acrylate	2681	1650 Y	64, 491
Ethylene Vinyl Acetate	2441	1000 T	64, 490, 504
Polyimide (Thermoset)	2460	6400 T	65, 491
Polymethylpentene	1160	2700 T	66, 491, 505
Polypropylene	810	4500 T	67, 493, 505
Polypropylene - Oriented	810	7500 T	67, 493, 505
Polystyrene	870	8000 T	72, 495, 505
Polysulphone	1800	8400 T	94, 498, 505
PVC (Polyvinyl chloride; Extrusion)	870	7000 T	84, 497, 506
PVC (Polyvinyl chloride; Solvent Cast)	870	6000 T	84, 497, 506
Vinyl chloride-acetate (Plasticized)	1160	2500 T	90, 473, 506
Polyvinylidene Chloride (Saran)	870	4900 Y	90, 497
<u>Sheets only:</u>			
ABS (Acrylonitrile-butadiene-styrene)	1810	5000 T	6, 27, 470
Olefin-modified styrene-acrylonitrile	870	11,200 Y	7, 494
Polymethyl Methacrylate-Stndrd (Acrylic)	1450	8200 T	14, 473, 505
Ionomer	1780	1700 Y	27, 478, 503
SAN (Styrene-acrylonitrile)	870	11,200 Y	93, 494
<u>Film only:</u>			
Polyvinyl Butyral	871	500 T	90, 501

*T (Tensile strength) is the lower limit of the range for a film, where data was available; Y (Yield strength) is the average value for bulk material.

differential. This would also correspond to the thermal resistance which could be compensated by a twenty percent increase in heat exchanger area.

The critical value of the increase in temperature differential was found using the model for a 100 ton chiller from Section 3. For each heat exchanger, the log mean temperature difference was calculated from the data in Table 3-5 (100 ton functional characteristics) and twenty percent of each of these values are displayed in Table 5-3 as "Incremental ΔT ", along with the thermal flux through each heat exchanger (from Tables 3-5 and 3-6), the allowable thermal conductance derived by dividing the thermal flux by the allowable increase in temperature differential, and finally, the range of expected allowable thickness in the polymer film, based on the range of thermal conductivities listed in Table 5-2.

Table 5-3
Range of Allowable Thicknesses for Polymer Films in Heat Exchangers

	Power (mBtu/ hr)	Area (ft ²)	Power Density (mBtu/ hr-ft ²)	Incr. ΔT (°F)	Minimum Thermal Conductance (mBtu/hr- ft ² -°F)	Film Thickness (mils)	
						(1)	(2)
<u>Component:</u>							
2nd Effect Generator	680	260	2.62	4.70	0.56	0.83	6.00
Condenser	660	300	2.20	2.38	0.92	0.50	3.61
Evaporator	1200	425	2.82	1.60	1.76	0.26	1.89
Absorber	1440	650	2.22	2.86	0.77	0.59	4.31
Hi Temp Recuperator	460	400	1.15	2.96	0.39	1.18	8.60
Lo Temp Recuperator	740	400	1.85	3.82	0.48	0.95	6.90
¹ Polyetherimide, 0.46 mBtu-mil/hr-°F-ft ²							
² HD Polyethylene, 3.34 mBtu-mil/hr-°F-ft ²							

Since films of less than one mil are difficult to find and are often unreliable with respect to thickness and strength, it is clear that this criterion will winnow out a great many candidates. In Table 5-4 we list, for each heat exchanger, only polymeric materials with sufficient thermal conductivity to permit the use of films one mil thick or thicker in one or more heat exchangers. Table 5-2 showed that a few materials were available only as extrusions, and could not be calendered or blown down to film thicknesses; these materials are all eliminated at this point since any

extruded components would have too much thermal resistance due to their thickness.

In these tables the yield or tensile strength from Table 5-2 has been used to calculate the differential pressure sustainable by a one quarter inch diameter tube of the material, and the candidates have been ordered by the magnitude of this rupture pressure. Even a modest allowance for irregularities and defects requires that we eliminate all candidates not capable of withstanding a differential pressure of three atmospheres (45 psi), and this criterion accounts for the absence of many materials not ruled out by the one mil thickness requirement. Finally, each heat exchanger has a typical operating temperature indicated in the heading of the table, and several materials have been dropped because their yield strength [5-2] is declining precipitously at that temperature, or because their maximum service temperature [5-2] is too low. We do not have complete data on the high temperature behavior of all candidates, and several more may be dropped on these grounds at a later date.

This process of elimination has left us with several serious candidates, most notably PET, Thermoset polyimide, Polysulphone, Acrylonitrile methyl acrylate, Acetal copolymer (tradename Celcon), Polymethyl methacrylate, high density Polyethylene and the fluoropolymers CTFE (tradename Aclar) and ECTFE (tradename Halar). All candidates are listed [5-2] as having excellent resistance to acids and, where data is available, to salts. All would appear to be economically acceptable candidates if they can be made to work technically, since the most expensive, the fluoropolymers, have base materials costs of \$24 per pound, which translates to 16 cents per mil-thick square foot for material, and perhaps 30 cents per mil-thick square foot for the most expensive rolled film. Unfortunately, we were not able to obtain samples of all of these in the time available. The results of our tests on the samples we did obtain are in Chapters 6 and 7.

5.4 Properties and Selection of Candidate Metal Foils

Various technical sources [5-4 to 5-6] were searched for metals that could be formed into corrosion resistant foils that might be suitable for use in various heat exchangers in LiBr absorption chillers. In particular, the high temperatures encountered in the directly fired first effect generator would seem to require that metal be used there; in the other components the metals could be used if they proved advantageous. As seen in Chapter 3, mild steel is used in several components in current designs; it is included here for reference and because we will see in Section 5.5 that there is reason to believe that a thin foil of mild steel with a polymer coating on it may well be a promising candidate.

The physical properties of the metals are summarized in Table 5-5. The first column gives the American Iron and Steel Institute type number (or comparable generic designation). The second column indicates the two or three principle secondary constituents; note that brevity demanded that we often omit mention of constituents crucial to the characteristics of the material.

Table 5-4
Maximum Wall Thickness and Differential Pressure for Candidate Polymers
[Wall Thickness, T (mils); Differential Pressure, DP (psi) in a 0.25 inch Tube]

	2nd Effect Generator		Condenser		Evaporator		Absorber		High Temp. Recuperator		Low Temp. Recuperator	
Thermal Flux (Btu/hr-ft ²):	2620		2200		2820		2220		1150		1850	
Temperature Change in Polymer (°F):	4.7		2.3		1.6		2.9		3.0		3.8	
	T	DP	T	DP	T	DP	T	DP	T	DP	T	DP
PET (Polyethylene terephthalate)	-	-	1.1	253	0.6	132	1.4	303	-	-	-	-
Polyimide (Thermoset)	4.4	226	2.7	137	1.4	71	3.2	165	6.4	329	5.1	259
Polysulphone	3.2	217	2.0	132	1.0	69	2.4	158	4.7	316	3.7	248
Acrylonitrile methyl acrylate	2.9	209	1.8	127	0.9	66	2.1	152	4.2	304	3.3	240
Acetal Copolymer	2.9	202	1.7	123	0.9	64	2.1	147	4.2	294	3.3	231
Polymethyl Methacrylate-Stndrd (Acrylic)	2.6	171	1.6	104	0.8	54	1.9	124	3.8	248	3.0	195
ETFE (Ethylene tetrafluoroethylene)	3.0	166	1.8	101	0.9	52	2.2	121	4.3	241	3.4	190
PVF (Polyvinyl Fluoride; Tedlar)	3.0	166	1.8	101	0.9	52	2.2	121	4.3	241	3.4	190
Polyethylene (Hi Mol Wt)	5.3	141	3.2	86	1.7	45	3.8	103	7.7	206	6.0	162
Olefin-modified styrene-acrylonitrile	1.6	140	0.9	85	0.5	44	1.1	102	2.3	203	1.8	160
SAN (Styrene-acrylonitrile)	1.6	140	0.9	85	0.5	44	1.1	102	2.3	203	1.8	160
ABS (Acrylonitrile-butadiene-styrene)	3.2	130	2.0	79	1.0	41	2.4	95	4.7	189	3.7	149
ECTFE (Ethylene Chlorotrifluoroethylene)*	2.0	126	1.2	77	0.6	40	1.4	92	2.9	184	2.3	145
Polyethylene (High Density)	6.0	115	3.6	70	1.9	36	4.4	84	8.7	167	6.9	132
Polyethylene (Ultra Hi Mol Wt)	4.6	110	2.8	67	1.5	35	3.3	80	6.7	160	5.3	126
PCTFE (Polychloro-trifluoroethylene; Aclar)	2.6	104	1.6	63	0.8	33	1.9	76	3.8	151	3.0	119
PFA (Perfluoroalkoxy; Teflon)	3.1	100	1.9	61	1.0	32	2.3	73	4.6	146	3.6	115
Polystyrene	1.6	100	0.9	61	0.5	32	1.1	73	2.3	145	1.8	114
PVC (Polyvinyl chloride; Extrusion)	1.6	87	0.9	53	0.5	28	1.1	64	2.3	127	1.8	100
Polypropylene - Oriented	1.5	87	0.9	53	0.5	28	1.1	63	2.1	127	1.7	100
PVC (Polyvinyl chloride; Solvent Cast)	1.6	75	0.9	46	0.5	24	1.1	55	2.3	109	1.8	86
PVDF (Polyvinylidene Fluoride)	1.7	73	1.0	45	0.5	23	1.2	53	2.4	107	1.9	84
Polyethylene (Linear Low Density)	4.2	70	2.5	43	1.3	22	3.0	51	6.1	102	4.8	80
Polyethylene (Medium Density)	4.2	67	2.5	40	1.3	21	3.0	48	6.1	97	4.8	76
Ethylene-methyl acrylate	4.8	63	2.9	39	1.5	20	3.5	46	7.0	92	5.5	73
FEP (Fluorinated Ethylene-Propylene; Teflon)	3.1	63	1.9	38	1.0	20	2.3	46	4.6	91	3.6	72
Polyvinylidene Chloride (Saran)	1.6	61	0.9	37	0.5	19	1.1	45	2.3	89	1.8	70

*Halar

The operating temperature indicates a regime where the structural properties of the material are essentially the same as at room temperature (except for the aluminum alloy, where the value of yield strength drops 35% over the range of temperatures indicated). Corrosion processes will accelerate at considerably lower, but less well determined temperatures. The foil thickness represents a value we chose as a minimum thickness reference case; thicker foils or strips may well be desirable. We have omitted data on yield strength after satisfying ourselves that a one inch diameter tube made of even three mil foil of any of these materials could easily withstand differential pressures of many atmospheres without permanent distortion. Conversely, none of the metals, at 20 mils, could support a horizontal tube of water of any useful length, a subject discussed in Chapter 8. The cost estimates refer to production quantity purchases of foil in the thickness shown and are based on telephone conversations with manufacturers.

