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CENTRAL COOLING—ABSORPTIVE CHILLERS

by

J. E. Christian

ICES
INTEGRATED COMMUNITY ENERGY SYSTEMS

TECHNOLOGY EVALUATIONS

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August 1977

Prepared for

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The four E's of the cover logo embody the goals of the Community Systems Program of the Energy Research and Development Administration, ERDA, namely:

- to conserve *Energy*;
- to preserve the *Environment*; and
- to achieve *Economy*
- in the design and operation of human settlements (*Ekistics*).

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FOREWORD

The Community Systems Program of the Division of Buildings and Community Systems, Office of Energy Conservation, of the United States Energy Research and Development Administration (ERDA), is concerned with conserving energy and scarce fuels through new methods of satisfying the energy needs of American Communities. These programs are designed to develop innovative ways of combining current, emerging, and advanced technologies into Integrated Community Energy Systems (ICES) that could furnish any, or all, of the energy using services of a community. The key goals of the Community System Program then, are to identify, evaluate, develop, demonstrate, and deploy energy systems and community designs that will optimally meet the needs of various communities.

The overall Community Systems effort is divided into three main areas. They are: (a) Integrated Systems, (b) Community Design, and (c) Commercialization. The *Integrated Systems* work is intended to develop the technology component and subsystem data base, system analysis methodology, and evaluations of various system conceptual designs which will help those interested in applying integrated systems to communities. Also included in this program is an active participation in demonstrations of ICES. The *Community Design* effort is designed to develop concepts, tools, and methodologies that relate urban form and energy utilization. This may then be used to optimize the design and operation of community energy systems. *Commercialization* activities will provide data and develop strategies to accelerate the acceptance and implementation of community energy systems and energy-conserving community designs.

This report, prepared by Oak Ridge National Laboratory, is part of a series of Technology Evaluations of the performance and costs of components and subsystems which may be included in community energy systems and is part of the Integrated Systems effort. The reports are intended to provide sufficient data on current, emerging and advanced technologies so that they may be used by consulting engineers, architect/engineers, planners, developers, and others in the development of conceptual designs for community energy systems. Further, sufficient detail is provided so that calculational models of each component may be devised for use in computer codes for the design of Integrated Systems. Another task of the Technology Evaluation activity is to devise calculational models which will provide part load performance and costs of components suitable for use as subroutines in the computer codes being developed to analyze community energy systems. These will be published as supplements to the main Technology Evaluation reports.

It should be noted that an extensive data base already exists in technology evaluation studies completed by Oak Ridge National Laboratory (ORNL) for the Modular Integrated Utility System (MIUS) Program sponsored by the Department of Housing and Urban Development (HUD). These studies, however, were limited in that they were: (a) designed to characterize mainly off-the-shelf technologies up to 1973, (b) size limited to meet community limitations, (c) not designed to augment the development of computer subroutines, (d) intended for use as general information for city officials and keyed to residential communities, and (e) designed specifically for HUD-MIUS needs. The present documents are founded on the ORNL data base but are more technically oriented and are designed to be upgraded periodically to reflect changes in current, emerging, and advanced technologies. Further, they will address the complete range of component sizes and their application to residential, commercial, light industrial, and institutional communities. The overall intent of these documents, however, is not to be a complete documentation of a given technology but will provide sufficient data for conceptual design application by a technically knowledgeable individual.

Data presentation is essentially in two forms. The main report includes a detailed description of the part load performance, capital, operating and maintenance costs, availability, sizes, environmental effects, material and energy balances, and reliability of each component along with appropriate reference material for further study. Also included are concise data sheets which may be removed for filing in a notebook which will be supplied to interested individuals and organizations. The data sheets are colored and are perforated for ease of removal. Thus, the data sheets can be upgraded periodically while the report itself will be updated much less frequently.

Each document was reviewed by several individuals from industry, research and development, utility, and consulting engineering organizations and the resulting reports will, hopefully, be of use to those individuals involved in community energy systems.

ABSTRACT

This technology evaluation covers commercially available single-effect, Lithium-Bromide absorption chillers ranging in nominal cooling capacities of 3 to 1,660 tons and double effect Lithium-Bromide chillers from 385 to 1,060 tons. The nominal COP measured at operating conditions of 12 psig input steam for the single-effect machine, 85°F entering condenser water, and 44°F exiting chilled-water, ranges from 0.6 to 0.65. The nominal COP for the double-effect machine varies from 1.0 to 1.15 with 144 psig entering steam. Data are provided to estimate absorption chiller performance at off-nominal operating conditions. The part-load performance curves along with cost estimating functions help the system design engineer select absorption equipment for a particular application based on lifecycle costs. Several suggestions are offered which may be useful for interfacing an absorption chiller with the remaining Integrated Community Energy System. The Ammonia-Water absorption chillers are not considered to be readily available technology for ICES application; therefore, performance and cost data on them are not included in this evaluation.

TECHNOLOGY EVALUATION SUMMARY SHEET OF

Central Cooling - Absorptive Chillers

By: J.E. Christian, ORNL January, 1977

A LITHIUM BROMIDE-WATER ABSORPTION CHILLERS

1 INTRODUCTION

To transfer heat, the absorption chiller utilizes a vapor compression cycle with evaporation and condensation of a refrigerant occurring at different pressure levels. The pressure differential is produced through a physio-chemical process. Two types of absorption chillers discussed here are: lithium-bromide and ammonia-water. The latter is mentioned only as a possibility for near-term technology.

2 PERFORMANCE

Single-effect, LiBr-water absorption chillers are available in nominal capacities from 3 to 1660 tons with attainable coefficient of performance, COP,* ranging from 0.6 to 0.65. Double-effect LiBr-water absorption chillers are available in nominal capacities from 385 to 1060 tons, with attainable COP ranging from 1.0 to 1.15.

The nominal capacity of the standard absorption chillers is based on the following operating conditions:

(1) 12 psig steam or equivalent hot water provided for the heat source temperatures for single-effect units; 144/123 psig steam for double-effect units;

(2) 85°F entering condenser water,

(3) 3.6 gpm/ton condenser water flowrate,

(4) 44°F chilled water,

(5) 2.4 gpm/ton chilled water flowrate, and

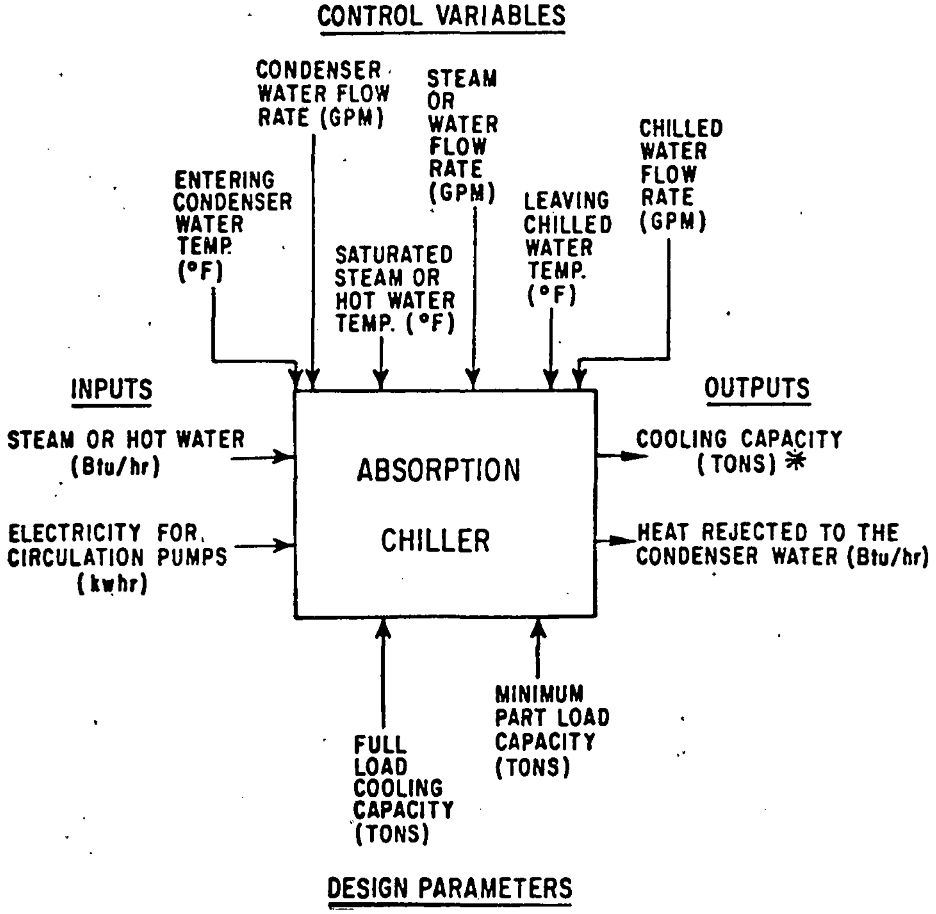
(6) standard lithium bromide flowrate

*COP = cooling capacity (Btu/h/amount of heat energy delivered to the absorption unit -- auxiliary electrical power not included).

The algebraic relationship between operating conditions and the absorption chiller capacity and COP is presented in a uniform manner by determining and listing the value of coefficients for a generalized polynominal of the form:

$$Y = A + BX + CX^2. \qquad \text{(Eq. DS-1)}$$

Any variation in the above operating conditions affects the absorption chiller capacity and COP. Figure DS-1 shows the input, output, and control variables, as well as the major design parameters necessary to simulate the operation of an absorption chiller.



* 1 TON = 12,000 Btu/hr

Fig. DS-1 Schematic of Absorption Chiller for Computer Simulation Purposes

Generalized equation coefficients A, B, and C are provided in Table DS-1 for substitution into Eq.DS-1 to show the algebraic relationship between each of the variable operating conditions and the percent of nominal cooling capacity or COP. The performance factors, capacity, and COP are represented by the dependent variable (Y) in Eq. DS-1; the independent variable (X) represents a particular operating condition. The data shown in Table DS-1 are to be used in conjunction with Eq. DS-1 to show the effect on performance by varying each of the operating variables independently, i.e., holding the other conditions constant. Data showing the effect on performance of changing two or more of the variables at the same time can be estimated from data in the main body of this report and from manufacturers' data sheets. The range of X shown in Table DS-1 represents the range of available data and of valid values of X to be used in the polynominal equation. The range given for X does not necessarily indicate the limits of operation for each control variable.

Table DS-1 Generalized Equation Coefficients for Substitution into Eq. DS-1 to Show Algebraic Relationship between Performance Factors and Operating Conditions

X	Y	Range of X	A	B	C	Reference Tables and Figures
		Single Effect				
Equivalent steam supply pressure (% of nominal)	Cooling capacity (% of nominal)	(30% ≤ X ≤ 100%)	65.3	0.347	0.0	Fig. 2.1 Table 2.4
Entering hot water temperature (°F)	Cooling capacity (% of nominal)	(173°F ≤ X ≤ 219°F)	239.0	1.65	-0.001	Fig. 2.3
Entering hot water temperature (°F)	COP (% of nominal)	(173°F ≤ X ≤ 219°F)	55.40	0.185	0.0	Fig. 2.3
Inlet condenser water temperature (°F)	Cooling capacity (% of nominal)	(75°F ≤ X ≤ 90°F)	286.60	-2.19	0.0	Fig. 2.4
Inlet condenser water temperature (°F)	COP (% of nominal)	(65°F ≤ X ≤ 95°F)	179	-0.93	0.0	Fig. 2.5
Condenser water flowrate (% of nominal)	Cooling capacity (% of nominal)	(65% ≤ X ≤ 135%)	44.4	0.776	-0.0021	Fig. 2.6
Condenser water flowrate (% of nominal)	COP (% of nominal)	(65% ≤ X ≤ 135%)	100	0.0	0.0	
Chilled water outlet temperature (°F)	Cooling capacity (% of nominal)	(40°F ≤ X ≤ 50°F)	-1.47	2.30	0.0	Fig. 2.1 Fig. 2.4
Chilled water outlet temperature	COP (% of nominal)	(40°F ≤ X ≤ 50°F)	100	0.0	0.0	

Table DS-1 (Cont'd)

