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A SURVEY OF ABSORPTION COOLING TECHNOLOGY IN SOLAR APPLICATIONS

Paul C. Auh

July 1977

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ABSTRACT

A comprehensive survey of the current state of the absorption cooling technology has been conducted. This survey discusses the basic and applied absorption cooling/heating technology, analyses the current state of the art including the discussion of limitations and possible solutions, identifies areas where promising developments are indicated, lists the current products and activities of the absorption industry, and presents the current RD & D efforts of the U. S. government.

The main subjects covered in the survey are as follows: Principles of absorption cooling technology (NH_3-H_2O cycle and H_2O-LiBr Cycle), Adaptation of absorption cooling technology for solar cooling applications, Thermal performance of absorption cooling units, Comparison of NH_3-H_2O absorption with H_2O-LiBr absorption, Commercially available solar absorption units, General trends of the absorption cooling industry toward solar application, Absorption cooling system performance in actual installations, Limitations of absorption cooling technology, Solar-powered absorption heat pumps, and U. S. ERDA activities relating to solar absorption cooling.

The treatment of the above subjects is intended to be basic and comprehensive in order that the general readers may understand the current aspects of absorption technology in solar cooling applications.

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Introduction

Solar cooling of buildings seems very attractive since solar energy is available at preferrable environmental conditions during the hot summer days. The commercial success of solar cooling systems, however, depends largely on its economic viability as well as its technical feasibility.

There have been several different approaches to solar cooling technologies. Although there are large numbers of variations to each approach (active system), they can be basically classified as: Rankine/vapor compression cycles, heat pump cycles, adsorption/desorption cycles, humidification/dehumidification cycles and absorption cycles. Numerous technical papers are available on the above subjects, some of which are listed in Reference 1.

Active R & D work is being pursured in all of the above categories in the U.S.A. Solar Division of ERDA is sponsoring at least one project in each category and plans to expand such activities vis RFPs and PRDAs. At the present time none of the cooling systems mentioned above is commercially available except solar absorption systems in a limited quantity.

The absorption technology has been a significant contribution to solar cooling since the technology has been well developed and its technical feasibility resulted in commercial products which are currently available.

In this paper, discussion will be focused on absorption systems only. The treatment of the subject will be basic and comprehensive so that the general readers could understand all the current aspects of the absorption technology in solar cooling applications.

1. What is "Absorption Cooling Unit" and How does it work?

Absorption cooling unit is a hardware built on the principles of one of the thermodynamic absorption cooling cycles. Although the comprehensive treatment of the subject can be found in References 2 and 3, two basic cycles will be vriefly described for the quick reference. These are NH_3-H_2O and H_2O-LiBr cycles, which have been commercially successful.

1.1 NH_3-H_2O Absorption Cycle

The cycle was originally developed for gas-fired units. Air-cooling of absorber/condenser components was possible due to higher grade energy input than that from the flat plate solar collectors. The schematic diagram of the cycle is shown in Fig. 1.1-1.

The cycle works as follows. Nearly pure ammonia vapor from the rectifier is liquefied in the condenser. Condenser heat is removed by the method of air cooling. Saturated liquid from the condenser is cooled down further through the refrigerant heat exchanger (RHX) and is expanded into the evaporator. Sudden pressure decrease across the expansion valve causes refrigerant boil inside the evaporator, thus absorbing heat from the chilling medium, (i.e., cooling effect).

Ammonia vapor from the evaporator is then absorbed by the ammonia-poor liquid in the absorber, thus releasing absorption heat. Ammonia-poor liquid from the generator gives up some heat to the analyzer while passing through the analyzer heat exchanger and further cools down by exchanging heat with ammonia-rich solution from the absorber. This ammonia-poor solution is eventually mixed with ammonia vapor in the absorber to produce ammonia-rich solution. It is then pumped via rectifier and solution heat exchanger to the analyzer.

Enthalpy of the ammonia-rich solution increases as it passes through the rectifier and the solution heat exchanger. Absorber heat is removed by the ambient air. The ammonia-rich solution of feed solution is distilled in the analyzer. The ammonia-rich vapor from the top of the analyzer is further purified in the rectifier by the condensation of water vapor in the mixture.

The results of the detailed analysis by the mass and energy balance are tabulated in Tables 1.1-1 and 1.1-2. The number at each point in Fig. 1.1-1 corresponds to that in Table 1.1-1. The numerical values in paranthesis in Table 1.1-1 are the initially given values and others are derived. The following assumptions are made reasonable and realistic.