Table 5-5
Physical Properties of Candidate Metal Foils

Material	Chemical/ Composition (%)	Operating Temperature (°F)	Density (lb/ft ³)	Cost (\$/ft ²)		
				5-mil	10-mil	20-mil
<u>Carbon Steel:</u>						
1010	1 Mn, 0.1 C..	600	489	0.22	0.41	
<u>Stainless Steels:</u>						
304	19 Cr, 10 Ni..	1800	500	0.60	0.87	1.23
316	17 Cr, 12 Ni..	1800	485	0.63	0.91	1.29
AL-6X ¹	20 Cr, 25 Ni..	1800	506	1.28	2.20	3.27
AL-6XN ¹	21 Cr, 25 Ni..	1800	503	1.28	2.20	3.27
AL 29-4C ¹	29 Cr, 4 Mo..	1100	478	0.95	1.56	2.21
S31803 ²	22 Cr, 6 Ni..	1100	489	-	-	2.04
(Alloy 2205)						
SEA-CURE ²	28 Cr, 3 Mo..	1100	480	-	-	2.40
<u>Aluminum Alloys:</u>						
3003	1.2Mn,.12Cu..	75-300	170	0.18	0.34	0.65
<u>Bronzes:</u>						
C61300	93Cu,7Al,0.4Sn		492			
C70400	92Cu,6Ni,1.5Fe		558			
<u>Nickel Alloys:</u>						
N04400	66Ni,32Cu,4Al	500	551	2.65		
(Monel ³)						
625	58Ni,21Cr,9Mo	1200	527	3.29	6.39	12.54
(Inconel ³)						

¹Trademark of Allegheny Ludlum Steel Corporation

²Trademark of Colt Industries

³Trademark of Huntington Alloys, Inc.

Almost all metals corrode to some extent, and the processes by which this occurs are complex and depend in detail on the chemistry of the environment. Since there is not a great deal of data on the corrosion process in LiBr, we have included data on sea water and other brine exposures in Table 5-6. The data is taken from references [5-4, 5-5] and from manufacturer's brochures and telephone presentations.

Table 5-6
Tabulated Corrosion Properties of Candidate Metals
[Fair (F), Good (G), Excellent (E) resistance or mils per year corrosion]

Environment	Sulphuric Acids 10%	Nitric Acids 65%	Lithium Bromide	Various Chlorides	Seawater	Comments
<u>Stainless Steel:</u>						
304	16,000	8	F	P		Ferric Chloride
316	850	11	G	F	F-G	" "
AL-6X ¹	120	31		E	E	" "
AL-6XN ¹	210(e)	29			E	" "
AL 29-4C ¹				E		" "
S31803 ²	4	5				
(Alloy 2205)						
SEA-CURE ²	1	1	G		E	
<u>Aluminum Alloy:</u>						
3003		G			F	
<u>Bronzes:</u>						
C61300	F	F			G	
C70400	F	F	G		G	In use.
<u>Nickel Alloys:</u>						
N04400	G	G		G	E	
(Mone1 ³)						
625	7	30		E	E	Ferric Chloride
(Inconel ³)						

¹ Trademark of Allegheny Ludlum Steel Corporation.

² Trademark of Colt Industries.

³ Trademark of Huntington Alloys, Inc.

In addition to generic data, there is quite a bit of specific work on the corrosion properties of specific metals in specific applications. Tests performed in LiBr on SEA-CURE, indicated that it performed "as well as Type 316 and better than Type 304" in a five day immersion test, with respect to pitting and crevice corrosion.[5-6]

The bulk of remaining reports deal with sea water exposure, a considerably more hostile environment than LiBr due to the presence of oxygen and of a large and variable number of other chemical agents. Deverell and Maurer showed AL6-X to be superior to 304 or 316 with respect to pitting corrosion in salt water cooled condenser applications [5-7] while Streicher, in a series of tests for Ocean Thermal Energy Conversion (OTEC) applications, found completely ferritic alloys, AL29-4C and Monit, exhibited the greatest resistance to crevice corrosion, the most likely mode of attack in sea water applications.[5-8]

An important lesson from these investigations is that the use of these "superferritic" stainless steels can result in enhanced galvanic attacks on non-ferrous components of the structure. This tendency can be contained by cathodic protection, but if care is not taken, hydrogen may be produced which will lead to embrittlement of the stainless steel.[5-9] In Japan, iron-based sacrificial anode has been similarly used to protect stainless steel from stress corrosion cracking.[5-10] These forms of "electrochemical maintenance" are similar to the chemical maintenance described earlier [5-1] and will be considered in depth in a later phase of this investigation.

5.5 Candidate Metals and Corrosion Inhibitors

The use of corrosion inhibitors in current absorption machines in the U.S. was discussed in Section 2.4; a substantial amount of testing has also been carried out on corrosion inhibition in Japan and the Soviet Union. We here summarise what we know of this work; much of it is available only in Japanese or Russian, and will take some time to obtain and translate.

Reference 5-11 selects optimal pairs from among five metals, including what appear to be super-ferritic stainless steels, and some forty inhibiting compounds, but we have as yet not been able to obtain it. Japanese investigators [5-12] found that a mixture of 0.145M LiOH, 0.03M Na₂B₄O₇ and 0.005M Benzotriazole provided effective corrosion protection for carbon steel, copper and cupronickel in hot LiBr. Other inhibitors recently found effective for mild steel and copper in LiBr were alkali metal tungstates with nitrates or nitrites [5-13], alkali metal vanadates [5-14], chromates [5-15] and molybdate [5-16]. A Russian study examined the value of LiOH and Lithium Chromate for various aluminum alloys as well as carbon steel, but the available abstract [5-17] does not reveal the results. A large number of older references (not included here) cover the effects of various inhibitors on the corrosion of mild steel and duplicate the state of the art discussed in Section 5.4.

It appears that the information in some of these references could be helpful in selecting candidate metals; however, the difficulty in obtaining the documents, and the fact that the particular alloys examined would not be available in the U.S., made it clear that more information on the specific behavior of our candidate alloys (from Table 5-11) in LiBr was necessary for realistic assessment. The experiments we undertook and their results are described in Chapter 6.

5.6 Composite Materials - Polymer Coatings

A third approach to heat exchangers of low material intensity and high corrosion resistance is to use a base metal, such as carbon steel, with a corrosion resistant polymer coating to protect it from the corrosive agents.[5-18] Such a coating can in principle be applied in several ways, including solution casting and electro-deposition. Heat transfer analysis of the sort used in Section 5.3 indicates that the coating should be one mil thick or, preferably, less, however, and this turns out to be a fairly restrictive condition when we also demand that the coating be flawless. Clearly nicks or bubbles will allow the LiBr access to the steel, and the ensuing corrosion will lift the polymer from the rest of the steel.

Most experts consulted agreed that it would be extremely difficult to obtain a flawless one mil coating by solution casting or painting techniques. Apparently tiny inhomogeneities or dirt specks on the surface prevent full coverage unless the coating is very thick. This negative assessment was born out by experiments discussed in Section 6.5 below, and we have accordingly dropped solution casting from further consideration.

This has led us to examine electrocoating techniques.[5-19, 5-20] Here the sample to be coated is immersed in a solution containing the polymer in an ionized state, and a voltage established between the solution and the sample leads polymer ions to deposit on the sample. As they build up an insulating layer, current and polymer deposition is directed to the areas that are not yet coated, resulting in an extremely uniform coating.

Electro-deposition can be carried out at a constant current density or at a constant voltage. When the current is applied, the electrochemical reactions start. In anodic deposition these are:

Anodic:

oxidation of water

dissolution of the substrate

Cathodic:

discharge of water

At constant voltage the anode electrode is completely covered with a film within a very short time. This is caused by the large currents flowing at the beginning of electro-deposition. According to the literature, the peak currents are on the order of 10 to 100 mA/cm², depending upon the applied voltage and the conductivity of the paint dispersions which usually range between 300 to 4000 micromhos/cm. If the film is an electronic insulator, it cannot transport the electrons which are required for the first two reactions shown above. The charge transfer, therefore, can only take place at the metal/film interface. The ions which are formed in these two reactions then carry the

current through the film due to a high electrical field. After passing through the film, the ions react chemically with the carboxylic ions of the polymer dispersed in the bath and give rise to formations of new layers of film. This causes the film thickness to increase. However, as the film thickness increases, the field strength across it decreases and the specific resistivity increases. When the current ceases to flow, the coating process is complete.

It should be noted that with the increase in resistance there is an increase in the film temperature. The point of maximum heating is critical for some coating because at this point the films may rupture. Such ruptures are caused by steam escaping from the water trapped in the film when it heats up and reaches the boiling point. It is obviously desirable to terminate electro-deposition before that point is reached.

When the coating process is complete, the anode (coated piece) is withdrawn and baked to form a durable coating. Sometimes the baking is replaced by induction heating.

The majority of commercial electrocoating processes in the U.S. are based on anodic deposition. However, cathodic deposition should not be overlooked because some coatings, epoxies and urethane-epoxies, deposit at the cathode and are very effective in protecting the substrates from corrosive conditions.

In the cathodic deposition process the metal surface does not ionize and remains passive during the coating operation. This means that no ions enter the coating dispersion even with easily oxidizable substrates. The electrolysis of water at the cathode produces the hydrogen radical, hydrogen, and hydroxyl ion.

Most of the cationic electro-deposition systems which have been described in the literature are based on nitrogen cations. These are protonated amines or quarternary ammonium compounds. Their handling and deposition is quite similar to the anodic deposition, except that the electrodes are reversed and the cathode becomes coated.

The suppliers of electrocoatings rate the materials in terms of throwing power which they relate to deposition efficiency. In more specific terms it is that characteristic which promotes uniform deposition of the film thickness at varying cathode-to-anode distances on the same deposition electrode. The resins which are employed in this anodic deposition process usually have pendant and/or terminal carboxylic acid groups on the polymeric chains which are neutralized with ammonia, amines, or alkanolamines, and are ionic and colloidal in nature. Sometimes surface active agents are also present in the coating materials.

During the anodic electrocoating process, the cations, i.e. ammonia, amines, or alkanolamines used to neutralize the resins are discharged at the cathode where they are redissolved as free bases. This causes a rise in the alkalinity of the coating bath as the process of electro-deposition proceeds. Many commercial coatings are specifically designed to compensate for this to improve stability and throwing power.

Among the commercial anodic electrocoating materials there are acrylic, epoxy, styrene copolymers, polyurethane, polybutadiene, and oleo-resinous systems. As mentioned above, these resins must have acid functionality which is usually through a carboxyl group. This functional group is introduced into the resins in a variety of ways. Most acrylic coatings are formed by copolymerization of various monomers with either acrylic or methacrylic acids. The other monomers are picked for their ability to contribute to the physical properties of the finished film or to crosslinking. Thus methyl methacrylate might be included to give hardness or toughness to the film, while butyl acrylate might be included to plasticize the film and give it flexibility. A third monomer, like N,N-dimethylol acrylamide, may be used for crosslinking.

The epoxy resins are usually partially esterified glycidyl ethers based on 2,2-bis (4-hydroxyphenyl) propane. They contain epoxy groups or hydroxyl groups produced by hydrolytic dissociation of the epoxy groups and residual carboxylic acid which is neutralized with strong nitrogen bases. Variations of that can be formations of adducts of maleic acid or anhydride with fatty acids which are in turn reacted upon in proper proportions by the epoxy resins. As a further variation of this these reaction products of epoxy resins are blended (or precondensed) with phenolic resins, usually prepared from p-ter-butyl phenol and formaldehyde.

Many styrene copolymers are similar to the acrylic coatings except that methyl methacrylate or another methacrylic acid ester, used to give the finished coating hardness, is replaced by styrene. Others are based on copolymers of styrene and maleic anhydride. Low molecular weight copolymers may often be half esterified with a fatty alcohol and then reacted with an epoxy resin to form electro-deposition epoxy esters. Styrene acrylic acid copolymers are also used in a similar manner.

The polyurethanes are usually prepared from low molecular weight reaction products of diisocyanates with glycols which are hydroxy terminated. These polyurethanes are reacted with stoichiometric amounts of polycarboxylic acid anhydrides like trimellitic anhydride to esterify both ends of the chains.