X	Y	Range of X	A	B	C	Reference Tables and Figures
Chilled water flowrate (% of nominal)	COP (% of nominal)	(33% ≤ X ≤ 168%)	100	0.0	0.0	
% of Nominal Design Load	COP (% of nominal)	(10% ≤ X ≤ 40% (40% ≤ X ≤ 135%)	48.30 100	1.56 0.0	-0.007 0.0	Fig. 2.8 Table 2.6
		Double Effect				
Equivalent steam supply pressure (% of nominal)	Cooling capacity (% of nominal)	(36% ≤ X ≤ 100%)	-6.0	1.91	-0.0085	Fig. 3.1 Table 3.2
Equivalent steam supply pressure (% of nominal)	COP (% of nominal)	(36% ≤ X ≤ 100%)	147.0	-0.69	0.0022	Fig. 3.1 Table 3.2
Inlet condenser water temperature (%)	Cooling capacity (% of nominal)	(75°F ≤ X ≤ 90°)	-410	14.5	-0.1	Fig. 3.3 Table 3.3
Inlet condenser water temperature (°F)	COP (% of nominal)	(75°F ≤ X ≤ 90°F)	See reference figure			Fig. 3.5
Condenser water flowrate (% of nominal)	Cooling capacity (% of nominal)	(65% ≤ X ≤ 135%)	43.35	0.776	0.0021	Fig. 2.6
Condenser water flowrate (% of nominal)	COP (% of nominal)	(65% ≤ X ≤ 135%)	100	0.0	0.0	
Chilled water outlet temperature (%)	Cooling capacity (% of nominal)	(40°F ≤ X ≤ 50°F)	-32.0	3.0	0.0	Fig. 3.1 Fig. 3.3
Chilled water outlet temperature (%)	COP (% of nominal)	(40°F ≤ X ≤ 50°F)	100	0.0	0.0	
Chilled water flowrate (% of nominal)	Cooling capacity (% of nominal)	(33% ≤ X ≤ 168%)	110	-0.1	0.0	Fig. 3.4
Chilled water flowrate (% of nominal)	COP (% of nominal)	(33% ≤ X ≤ 168%)	100	0.0	0.0	
Design load (% of nominal)	COP (% of nominal)	(10% ≤ X ≤ 100%)	63.0	1.2	-0.0083	Fig. 3.5 Table 3.4

The COP of absorption systems is seen to be relatively insensitive to variations in generator, absorber, and condenser temperatures as soon as the temperatures are sufficient to sustain operation.

3 MANUFACTURER-SUGGESTED OPERATING CONDITIONS

Manufacturer-suggested operating conditions for central absorption chillers are:

(1) minimum chilled-water temperature: 40°F;

(2) maximum generator inlet steam pressure:
(a) 14 psig for single-effect units, and
(b) 144 psig for double-effect units;

(3) maximum generator inlet steam temperature:
(a) 340°F for single-effect units, and
(b) 400°F for double-effect units;

(4) maximum generator inlet hot water temperature:
(a) 270°-300°F for single-effect units,* and
(b) 400°F for double-effect units;

(5) maximum permissible load in tons, as a function of entering condenser water temperature, as tabulated below:

Entering Condenser Water Temperature (°F)	Single-Effect Unit <1000 tons (%)	Single-Effect Unit >1000 tons (%)	Double-Effect Unit (%)
45	50	43	45
55	70	58	60
68	100	79	92
74	-	100	100

(6) fouling factor on the interior of all tube surfaces of 5×10^{-4}.

The use of steam pressures or water temperatures higher than those recommended can result in overfiring the machine and may lead to operating difficulties or premature machine failure. Machines should not be expected to operate above 113% of the nominal capacity for the size machine chosen.

Water flows should not exceed 10-12* fps in copper tubes or 12 fps in cupronickel tubes. Use of higher water quantities will result in a proportionate increase in tube water velocities and possible tube erosion. Capacity control is achieved by sensing the exciting chilled water temperature, which is used to control automatically the steam or water flow.

*depends on manufacturer

4 ABSORPTION CHILLER RELIABILITY

Large tonnage, single- and double-effect lithium bromide-water absorption machines are comparatively trouble-free and simple to operate. Today's units are manufactured to rigid standards of vacuum integrity and internal cleanliness. Equipment, such as electronic halide leak detectors and helium mass spectrometers, ensures the leaktightness of machines before shipment from the factory.

The pressures within the shell are the vapor pressures of the liquids used in the cycle at their respective temperatures. In operation, the pressure in the absorber and evaporator sections is about 0.01 atmosphere. Pressure in the concentrator and condenser sections is about 0.1 atmosphere. To illustrate the importance of maintaining the machine leak free, introduction of sufficient air to raise the pressure just 0.06 psi will increase the exiting chilled-water temperature by 10°F.

According to one absorption chiller manufacturer, the most common cause of unscheduled shutdown is crystallization caused by: (1) malfunction of system controls, (2) failure of a pressure reducing valve, or (3) inadvertent introduction of air into the machine. Also, interruption of electric power, which will cause the machine to shut down without the normal dilution cycle, may result in crystallization.

5 SAFETY REQUIREMENTS

All absorption machines must operate in compliance with applicable *ASME* codes.

6 ENVIRONMENTAL EFFECTS

Absorption chillers are relatively quiet and vibration-free.

7 DOLLAR COST

Using 1976 dollars, Eq. DS-2 estimates the total installed cost of single-effect LiBr-water absorption chillers, and Eq. DS-3 estimates the total installed cost for double-effect absorption chillers in mid-1976 dollars.

$$\text{Single effect desired capacity total cost} = 96{,}000 \left(\frac{\text{desired capacity}}{500}\right)^{0.66} \qquad \text{(Eq.DS-2)}$$

$$\text{Double-effect desired capacity total cost} = 135{,}000 \left(\frac{\text{desired capacity}}{500}\right)^{0.7} \qquad \text{(Eq.DS-3)}$$

The operating and maintenance (O&M) costs derived by Eq. DS-4 are based on full-service contract costs in dollars per year and are representative of both single- and double-effect absorption chillers.

$$\text{desired capacity O\&M cost} = 3400 \left(\frac{\text{desired capacity}}{500}\right)^{0.56} \qquad \text{(Eq.DS-4)}$$

The projected economic life of an absorption chiller is assumed to be 20 years.[1]

B AMMONIA-WATER ABSORPTION CHILLERS

A preliminary survey of ammonia-water absorption chiller technology has led to the conclusion that this is not a technology of high-priority interest for ICES application. Compared to the lithium-bromide-water units operating at heat source temperatures above 180°F, ammonia-water absorption chillers have four major disadvantages:

(1) Internal pressures and associated pumping power requirements are higher.

(2) The system is more complex, e.g., a rectifier is required for the separation of ammonia and water vapor at the generator outlet rather than the simple still used in LiBr units.

(3) The COP is lower than for LiBr units.

(4) Ammonia is classified in the ANS B9 Safety Code Group 2, which restricts its use inside dwellings.

Prepared by J.E. Christian

Date January 1977

A LITHIUM BROMIDE-WATER ABSORPTION CHILLERS

1 INTRODUCTION

To transfer heat, the absorption chiller utilizes a vapor compression cycle similar to that of the mechanical compressive chiller, with evaporation and condensation of a refrigerant occurring at different pressure levels. The difference between the two cycles is that the absorption cycle uses a heat-operated generator to produce the pressure differential, (physio-chemical process), whereas the mechanical compression cycle uses a compressor. Although two types of absorption chillers -- lithium-bromide and ammonia-water are discussed here, the latter is mentioned only as a possibility for near-term technology.

1.1 ABSORPTION REFRIGERATION PROCESS

The basic single-effect absorption system is shown schematically in Fig. 1.1. The working fluid for the system is a solution of refrigerant

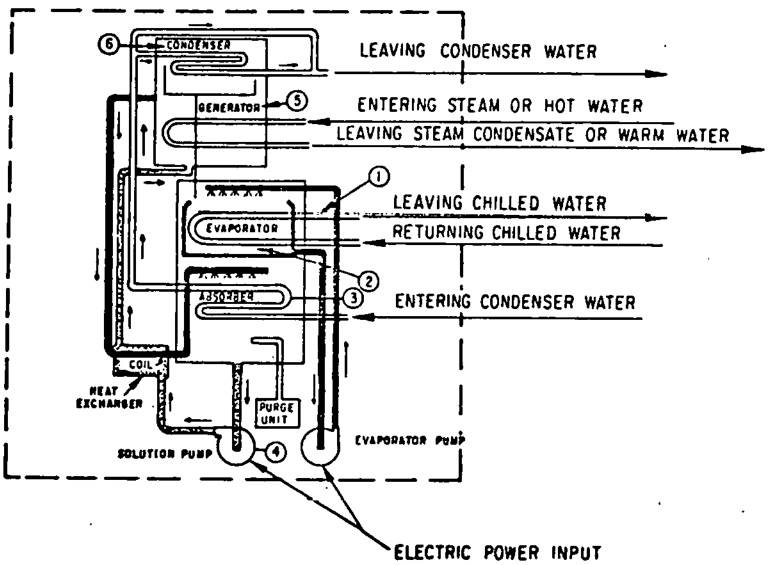


Fig. 1.1 Typical Absorption Chiller

Source: ASHRAE Equipment Handbook (Ref. 2) and absorbent that have strong chemical affinity for each other. Heat is added to this solution in the generator where the refrigerant and the absorbent are separated; i.e., regenerated by a distillation process. A simple still is adequate for the separation when the pure absorbent material is nonvolatile, as in the LiBr-water system. However, fractional distillation equipment is required when the pure absorbent material is volatile as in the ammonia-water system, because the refrigerant should be essentially free of absorbent; otherwise, vaporization in the evaporator is hampered. The regenerated absorbent normally contains substantial amounts of refrigerant. If the absorbent material tends to solidify, as in the LiBr-water system, it is necessary to have enough refrigerant present to keep the pure absorbent material in a dissolved state at all times. Certain practical considerations, particularly the avoidance of excessively high temperatures in the generator, allow a moderate amount of refrigerant in the regenerated absorbent to inhibit crystalization. A double-effect absorption system uses a 2-stage generator that permits higher heat source temperatures. (See Sect. 3 for details.)

The vaporized refrigerant, which is liquified in the condenser by removing heat usually through a cooling tower, then expands from the high-pressure portion of the system (generator and condenser) into the low-pressure evaporator. Here vaporization of the refrigerant chills water that is circulated to cool the building air.

In the absorber, refrigerant vapor is recombined with the absorbent mixture from which it was initially obtained. Because this recombination reaction is exothermic, heat must be removed from the absorber to maintain its temperature at a sufficiently low value to ensure a high-chemical afinity between the refrigerant and the solution. The absorber solution, now rich in refrigerant, can be pumped back into the generator to continue the cycle. A more detailed description of the absorption process is available in Refs. 1--3.

1.2 PERFORMANCE FUNCTIONS

The cooling capacity and COP of an absorption chiller varies as a function of operating conditions, such as chilled water and heat sink (cooling tower water) temperature. Figure 1.2 shows a schematic of the major variables needed to describe the full- and part-load performance of absorption chillers. In subsequent sections, such performance functions are illustrated graphically. The performance data presented is from manufacturers' catalog data. No independent performance verification was carried out for this technology evaluation.