(a) Equilibriums at points 1, 4, 9, 10 and 11

(b) Saturated liquids at points 2 and 6a

(c) RHX temperature approach --- $30^{o}F$

(d) SHX temperature approach --- $5^{o}F$

(e) No pressure drop except across valves

The cycle performance (CP) is the ratio of QEVA to QGEN and the coefficient of performance (COP) is based on the 75% generator efficiency. The COP of about 0.4 is the typical value of the commercially available, air-cooled gas absorption unit. The decrease in COP due to flue losses in the gas-fired unit is not encountered in solar-powered unit. For the cycle described above, for a typical gas-fired unit, it is evident that the concentrating solar collector is needed to operate the air-cooled NH_3-H_2O cycle. The same analyses can be also performed at lower generator temperatures.

1.2 H_2O-LiBr Absorption Cycle

Although the operational principles are the same as the previous one, H_2O-LiBr cycle is less complicated than the NH_3-H_2O cycle; i.e. a simple boiler could replace the distillation column (generator, analyzer and rectifier) in NH_3-LiBr cycle because of the non-volatile nature of the LiBr. Here, water is the refrigerant and solution contains lithium bromide dissolved in water.

Referring to Figure 1.2-1, heat is applied to the generator to vaporize water (refrigerant) from the mixture of H_2O-LiBr solution. Water vapor from the generator then passes to the condenser, where it condenses to liquid water, liberating the heat of condensation. Cooling tower water is generally used for the dissipation of condenser heat. Liquid water from the condenser passes through the expansion valve, and it immediately vaporizes in the evaporator due to the sudden pressure drop across the expansion valve. The requirement of evaporation heat of water in the evaporator, thus, produces the cooling effect.

At the absorber, water vapor from the evaporator is absorbed in the water-poor solution coming from the generator via the solution heat exchanger (SHX). Here, heat of absorption is also dissipated to the cooling tower water. Resulting water-rich solution from the absorber is then pumped back to the generator via the solution heat exchanger. The solution heat exchanger (SHX) is employed to reduce the generator heat input and to decrease the amount of absorber heat dissipation, thus, improving the cycle COP. Pressure reducing valve is also necessary between the solution heat exchanger and the absorber to maintain the high and low pressure sides of the absorption cycle.

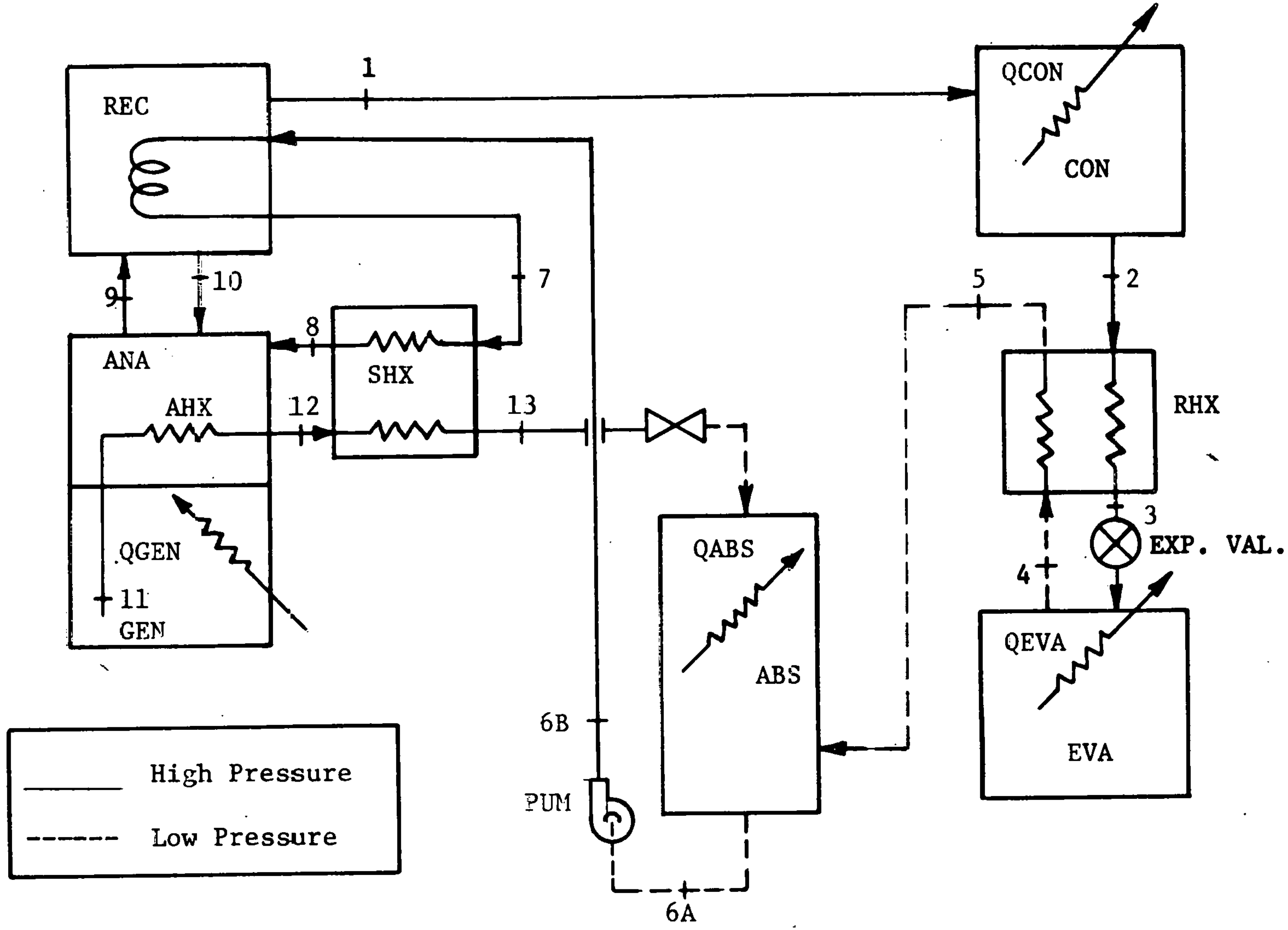
The cycle diagram is shown in Fig. 1.2-1 and the corresponding analysis is included in Tables 1.2-1 and 1.2-2 with the following assumptions.