The butadiene resins are usually low molecular weight liquid copolymers containing carboxyl groups. Most of the literature on liquid polybutadiene coatings for electro-deposition appears to come from Japan.

One reported preparation of a cationic epoxy-urethane coating consists of reacting a resinous polyepoxide with less than stoichiometric amounts of a polyisocyanate to form a hydroxy-functional polyurethane. Tertiary amine functionality is then generated by reacting it with a monosecondary amine to

provide dispersability in water with the help of an acid. Prior to use, the resin solution is combined with an aminoplast for crosslinking, like a benzo-guanamine formaldehyde condensate. Good salt resistance results.

A cationic acrylic-urethane resin is formed by preparing an acrylic interpolymer of butyl acrylate, styrene, N,N-dimethylaminoethyl methacrylate and hydroxyethyl acrylate by a typical free-radical chain-growth polymerization procedure. The interpolymer is neutralized by an acid and then reacted with a diisocyanate and after that reacted again with more hydroxyethyl acrylate and ethylhexanol. The product, after electrodeposition and baking, reportedly gives good protection to bonderized steel from corrosive effects of a salt spray.

At a commercial level, we have contacted two manufacturers, Glidden Paint Co. and Pittsburgh Plate Glass, who both offer epoxy coatings for electroplating. In the case of Glidden, the coating has already been used to coat heat exchangers. We are very enthusiastic about the potential of this technology, but unfortunately the establishment of a coating facility was too large an endeavor for the current phase of the project. Appendix 2 describes a coating bath we will propose setting up in the second phase of this effort; in this bath we will test both the commercial and other, custom-prepared polymer coatings, chosen from the possibilities discussed above.

6. TESTING OF CANDIDATE MATERIALS

6.1 Introduction

As indicated in Chapter 5, adequate data does not currently exist on the behavior of either the polymeric or the metallic candidate materials in the LiBr environment to permit definitive choices among the sets of candidates examined. We therefore undertook a series of experimental examinations of the abilities of these materials to withstand corrosion, erosion or degradation due to immersion in LiBr solutions. In this chapter we report the results of these experiments for the materials without regard to questions of fabrication; in Chapter 7 we examine the impact of LiBr immersion on materials when they have been shaped into configurations similar to those that will actually be used to make heat exchangers.

6.2 Alteration of Physical Properties of Polymers

The polymers actually tested are listed in Table 6-1; they were selected on the basis of position in Table 5-4, after making allowances for the availability of samples or other considerations. In some cases we could only find vendors for resins, in pellet form, which we would have had to calender or solution cast into film. In a few other cases we found no vendor who could deliver samples in a suitable time frame. Acetal copolymer was only available in 30 mil samples in the time frame in which we were working. In any event we worked, for this first phase, with the best materials we were able to obtain in the form of suitable films. In later phases we will examine other materials as we obtain samples.

All tests involved a comparison of virgin samples with samples which had been immersed for an extended period in a 100°C (212°F) LiBr solution. The temperatures were monitored by a thermocouple which activated a differential controller driving a heat lamp directed at the insulated beaker. The insulation was attached in such a way as to encourage the formation of convection currents, ensuring a steady stirring action. In some cases the heat lamp was simply powered through a variable transformer and the temperature monitored. The small variations in temperature which ensued did not appear to cause any unusual behavior in the samples. The brines were held near a concentration of 0.62 pounds of LiBr per pound of brine by the addition of distilled water; at these concentrations the brines do not boil and evaporation is slow enough to permit effective concentration management.

Usually the first sign of deterioration or change in appearance of a plastic material is noticed through visual inspection. This is especially true in cases where demanding environmental conditions result in changes in surface appearance such as fading, discoloration, cracking and gloss retention. Physical properties other than appearance which may be important parameters in evaluating the durability of plastic components include hardness and softening point, permeability, density changes and others.

In this laboratory test program, the densities and visual appearance of selected polymeric samples was noted before and after treatment in hot LiBr solution. Data obtained from this study are listed in Table 6-1.

Table 6-1
Densities of Candidate Polymer Films Before and After LiBr Treatment

Material	Thickness (mils)	Density (gm/cc)		Treatment Duration (hrs)
		Untreated	Treated	
ECTFE - Halar	4	1.79	1.88	176
HD Polyethylene	16	1.01	0.95	176
PCTFE - Aclar 22C	2	2.09	2.21	136
FEP - Teflon	2	2.19	2.32	120
PVF - Tedlar	4	2.24	2.17	216

These differences in density are not large enough to be considered significant. Visual and microscopic inspection of these samples also did not reveal any observable differences before and after treatment in the hot LiBr solution.

It should be noted that although no notable difference in physical properties was observed in short term tests, long term tests ranging from six months to a year should be conducted to detect slow chemical attack. Moreover, because of the interaction of exposure time, exposure temperature, reagent concentration and reactivity of individual reagents with different polymeric material, it is often difficult to judge from short-term exposure data the true significance of property and appearance changes, unless they are extremely large. Therefore, long term immersion testing, lasting several months, is being considered for implementation in the next phase of the research program.

6.3 Mechanical Testing of Candidate Polymer Materials

Manufacturer's data on the strength of the candidate materials is adequate to indicate that the virgin materials are sufficiently strong to withstand the two to three atmospheres of pressure they might be subject to in the heat exchanger application. Nonetheless, the polymers were tested on our strain gauge so that a direct comparison could be made with their performance after being bathed in hot LiBr to determine any adverse impact the hot brine might have on the materials.

Tensile strength of the polymeric materials was determined using a modified ASTM E345 test procedure. An Instron tensile strength tester Model 4202 was utilized in this measurement. The tester applies a uniform stress

and retains an indication of the highest stress achieved in the course of the individual measurement. Samples are one half inch wide and shaped as shown in Figure 6-1. After measurement of the highest actual force reading (in pounds) the measurement is converted to pounds per transverse square inch. The highest stress reading is followed either by tearing or by irreversible elongation; either is unacceptable from the point of view of component fabrication. Data on untreated samples and those treated with 212°F LiBr solution are listed in Table 6-2.

Figure 6-1. The shape of samples undergoing stress testing.

An examination of the results obtained in the tensile strength tests indicates that Aclar 22C possesses the highest tensile strength; however, it also demonstrated the largest decrease in tensile strength, approximately 6% after immersion into hot LiBr solution. Halar, a copolymer of Aclar and Ethylene, has the next highest tensile strength with no significant difference in tensile strength between the treated and virgin film. Comparison of our results with those published in reference 5-2 indicates substantial differences in tensile or yield strength. Thus the latter should not be used to predict the strength of particular samples or production runs, and the ratings displayed in Table 6-2 should be used as a guide, but not as a firm constraint.

6.4 Assessment of Polymer Chemical Resistance by Infrared Spectroscopy

In addition to measuring physical and mechanical properties, it is also possible to measure chemical properties and attempt to relate them to service life. Aging in polymeric materials is normally attributed to chemical degradation mechanisms, and by measuring the rate of the degradation reaction one can then extrapolate the results to estimate long-term performance. Thus measures of molecular and microscopic level changes, both of which occur early in the degradative process than macroscopic changes, are quite sensitive early predictors of degradation.

Table 6-2
Tensile Strength Tests for Selected Polymeric Materials

Material	Thickness (mils)	Tensile Strength (lbs/in ²)		Published
		Untreated	Treated	
ECTFE - Halar	4	4646	4557	7000
	4	4735	4778	
	4	4942	4513	
HD Polyethylene	16	3965	4192	2400
	16	4334	4059	
	16	4539	4092	
PCTFE - Aclar 22C	2	6843	6049	5000
	2	6182	6347	
	2	6446	5818	
FEP Teflon	2	2540	2248	2500
	2	2459	2480	
	2	2377	2325	
PVF - Tedlar	4	6550	6775	7000
	4	7150	6900	

Infrared (IR) spectroscopy is one analytic method suitable for detecting molecular degradation in polymeric material. Absorption of electromagnetic radiation takes place for frequencies (or, equivalently, wavelengths) which are resonant with vibrations in the molecular groups that make up the polymer. Changes in the frequency of these resonant absorption "lines" indicate variations in the chemical bonds holding the molecular groups together in the larger polymer molecules. IR spectra of the selected polymer films measured before and after treatment in hot LiBr solution are shown in Figures 6-2 through 6-5. Without exception, no difference could be found in the frequency (measured in wave numbers, the inverse of wavelength) of absorption lines and bands between treated and untreated polymer films, indicating that no change occurred in the chemical structure of the polymers. In all the figures, the intensity of the transmitted beam was lowered for the after-treatment spectrum to allow the two spectra to be distinguished. After allowing for this, their remain changes in the relative magnitude of the absorption bands in the treated Aclar 22C sample which suggest some physical change such as swelling may have occurred in the polymer material during the hot LiBr treatment.

Figure 6-2. Comparison of infrared spectra of Halar (A) before and (B) after treatment with 100°C LiBr solution for 136 hours.

Figure 6-3. Comparison of infrared spectra of high density polyethylene (A) before and (B) after treatment with 100°C LiBr solution for 176 hours.

Figure 6-4. Comparison of infrared spectra of Aclar 22C (A) before and (B) after treatment with 100°C LiBr solution for 136 hours.

Figure 6-5. Comparison of infrared spectra of Teflon (A) before and (B) after treatment with 100°C LiBr solution for 120 hours.

6.5 Tensile Testing of Candidate Metal Foils

To determine the relative strengths of the candidate materials in foil form, tensile testing was performed on the following stainless steel alloys: 304, 316, S31803, Sea Cure, 29-4C, AL6X and AL6X+N. In addition, 3003 aluminum, CDA 614 bronze, and M-400 Monel were also tested. The tensile tests were performed to ascertain tensile strengths of the materials in thin foil form and provide a basis for comparing weld seam strengths with base material strengths on specimens with identical preparation histories.

A number of specimens of each material were purchased from Metal Samples Corporation. All were in the form of 3 in. long by 1/2 in. wide strips, 13, 14, or 20 gauge. One strip of each material except the 29-4C alloy was cold rolled to 0.003 in. thickness at BNL. The 29-4C was cold rolled to 0.005 in., the minimum thickness recommended by the producer. All of the newly rolled foils were predictably hard, and the 304 stainless steel and S31803 foils were deeply notched along the edges. Since better welds usually follow reductions in the hardness and brittleness of specimens, we decided to anneal the stainless steel specimens.

Our initial attempt at annealing the stainless steel foils was unsuccessful. The oven temperature was set to the annealing temperature specified by the manufacturer for each material and the baking was done in an Argon atmosphere. Apparently either the foils overheated or the Argon was impure, however, because the specimens came out totally blackened and more brittle than ever. Since these specimens were unuseable, we decided to temporarily abandon this treatment and to go ahead with testing the foils "as rolled".

A second set of the stainless steel specimens were rolled into 0.003 in. thick foil (except the 29-4C was rolled to 0.005 in.) and coupons of all the listed candidate materials were prepared for tensile testing.

Three coupons of each material were prepared for tensile testing in shapes comparable to those shown in Figure 6-1. The test section of each coupon was machined to the width listed to eliminate inconsistencies due to edge imperfections which resulted from rolling.

The greatest deviation in pounds pull at failure between coupons of the same material was 6% for the 316 stainless steel. All other deviations were less than 5%. This consistency of readings verifies the procedure. The calculated tensile strengths were much higher than expected for many of the materials, and this is probably due to the extensive cold rolling and lack of annealing. In any event, as shown in Table 6-3, the tensile strengths are all far in excess of our requirements.