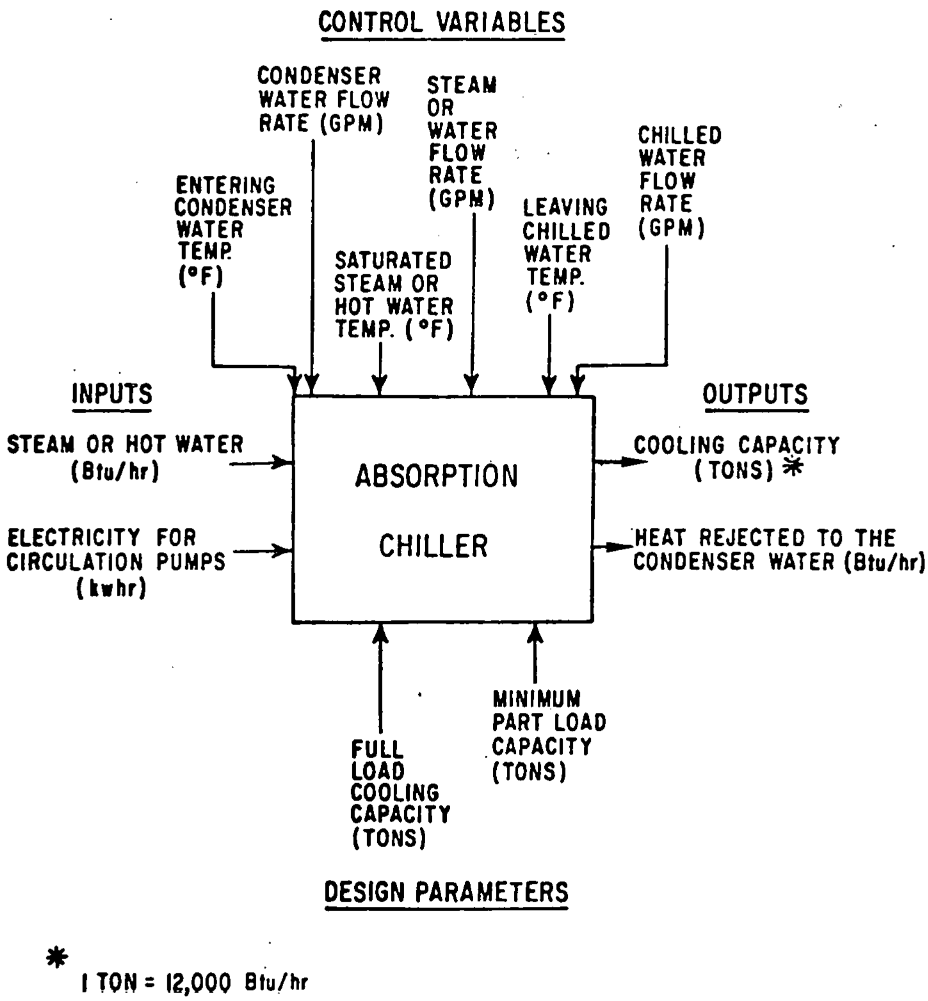


Fig. 1.2 Schematic of Absorption Chiller for Computer Simulation Purposes

*1 ton = 12,000 Btu/h

A polynomial in the form of Eq. 1.1 that fits each data set was found, and the value of coefficients A, B, and C are listed for each performance function. The coefficients A, B, and C are generated through a computer program that fits least-squares polynomials to bivariate data using an orthogonal polynominal method. The generalized empirical equation if of the form:

$$Y = A + BX + CX^2, \qquad \text{(Eq.1.1)}$$

where Y represents either the percentage of nominal cooling capacity or COP of the absorption chiller. The absolute values of capacity or COP can be found by multiplying Y by the nominal values based on manufacturers' suggested data given in this report and dividing by 100.

The X variable represents independent operating conditions, e.g. condenser water temperature or chilled-water flowrate.

2 SINGLE-EFFECT, LITHIUM BROMIDE-WATER ABSORPTION CHILLERS

2.1 DESCRIPTION

The single-effect, absorption chiller unit may be contained in one or two shells. Generally, one shell contains the high-pressure (absorber and evaporator) sections, while the other shell consists of the low-pressure (generator and condenser) sections. Units are shipped completely factory-assembled when transportation methods permit. However, when shipping regulations require large absorption units to be shipped in two sections, the units are assembled at the factory, checked for leaktightness, and then disassembled for shipment.

2.1.1 Manufacturers and Available Size Ranges

Single-effect, lithium bromide-water absorption chillers are available in nominal capacities ranging from 3 to 1660 tons. Nominal operating conditions consist of an incoming equivalent steam pressure of 12 psig (∿244°F), cooling tower water of 85°F, outgoing chilled water at 44°F, and a 5×10^{-4} fouling factor.

A list of U.S.-based absorption chiller manufacturers and size ranges are shown in Table 2.1. The address of each manufacturer is available in the *Air-Conditioning, Heating, and Refrigeration News*

Table 2.1 Manufacturers and Sizes of Single-Effect, Absorption-Type, Liquid Chillers

Company	Nominal Capacity Range (ton)
Arkla Industries, Inc.	3, 25
Carrier Air Conditioning Corp.	100-815
The Trane Co.	101-1660
York Division, Borg-Warner Corp.	120-1400

Directory, issued December 27, 1976. Table 2.2 shows a partial list of common, nominal cooling capacities currently available. Larger capacities can be met by multiple-unit installations.

Table 2.2 Nominal Manufactured Sizes of Single-Effect LiBr-Water Absorption Chillers (Tons)

3	235	565	906
25	256	617	960
100	270	665	1125
120	311	704	1250
155	410	750	1377
172	455	794	1465
200	520	852	1660

2.1.2 Technical Data

- *Dimensions and weight.* The space requirements and operating weight of a variety of absorption chillers are shown in Table 2.3, but the dimensions shown do not include service access areas. Sufficient space should be allowed at one end of the unit to permit tube and spray header removal for periodic tube inspection and cleaning.

Table 2.3 Single-Effect Absorption Chiller Physical Dimensions

Nominal Capacity (ton)	Length x Width x Height (ft)	Weight (lb)
3	5.0 x 3.0 x 6.5	1,115
25	9.0 x 3.5 x 7.5	3,508
101	11.0 x 5.0 x 7.5	11,260
200	14.0 x 6.0 x 8.0	16,350
397	17.0 x 7.0 x 9.0	27,800
665	22.0 x 7.5 x 10.0	44,300
1125	28.0 x 8.5 x 11.0	70,900
1660	33.0 x 10.0 x 12.0	107,800

• *Electrical requirements.* Absorption chillers are available in standard AC voltages of 200, 380, 415, 460, or 575 at 50- or 60-Hz frequencies. Units operating with nonstandard voltages also are available. A prewired control circuit transformer supplies 115-V control power from line voltage. The 3- and 25-ton units are available in 115- and 230-V, 60-hz, single-phase.

2.2 MATERIAL AND ENERGY BALANCE

Figure 1.1 shows the essential inputs and outputs to the absorption chiller. The heat sink, which may be a cooling tower or cooling pond, is considered a separate component in the ICES technology Evaluation, and therefore will not be discussed here. A separate evaluation on heat rejection provides the additional data required to carry out a complete evaluation of absorption air-conditioning systems.

The inputs are: (1) the thermal energy received from the energy source which is usually in the form of low-pressure steam or hot water; (2) the thermal energy received at a lower temperature from the buildings; and (3) the electrical energy required to circulate the refrigerant and absorbent through the absorption cycle.

The output is the thermal energy rejected to a heat sink at some intermediate temperature. A simple energy balance shows that the output to a heat sink must be equal to the sum of the thermal energy inputs from the steam or hot water source, from the buildings, and from the thermal equivalent of the auxiliary equipment electrical demand.

The cooling capacity, measured in Btu/h divided by the amount of energy delivered to the absorption unit converted to Btu/h, is equal to the coefficient of performance of the unit. The *ASHRAE Handbook*[2] states that single-effect, absorption chillers have full-load COPs in the range of 0.6 to 0.65, which is in good agreement with commercial literature. The COP does not include energy for auxiliary requirements such as absorbent and refrigerant circulation, which amounts to less than 0.5% of the heat input from the heat source. The COP does not include energy for chilled and condenser water circulation nor cooling tower fans.

2.2.1 Performance as a Function of Variable Operating Conditions

The nominal capacity rating of most single-effect absorption units is based on the following operating conditions:

(a) 12 psig saturated steam or equivalent hot water from the heat source, and

(b) 85°F entering condenser water,

(c) 3.6 gpm/ton condenser water flowrate,

(d) 44°F chilled-water flowrate, and

(f) standard lithium bromide flowrate.

Performance variations resulting from changes in each of the six standard operating conditions are discussed below.

• *Steam Equivalent Heat Source.* Standard, single-effect, absorption air-conditioning units use steam ranging in pressure from 2 to 14 psig (219° to 248°F).[3] Figure 2.1 shows that the full-load capacity drops as the steam supply pressure (nominally 12 psig) is reduced. At lower steam pressures, the steam contains less latent heat (Btu/lb); therefore, less energy

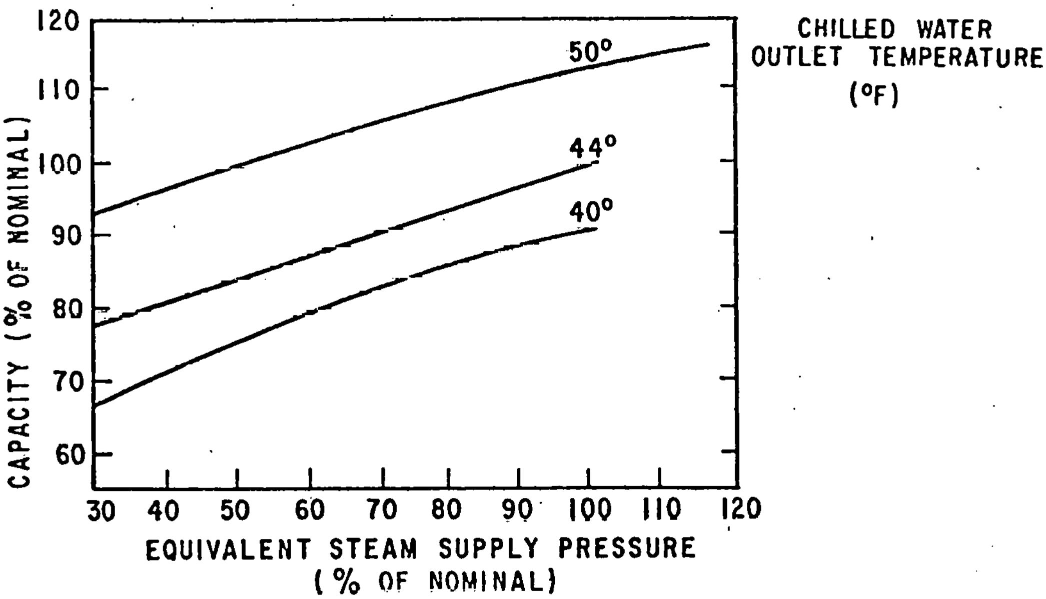


Fig. 2.1 Effect of Steam Supply Pressure on Capacity for Chilled Water Outlet Temperatures from 40° to 50°F*

*Based on: ∿500 ton single-effect abosrption chiller, 85°F inlet condenser water, and nominal chilled water and cooling tower water flowrates stated in Sect. 2.2.1.

is transferred to the generators resulting in less water evaporation. The exit temperature is assumed fixed at atmospheric pressure, 0 psig. The algebraic relationship can be represented by substituting the generalized equation coefficients A, B, and C shown in Table 2.4 into Eq. 1.1, where X is the percent of nominal steam supply pressure, and Y is the percent of nominal cooling capacity at 40°, 44°, and 50°F exiting chilled-water temperature.

Table 2.4 Generalized Equation Coefficients -- Percent of Nominal Capacity (Y) Vs Percent of Nominal Steam Supply Pressure (X)*

Chilled Water Outlet Temperature	Coefficients A	B	C
50	77.55	0.509	-0.0015
44	65.30	0.347	0.0
40	44.33	0.774	-0.0032

*$33 \leq X \leq 100$ for chilled-water outlet temperatures from 40–45°F and $33 \leq X \leq 115$ for chilled-water outlet temperatures from 45–50°F

If the heat source is in the form of hot water, the equivalent hot water temperature and flow required to provide equivalent full-load cooling capacity performance can be determined from Fig. 2.2. The equivalent hot

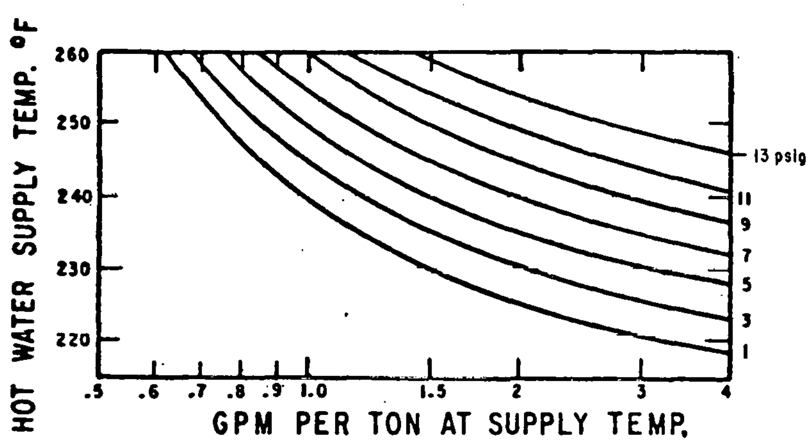


Fig. 2.2 Equivalent Steam Pressure to Generator Flange for Hot Water Based on ∿500-Ton Unit.[4]

ICES TECHNOLOGY EVALUATION

water supply temperature can be estimated by using the coefficients A, B, and C (shown in Table 2.5) in Eq. 1.1 for given hot water flowrate (X) and desired equivalent steam pressure to the generator flange. The equivalent steam pressure curves in Fig. 2.2 are from pressure measurements at the generator flange. A 3-psig pressure drop through the steam valve is assumed. Thus, the nominal 12 psig supply steam source provides 9 psig at the generator flange. Figures 2.1 and 2.2 offer a method of estimating the full-load hot water source temperatures, which results in the same cooling capacity as a given saturated steam heat source.