(a) Equilibriums at points 2, 4, 5, and 7

(b) SHX temperature approach --- $10^{o}F$

(c) No pressure drop except across valves

The COP of the given cycle, 0.78, can be comparable with the COP of 0.72 of the commercially available solar-powered H_2O-LiBr units. The absorber/ condenser components of this cycle can not be air-cooled due to the low operating absorber temperature at 90^oF.



REC	(Rectifier)	CON	(Condenser)
ANA	(Analyzer)	RHX	(Refrigerant Heat Exchanger)
AHX	(Analyzer Heat Exchanger)	EVA	(Evaporator)
GEN	(Generator)	PUM	(Solution Pump)
SHX	(Solution Heat Exchanger)	ABS	(Absorber)

EXP. VAL. (Expansion Valve)

Fig. (1.1-1) Standard NH_3-H_2O Absorption Chiller[19]

Table 1.1-1

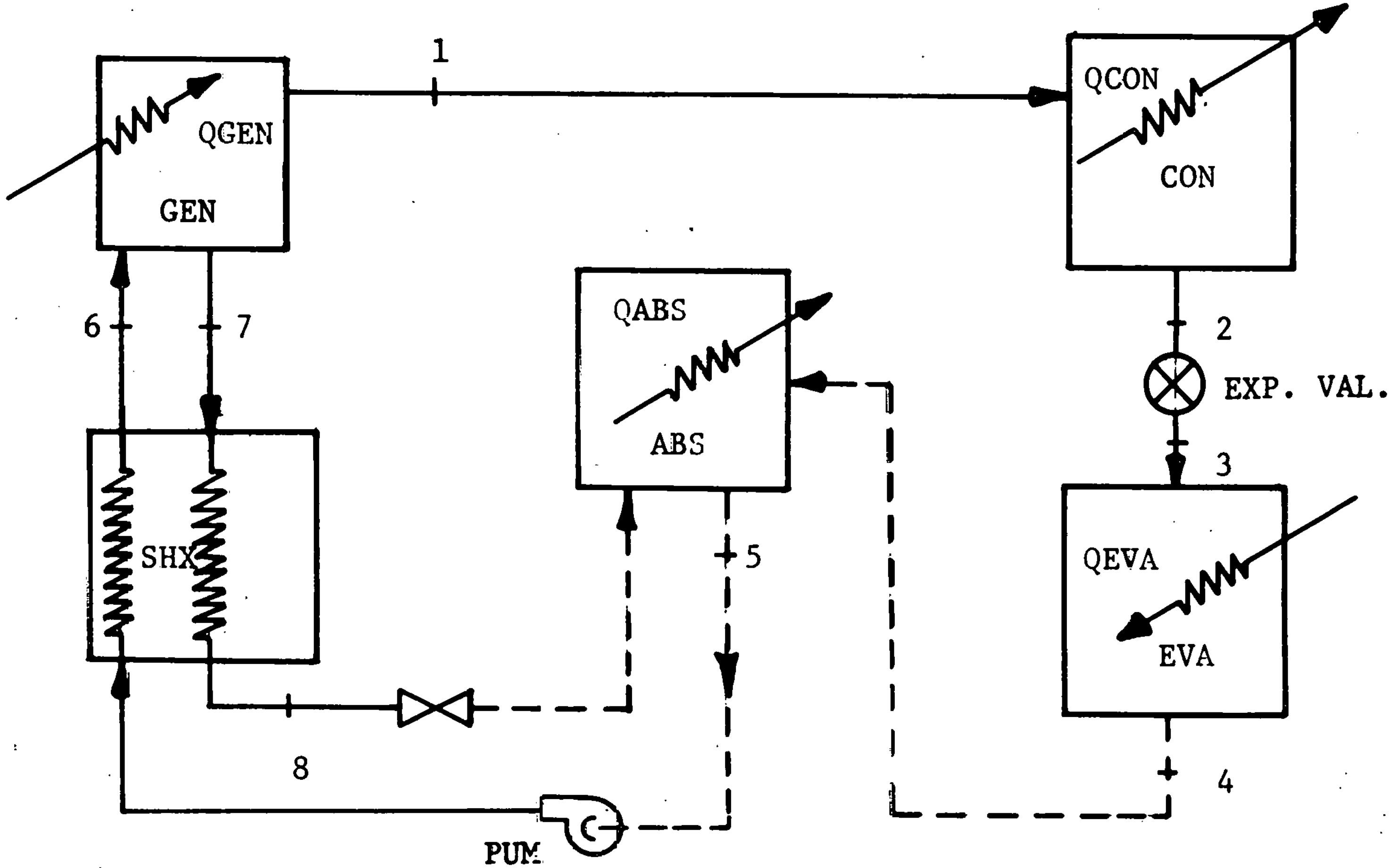
Mass and Energy Balances of Standard NH_3 -H_2O Absorption Cycle [19]