6.6 Corrosion Testing of Candidate Metals

Electrochemical corrosion testing is based on the fact that the corrosion of metals involves the oxidation of the metal to its ionic state in solution and, in consequence, the flow of electric current between metal and solution

Table 6-3
Tensile Testing of Candidate Metals

Candidate Material	Test Section Thickness (in.)	Test Section Width (in.)	Average Pull at Failure (lbs)	Tensile Strength (psi)
304 SS	0.003	0.290	241.8	277,988
316 SS	0.003	0.345	246.8	238,487
S31803	0.003	0.290	236.2	271,494
Sea Cure	0.003	0.345	192.0	185,507
29-4C	0.005	0.345	290.4	168,368
AL6X	0.003	0.347	221.5	212,776
AL6X + N	0.003	0.347	256.4	246,301
3003 aluminum	0.003	0.347	13.8	13,256
CDA-614 bronze	0.003	0.347	95.7	91,960
M-400 Monel	0.003	0.347	83.4	80,096

under the influence of electrochemical potentials. Thus measurement of the electrochemical potentials (in particular, of the excess above thermodynamic equilibrium potential) can serve as a measure of the magnitude of the chemical forces driving the corrosive reaction, while the currents serve as indicators of the rate of corrosive activity. Since electric currents and potentials can be measured with extreme sensitivity and accuracy, their detection can serve as an extremely sensitive probe of the presence of corrosive activity, comparable to microscopic examination after weeks or months of exposure in aggressive solutions.

Even more powerful, however, is the technique of initiating corrosion in a controlled manner by regulating the electrochemical potential between the metal sample and the corrosive solution. The experimental set up used to do this is shown in Figure 6-6 and may clarify the activity. A reference electrode measures the potential of the solution within an additive constant; no appreciable current ever flows through the reference electrode. However, the voltage of the solution with respect to that of the metal sample, that is, the voltage across the metal-solution interface where corrosion occurs, may now be measured by measuring V_r , the voltage of the reference electrode with respect to the voltage of the metal sample. If the voltage of another electrode, the "counter" electrode, is varied in a controlled way, an electric current can be made to flow from the metal sample through the solution to the counter electrode. As this current is adjusted by adjusting the voltage of the counter electrode, the voltage between metal sample and the reference electrode can be monitored, revealing variations in the potential difference between the sample and the solution.

Figure 6-6. Electrolysis cell and potentiostat schematic diagram.

In practice, the voltage on the counter electrode, and hence the current, are controlled automatically by an electronic device, the potentiostat, to maintain a pre-set or swept value of V_R . Thus the entire system produces a plot of current as a function of V_R . The counter electrode is a platinum wire, surrounded by a glass sheath to confine any reaction products. The reference electrode is calomel, connected to the solution through a salt bridge.

Recall now that the current flow is a measure of corrosion as electrons flow through the circuit, ions leap from metal to solution or back, and this is the process by which corrosion occurs. Starting with a negative voltage from metal to solution, no current flows and there is no corrosive activity - the sample is protected by a passivating oxide film. If the voltage on the counter electrode is now increased, the voltage from metal to solution (as measured by the voltage from metal to reference electrode) will increase and a point will be reached where a small current starts to flow; breakdown of passivity has been initiated. At this point, a small increase in the metal to solution voltage will bring about a large increase in the current. This behavior is shown for a particular sample, 316 stainless steel at 23°C, in Figure 6-7; spectra are shown for other representative samples in Figures 6-8 through 6-10. The voltage at which significant current appears (taken here as 100 microamperes above the baseline) we will call E_p (E "pitting") as it marks the onset of pitting corrosion.

Figure 6-7. Sample current versus sample-solution voltage for 316 stainless steel at 23°C.

Figure 6-8. Sample current versus sample-solution voltage for 316 stainless steel at 100°C.

Figure 6-9. Sample current versus sample-solution voltage for AL6X at 23°C.

Figure 6-10. Sample current versus sample-solution voltage for Aluminum 3003 at 23°C.

The larger E_p , the more resistant the metal surface is to pitting or other forms of localized corrosion and the less likely it is that corrosion will be initiated. The values of E_p we obtained for various samples are shown in Table 6-4. But more information is available from these curves: when current flow reaches a few milliamperes, the reaction becomes momentarily self sustaining as the pits grow, so that even though the voltage is reduced (the point where the curve starts to bend back, at its rightmost point) the current continues to increase until the voltage has dropped below the point where pitting was initiated. This tendency for the current to continue to grow indicates the autocatalytic nature of these corrosion reactions.

Table 6-4
Pitting Potential of Candidate Metals (mV)

Candidate Material	23°C = 73°F		100°C = 212°F	
	E_p	E_{p2}	E_p	E_{p2}
AL6X	420	40	280	-220
AL6X + N	400	0	120	-160
S31803	80	-140	20	-300
29-4C	100	-80	-60	-220
316 SS	120	-117	-40	-300
SEA CURE	110	-260	-60	-280
M-400 Monel	20	-190	-250	-250
304 SS	-80	-220	-160	-340
CDA-614 bronze	-360	-420	-410	-430
3003 aluminum	500	-560	-820	-820

Furthermore, the spread of pitting corrosion has "activated" the surface; it is now much easier to initiate the breakdown of passivity. In the second sweep (labelled "2"), the pitting potential E_{p2} is substantially below the initial E_p , a pattern repeated for all samples. This secondary pitting potential is a measure of the tendency of the surface to corrode once it has been damaged by a prior bout of corrosion. As such, it serves as an important indicator of long term corrosion resistance, and is also presented in Table 6-4.

After pitting corrosion has occurred, as indicated by high currents on the potentiometer, the surface of the metal is etched and rough, usually only in a few areas. Microscopic examination reveals a multitude of small pits and other complications. For example, in the case of Monel metal, shiny spots were visible which may have been nickel redeposited by a negative voltage after being dissolved out into solution during the pitting episode.

The temperature dependence of the pitting potential is also of great interest. In all cases, raising the temperature lowers E_p , sometimes substantially, corresponding to the increased chemical activity and tendency towards corrosion which generally obtains at higher temperatures. The data for E_p and E_{p2} taken at 100°C (212°F) are presented in Table 6-4 along with those for 23°C (73°F).

All of these tests were performed on small (1 cm x 1 cm) surfaces of foil which were prepared as described in Section 6.5. They were painted with an insulating polymer in such a way that all sharp edges were concealed and the exposed metal surface lay completely below the surface of the solution so that it was not exposed to a solution-air interface. All experiments were carried out in a specially designed test cell using 98% pure LiBr at 62 grams per 100 grams of solution.

The behavior of 3003 Aluminum is sufficiently striking to merit separate discussion. At room temperature it appears to be remarkably resistant to corrosion, with by far the largest value of E_p . A few sharp current spikes below E_p indicate that small breaks in the oxide layer occurred, but immediately repaired themselves. However, at higher temperatures, this alloy's pitting potential appears to be the lowest of the metals tested, so the dependence of 3003 Aluminum's pitting potential on temperature is by far the strongest. If ways are found to exploit its high corrosion resistance in, say, the evaporator, the design must ensure that it can never be heated, even during transient operating or shutdown conditions.

In general, however, the Allegheny Ludlum alloy AL-6X emerged as clearly superior in a LiBr environment on the basis of the tests performed to date. It is perhaps surprising that AL-6XN did not perform better, and continued testing will determine if the particular foil configuration had an influence on the results.

It is also important to note that there is much that we do not understand in these systems. The negative and positive current loops at the low voltage (left hand side) of the spectra indicate electrochemical reactions probably associated with redeposition of metal ions from the solution, which could strongly affect the longevity of these materials in realistic LiBr environments. Such environments encourage far more complex reactions than we have yet investigated, since ions released in the corrosion of one material can increase the rate of corrosion of other materials, while the presence of corrosion inhibitors may ameliorate the problems, but will certainly make the chemistry more complex. In addition, the behavior of the promising materials in the pollutants, mild acids and other chemicals that occur in system water and in the condensate of flames must be investigated, although tabulated data for the stainless steels and nickel alloys indicates that they should withstand attack in the anticipated environments. In short, substantially more work is needed on the most promising candidates.

6.7 Corrosion Testing of Candidate Composite Materials

We were not able to obtain suitable samples of electro-deposited polymer coatings within the time constraints of this phase of the work. We did obtain three samples of solution coated steel foil, nominally of coatings of various thicknesses, from the Polymer Studies Group at BNL. The appearance of these samples was so poor, however, pockmarked and of non-uniform thickness, that it was apparent this route could not provide the level of protection we needed. This evaluation was born out in conversations with members of the Polymer Studies Group, so we performed no further tests on composite materials in this phase, and have ruled out solution-cast coatings as a candidate.

7. TESTING OF CANDIDATE MATERIALS IN FUNCTIONAL CONFIGURATIONS

7.1 Introduction

In this section we report the fabrication of miniature test structures from the various candidate materials, the tests to which these structures were subjected and the results of the tests. The miniature test structures are designed to duplicate the stresses and conditions that would be experienced by the materials in heat exchangers, but to do it on a scale amenable to bench top experiments and without the complex topological constraints required of actual heat exchangers. The tests are often duplicates of the materials tests described in Chapter 5, since the methods of joining the materials to make components - usually welding - can have a deleterious effect on the material properties and enhance corrosion or produce a weak area in the sample.

Unfortunately it was not possible to fabricate miniature test structures from the composite materials. Because the polymer coating must be electroplated on after the component has been fabricated, the plating apparatus must be available on site. While this will not be a problem in later phases of this work, it was not possible to set up a plating bath under the scope of the current contract.

7.2 Testing of Polymer Welds

Bonding techniques lie at the core of component fabrication. At an early stage, we chose melt bonding as the method of choice for all polymers to which it was applicable, due to its simplicity, speed, potential for automation and, in particular, its great strength advantage over almost all adhesives. Bond tests between polymer films were carried out to identify, from among the polymer candidates identified in the materials assessment, those polymers that produced the highest bond strength and that were resistant to delamination and other forms of failure during exposures to 212°F LiBr solutions. No welding or bond tests were carried out for PVF (Tedlar), since the type we had in stock is formulated for adhesive bonding, and melt bonding samples did not arrive in time.

In all cases in this study, linear welds on the polymeric materials were formed by employing a Vertrod Thermal Impulse Heat Sealer, model 14B/BHP-WC. Heat seal temperatures and seal time were adjusted experimentally for each polymer to produce the strongest bond. All bonds were one quarter inch in width; it is probable that much narrower bonds would perform just as well for most materials, and we will experiment further with this in later phases of the project.

Bond strength of these welds was determined using a modified ASTM D903-49 "T" Peel Test procedure. Six sample bonds for each polymer were made. Three of these were then "treated" in hot LiBr solution exactly as described in Section 6.2, for the lengths of time indicated in Table 6-1. Peel tests on all six samples were then conducted to determine the bond strength and

possible degradation of the bond due to treatment in the hot LiBr solution. The peel test results are listed in Table 7-1.

Table 7-1
Peel Tests of Bonds for Selected Polymers

Material	Thickness (mils)	Heat Seal Temperature (°F)	Peel Strength (lb/in.)		Weld Strength/ Material Strength
			Untreated	Treated	
ECTFE - Halar	4	300	17	18	0.92
	4	300	17	16	
	4	300	19	17	
HD Polyethylene	16	250	35	44	0.62
	16	250	31	44	
	16	250	49	47	
PCTFE - Aclar22C	2	300	8.3	4.3	0.40
	2	300	5.5	2.6	
	2	300	4.6	4.5	
FEP - Teflon	2	450	5.7	4.2	1.09
	2	450	6.2	4.6	
	2	450	5.7	5.1	

Visual observation of the weld samples showed that the failure on almost all the welds occurred at the edge of the weld. A microscopic examination of these samples showed a thinning of the material at a location above the weld. With the exception of Aclar 22C, no significant difference was observed in the peel strength of these polymers attributable to the LiBr treatment, and it is not at all clear that the differences in the results for Aclar are significant.