Table 2.5 Generalized Equation Coefficients -- Hot Water Supply Temp. (Y, °F) Vs Hot Water Source Flowrate (X, gpm/ton)

Equivalent Steam Pressure to the Generator Flange (psig)	A	B	C	Range of X
1	264.5	-28.0	4.215	0.6 - 4
3	269.2	-27.3	4.004	0.65 - 4
5	275.0	-28.2	4.171	0.75 - 4
7	273.4	-22.6	3.092	0.85 - 4
9	279.7	-23.9	3.333	1.0 - 4
11	260.2	- 5.0	0.0	1.25 - 4
13	265.2	- 5.0	0.0	1.45 - 4

Figure 2.2 indicates the capacity rating at equivalent hot water supply temperatures from 219°F to 260°F. The exit hot water tamperature is assumed to be fixed at 212°F. For source temperatures above 260°F, consult absorption chiller manufacturers for size selection.

Figure 2.3 shows the effect of a lower entering hot water temperature on both the absorption chiller capacity and the COP.[5] The hot water flow-

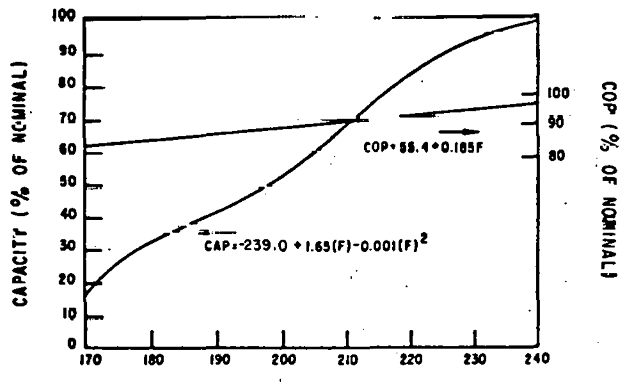


Fig. 2.3 Approximate Absorption Chiller Performance as a Function of Hot Water Source Temperatures Below 240°F.[5]

rate is assumed to be fixed at 2.8 gpm/ton of nominal capacity.

The algebraic relationship between the percentage of nominal capacity or COP and the entering hot water temperature (°F) can be derived by the equations shown in Fig. 2.3.

Most manufacturers recommend contacting local representatives when system design calls for delivery temperatures to the absorption units below 240°F for specific installations. The COP of absorption systems, as shown in Fig. 2.3, is relatively insensitive to variations in generator temperatures after the temperatures are sufficient to sustain operations. This characteristic is a consequence of the fixed heat of vaporization of the refrigerant.[6]

- *Condenser Water Temperature.* A lower condenser water temperature increases the potential full-load cooling capacity of the absorption chiller, as shown in Fig. 2.4. However, as the entering condenser water temperature drops below 75°F, the maximum capacity is reduced by the machine's control logic because of the danger of overfiring the machine.

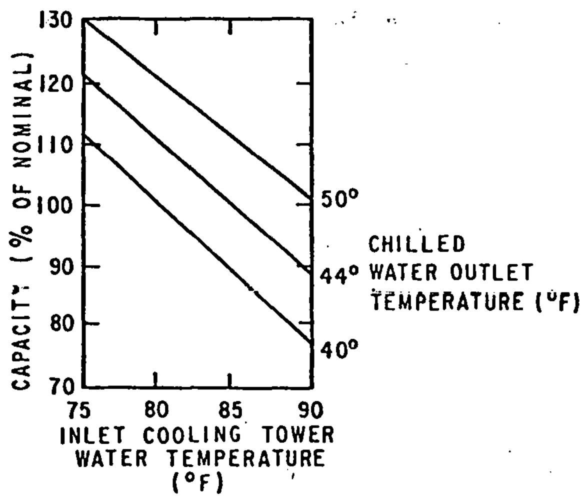


Fig. 2.4 Effect on the Single-Effect Absorption Chiller Cooling Capacity* of the Cooling Tower Inlet Water Temperature[3]

*Based on ∿500-ton unit at operating conditions listed in Sect. 2.2.

The leaving condenser water temperature can be estimated, for any given entering condenser water temperature greater than 75°F, by Eq. 2.1, since the heat rejected from the absorption chiller condenser is the sum of the input heat sources. The condenser water flow rate is assumed to be held constant at 3.6 gpm per ton of nominal rated capacity.

$$T_1 = T_2 + \left(\frac{Cap + Heat + Aux}{1800 \times TON}\right) PL \qquad \text{(Eq.2.1)}$$

where:

T_1 = leaving condenser water temperature, °F

T_2 = entering condenser water temperature, °F

Cap = cooling capacity, Btu/h

Heat = energy from heat source, Btu/h

Aux = thermal equivalent of the auxiliary equipment electrical requirements, Btu/h (Sect. 2.2.3)

TON = nominal rated capacity, tons

PL = fraction of full load, i.e., 0.9 = 90% of full load.

The relationship between the entering condenser water temperature, the leaving chilled water temperature, and the percent of nominal cooling capacity can be estimated by Eq. 2.2.

$$Y = (323 - 0.838X_2) - (3.77 - 0.0358X_2)X_1, \qquad \text{(Eq.2.2)}$$

where:

Y = percent of nominal cooling capacity,

X_1 = entering condenser water temperature, °F ($75 \le X_1 \le 90$)

X_2 = leaving chilled water temperature, °F ($40 \le X_2 \le 50$)

Equation 2.2 fits the three curves shown in Fig. 2.4 with an accuracy of ±2% of the percent of nominal capacity.

The three curves are for a nominal leaving chilled water temperature of 44°F, and the expected minimum and maximum leaving chilled water temperatures. The entering chilled water temperature is assumed to float because it is dependent on the actual cooling load provided.

The effect of a lower condensing water temperature results in a slightly higher COP, as shown in Fig. 2.5. Usually the entering condenser water temperature is allowed to "float" down as the wet bulb temperature drops, for absorption units with heat source water temperatures below 210°F and for steam applications. With a heat source of hot water entering the absorption generator at 210°F to 240°F, and the minimum allowable entering condenser water is frequently set greater or equal to 75°F.

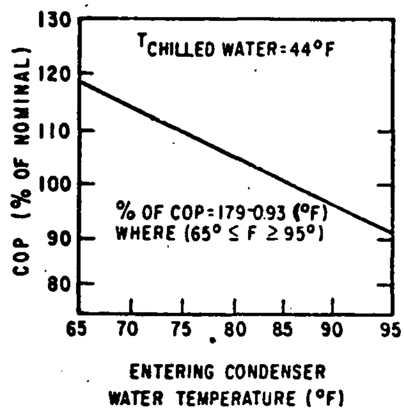


Fig. 2.5 Effect of Condenser Water Temperature (X, °F) on Percentage of Nominal COP (Y) Based on Other Operating Conditions*

*65°F ≤ X ≤ 95°F

o *Condenser Water Flowrate.* Figure 2.6 shows that as the condenser water flowrate increases, the cooling capacity increases as well. Generally, once a design flowrate is specified for a given absorption installation, it remains fixed at all part-load operating levels. The design condenser water

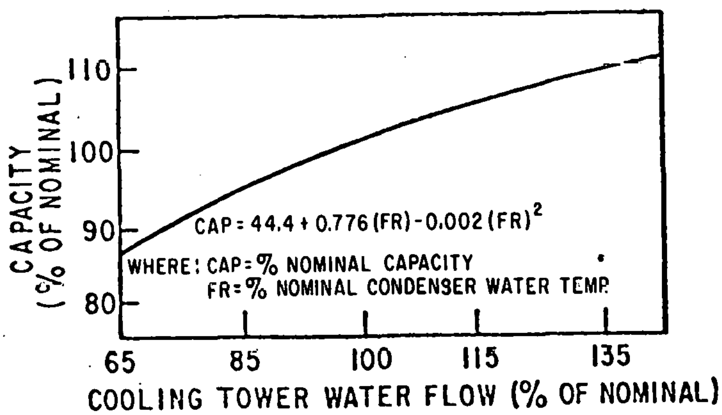


Fig. 2.6 Capacity Correction Factor as a Function of Cooling Tower Water Flow

requirements usually are more a function of the steam rate and condenser water temperature rise, as shown in Fig. 2.7.

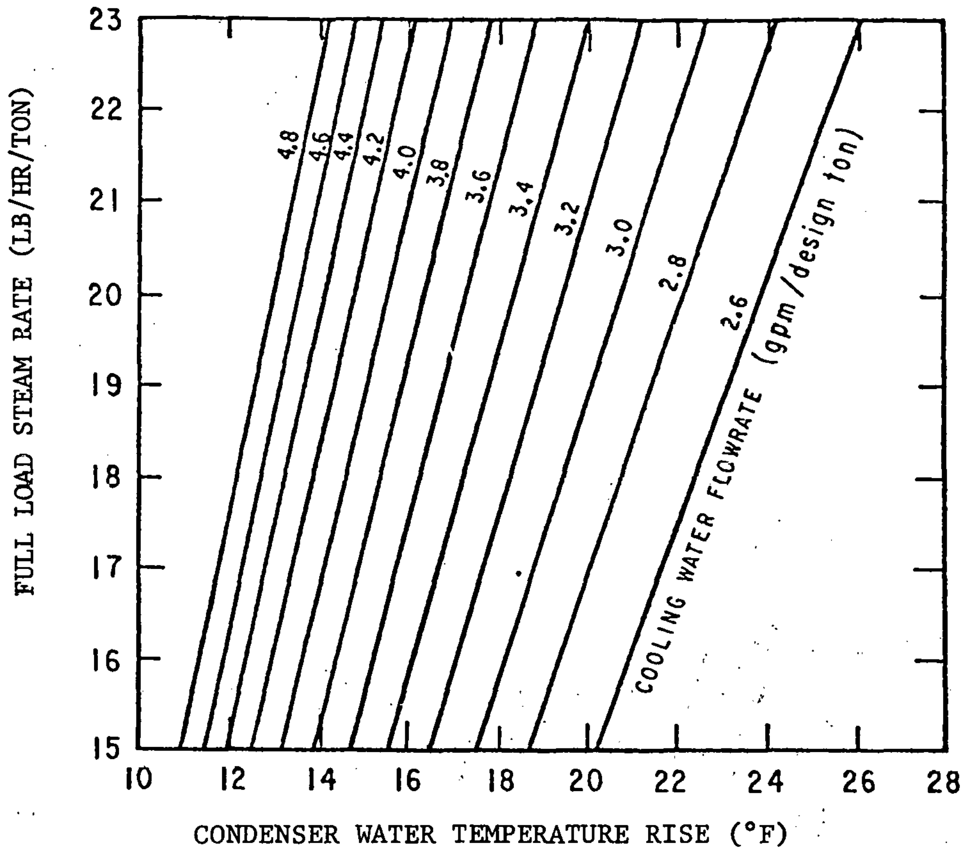


Fig. 2.7 Condenser Water Requirement as a Function of Cooling Water Temperature Rise and Steam Rate

The entering condenser water temperature to the absorption unit is specified by assuming a design approach temperature of the cooling tower (leaving condenser water temperature from the tower - ambient wet bulb temperature). Then the exiting condenser water temperature can be estimated using Eq. 2.1 on page 12.