pt	P (psia)	T (^oF)	W (lb/min)	Z (lb-NH3)/(lb-Sol'n)	E (Btu/lb)	TE (Btu)
1	300	(157)	(1.000)	0.9950	589	589
2	300	123	1.000	0.9950	103	103
3	300	73	1.000	0.9950	43	43
4	(60)	(35)	1.000	0.9950	522	522
5	60	93	1.000	0.9950	582	582
6A	60	(134)	3.646	0.360	10	36
6B	300	136	3.646	0.360	11	40
7	300	177	3.646	0.360	57	208
8	300	248	3.646	0.360	137	500
9	300	(248)	1.136	0.922	680	772
10	300	(238)	0.136	0.385	123	17
11	(300)	(354)	2.646	0.120	300	794
12	300	(283)	2.646	0.120	225	595
13	300	182	2.646	0.120	115	304

Table 1.1-2

Mass and Energy Balances of Standard NH_3-H_2O Absorption Cycle [19]

	Q (Btu/min)		Q (Btu/min)
QEVA	479	QAHX	198
QGEN	850	QSHX	291
QPUM	4	QRHX	60
QABS	850		
QCON	486	CP	0.564
QREC	167	COP	0.423



High Pressure

Low Pressure

GEN (Generator)
CON (Condenser)
EVA (Evaporator)
ABS (Absorber)
SHX (Solution Heat Exchanger)
PUM (Solution Pump)
EXP. VAL. (Expansion Valve)

Fig. (1.2-1) Standard H_2O-LiBr Absorption Cycle

Table 1.2-1

Mass and Energy Balances of Standard H_2O-LiBr Absorption Cycle[2]

pt	P (mmHg)	T (^oF)	W (lb/min)	Z $\left(\frac{\text{lb-LiBr}}{\text{lb-Sol'n}}\right)$	E (Btu/lb)	TE = (W x E) (Btu)
1	66	192	(1.0)	0.0	1147	1147
2	66	(110)	1.0	0.0	78	78
3	6.3	40	1.0	0.0	78	78
4	6.3	(40)	1.0	0.0	1079	1079
5	6.3	(90)	12.2	0.56	-75	-915
6	66	163	12.2	0.56	-38.3	-467
7	66	(192)	11.2	0.61	-30	-336
8	66	100	11.2	0.61	-70	-784

Table 1.2-2

Mass and Energy Balances of Standard H_2O-LiBr Absorption Cycle

	Q (Btum)		Q (Btum)
QEVA	1001	QCON	1069
QGEN	1278	QSHX	448
QABS	1210	COP	0.78

2. How can the absorption unit be adapted to solar cooling?

2.1 NH_3-H_2O Absorption Units

Gas-fired absorption units have been developed exclusively in the past for the residential or light commercial applications. Naturally, the choice was NH_3-H_2O unit since the absorber/condenser components can be air-cooled at the generator temperature of the gas-fired units. Although air-cooling reduces COP of the unit significantly, economic considerations of the residential-size unit favors the elimination of the cooling tower, which is required for water-cooled units.