In the last column we compare the strength of the welds (in pounds per inch) with the strength of one inch of the virgin material at the specified thickness, taken from Table 6-2. For the welds and for the virgin materials we took the average of all six tests, since any difference between the treated and untreated samples appears to lack statistical significance. The peculiar result for Teflon is undoubtedly produced by statistical fluctuations; the other results indicate that, as one would expect, the welds will be significantly weaker than the parent material in many cases. This will of course affect design specifications. Unless we find a systematic error in our bonding methods, these results will weigh against the adaptation of Aclar, due to the large design allowances that would be necessary. A tendency of PCTFE to degrade when heated has been noted in the literature.[7-1]

7.3 Fabrication and Testing of Polymer Miniature Test Structures

The burst strength of these polymer films was also determined by measuring the maximum fluid pressure that tubes constructed of polymer film, typically less than 1 inch in diameter by 6 inches long, can withstand before distortion or rupturing. In this test, the rupture occurred at the edge of the polymer film next to the seam weld for all the samples tested with the exception of Halar. Sample test results are summarized in Table 7-2.

Table 7-2
Burst Strength Tests for Selected Polymers

Material	Thickness (mils)	Diameter (in.)	Burst Strength (psi)	Failure Mode	Burst Strength/ Weld Strength
ECTFE - Halar	4	0.39	90	material	1.01
HD Polyethylene	16	0.40	75	weld	0.36
PCTFE - Aclar 22C	2	0.44	10	weld	0.44
FEP - Teflon	2	0.51	10	weld	0.49

The last column compares the strength of these 6 inch long welds to the strength of 1/2 inch welds reported in Table 7-1. These results indicate that it is difficult to get long welds that are uniformly good, and of course the configuration breaks down at the weakest point. That Halar was the only polymer material tested that did not tear at the weld may indicate that it is easier to get strong, uniform welds with this material.

Permeability of the polymer film itself or of the welds to LiBr was also determined. In this test, distilled water was sealed in approximately 1 in. x 1 in. pockets fabricated by heat sealing the polymeric material. These pockets were then immersed in hot LiBr solution and kept there for the same period as for the other treatments, indicated in Table 6-1. Permeability was measured by measuring the concentration of bromine in the distilled water after the test period. The data are listed in Table 7-3.

Pockets made from high density polyethylene and Aclar 22C developed leaks during the immersion and thus permeability data could not be obtained for them. Since the LiBr bath can reach temperatures well above 100°C without boiling, it is possible that these pockets were subject to large internal pressures if the bath took an unnoticed upward temperature excursion.

Permeability test results on Teflon and Halar indicate that Halar and/or the welds on this material were less permeable to LiBr penetration. The reading of less than 100 parts per million is very difficult to differentiate from possible accidental contamination despite our best efforts at cleanliness.

Table 7-3
Permeability Data for Selected Polymeric Materials

Material	Thickness (mils)	Heat Seal Temperature (°F)	Permeability
ECTFE - Halar	4	300	<100 ppm
HD Polyethylene	16	250	Leak
PCTFE - Aclar 22C	2	300	Leak
FEP - Teflon	2	450	16,000 ppm

Note that the Halar used in this study was twice as thick as the Teflon, and would therefore allow less diffusion in any event, although the difference appears large enough to be significant. In any event, we need to follow up by re-testing the materials for which the leaks developed.

Of the polymer materials tested to date, ECTFE-Halar has emerged as the best candidate, especially after reliable welding is taken into account. Halar also only costs about one half of FEP Teflon costs, making it a bargain as well as a good performer. We will, however, continue a testing program for other candidates as samples become available to us.

7.4 Metallic Welds and Tests

The key to the fabrication of components or test structures from the metallic candidates is the ability to weld them. If heavier gauge materials are used - say 50 mils and up, as in standard contemporary machinery - then arc welding or brazing are simple, reliable and well known. When we attempt to weld thinner strip and foil, however, it becomes increasingly difficult to produce reliable bonds or to avoid burning out the material.

These problems had beset the BNL solar collector project, and in the course of that work two solutions were identified: laser welding and ultrasonic welding. Further investigation revealed that laser welding would be extremely expensive, either with the work farmed out or if we attempted to set up a system at BNL. Ultrasonic welding showed almost equal promise, and even had distinct advantages, such as producing almost no alteration to the grain structure of the metal during the welding process. This is important both to maintain strength close to that of the virgin material and because the corrosion resistance properties of many alloys depends strongly on the grain structure imparted during the annealing process.

We therefore had installed an ultrasonic welding rig and had developed experience in using it which was available for this project. The system had been used extensively with copper, and in this project was exercised with aluminum and Monel metal, with the results described below. Unfortunately, a

problem which has seemed close to solution for some time now, but which has not quite been solved continued to impede progress. Certain metals are easily welded with the ultrasonic seam welder, and we have had excellent results in welding all of them. They include copper, brass and aluminum, including each to any of the others. Some other materials are difficult to weld ultrasonically, particularly steel, stainless steel and tin bearing alloys.

The problem is essentially that the welding tip or head must be "incompatible with" and harder than the materials being welded, lest it weld itself to the work. Since the welding tip is made of steel, this has meant that we have been unable to weld stainless steel foils together in a reliable way without soon damaging the welding tip.

Several solutions have been attempted, the most recent being the application of a coating of Tungsten Carbide to the surface of the steel tip by "detonation bonding", a process in which the Union Carbide salespeople were extremely confident. The first round of testing of this new tip occurred in June, and unfortunately it too was a failure. The tip is now back at Union Carbide getting a second generation tungsten carbide surface, which will certainly work better than the current surface, since several specific problems have been identified. However, to date we have not been able to produce satisfactory welds in the stainless steel and bronze foils.

Using our standard weld tip we prepared a number of welded coupons of 3003 Aluminum and Monel 400 for peel testing. After tuning the welding tip on a proxy work piece of the same materials as the real work piece, it is necessary to establish three settings on the welding machine which will produce optimum welds. The settings are tip speed, tip pressure, and power level; and they depend on the materials being welded and the thickness of the piece which the welding tip contacts. These settings must be established by trial and error for specific material combinations and thicknesses, but are thereafter repeatable.

Strips of the 0.531 in. wide, 0.003 in. thick, "as rolled", foil of the 3003 aluminum and M-400 Monel were cut into 2 in. long segments. After tuning and adjusting the welder on remaining scrap, two foil segments per coupon were aligned face-to-face on the welding anvil and welded together across their width near one end. The completed coupons resemble two face-to-face sticks of gum which have been pressed together with a dull knife edge near one end.

Peel testing was accomplished by placing the "legs" of the coupons in the opposing jaws of a tensile testing and pulling them apart. The results are shown in Table 7-4. The last column, indicating the ratio of the pull at failure in pounds per inch for these welds to that of the parent material from Table 6-3 indicates that to date the aluminum is more amenable to our welding than is the Monel.

Two conditions adversely affected the quality of the welds and their apparent strength. The first was lack of enough scrap material to permit making optimum adjustments on the welding machine. The second was the

Table 7-4
Peel Test Results

Material	Lowest Tension to Cause Weld Failure		Average Tension to Cause Weld Failure		Weld Strength/ Material Strength
	(lb)	(lb/in.)	(lb)	(lb/in.)	
3003 aluminum	12.6	23.7	13.5	25.4	0.64
M-400 Monel	18.2	34.3	22.4	42.2	0.18

brittleness of the unannealed coupon materials. (The welds themselves did not pull apart. Instead, the coupons broke adjacent to the welds. This is indicative of excess weld power or of material brittleness.)

It is clear that we need more experience in using our welder on these unfamiliar metals, and in particular that development of a tip for stainless steel is a top priority. Several routes to this goal are currently being pursued, for other research programs as well as this one, and we are confident that one of these will bear fruit in the near future.

8. OPTIONS FOR COST REDUCTION FOR SPECIFIC COMPONENTS

8.1 Introduction

In Chapters 2 and 3 we discussed the characteristics of a current generation of absorption chillers, including estimates of the current costs of components. In Chapter 4 we established cost goals for these components which would enable absorption chillers to offer significant economic advantages compared to compression machinery. In Chapters 5, 6 and 7 we discussed the properties of various materials which made them candidates for the construction of such lower cost components and found several materials which appeared promising. In this section we match these materials with various heat exchangers to suggest specific design directions in each case.

In thinking about new materials and component configurations, it is important to keep several design constraints in mind, including thermal conductivity, ability to operate at the requisite temperature and structural considerations. These have all been discussed in the chapters on material properties, but our design approach with respect to materials strength and component structures needs to be spelled out.

Currently, the thickness of metals used in the various heat exchangers is determined by structural requirements. A horizontal tube bank must be able to support itself and the water it contains, and all systems must be able to withstand pressure differentials of one to a few atmospheres over areas of up to several square feet. Construction of this sort requires the use of heavy materials and, because the stresses at welds and points of levered support can be quite high, promotes stress corrosion, as discussed in Section 2.4.

We attempt to ameliorate these problems in two ways: first, we design components so that the differential pressures are placing as many components as possible under tension. This produces only modest internal strains in the materials, compared to the strains produced in tubesheets or at the ends of other long, horizontal components in current designs. Second, we specifically construct racks which will hold, under tension, components which could not support themselves, such as a polymer heat exchanger of tube bundle or air mattress design. In the end, we hope to be able to produce systems which are strung on frames in the manner of box kites, with only the frame bearing any compressive forces, and in which all barriers to material flow are thin layers of material held in tension by a balance of pressure and support by the frame. In the following sections, we make this philosophy more precise through examples.

Finally, it should be made clear that we are not, in this chapter or this report, designing heat exchangers and demonstrating that specific designs will meet our cost goals. We are assessing feasibility, by which we mean that we will now show that materials, which appear capable of meeting the cost goals of Chapter 4 on the basis of cost data from Chapter 5, can in reality be formed into suitable heat exchangers. Having shown the plausibility of these designs, we will have demonstrated that absorption chillers meeting our cost goals appear feasible.

The conceptual designs we offer also contain numerous cost advantages in addition to permitting lowered material intensity, notably with respect to the consolidation of components; we do not evaluate these savings here, since our designs are at far too preliminary a stage for this to be meaningful. Tables 3-2, 3-4, 3-6 and 3-8 indicate the current costs of components we hope to eliminate or combine, and these numbers may be indicative of potential savings.

8.2 Generators and Condenser

One possibility for this subsystem is to use a series of concentric cylinders for the first and second effect generators, as shown in horizontal cross section in Figure 8-1. This system is in some respects similar to one proposed by Batelle for a two fluid system, here designed for LiBr and with the outermost structure, the condenser, designed to be under tension as discussed above. It is also similar to other "falling film" designs. With this design, natural pressure differentials are tending to maintain the shapes of all the cylindrical components, so they are also kept under tension. The outer shell and condenser have clearly been combined into one structure, with attendant materials and cost savings. The outer shell must of course be sufficiently strong to withstand normal abuse from the environment, but this will still result in a much thinner shell than is currently in use.

Figure 8-1. Generators and condenser assembly - conceptual schematic.

Hot flue gas flows through the innermost cylinder, transferring heat through the pipe wall to the first generator solution, which is flowing vertically down the outside diameter. The liberated water vapor then condenses on the inner diameter of the next concentric cylinder, and transfers its heat through the pipe wall to the second generator solution, which is flowing down the outside diameter. Water vapor from generator 2 condenses on the inner surface of the condenser structure, transferring heat to the condensing water flowing in the outer sheath. As in the Battelle system, each surface with a falling film of water employs special fin arrangements to control the fluid flow and to enhance heat transfer.