• *Chilled-Water Temperature.* The exiting chilled-water temperature provided by a standard absorption unit affects capacity as shown in Figs. 2.1 and 2.4. The COP is assumed by most manufacturers to be unaffected by various chilled-water temperatures between 40° and 50°F.

• *Chilled-Water Flowrate.* A reduction in the chilled-water flowrate to 50% of nominal has a negligible effect on the COP for a given condenser water flowrate.[7]

• *Lithium Bromide Flowrate to the Generator.* The quantity of solution circulated between the absorber and generator changes the concentrations of the weak and strong solutions throughout the unit, and likewise the operating efficiency. Generally, the higher the concentration, the better the COP. A decrease in solution flowrate from the absorber to the generator increases the concentrations and likewise the COP. However, this increases the required equivalent heat source temperature. It also places some additional limits on the range of variables because of the increased tendencies of bordering on crystallization.[7]

The optimum percent solution flowrate for improved COP generally is somewhat greater than the percent nominal capacity at which the unit must perform. The lowest required equivalent heat source temperatures can be achieved with 100% of solution flow. Reference 7 shows quantitative results from a computer analysis of the effects of solution flow on the maximum capacity and the COP for a standard water-cooled, lithium bromide absorption unit.

2.2.2 Performance at Part-Load and Capacity Control

Figure 2.8 shows that the COP drops with cooling load below 40% of full design capacity for a standard unit at nominal operating conditions.[8]

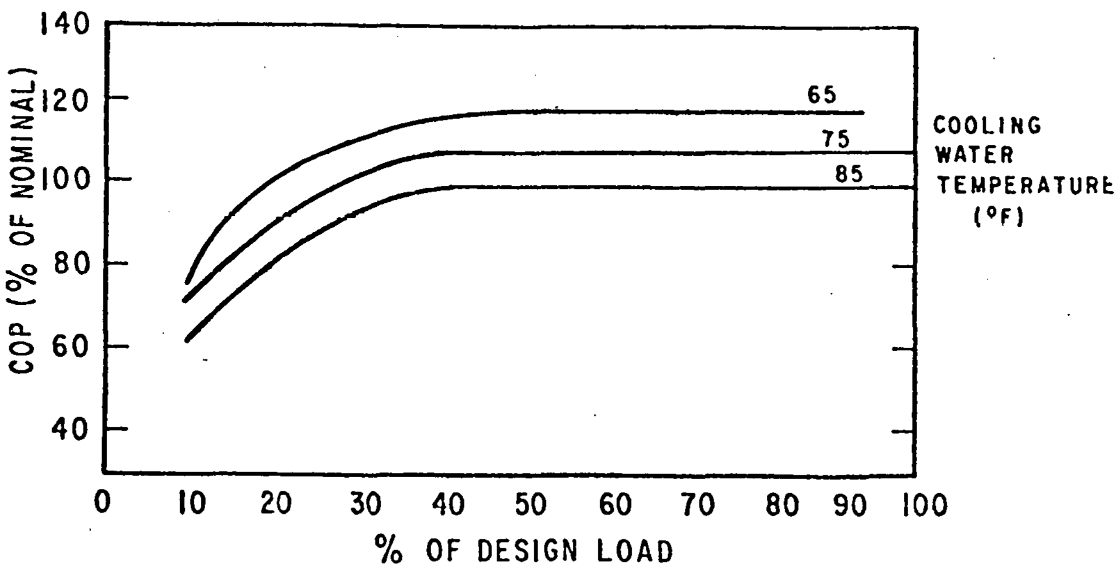


Fig. 2.8 Percent of Nominal COP Vs the Percent of Full Design Load for Various Cooling-Tower Water Temperatures.[4]

Part load capacity is met by simply reducing the flow of hot water or steam to the generator by monitoring the leaving chilled water temperature. The input temperature remains the same. The entering condenser water temperature is allowed to float downward for all heat sources except hot water delivered between the temperatures of 210°F and 240°F. If the absorption unit cooling tower was selected to provide 85°F water at 78°F WB (a 7°F approach) at 100% load, then at a part load of 75% and 65°F WB, the entering condenser water will be 65°F + (0.75 x 7) = 70.25°F. The leaving chilled water temperature generally remains fixed.

The algebraic representation of the part load COP curves from 10% to 40% can be obtained by using generalized equation coefficients A, B, and C shown in Table 2.6. The independent value is the percentage of full load.

Table 2.6 Generalized Equation Coefficients — Percentage of Nominal Capacity (Y) Vs Percent of Design Load (X)*

Cooling Tower Water Temperature (%)	A	B	C
65	49.25	3.21	-0.038
75	45.75	2.86	-0.033
85	48.80	1.56	-0.007

*$10 \leq X \leq 40$

Figure 2.8 must be used with caution since part-load input energy requirements are influenced by several factors such as load, chilled water flow, exiting chilled water temperature, condenser water flow, entering condenser water temperature, available steam pressure, and equipment components. Therefore, appreciable inaccuracies could result from applying a "typical part-load curve" (percent design load vs percent energy input) for a family of machines to an owning-operating cost estimate of a particular job.

The maximum design load recommended by most absorption chiller manufacturers is about 113% of the nominal capacity derived with operating conditions similar to those listed in Sect. 2.2.1.

The maximum operating capacity of absorption units varies between 115% and 140% of the nominal capacity. The maximum operating capacity is obtainable by increasing the number of evaporator passes, increasing the number of absorber passes, increasing the generator steam supply pressure to the maximum allowable (∿15 psig), decreasing the condenser water to 80°F from 85°F, increasing the condenser water flowrate 15% to 35%, and increasing the chilled water temperature from 44°F to 50°F. The COP at machine operation greater than nominal capacities hold relatively constant.

2.2.3 Effect of Fouling

The size selection and performance estimation data on the absorption chiller suggested in this report assume an average fouling factor of 0.0005. Table 2.7 shows that higher fouling factors mean more scale buildup on heat exchanger surfaces and a resulting reduction in the overall heat transfer coefficient. To obtain the same rated COP of 0.6 to 0.65 for an absorption chiller with higher condenser fouling as a result of poor quality condenser water, additional heat transfer area is required. The percentage of additional heat transfer area needed is shown in Table 2.7.

Table 2.7 Heat-Transfer Surface Required to Offest Fouling

Fouling Thermal Resistance (hr) (sq ft) (°F temp. diff)/Btu[c]	Over-All Heat Transfer Coefficient[b] Btu/(hr) (sq ft) (° F temp. diff)[c]	Thickness of Scale[a] Approximate (in)	Increase of Required Heat Transfer Area[b] (approximate %)
clean tubes	850	.000	-45
0.0005	595	.006	0
0.001	460	.012	40
0.002	313	.024	125
0.003	240	.036	205

[a] Assume a mean value for the thermal conductivity of the scale of 1.0 Btu/(h) (ft^2) (°F/ft).

[b] The overall heat transfer coefficient U selected for this illustration is typical for a water-cooled refrigerant condenser. However, because it is possible to have different overall heat transfer coefficients depending on the systems, the effect on the overall heat transfer by the scale will vary.

[c] Sq ft of inside surface of tube in heat exchanger.

Source: Carrier Air Conditioning Co., *Handbook of Air Conditioning System Design*, p. 5-3, McCraw-Hill, 1965.

2.2.4 Auxiliary Electric Energy Inputs

Figure 2.9 shows the auxiliary electrical energy inputs for the circulating pumps required in a single-effect lithium bromide absorption unit.[3] The electrical power input vs absorption chiller size can be estimated by the equation shown in Fig. 2.9.

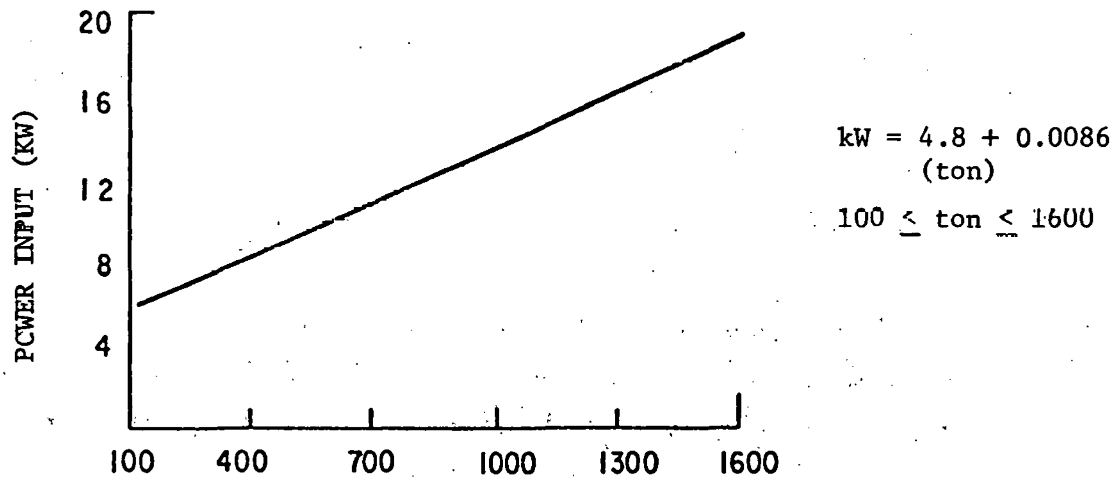


Fig. 2.9 Auxiliary Power Requirement for Absorption Chiller LiBr-Water Circulating Pumps

The cooling-water system is actually a much larger consumer of energy than the circulation pumps and must be considered in an evaluation of the total absorption air-conditioning system. However, the actual power consumption of cooling tower pumps and fans is not provided in this evaluation, as explained in the beginning of Sect. 2.2. The Arkla 25-ton absorption unit draws a maximum of 120 W, and the Arkla 3-ton unit, has a typical power input requirement of about 450 W. The 3-ton unit power input includes a circulation fan which delivers cooled air directly to the conditioned space.

2.3 OPERATING AND MAINTENANCE REQUIREMENTS

The most important aspect of absorption machine operation and maintenance is the requirement that internal cleanliness and leaktightness be sustained throughout the life of the unit.[2] By either automatic or manual purgers, furnished with the machines, purging must be accomplished with sufficient regularity to prevent the accumulation of air and other noncondensible gases within the unit. Air-tight units require purging only about once a week. High purge rates indicate air leakage. The most common source of air leakage is through the solution seals on the circulating pumps. These require about three man-days to replace. A complete set of solution seals should be installed about every 4 to 5 years.[9]

- *Location*. The units should be installed in an area protected from the weather and maintained at a temperature above freezing.

- *Piping*. All piping connections should be arranged for easy removal of water heads for service and cleaning of tube bundles.

- *Steam Supply*. A steam separator should be installed in the steam supply line to the steam valve to ensure dry steam to the generator at all times. When the steam supply pressure exceeds 15 psig, a pressure regulator for dead-end service should be installed ahead of the control valve, and a 15 psig relief valve, sized to relieve maximum steam flow, should be installed between the control valve and generator flange. A relief valve is not required if the steam source relief valve is set at 15 psig.

Shut-off valves should be installed in both the feed and return lines of the steam piping to permit unit servicing.

- *Hot Water Supply*. During hot water unit shutdown, the water in the generator contracts as it cools. This may form a vacuum that can be avoided by installing a check valve in the return hot water piping with a 3/4-in. bypass around the check valve.

• *Operation*. Proper operation and control of the absorption system requires that the condenser water flow be practically constant at all times. Cooling tower water temperature generally must be kept above 45°F. In all steam applications, superheat should be limited so that the maximum steam temperature does not exceed 340°F. In hot water applications, a water supply temperature of 270°F maximum is recommended.

Use of steam pressures or water temperatures higher than those recommended can cause overfiring the machine and premature failure. Also, if the operating steam pressure to the generator flange exceeds 15 psig, the steam heads fall under the control of the ASME Codes which require having qualified operating engineers at hand and preventing unattended operation.