The generator temperature of the gas fired unit ($\sim 350^{\circ}F$) is usually too high to be adapted to solar cooling system using flat plate solar collector whose collection temperature is limited up to about $210^{\circ}F$. Lower generator temperature is possible in the air-cooled mode by increasing the ammonia concentration of the weak and strong solutions. Lowering of generator temperature, however, would require larger heat transfer surfaces in condenser and absorber, and more electrical power for solution pump and fan.

The practical range of the generator temperature of the air-cooled unit may lie between $250^{\circ}F$ and $350^{\circ}F$. In the case of water-cooling of absorber/ condenser components, generator temperature may range from $200^{\circ}F$ to $250^{\circ}F$. Water-cooled NH_3-H_2O units, however, are not preferred over water-cooled H_2O-LiBr units because of its lower COP, higher pump work and toxic nature of NH_3 refrigerant (i.e., direct expansion coil is not allowed) as well as the higher complexity of the unit. Table 2.1-1 shows the theoretical comparison between H_2O-LiBr unit and NH_3-H_2O unit both water-cooled).[4] At the same operating conditions, COP of the NH_3-H_2O unit is 0.57 compared with 0.75 of

Table 2.1-1

Comparison between H_2O-LiBr and NH_3-H_2O Absorption Units (both water-cooled)[4]

	H_2O-LiBr Unit		NH_3-H_2O Unit	
	pt in Fig. (1.2-1)		pt in Fig. (1.1-1)	
Absorber	5	92°F (54%)[b]	6A	90°F (55%)[c]
Evaporator	4	42°F	4	42°F
Condenser	2	105°F	2	105°F
Generator	7	179°F (58%)	11	179°F (50%)
Recirculation Rate[a]		13.5		9.0
HX Effectiveness		75%		75%
COP		0.75		0.57
Pump Head		6 ft-H_2O		444 ft-H_2O
Pump Work (Theor.)		0.0005 HP/ton		0.055 HP/ton

(a) Recirculation Rate = rate of weak solution circulated per unit rate of refrigerant generated

(b) Unit = lb-LiBr/lb-sol'n

(c) Unit = lb-NH_3/lb-sol'n

H_2O-LiBr unit and the pumping power requirements of the NH_3 unit is more than 100 times that of the LiBr unit.

In summary, air-cooled NH_3-H_2O unit can be operable at the generator temperature range of 250 to 350°F, which suggests that the concentrating-type solar collector should be used. The attainable COP of the unit may range from 0.5 to 0.7 at the generator temperature range mentioned above. Major modifications necessary to change from gas-fired to hot-water powered, occurs in generator where heat exchange mode changes from gas-to-liquid to liquid-to-liquid.

2.2 H_2O-LiBr Absorption Units

H_2O-LiBr units have been developed for both small residential (3 tons of cooling) and large industrial applications (up to 1700 tons of cooling). These units can either be hot-water or steam powered.

Due to the unit operation at lower generator temperatures as seen in Table 1.2-1, the LiBr unit has been the primary candidate for solar cooling applications. The major disadvantage of LiBr unit is the requirement of the cooling tower since the lower absorber/condenser temperature is required to avoid the crystallization of LiBr. Crystallization occurs at the discharge end of the solution heat exchanger to the absorber, and it would hinder solution flow through the said heat exchanger. Referring to Table 2.1-1, if we assume absorber/condenser temperature of 125°F for air-cooling while the evaporator temperature is fixed at 42°F, the concentration of the saturated strong solution at absorber outlet is 64% and the saturation temperature of 64% strong solution at the condenser pressure (100 mmHg), is 225°F. Assuming the same recirculation rate of 13.5, the calculated concentration of the weak solution saturated at 100 mmHg should be 69%, which exceeds the crystallization concentration of 68% at 100 mmHg. At the recirculation rate of 16, the weak solution concentration becomes 68% and its corresponding equilibrium temperature would be 245°F at 100 mmHg.

In summary, air cooling of LiBr unit is not possible due to the crystallization of LiBr and also increased generator input temperature. Water-cooled LiBr units can be operable at the generator input temperature of 170 to 210°F from the flat plate solar collector. Attainable COP of such units may range from 0.6 to 0.8.

Most industrial H_2O-LiBr units have been designed to have the generator input temperature greater than that expected from flat plate solar collectors. Typical low pressure steam temperature from gas or oil boilers is 244°F at 12 psig. Temperatures of other sources, such as hot water could be about 250°F.