Simplicity of construction, ease of manufacture, and ease of controlling part load operation are some of the advantages of this concept. A potential disadvantage is the heat exchange areas are somewhat specified by the concentric nature of the design. However, this does not seem to be a major problem for chillers. As an example, consider the 25 ton system with the heat exchange areas given in Table 3-4. Assuming five modular units and a length of four feet, we find that the inner cylinder has a diameter of 9.5 inches, the next cylinder 12.4 and the condensing surface an average diameter of 14.3 inches. The exact configuration of the outer surface is a future design choice involving the size and shape of the frame, the curvature of the tensioned surface and the internal stress in the surface material.

It may be possible to substantially reduce the cost of the modules by employing either thin metal foils or polymer coated foils for the cylinders. In addition, heat transfer enhancing surfaces may be stamped directly onto the surface at low cost. The first effect generator surface, with a diameter of 9.5 inches, will have tearing forces on the order of 70 lb/in.; examination of Tables 5-5 and 6-3 will indicate that this can easily be tolerated by 3 mil foils of AL-6X or the other stainless steels, that the Monel or bronze could tolerate these forces at 10 mils or so and that only the aluminum, already ruled out for the generator by its poor performance at elevated temperatures, would have to be made quite thick in this application.

The forces on the intermediate surface are quite small, corresponding to a pressure difference of some 60 mm (1.2 psi). The forces on the outer surface depend on the selected geometry; they are maximal where the sheet touches the frame, and become greater, for a given circumference, as the frame is made larger, drawing the surface more taut. For the minimum force case, with a one atmosphere pressure differential, we find an internal force of about 100 lb/in., still well within the reach of modest thickness of most metals (again, aluminum may also be unsuitable depending on its corrosion susceptibility at 100°F). Note that we have indicated an "air mattress" configuration for a second layer of material on the outside, so that system water for cooling the condenser flows through the flattened tubes. (The actual geometry of the tubes will depend on the pressure in the system water.)

The innermost cylinder receives heat from the high temperature flue gasses and is cooled by the generator solution on the outer diameter. Potential stress problems may arise if the normal heat transfer to the solution is

locally reduced, resulting in hot spots caused by an unwetted patch on the solution side. The problem is alleviated by ensuring that full wetting always occurs, or by allowing for sufficient lateral heat transfer in the foil. Based on simple thermal stress and heat transfer calculations we estimate in Appendix 3 that a thickness of at least 6 mils is required for safe operation of a stainless steel foil cylinder with a nonwetted area of one square inch. While this suggests that thin foils may be practical, refined calculations and prototype testing are clearly required.

Many issues remain to be addressed, such as the use of polymer coatings to protect mild steel surfaces from system water, other aspects of the choice of metals and the possibility of using other designs, such as some discussed below, for some parts of the generators/condenser system. However, a glance at Tables 4-5 and 5-5 will show that the use of metals at these thicknesses will certainly enable us to meet or exceed our cost goals.

8.3 Evaporator and Absorber

The evaporator and generator can be combined in a structure similar to the one outlined in Section 8.2, or constructed as more conventional heat exchangers inside a standard shell. The primary disadvantage to the concentric structure is that a substantial amount of space within the inner cylinder is not used; recuperators can be designed to fit within it, but are too small to result in efficient use of space by themselves.

It may also be possible to design systems where the generators and condenser are inside and concentric with the evaporator and absorber. A horizontal cross section of such a system is indicated schematically in Figure 8-2. In this case the absorber would form the inner wall of the outer pair, so that cooling water would serve the condenser and absorber simultaneously by flowing in the intervening wall. Then only the outermost wall would need to be supported, since all other pressures (and temperatures) decrease from the center of the system out. Unfortunately, this design results in a condenser and evaporator which are both oversized by about a factor of two if the absorber area is held constant (see Table 3-4), so a substantial amount of assessment must be done to determine the economic feasibility of this particular design. G. Vliet and colleagues [8-1, 8-2] have examined optimal heat exchanger sizing and found current practice to diverge significantly from the optimal. The impact of cooling the absorber and condenser in parallel must also be evaluated.

More conventional designs are also possible, based on a large cylindrical containment vessel as in the designs in Chapter 3, with a variety of heat exchanger designs within it. One possibility is evaporators and condensers made of thin wall metal tubing, but this would continue to require expensive brazing techniques during assembly. More exotic possibilities include air mattress designs of either polymer or polymer-coated metal materials or polymer tube bundles. In these designs the system water flows in the passages which are distended by pressure as is an air mattress; they can be quickly and easily assembled using heat sealing techniques for polymers or ultrasonic

Figure 8-2. Concentric, tensioned absorption chiller - conceptual schematic.

welding for metals. Alternating sheets of evaporator and condenser material would provide a more direct path for water vapor than is found in conventional designs.

Tube bundles can also be made of polymers. Du Pont has recently been marketing a heat exchanger made of flexible bundles of 0.1 inch diameter Teflon tubes with 10 mil walls. Although many design changes should be considered, this route may also be of interest. The current Du Pont systems will inhibit heat flow and increase the temperature differential across the heat exchanger due to their thick walls, and our assessment indicates that on the basis of thermal conductivity and yield strength, Teflon FEP is far from an optimal choice for this application. The Du Pont system is also too expensive for our applications as currently marketed. Nonetheless, the thermal resistance can be compensated for by squeezing more heat transfer area into the system, which is accomplished by using many small tubes, a configuration which has larger surface to volume ratios.

However, as heat transfer area is increased, so is the pumping energy. In Appendix 4 we address this problem and show that for a constant volume of system water inside the chiller vessel, the heat transfer per degree scales with tube radius to the -0.8 power, while pumping backpressure increases with radius to the 1.25 power, leading to an increase in parasitic power consumption. These considerations apply not only to the tube bundle approach, but also to optimal sizing of the tubes in the air mattress design, whether of metal or polymer. It is this tradeoff, where the tensile strength to withstand one to three atmospheres of pressure determines the thickness of material required, thus establishing heat transfer capacity per square foot, in turn establishing the required heat transfer area and hence the pumping power, that will determine the suitability of polymers to the evaporator and condenser. At this point, we have certainly not ruled the possibility out.

Evaporators run at low temperatures, around 50°F, so if the tensile forces are not too great, Aluminum 3003 would appear to be a potential candidate. If the system water seems capable of corroding this material, a polymer coating on that side would still result in a less expensive material than the copper now in wide use. The excellent ultrasonic welds we were able to make with the aluminum alloy indicate that there would be little trouble forming it into convenient structures, such as air mattresses suspended on frames. If strength is a problem, mild steel foil with a polymer coating against the system water is another possibility. Designs which would insure that Lithium Bromide solution could not splash on the evaporator would also be of value.

8.4 Recuperators

Current designs for recuperators are not expensive, involving mild steel in flat plate configurations. It is quite possible that the thickness of the steel in these devices could be reduced considerably, but the savings would not be large. There are, moreover, disadvantages to mild steel, discussed in Section 8.5, and selected polymer designs may be preferable. Two sorts of designs appear promising; what we call a "multiple air mattress" configuration and tube bundles.

The multiple air mattress configuration is shown in cross section in Figure 8-3 for a counterflow recuperator. Layers can be added to such a structure indefinitely, and the longitudinal welds can be produced from rolls continuously. If the entire system is designed so that the recuperators are within the evacuated vessel, then the pressure stresses are small (1 or 2 psi) and quite thin films can be used, allowing heat exchange areas comparable to current designs.

Figure 8-3. Multiple air mattress counterflow heat exchanger (F1 = Fluid 1, flowing into page, F2 = Fluid 2, flowing out of page).

Situating the recuperators within the vacuum vessel is also important because although there are many polymers which will not degrade at the temperatures involved here (up to 300°F), most polymers do suffer from substantial reductions in yield strength at these temperatures, and careful design and testing will be necessary to ensure long term structural stability. That this can be done seems certain; on the basis of material cost and the maintenance considerations raised below it seems likely that it will be worthwhile if suitable fabrication techniques can be identified.

8.5 Maintenance Considerations

The tables of Chapter 3 indicate that mild steel is used in all components of the system which do not come in contact with system water, cooling tower water or the gas flame. As discussed in Chapter 2, although the

mild steel can be oxidized by the brine, the presence of corrosion inhibitors seems to encourage the formation of a less permeable layer of oxide or hydroxide on the steel surface, leading to a gradual lessening of corrosion over a period of a few years. Operationally, this is observed in the evolution of hydrogen, a non-condensable gas, within the evacuated vessels. This hydrogen is pumped out at regular intervals, as part of routine maintenance, and as the machine ages, the amount of hydrogen evolved decreases essentially to zero over a period of a few years. This seems to be the major corrosion process attributable to the brine, and it is being brought more and more under control as increasingly effective corrosion inhibitors are developed.

The presence of the non-condensable gasses does lessen the system efficiency, however, and pumping it out is a maintenance procedure involving significant, although not overwhelming, expense; larger systems have installed pumps, but savings would be incurred if the need for the pumps could be obviated. Through the use of all polymer heat exchangers, polymer coatings on the inside of pressure vessels, the replacement of pressure vessels by active components as discussed above and comparable steps to minimize or eliminate the use of mild steel, substantial savings in maintenance may be possible.

8.6 Options for Post-Development Cost Reduction

Component product cost is affected by a number of factors that include raw material cost, production cost, tooling cost, warehousing cost, and others. To achieve the maximum cost reduction requires addressing all these fundamental factors and their interrelationships so that the total cost is uniformly reduced. Additionally, simplification of component design contributes to cost reduction by reducing manufacturing requirements and improving production yields.

Other opportunities for cost reduction occur with the creation of subtler designs that allow a material to serve more than one purpose thereby obtaining more utility from a given material. For example, a material might be used as a thermal subcomponent as well as a structural subcomponent, eliminating some materials and simplifying construction. Cost reduction potential also exists in choosing new materials that improve reliability and durability of the product by reducing maintenance and repair cost. Some of the polymer systems under investigation here may offer such an opportunity. This approach is in use in the electric vapor compression cooling industry as hermetic compressor designs become common in increasingly larger machines. Such systems have become standards of reliability and durability for the cooling industry.

Once an advanced design is established, increased production rates also reduce cost by amortizing development and tooling investments over more units. This is often accomplished by improved methods of mass production including automation. Bold and aggressive long term planning is necessary and could be inspired by the potential of new low cost designs. It is unlikely that a conservative approach to marketing can reclaim the initiative now held by the compression chiller industry. In particular, dealer and distributor markups are now very high, due in part to low sales volume. Reducing these

incremental cost margins of bringing a gas fired absorption product to the customer can improve the market position of absorption machines significantly. Marketing of packaged, modular systems of flexible layout requirements and simple installation and set up procedures can help greatly, as can the use of either centralized national marketing or piggy-back distribution, where the new chiller systems are associated with a well established distribution network for a compatible product. These and similar considerations will merit thought at a much later stage in this effort.

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Appendix 1

Design Procedures for Directly Fired Double Effect LiBr Chillers.

The chillers described by the matrices of Chapter 3 were designed by traditional and widely accepted principles. An introduction may be found in reference 3-1 or in engineering texts. In this appendix we summarize the actual procedures used in developing the "Functional Requirements" matrices so that the results may be extended to other circumstances if desired. We start with the requirements that our chilled water temperature places on the evaporator, derive the absorber operating characteristics from this, and then proceed to the generator section.