To reduce fouling, it is necessary to pay attention to good water treatment practices. Periodic cleaning of the internal condensing or cooling water tube surfaces is required by mechanical and/or chemical means. This is usually true, regardless of the effectiveness of the water treatment practices followed. However, the more effective the water treatment, the longer the allowable period of time between tube cleaning.[2]

• *Condensate*. Some condensate may flash at full load; a subcooler may be installed ahead of the condensate receiver to cool the condensate to a temperature below the saturation temperature at atmospheric pressure, thus eliminating flashing entirely. It is recommended that a cooling medium such as heat recovery feed water be used to keep this energy within the system.

The condensate system must not draw supply steam through the machine because this reduces the machine efficiency and may offset any potential energy savings which might otherwise be realized by the use of the condensate return system. It also reduces the tube life because of erosion.

• *Absorber*. Unless the refrigerant-absorbent solution is charged with some type of corrosion inhibitor such as lithium chromate, the absorber tubes should be constructed out of a highly corrosion-resistant material, such as cupronickel.

3 DOUBLE-EFFECT, LITHIUM BROMIDE-WATER ABSORPTION CHILLERS

3.1 DESCRIPTION

The double-effect, lithium bromide absorption chillers operate on basically the same principle as the single-effect, except that a two-stage generator is used. The first stage generator accepts steam at 125 to 150 psig to provide the heat necessary to boil off refrigerant (water) from a dilute solution of distilled water and lithium bromide salt. This refrigerant vapor boiled from the solution then is used as 5 psig steam to heat the second-stage generator. Dilute lithium bromide solution from the absorber section of the machine is sent to the first-stage generator, where it is partially concentrated, and then to the second-stage concentrator where the concentration process is completed.

As the refrigerant from the first stage heats the solution in the second stage, it condenses and is piped directly into the condenser section. Because refrigerant from the first-stage generator condenses in the second stage, it can be introduced as a liquid directly to the condenser. This reduces by approximately 30% the amount of heat per ton of refrigeration that is rejected to the cooling water, as compared to a single-stage design.[9]

3.1.1 Manufacturers and Available Size Range

Double-effect, lithium bromide-water absorption chillers are available in this country in nominal capacities ranging from 385 to 1060 tons. Nominal operating conditions consist of an incoming steam pressure to the machine of 144 psig, cooling tower water at 85°F with a 3.6 gpm/ton flowrate, and outgoing chilled water at 44°F with a 2.4 gpm/ton flowrate.

Only one U.S. company, the Trane Co., markets a two-stage absorption unit with capacities from 385 to 1060 tons.

3.1.2 Technical Data

• *Dimensions and Weight.* The space requirements and operating weight of three double-effect absorption chillers are shown in Table 3.1.

Table 3.1 Double-Effect Absorption Chiller Physical Dimensions

Nominal Capacity (Ton)	Length x Width x Height (ft)	Weight (lb)
385	18 x 10 x 12	41,850
656	23 x 10 x 13	63,200
1060	30 x 10 x 14	94,700

• *Electrical Requirements.* Double-effect, absorption chillers are available in standard voltages of 200, 460, or 575 volts, with three-phase, 60-Hz power. A separate control transformer and control circuit requires 115 volts, single-phase electric power.

3.2 ENERGY AND MATERIAL BALANCE

Commercial literature on the double-effect absorption chillers indicates a nominal COP of about 1.0.[10] The COP values discussed for absorption chillers include energy for refrigerant and solution circulation, which is less than 1% of the total heat source input energy but does not include electrical requirements for chilled water distribution nor cooling water pumps and cooling water pumps and cooling tower fans.

3.2.1 Performance as a Function of Variable Operating Conditions

• *Steam Equivalent Heat Source.* Three separate double-effect absorption models are available for application with: (1) 150 psig steam, (2) 125 psig steam, and (3) hot water systems. Figure 3.1 shows the effect of steam supply pressures on cooling capacity of double-effect, absorption refrigeration units.[10] The generalized equation coefficients, A and B, are provided in Table 3.2 to represent mathematically the relationship between the percentage of nominal capacity, Y, and the equivalent steam supply pressure, X, at chilled-water temperatures between 40° and 50°F. As the steam supply

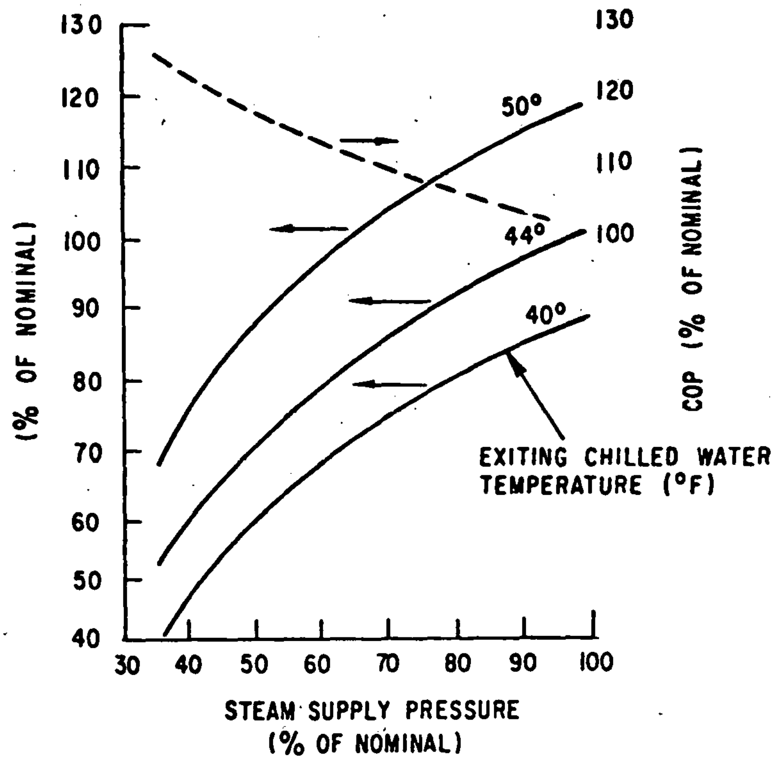


Fig. 3.1 Effect of Steam Supply Pressure on Capacity and COP for Chilled Water Outlet Temperatures from 40° to 50°F*

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*Based on ∿525-ton unit with nominal heat input of 144/123 psig steam corresponding to 12,100 Btu/ton.[10]

Table 3.2 Generalized Equation Coefficients - Percent of Nominal Capacity or COP (Y) Vs Percent of Nominal Steam Supply Pressure (X)**

Chilled Water Outlet Temperature (°F)	Coefficients A	B	C
Capacity, 50	-2.2	2.28	-0.011
Capacity, 44	-6.0	1.91	-0.0085
COP, 44	147.0	-0.69	0.0022
Capacity, 40	-15.8	1.87	-0.0085

**$36 \leq X \leq 100\%$.

pressure is reduced, the maximum available capacity drops; however, the COP at lower steam pressures is higher, as will be discussed in Sect. 3.2.2.

If the heat source is hot water, the equivalent entering and exiting hot water temperatures to provide equivalent capacities at various steam pressures can be determined from Fig. 3.2. The hot water supply flowrate is a function of the required heat input, the difference between the entering and exiting hot water temperature, and the specific heat and density of the hot water at its average temperature in the machine.

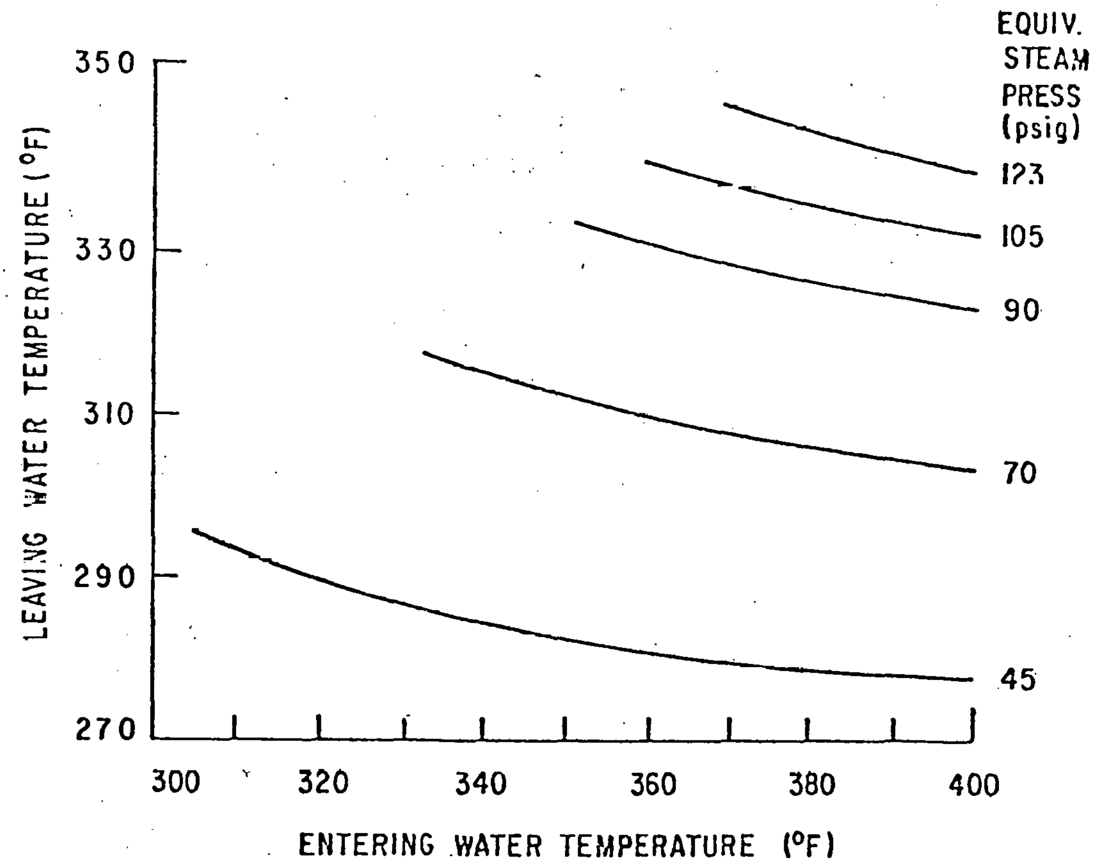

Fig. 3.2 Equivalent Steam Pressures of Various Combinations of Entering and Exiting Water Temperatures[10]

• *Condenser Water Temperature.* A lower condenser water temperature normally will tend to increase the capacity of the absorption machine, as shown by Fig. 3.3. Table 3.3 provides the generalized equation coefficients A, B, and C for Eq. 1.1 to show algebraically the relationship between the cooling tower water temperature, X, and the double-effect absorption chiller nominal cooling capacity, Y.

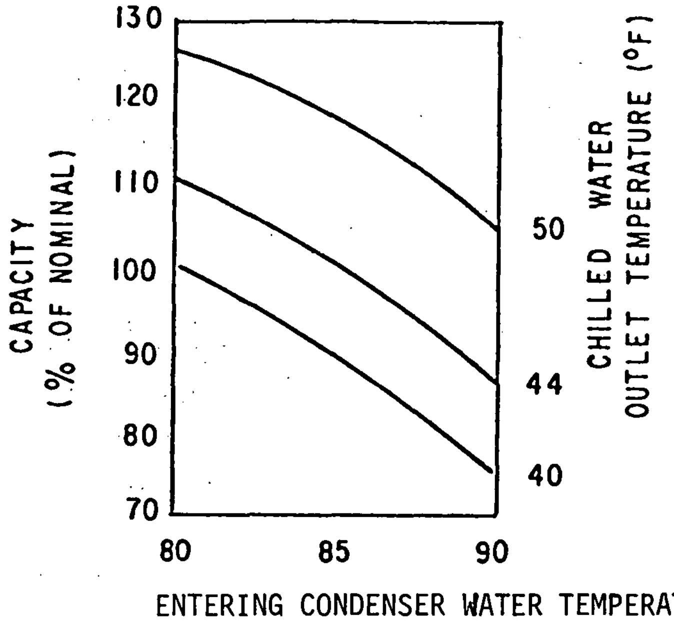

Fig. 3.3 Effect of Entering Condenser Water Temperature on Cooling Capacity of Double-Effect Absorption Chiller*

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*Based on: ∿ 500-ton unit[10] and nominal conditions stated in Sect. 3.1.1.