Although commercially available LiBr-H_2O units were designed for higher heat input temperatures, they can be used for solar application without major modification. Capacity of the unit, however, is decreased as the temperature level of the hot water to the generator is reduced. It, in turn, reduces the amount of energy input to the generator.

Figure 2.2-1 shows the cooling capacity of non-residential units as a function of generator inlet water temperature.[5] Although the derated capacity of the units varies among different manufacturers, the average values were used in the plot to merely show the general trends. The unit has 100% cooling capacity at the generator inlet water temperature of 240°F. With the 43°F reduction in generator inlet water temperature, 50% of the nominal capacity is lost or derated.

In summary, industrial H_2O-LiBr units today could be used for solar applications with no equipment modifications, but with derated capacity. Major R&D work is, however, needed to modify the existing absorption units for the optimum solar applications.

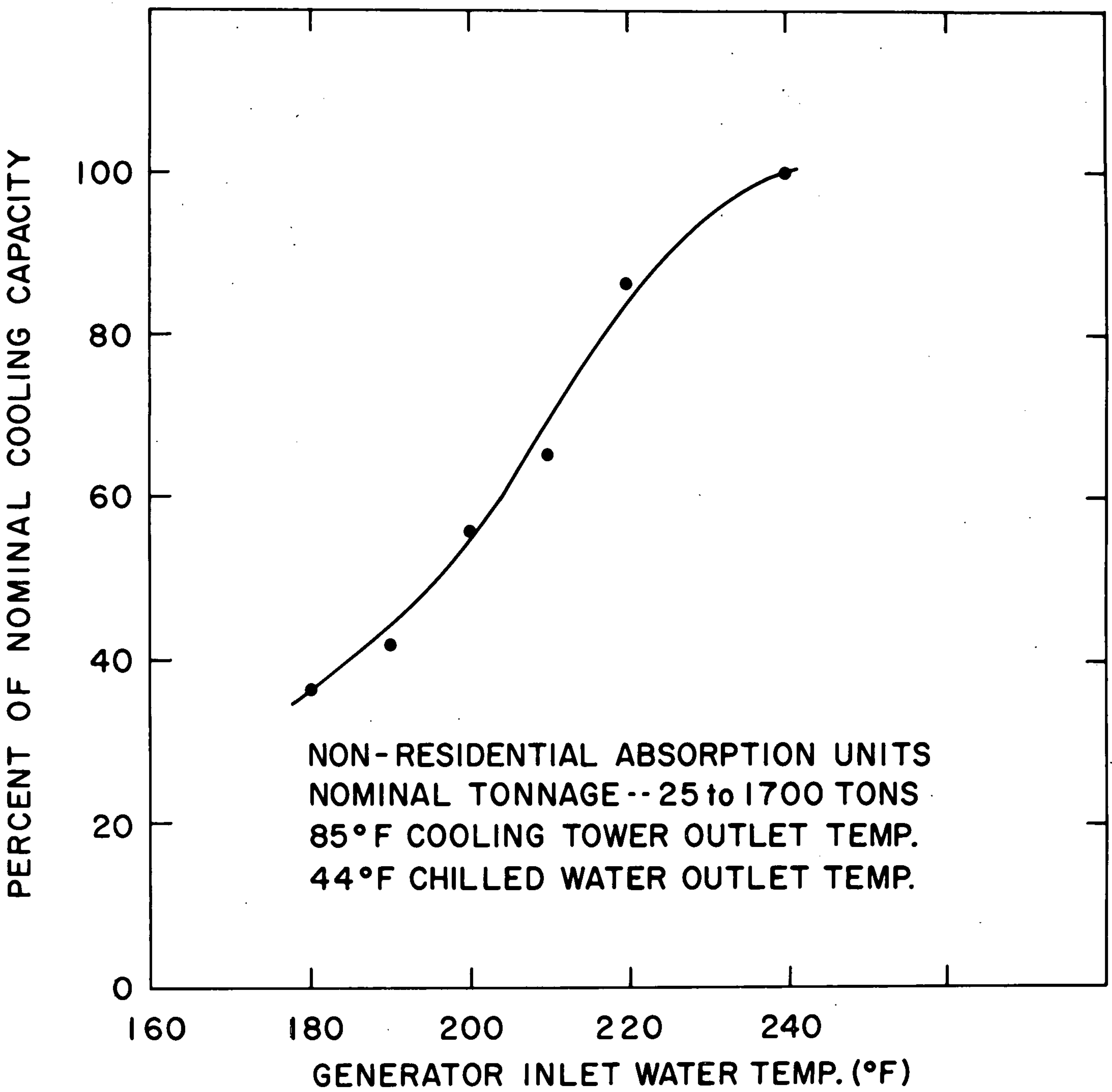


Fig. (2.2-1) Performance of Non-Residential Absorption Units as a Function of Generator Inlet Water Temperature[5]

3. What could be the maximum performance of the absorption unit?

3.1 Carnot Cycle

The heat operated absorption refrigeration cycle can be regarded as a combination of heat engine cycle and mechanical refrigeration cycle. The ideal coefficient of performance for the combination of the above cycle is:

$$COP = \left(\frac{Tgen - Tsink}{Tgen}\right)\left(\frac{Tevap}{Tsink - Tevap}\right) \qquad \text{Eq. 3.1-1}^2$$

The above equation applies to the Carnot Cycle with the following assumption.