Using the pressure-temperature-concentration (P-T-X) chart after assuming a 3°F approach to the leaving chilled water, a 41°F refrigerant vaporization is selected. Thus the evaporator must operate at 6.5 mmHg.

As some pressure drop is required to cause vapor flow, a pressure less than 6.5 mmHg is required. For the initial selection, pick 6.0 mmHg. The absorber must operate at this pressure, a function of the solution concentration and its temperature. As one parameter is 85°F cooling water, and a 12°F approach is assumed for the leaving solution, then 97°F is determined. From the P-T-X chart we select, initially, 57% solution.

As the cooling water will rise from its 85°F entering condition due to heat rejection, from the absorber, we initially select 100°F as the condensing temperature for the refrigerant.

We want a double effect cycle as opposed to single effect, so we initially assume an equal concentration change in both the first effect generator and the second effect generator. As the concentration out of the second effect generator flows to the absorber via a recuperative heat exchanger it is desirable to have its vapor pressure very near that of the leaving solution's 6.0 mmHg. Low concentration changes inherently cause high thermal losses, while high concentration changes require low flows that make solution distribution more difficult, such as in the absorber and, in the case of direct fired units, in the very hot first effect generator. Initially select a 5% change, so the strong solution returns at 57 + 5 or 62%. From the P-T-X chart, this solution must enter the absorber at 117°F. This allows a 20°F approach at the recuperative heat exchanger. Note that crystallization occurs at 65.5% and 130°F.

Using the P-T-X chart, the 62% solution at 192°F has a boiling point equivalent to water at 103°F, 58 mmHg. With the initial condensing temperature selection of 100°F, the 103°F allows 3°F equivalent pressure drop to move the vapor from the generator to the condenser. As a first approximation we selected a 5% change equally divided, 2.5% change in each generator.

From the above paragraph, the solution entering the second effect generator must be 59.5%. At 58 mmHg, this solution starts boiling at 183°F. To cause boiling in the second effect generator its energy supply must be above the 183°F and 192°F. Therefore, the first effect vapor, steam, must be above 192°F.

As the vapor from the first effect must be above 192°F we estimate a 13°F differential driving force arriving at 205°F vapor required as a first approximation. As the second effect generator is also the first effect condenser, this 205°F saturated vapor, 670 mmHg, must be delivered after flow pressure drops occurring in the piping and separator imposed between the first effect generator and condenser. (A "solution-vapor" separator is required because bubbles of low specific volume are easily entrained in the "frothy" solution.) Note that any Li-Br solution carried over with the refrigerant vapor creates suppression of the refrigerant's boiling point in the evaporator. Also, any refrigerant bubbles that stay in solution will condense as the solution passes through the high temperature recuperative heat exchanger.

Allowing a 50 mmHg pressure loss from the first effect generator to the first effect condenser, a pressure of 725 mmHg is established as the first effect generator's boiling point, 210°F. With 57% entering, boiling begins at 295°F and, at 59.5%, the intermediate 307°F solution temperature leaving the first effect generator. The 307°F stream passes through the hot recuperative heat exchanger dropping its temperature, ideally, to 183°F, entering the second effect generator. The 307 to 183°F drop is transferred to the weak 57% stream flowing to the first effect generator, heating this stream. As the 57% mass flow is greater than the 59.5% stream's and the streams have different thermal properties some of the gas energy supply is used to preheat this stream.

Now a series of iterative calculations must be made to balance the heat transfer requirements of the second effect generator to that available to completely condense the first effect vapor supplied plus its superheat. The calculations should proceed in slight increments of a larger change in the intermediate solution's concentration and an equal decrement in the strong solution's concentration. We assume that the entering intermediate solution temperature is at its boiling point, and does not have to be preheated or cooled at 53 mmHg, the first arbitrarily selected starting point.

When a thermal balance has been achieved by this iteration, the entire cycle can be calculated using the thermal properties of LiBr solutions with the solution's mass flows determined from the 100 ton refrigerant load, vaporized at 41°F, plus the flash loss from the condenser at the first selection of 100°F.

Once the cycle is completely calculated the initial selection of a 57% solution and the total concentration change can be reselected to further optimize conditions. Note that in both recuperative heat exchangers the more concentrated of the streams establishes the heat transfer available.

After the cycle calculations have been completed and the net COP determined, the gross COP, the ratio of the evaporator work to that of the input to the first effect, in common units of Btu/hr, is calculated on the basis of a 3% jacket loss and 78% combustion to thermal conversion, realized with a flue gas temperature of 450°F and 7.5% CO₂. (As the solution entering the first effect generator will need to be preheated it could be put through a flue gas exchanger to improve the cycle efficiency. However, the added complexity and costs associated are open to question. Efforts to decrease the 450°F flue temperature may be a better option.

Appendix 2

A Laboratory Polymer Electroplating Facility

The laboratory equipment, which can be used for both anodic and cathodic depositions, usually consists of a small tank for holding the coating dispersion. The tank must be large enough to allow the piece, or a number of pieces, to be submersed approximately 6 inches below the coating surface with 6 inches clearance on all sides. The walls of the tank are sometimes used as one of the electrodes. More often, however, the tank walls are lined with an electrically insulated coat, or made from an insulating material like glass or plastic, while the counter electrodes (often made out of graphite) are inserted in the coating liquid and positioned according to the size or shape of the work pieces. The work pieces, of course, are connected to the source of the opposite current and act as the opposite electrodes.

Practically all the electrical input is converted into heat which should be removed by cooling to maintain constant bath temperature, usually at 20 to 30°C. To accomplish this the coating tank, if small enough, can be positioned inside a larger constant temperature bath or it can be equipped with a water circulating cooling coil. The temperature and the rate of circulation of the cooling water should be controlled.

Current consumption ranges from about 15 coulombs/gram of finished coat up to 150 coulombs/gram, resulting in an overall requirement of 2 to 4 ma/cm² for 1 to 30 minutes. Electrocoating baths are usually operated between 200 to 400 volts. However, a d.c. power supply which operates at a maximum of 1000 volts and 25 amps is often used. Proper safety precautions must, of course, be maintained.

It is desirable to agitate the coating material by either mechanical stirring or pumping. The rate of agitation should be such as to cycle the entire tank volume in 6 to 30 minutes. If the concentrations of the non-volatiles in the coating dispersions are low, the rate of agitation and the pumping stability are less critical. For that reason most baths are maintained at 5 to 15% dispersion solids.

Figure A2-1 illustrates the coating process and equipment.

The general practice is to clean and treat metal surfaces before applying industrial paint finishes. This can include metal preparation and formation of a surface conversion coating prior to painting. However, if a material like steel foil, for instance, already has a thin tin coat then putting a conversion coating, like zinc phosphate, over it would not be practical. However, there may still be dust on the surface and perhaps rolling oils and/or some microscopic puddles of an organic lubricant, depending upon the fabrication of the foil. The surface impurities interfere with adhesion of the electrocoat and are best removed. Cathodic cleaning of the surface does a thorough job of removing the impurities. When such cleaning is not practical, however, it might be replaced by a thorough rinse. Usually the rinse consists

Figure A2-1. Illustration of the coating bath.

of a wash with a heavy duty alkaline cleaner followed by rinsing with ion-free water. A formulation is presented below of a typical heavy duty alkaline cleaner:

- 10 to 20% sodium carbonate
- 15 to 25% tetrasodium pyrophosphate
- 50 to 70% sodium metasilicate
- 2 to 8% non-ionic, low foaming, surfactant system

A cleaner like the one shown above would be used at concentration of 3.5 to 15 gm/liter in ion-free water at 65 to 75°C. The rinsing in such a solution can last 1 to 1-1/2 minutes. This same cleaning solution can also be used for aluminum surfaces, though sometimes a specially formulated sodium hydroxide etch is preferable. There is no practical reason to dry the metal surfaces after the cleaning rinse and wet pieces can be placed directly into the electrocoating bath.

Appendix 3
Calculation of Burnout Potential for the First Effect Generator Cylinder

In this appendix we estimate the maximum thermal stress due to a hot spot formed in the foil cylinder of the first effect generator. The incident heat flux from the flame is q_1 , and for a region of characteristic length, $2a$, no normal heat transfer to the generator solution is allowed. For the remaining surface, the outward heat flux is q_2 . In steady state with no hot spots, or end effects, $q_1 = q_2$.

A differential element of the cylinder is:

A heat balance gives:

$$t q_{x+dx} - q_x t + q_2 dx - q_1 dx = 0$$

or

$$t \frac{dq}{dx} + q_2 - q_1 = 0$$

Since $q = -k \frac{dT}{dx}$, $q_2 = h_2(T - T_s)$, $q_1 = h_1(T_f - T)$ where k is the thermal conductivity, h_1 and h_2 are film coefficients, or

$$\frac{d^2 T}{dx^2} - \frac{h_2 + h_1}{kt} T = - \frac{h_2 T_s + h_1 T_f}{kt}$$

Note as $x \rightarrow \infty$, the solution is:

$$T = \frac{h_2 T_s + h_1 T_f}{h_2 + h_1}$$

Let

$$\theta = T - \frac{h_2 T_s + h_1 T_f}{h_2 + h_1}$$

Then

$$\frac{d^2 \theta}{dx^2} - \gamma \theta = 0, \quad \gamma = \frac{h_2 + h_1}{kt}$$

as $x \rightarrow \infty$, $\theta \rightarrow 0$

$$x \rightarrow \infty, \text{ tk } \frac{dT}{dx} = -q_1 a \quad \text{or} \quad \frac{d\theta(0)}{dx} = -\frac{q_1 a}{kt}$$

The solution is:

$$\theta = \frac{q_1 a}{\sqrt{\gamma} kt} e^{-\sqrt{\gamma} x}$$

or

$$T = \frac{h_2 T_s + h_1 T_f}{h_2 + h_1} + \frac{q_1 a}{\sqrt{\gamma} kt} e^{-\sqrt{\gamma} x}$$

The max stress, σ , is nearly

$$\sigma \approx \alpha E \theta = \alpha E \frac{q_1 a}{\sqrt{\gamma} kt} e^{-\sqrt{\gamma} x} \Big|_{x=0}$$

where α = coefficient of thermal expansion and E = modulus of elasticity, or:

$$\sigma \approx \frac{\alpha E q_1 a}{\sqrt{\gamma} k t} = \frac{\alpha E q_1 a}{\sqrt{(h_1 + h_2) k t}}$$

We can now estimate the minimum allowable thickness of metal that will keep the maximum stress below the yield point. From the 25 ton absorption data (Table 3-3), $q_1 = 4500 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$. Using stainless steel, with $E = 28 \times 10^6 \text{ psi}$, $\alpha = 10^{-5}\text{ }^\circ\text{F}^{-1}$, $k = 10 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$, $\sigma_{\text{yield}} = 78,000 \text{ psi}$ and taking $h_2 = 400 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$, $h_1 \ll h_2$, and $a = 1/2 \text{ inch}$, gives a minimum thickness of 1.4 mils. Reliability, credibility and corrosion resistance will lead us to use foils or strips of considerably greater thickness, so it is unlikely that isolated hot spots will present a major problem.

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Appendix 4

The Influence of Tube Size on Heat Transfer and Pressure Drop

In this appendix we consider the advantages and disadvantages of the number and size of heat transfer tubes that are used in a heat exchanger of a given overall volume. Specifically we will estimate the dependence of heat transfer and pressure drop on tube radius, holding the overall heat exchanger volume constant. As a further restriction, the cross sectional flow area inside the tubes is set equal to the flow area outside the tubes. That is, each fluid sees the same average area for flow. For this estimate, phase and compositional changes are neglected, making the results strictly applicable to the recuperators only, although the insights from the calculations will be useful in considering the other heat exchangers also.