Table 3.3 Generalized Equation Coefficients - Percent of Double-Effect Absorption Chiller Nominal Cooling Capacity (Y) Vs Cooling Tower Water Temp. (X, °F)** for Three Chilled-Water Temp.

Chilled Water Outlet Temperature (°F)	Coefficients A	B	C
50	-426	14.9	-.1
44	-410	14.5	-.01
40	-556	17.8	-.12

**80 $\leq$ X $\leq$ 90

The effect of entering condenser tower water temperature on the double-effect absorption chiller COP is about the same as the effect on the single-effect chiller COP shown in Fig. 2.5. Based on entering condenser water temperature, the double-effect absorption design takes advantage of the improved efficiency by using automatic controls that limit the steam input to the machine and prevent overfiring. The improvements in the COP as the condenser water temperature drops are shown and discussed in Sect. 3.2.2.

- *Condenser Water Flowrate.* The required flowrate usually is determined on the basis of design tons required and evaporator tons available. The effect of the cooling water flowrate on capacity for the double-effect absorption chillers is similar to that of cooling water flow on single-effect units, as shown in Fig. 2.6.

- *Temperature.* The exiting chilled-water temperature provided by a double-effect absorption unit affects full-load capacity and COP, as shown in Figs. 3.1 and 3.3.

- *Chilled-Water Flowrate.* The chilled-water flowrate, coupled with the number of passes through the evaporator, can alter the avialable capacity anywhere from 90 -- 110% as shown in Fig. 3.4.

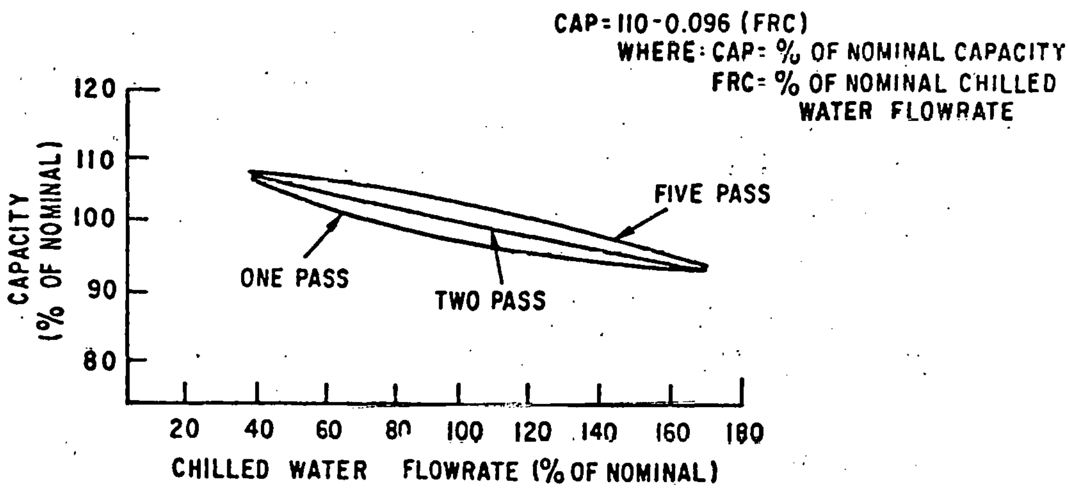


Fig. 3.4 Percentage of Nominal Capacity (CAP) as a Function of the Chilled Water Flowrate Through the Evaporator (FRC)*

*34 $\leq$ FRC $\leq$ 168 (based on ~500-ton unit[10] and other nominal operating conditions listed in Sect. 3.1.1

The wider variance in the flowrates near the nominal value reflects the larger number of possible passes that can be made through the evaporator. The number of passes through the evaporator can vary from one to five.

The COP of the absorption chiller remains relatively constant across ±50% of the nominal chilled-water flow.[8]

3.2.2 Performance at Part-Load and Capacity Control

Figure 3.5 shows that the peak COP occurs when the double-effect absorption machine is operating at 60 -- 70% of the design load. The COP begins to drop off rapidly at about 40% of full load. Control of part-load capacity is accomplished by sensing a decrease in the exiting chilled-water temperature which activates the heat source throttle valve and decreases the absorption chiller generator.

Figure 3.5 should be used with caution (see Sect. 2.2.2). Table 3.4 provides the coefficients A, B, and C for substitution into Eq. 1.1 to represent the algebraic relationship between the percentage of nominal COP (Y) and the percent of full load (X) at various condenser water temperatures.

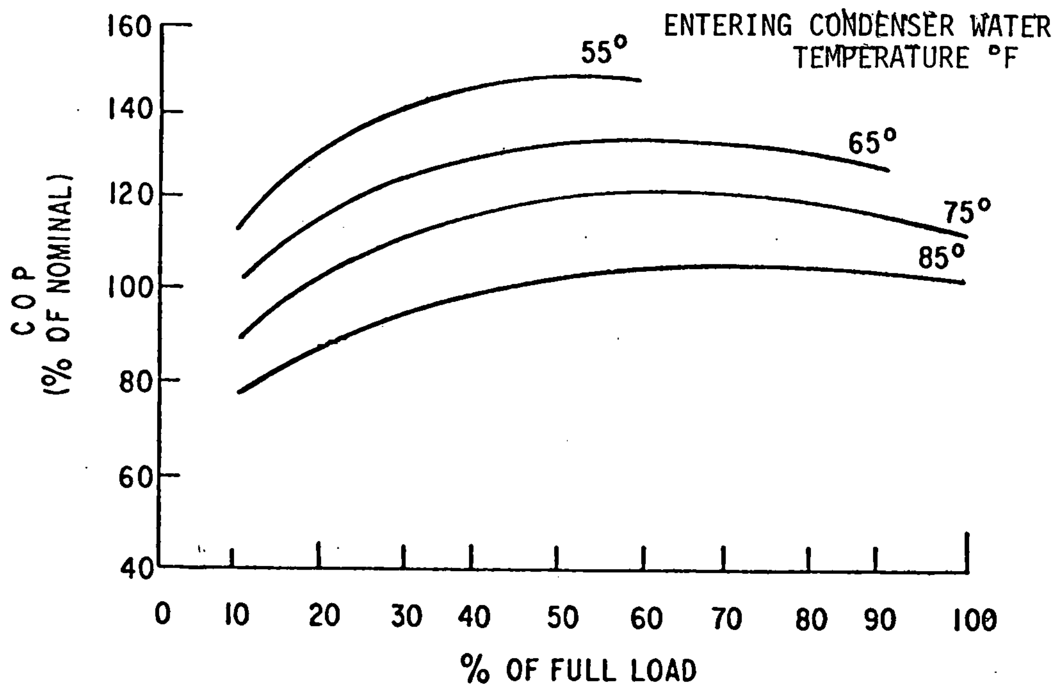


Fig. 3.5 Part-Load Performance at Various Cooling Water Temperatures

Table 3.4 Generalized Equation Coefficients - Percent of Nominal COP Vs Percent of Full Load (X) at Various Entering Condenser Water Temperatures

Cooling Tower Water Temperature (°F)	A	B	C
55	87.00	2.57	-.026
65	83.34	1.68	-.013
75	70.14	1.60	-.012
85	63.00	1.20	-.0083

Manufacturers recommend that machine selection should be made with a combination of equivalent steam pressure, cooling-water temperature, and cooling-water flows so as not to exceed 113% of nominal capacity for the size involved. Operation at higher capacities is not recommended.

3.2.3 Auxiliary Electric Energy Inputs

Figure 3.6 shows the auxiliary electrical energy inputs for the lithium bromide-water circulating pumps required in a double-effect lithium bromide absorption unit.

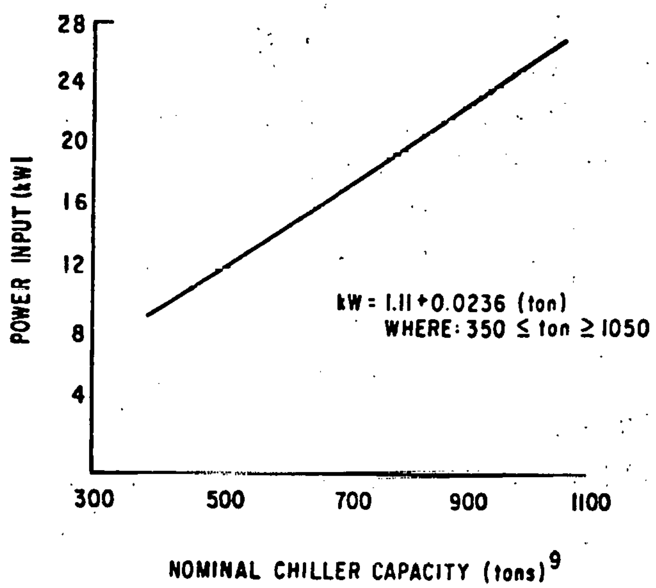


Fig. 3.6 Auxiliary Power Requirement for Double-Effect Absorption Chiller Circulating Pumps; Power Input (kW) Vs Capacity (Tons)

3.3 OPERATING AND MAINTENANCE REQUIREMENTS

- *Automatic Control.* A control sensor is provided to override the unit capacity control circuit and limit power input under conditions of reduced condenser temperature. If energy input were not limited, overfiring of the machine could result under certain conditions. Capacity of the double-effect absorption chiller is limited to 60 percent of nominal with a cooling-water temperature of 55°F. This limit increases to 100 percent of nominal capacity with a condenser temperature of 75°F or greater, as shown in Fig. 3.5.

- *Chilled-Water Flow and Condenser-Water Flow Interlock.* The condenser water pump should be wired such that when the chilled-water flow stops, the condenser-water pump shuts off. This is done as added protection against freeze-up in case of a chilled-water flow failure.

- *Purge Unit.* A purge pump, separated from the machine by a hand valve and a normally-closed solenoid valve, should be provided. This pump must be operated periodically to remove any noncondensible gases that may build up in the machine.

- *Demand Limits.* In typical applications, unless demand is limited, the two-stage generator may draw as much as 150 percent of design steam input upon startup. If the full-load demand is close to heat source availability, a demand limiter is recommended. A demand limiter provides a time delay on the incoming steam or water valve that restricts the maximum demand to about]20 percent of full load.

- *Heat Source.* In all applications, superheat should be limited so that steam temperature does not exceed 400°F. Hot water machines can utilize up to 400°F hot water. Use of steam pressures higher than those recommended may result in overfiring the machine and may lead to operating difficulties or premature machine failure.

• *Water Flows*. Tube water velocities should not exceed 10 ft/sec in copper tubes and 11 ft/sec in cupronickel tubes.

• *Water Treatment*. Good water treatment is necessary to protect the unit from possible tube failure and to maintain design capacity. A fouling factor of 5×10^{-4} is assumed in the performance estimates found in this technical evaluation.

4 ABSORPTION CHILLER RELIABILITY

Large tonnage, single- and double-effect lithium bromide-water absorption machines are comparatively trouble-free and simple to operate. Today's units are manufactured to rigid standards of vacuum integrity and internal cleanliness. Equipment, such as electronic halide leak detectors and helium mass spectrometers, ensures the leaktightness of machines before shipment from the factory.[2]

The pressures within the shell are the vapor pressures of the liquids used in the cycle at their respective temperatures. In operation, the pressure in the absorber and evaporator sections is about 0.01 atmosphere. Pressure in the concentrator and condenser sections is about 0.1 atmosphere. To illustrate the importance of maintaining the machine leak free, introduction of sufficient air to raise the pressure just 0.06 psi will increase the exiting chilled-water temperature by 10°F.

According to one absorption chiller manufacturer, the most common cause of unscheduled shutdown is crystallization caused by: (1) malfunction of system controls, (2) failure of a pressure-reducing valve, or (3) inadvertent introduction of air into the machine. Also, interruption of electric power, which will cause the machine to shut down without the normal dilution cycle, may result in crystallization.

5 SAFETY REQUIREMENTS

All absorption machines must operate in compliance with applicable ASME codes.

6 ENVIRONMENTAL EFFECTS

Absorption chillers are relatively quiet and vibration-free. The major indirect impact is noise and thermal pollution from the cooling towers.