Tsink = Tabsorber = Tcondenser

Figure 3.1-1 is a plot of Eq. 3.1-1. The Carnot Cycle COP of absorption unit was plotted against the generator temperature with the heat sink temperature as a parameter. The evaporator temperature of the absorption chillers could range from 35°F to 40°F. The cross-hatched block in the figure represents the region of possible solar application. The generator temperature range ($\sim$180°F to 250°F) was so chosen that it includes the operating range of H_2O-LiBr and NH_3-H_2O units. The heat sink temperature range of 85°F $\sim$ 125°F could cover both air-cooling and water-cooling. From Figure 3.1-1, it is easily seen that the Carnot Cycle COP of absorption unit could range from 0.5 to 2.5 at the realistic operating conditions for solar applications. (See the cross-hatched block).

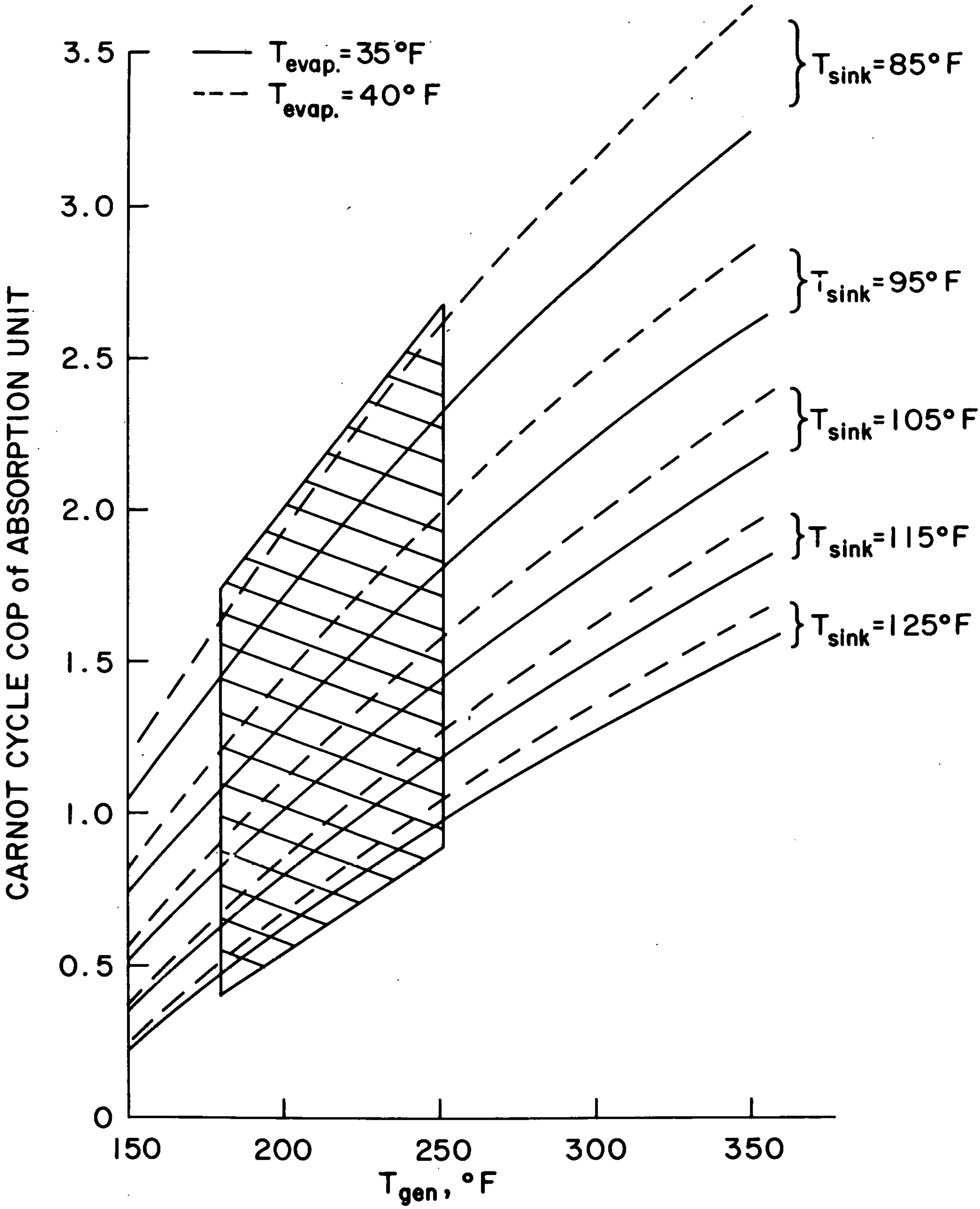


Fig. (3.1-1) Carnot Cycle COP of Absorption Unit as a Function of Generator Temperature

3.2 Non-Ideal Cycles

The Carnot Cycle discussed in Section 3.1 is an ideal, reversible one, thus achieving maximum theoretical COP.

The practical absorption cycles employing either NH_3-H_2O or H_2O-LiBr can not achieve Carnot performance due to the hardware limitation, irreversible process, working fluid properties, etc. The COP of absorption unit is defined as:

$$COP = \frac{Qevap}{Qgen} = \frac{\text{Cooling Effect}}{\text{Generator Heat Input}} \qquad \text{Eq. 3.2-1}$$

or

$$COP \doteqdot \frac{\Delta Hvap}{\Delta Hvap + (\Delta Hdil)(Rr + 1)} \qquad \text{Eq. 3.2-2}$$

where ΔHvap = heat of vaporization of refrigerant (NH_3 or H_2O), Btu/lb of refrigerant

ΔHdil = heat of dilution, Btu/lb of solution

Rr = recirculation rate

= mass rate of solution leaving generator per unit mass rate of refrigerant leaving generator

Equation 3.2-2 states that the value of COP is always less than 1.0 unless the value of ΔHdil is zero or negative. Heat of dilution (ΔHdil) of ideal solution is zero. Therefore, the theoretical COP of the cycle employing ideal solution, could be 1.0. Non-ideal solution has either negative or positive deviation from the Raoult's law. Both NH_3-H_2O and H_2O-LiBr solutions have negative deviations and the heats of solution or heats of dilution are exothermic. i.e.: $\Delta Hdil > 0$

Therefore, the theoretical maximum COP of the absorption cycle using conventional working fluids such as NH_3-H_2O and H_2O-LiBr can not exceed 1.0.