Consider a heat exchanger of square cross section with cross sectional area L^2 and a unit length. If the flow areas are equal for each fluid, the number of tubes of inner diameter r_i and outer diameter r_o that can fit into the area is:

$$N = \frac{L^2}{\pi(r_o^2 + r_i^2)} \quad (A4-1)$$

Note that if $r_o \approx r_i$, the spacing between tube centers is $\sqrt{2}\pi r_o$. We will estimate the heat transfer rate per degree on the inner tube surface, which is the heat transfer coefficient, h , times the total surface area on the inside of the tube. Then we assume that the film coefficient on the outside of the tube is nearly the same. The total resistance to heat transfer is therefore twice the film coefficient plus the tube wall resistance.

The area on the inside of the tube is

$$A_i = 2\pi N r_i = \frac{2r_i L^2}{r_o^2 + r_i^2} \quad (A4-2)$$

The heat transfer coefficient can be estimated from the well known relationship of Dittus and Boelter for fully developed turbulent flow in smooth tubes:

$$h_i = 0.023 \frac{k}{r_i} \left(\frac{2Vr_i}{v} \right)^{0.8} Pr^{0.4} \quad (A4-3)$$

where k is the thermal conductivity, V is the fluid velocity v is the kinematic viscosity and Pr is the Prantl number. The velocity, V , is simply the flow rate, Q , divided by the flow area, or:

$$V = \frac{Q}{\pi r_i^2 N} = \frac{Q}{L^2} [r_o^2/r_i^2 + 1] \quad (A4-4)$$

For thin wall tubes with r_o approximately r_i , V is nearly independent of r_i . Substituting into Eq. (8-3) gives:

$$h_i = \frac{0.023K}{r_i} \left[\frac{2Q}{L^2} \left(\frac{r_o^2}{r_i^2} + 1 \right) \frac{r_i}{v} \right]^{0.8} Pr^{0.4} \approx r_i^{-0.2} \quad (A4-5)$$

showing that h_i is only weakly dependent on r_i . The heat transfer per degree is then $h_i A_i$, or:

$$h_i A_i \approx r_i^{-0.2} r_i^{-1} = r_i^{-0.8}$$

So the heat transfer per degree scales with $r^{-0.8}$. The pressure drop, ΔP , is:

$$\Delta P = f \frac{L}{2r_i} \rho \frac{N^2}{2g_c}$$

where, for turbulent flow

$$f = \frac{0.266}{Re_{r_i}^{0.14}} \text{ or } 0.266 \left(\frac{v}{r_i V} \right)^{1/4}$$

So, the pressure drop scales with $r^{5/4}$, if ρ and U are constant.

As an example, if the tube size is reduced to 0.1 inch in diameter from one inch (a typically used value) and we neglect the effect of wall thickness, the heat transfer per degree on the inside wall increases by a factor of 6.3, but the pressure drop is increased by a factor of 17.8. Consequently, the use of many small tubes can increase the heat transfer, but the parasitic losses rise substantially and the optimum tube diameter is one which balances these competing effects.

Appendix 5
BASIC Program for Breakeven Costs
(Cromemco Structured Basic)

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100 *Datain
105  @" 400 Ton Absorption Chiller Breakeven Costs"
110  Rem:
115  Rem: Absorption System Characteristics
125  Rem- Copa =Net Design Point COP of absorption system
127  Seas=0.93 : Rem- Seasonal COP/Design Point COP
130  Ela=25+65 : Rem- kW to absorption unit, parasitics+cooling tower
135  Rem- Bec : Rem- Breakeven capital cost of absorption system
140  Kctf=46300.0 : Rem- Capital for cooling tower if COP=1
145  Kctv=33.6 : Rem- Variable $/thermal ton if COP<>1
150  Kiaf=43400.0 : Rem - Installation cost ($) for absorption system.
155  Cog=5.7/1.026 : Rem- cost of gas, $/mcf
160  Cea(1)=0.16 : Cea(2)=0.08 : Rem- $/kWh to absorption system
165  Coma=3600 : Rem- Cost of O&M in $/year.
167  Bef=0.75 : Rem- Boiler Efficiency for Variable Cooling Tower Cost
170  Rem
175  Rem- Compression System Characteristics
180  Copc=5.3 : Rem- COP of compression system
182  Elc=51 : Rem- Compression Cooling tower power.
185  Kc=122000.0+32000.0 : Rem- Capital Equipment $ + cooling tower cost
190  Cec(1)=0.13 : Cec(2)=0.08 : Rem- $/kWh to compression system.
195  Comc=3700 : Rem- Cost of O&M in $/year
200  Rem
205  Rem- Common Elements
210  Hpy(1)=420 : Hpy(2)=2000 : Rem- Hours per year of operation
215  D=0.06 : Rem- Constant $ discount rate.
220  N=7 : Rem- Years in economic life.
225  E0=0.0 : E3=0.03 : Rem - Low (high) escalation rates.
230  Kwpt=3.516 : Rem: kW/ton
235  Mcfpt=12000.0/1026000.0 : Rem- mcf of gas per hour per ton.
240  Cap=400 : Rem- Capacity of both systems
245  Ot$="-----"
400 *Prep
405  Rem- S=1 picks cheap gas, costly electricity while
410  Rem- S=2 picks cheap electricity, costly gas.
415  Def Fnpv(E)=((1+E)/(D-E))*(1-((1+E)/(1+D)) N)
420  Npve(1)=Fnpv(E3)
425  Npve(2)=Fnpv(E0)
430  Npvg(1)=Fnpv(E0)
435  Npvg(2)=Fnpv(E3)
440  Npvom=Fnpv(0.0)
445  Gasph=Cap*Mcfpt/Seas
450  Elph=Elc+Cap*Kwpt/Copc : Rem- Electricity use by compression unit.

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500 *Loops
505  Copa=1.0 : Rem- Design point COP= Cooling power/Gross gas input.
510  Rem- Can also loop over Copa to calculate BEC versus COP.
515  Actc=Kctf+Kctv*Cap*Bef*(-1.0+1.0/Copa)
520  Rem- Now loop on duty cycle
530    For H=1 To 2
535      Gasuse=Gasph*Hpy(H)/Copa
537      @ : @"Hours per year=",Hpy(H)
540      Eluse=Elph*Hpy(H)
545      Rem- and finally loop on price scenario
550        For S=1 To 2
555          @ : @" Elec. cost=",Cea(S),Cec(S)
560          Bec=Kc/Cap : @"  Cmp eqp+inst+CT=",Kc/Cap
565          Bec=Bec-Actc/Cap : @"  Abs CT=", -Actc/Cap,Bec
570          Bec=Bec-Kiaf/Cap : @"  Abs instal=", -Kiaf/Cap,Bec
575          T=Comc*Npvom/Cap : Bec=Bec+T : @"  Comp O&M=",T,Bec
580          T=-Coma*Npvom/Cap : Bec=Bec+T : @"  Abs O&M=",T,Bec
585          T=Eluse*Cec(S)*Npve(S)/Cap : Bec=Bec+T : @"  Comp Elec.",T,Bec
590          T=-Gasuse*Cog*Npvg(S)/Cap : Bec=Bec+T : @"  Abs Gas=",T,Bec
595          T=-Ela*Hpy(H)*Cea(S)*Npve(S)/Cap : Bec=Bec+T : @"  Abs Elec=",T
600          @"Total Breakeven Cost=",Bec
610        Next S
615      Next H
620    Stop

```

```

100 *Datain
105  @" 100 Ton Absorption Chiller Breakeven Costs"
110  Rem:
115  Rem: Absorption System Characteristics
125  Rem- Copa =Gross Design Point COP of absorption system
127  Seas=0.93 : Rem- Seasonal COP/Design Point COP
130  Ela=6.2+16.3 : Rem- kW to absorption unit, parasitics+cooling tower
135  Rem- Bec : Rem- Breakeven capital cost of absorption system
140  Kctf=14900.0 : Rem- Capital for cooling tower if COP=1
145  Kctv=63.6 : Rem- Variable $/thermal ton if COP<>1
150  Kiaf=23000.0 : Rem - Installation cost ($) for absorption system.
155  Cog=5.7/1.026 : Rem- cost of gas, $/mcf
160  Cea(1)=0.16 : Cea(2)=0.08 : Rem- $/kWh to absorption system
165  Coma=2350 : Rem- Cost of O&M in $/year.
167  Bef=0.75 : Rem- Boiler Efficiency for Variable Cooling Tower Cost
170  Rem
175  Rem- Compression System Characteristics
180  Copc=4.3 : Rem- COP of compression system
182  Elc=13 : Rem- Compression Cooling tower power.
185  Kc=38000.0+15000.0+10000.0
186  Rem- Capital Equipment $ + cooling tower cost
190  Cec(1)=0.13 : Cec(2)=0.08 : Rem- $/kWh to compression system.
195  Comc=1900 : Rem- Cost of O&M in $/year
200  Rem
205  Rem- Common Elements

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210 Hpy(1)=420 : Hpy(2)=2000 : Rem- Hours per year of operation
 215 D=0.06 : Rem- Constant \$ discount rate.
 220 N=7 : Rem- Years in economic life.
 225 E0=0.0 : E3=0.03 : Rem - Low (high) escalation rates.
 230 Kwpt=3.516 : Rem: kW/ton
 235 Mcfpt=12000.0/1026000.0 : Rem- mcf of gas per hour per ton.
 240 Cap=100 : Rem- Capacity of both systems
 245 Ot\$="-----"

100 *Datain
 105 @" 25 Ton Absorption Chiller Breakeven Costs"
 110 Rem:
 115 Rem: Absorption System Characteristics
 125 Rem- Copa =Gross Design Point COP of absorption system
 127 Seas=0.9 : Rem- Seasonal COP/Design Point COP
 130 Ela=1.6+2.0 : Rem- kW to absorption unit, parasitics+cooling tower
 135 Rem- Bec : Rem- Breakeven capital cost of absorption system
 140 Kctf=5200 : Rem- Capital for cooling tower if COP=1
 145 Kctv=55.2 : Rem- Variable \$/thermal ton if COP<>1
 150 Kiaf=7000 : Rem - Installation cost (\$) for absorption system.
 155 Cog=5.7/1.026 : Rem- cost of gas, \$/mcf
 160 Cea(1)=0.17 : Cea(2)=0.08 : Rem- \$/kWh to absorption system
 165 Coma=1200 : Rem- Cost of O&M in \$/year.
 167 Bef=0.75 : Rem- Boiler Efficiency for Variable Cooling Tower Cost
 170 Rem
 175 Rem- Compression System Characteristics
 180 Copc=4.1 : Rem- COP of compression system
 182 Elc=0 : Rem- Compression Cooling tower power.
 185 Kc=12750.0+6150 : Rem- Capital Equipment \$ + cooling tower cost
 190 Cec(1)=0.16 : Cec(2)=0.08 : Rem- \$/kWh to compression system.
 195 Comc=1200 : Rem- Cost of O&M in \$/year
 200 Rem
 205 Rem- Common Elements
 210 Hpy(1)=300 : Hpy(2)=1200 : Rem- Hours per year of operation
 215 D=0.06 : Rem- Constant \$ discount rate.
 220 N=7 : Rem- Years in economic life.
 225 E0=0.0 : E3=0.03 : Rem - Low (high) escalation rates.
 230 Kwpt=3.516 : Rem: kW/ton
 235 Mcfpt=12000.0/1026000.0 : Rem- mcf of gas per hour per ton.
 240 Cap=25 : Rem- Capacity of both systems