7 COST CONSIDERATIONS

7.1 ESTIMATED CAPITAL COSTS

Figure 7.1 shows the total, installed cost (1976 dollars) of single-effect LiBr absorption chillers ranging in nominal capacities from 100 to 1660 tons. Figure 7.2 shows the total, installed cost of double-effect LiBr absorption units ranging in nominal capacities from 385 to 1060 tons. The equipment costs are from manufacturer representatives in the Knoxville, Tennessee area, and installation costs are from Richardson's *Process Plant Construction Estimating Standards*.[11]

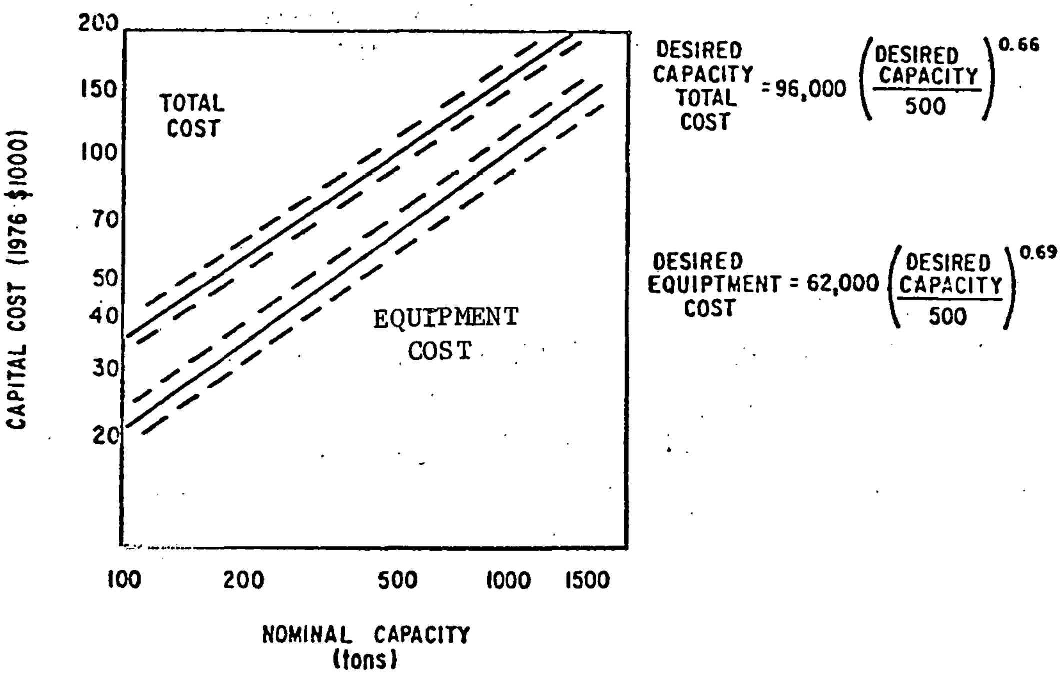
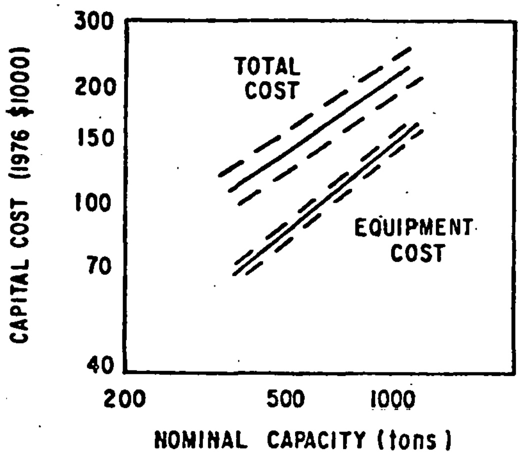


Fig. 7.1 Capital Cost of Single-Effect LiBr Absorption Chillers (Mid-1976 $)



$$\text{DESIRED CAPACITY TOTAL COST} = 135{,}000 \left(\frac{\text{DESIRED CAPACITY}}{500} \right)^{.7}$$

$$\text{DESIRED EQUIPMENT COST} = 85{,}000 \left(\frac{\text{DESIRED CAPACITY}}{500} \right)^{.01}$$

Fig. 7.2 Capital Cost of Double Effect of LiBr Absorption Chillers (Mid-1976$)

The cost of a 3-ton capacity unit at 210°F supplied heat source from Arkla is available for about $3000.[12] The total installation is estimated at about $5000. A 25-ton unit at a design hot water supply temperature of 225°F is available from Arkla for about $11,000.[12] The total installation cost of this unit is estimated at about $18,000. Costs for these two units cannot be estimated by the equations shown in Fig. 7.1.

The equipment costs include manufacturer's recommended auxiliary equipment such as economizer, demand limiter (see Sect. 3.3), and positive concentration limiter (control that shuts off the machine when the conditions tend toward the possibility of crystallization.) The piping and controls feeding in and out of the absorption units are assumed to be similar to the piping and control arrangement suggested in manufacturers' data sheets on absorption systems. A total cost is equal to the equipment cost plus the installation cost.

The installed costs include the cost of: (1) concrete mounting pads; (2) condenser piping; (3) mounting of steam or hot water control valve;

(4) necessary insulation of refrigerant water box, pump casings and connections; and (5) electrical hookup for power and control. The total cost does not include: (1) openings into buildings for admission of equipment and patching of any opening(s) made, or (2) proper rigging equipment.

7.2 OPERATING AND MAINTENANCE COST

The O&M cost for both single- and double-effect absorption chillers is assumed to equal the cost of a full service maintenance contract given in Fig. 7.3.[13] The service contract costs include a preventive maintenance program consisting of three annual inspections, machine lubrications, cleaning, and annual removal of the heads to inspect the absorber and condenser tubes.

The economic life of an absorption chiller is assumed to be 20 years.[14]

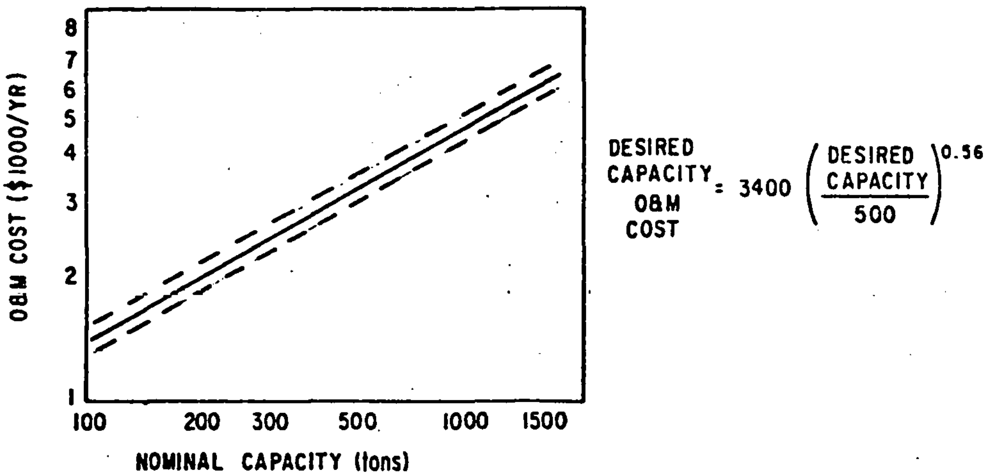


Fig. 7.3 Operating and Maintenance Cost for Both Single- and Double-Effect Absorption Chillers (Mid-1976$)

8 STATUS OF DEVELOPMENT AND POTENTIAL FOR IMPROVEMENT

8.1 GENERAL

Improvements in absorption systems which might be sought include: (1) reducing the generator temperature required for operation: (2) increasing the allowable cooling water temperature; and (3) increasing the COP. These objectives tend to be mutually exclusive; although some improvements can be achieved in each of these areas by increasing the sizes of heat exchange surfaces, this is generally unacceptable because of the associated increase in cost.[6]

Theoretical analysis of the absorption cycle can be found in Refs. 1 and 6. The theoretical limit on performance is the ideal Carnot system (Carnot engine and refrigerator). The theoretical analysis reveals that the COP of absorption systems is relatively insensitive to variations in generator, absorber, and condenser temperatures when the temperatures are sufficient to sustain operation.[6] The potential improved performance of absorption chillers coupled to a low temperature heat source by utilizing a refrigerant storage system has been reported.[6] The concept has potential for storing liquid refrigerant when the waste heat from prime movers in total energy systems is greater than the heat requirements elsewhere.

The stored liquid refrigerant can be released into the evaporator, as necessary, to satisfy more of the total cooling demand with absorption units rather than compressive chillers. This scheme is attractive because:[15]

(1) the volume required to store the cooling capability is small due to the high heat of vaporization of the water; and

(2) the water is stored at near ambient temperatures where heat losses to the stored refrigerant are minimal.

Finally, additional analyses of refrigerant cold storage systems are required to determine the most cost-effective heat exchanger and heat dissipation capacity as well as most-effective mass flowrates.[6] Such analyses should also examine other working fluids, such as ammonia-water, which may be better suited to the refrigerant storage mode of operation.

A 1975 assessment of solar-powered cooling of buildings,[16] recommend that R&D studies of double-effect machines driven by medium concentration collectors should be carried out because of the higher COPs attainable.

Another study[17] suggests that there are not enough data to analyze different refrigerant and absorber combinations other than ammonia and water and LiBr and water. Other combinations may look promising, but combinations of organic substances have been eliminated in the past because, in workable temperature ranges, these combinations decompose. However, with lower heat source temperatures they may be very good. Much of this information is proprietary, and therefore not readily available. A brief discussion on other refrigerant-absorbent pairs that have been investigated and the desirable characteristics of each can be found in Chapt. 1 of Ref. 1.

8.2 INTEGRATION INTO ICES

The classical application of absorption chillers in total energy systems is waste-heat application from onsite electric generator-prime mover sets. When waste heat is available and chilled water is called for, the heat is run through the absorption chiller generator. If additional cooling is required, generally some type of compressive chiller is turned on. To employ an absorption chiller in an ICES more effectively, the object must be to cut into the percentage of cooling capacity provided by mechanical-compressive processes. There are four major ways of doing this:

(1) *Thermal Storage* - both high temperature waste heat source storage and low temperature refrigerant storage.

(2) *Solar assisted absorption systems.* Since the heavy cooling load generally occurs on hot sunny days, the load match is not as great a problem as with solar heating.

(3) *Utilizing the absorption unit as an intermediate heat exchanger between hot and cold zones within buildings.* The application of this system offers energy savings during those times in which part of a building is calling for heat, while others are calling for cooling. The condenser water can be circulated through the building spaces calling for heat while the chilled water is sent to the inner core or other building zones calling for cooling. This application is called "Thermal Gain Absorption" by the York Corporation.[18]

(4) *Integration with compressive chillers*. A number of absorption units have been installed with compressive chillers run by back pressure steam turbines. The base cooling load is met by the compressive chiller and the waste heat in the form of back pressure steam (12 psig) is run through the single effect absorption unit. Systems of this sort are attractive for application of very large tonnages and where very little waste heat is available when space cooling is called for by the community.

B AMMONIA-WATER ABSORPTION CHILLERS

A preliminary survey of ammonia-water absorption chiller technology has led to the conclusion that this is not a technology of high-priority interest for ATMES application. Thus, this survey will be completed at a later date as a part of near-term technologies. Some characteristics of ammonia-water units which form a basis for this decision are summarized below.

The only U.S. commercially available ammonia-water absorption systems are direct gas-fired units. A number of European companies are reported to manufacture ammonia-water absorption units but, with ammonia classified in ANS B9 Safety Code Group 2 which restricts its use inside dwellings, ammonia-water absorption systems have been unable to compete with the lithium-bromide units in this country.

Compared to LiBr-water units operating at heat source temperatures above 180°F, the ammonia-water system has the following additional disadvantages which have inhibited commercialization for space conditioning:

(1) internal pressures and associated pumping power requirements are higher,[18]

(2) the system is more complex - a rectifier is required for the separation of ammonia and water vapor at the generator outlet rather than the simple still in LiBr units,[18] and

(3) COP is lower.[6]

The ammonia-water refrigerant-absorbent combinations for solar powered absorption air conditioning do appear to be a prime candidate, since the generator may be operated with a heat source temperature less than 180°F.[6]

In conclusion, ammonia-water abosrption chillers is not considered to be a commercially available technology applicable to the ATMES program. Further evaluation will be completed at a later date as a part of near-term technology.

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