The fluid combination exhibiting positive deviation from the Raoult's law has endothermic heat of solution and the term, ΔHdil, becomes negative. Then, theoretically, the COP of Eq. 3.2-2 could exceed 1.0. At the present time, no such pairs have been identified for commercial use.

In brief, maximum theoretical COP of the absorption unit using presently available working fluid could not exceed 1.0 and the practical COP of the commercially available single-effect units could range from 0.5 to 0.8, mainly depending on the type of absorption cycle employed, air-cooling or water cooling of absorber/condenser components, and the kind of working fluids used.

4. What is the difference between NH_3-H_2O absorption units and H_2O-LiBr absorption units?

Both NH_3-H_2O and H_2O-LiBr absorption units have been commercially available. NH_3-H_2O units are usually gas-fired and are favoured for residential applications since the absorber/condenser components can be air-cooled. H_2O-LiBr units, on the other hand, use hot water or low pressure steam as their power source, and industrial application of such units have been common due to the requirement of cooling towers.

For the solar application of absorption units, H_2O-LiBr units have been preferred to NH_3-H_2O units mainly because the former units can be operable at lower generator temperature, which can easily be obtained from flat plate solar collectors.

4.1 Comparison between NH_3-H_2O units and H_2O-LiBr units

a) Water-cooling versus Air-cooling

Ammonia-water absorption units are the only commercially available ones that can be air-cooled. As mentioned earlier in Section 2.1, air-cooling of NH_3-H_2O units may be practical in the generator temperature range of $250 \sim 350^{\circ}F$. Again, water-cooling of NH_3-H_2O units is possible in the generator temperature range of $200 \sim 250^{\circ}F$, but is not preferred over water-cooled H_2O-LiBr units due to the reasons explained in Section 2.1.

Water-Lithium bromide units can <u>not</u> be air-cooled because the LiBr concentration in the generator could exceed the crystalization limit of LiBr when the temperature of absorber/condenser components is increased for air-cooling. The detailed explanation is contained in Section 2.2. The H_2O-LiBr units, thus, require water-cooling of their absorber/condenser components and the cooling tower is normally employed. Maintenance of the cooling tower is also essential to control the scaling and corrosion problems. Addition of the cooling tower to the small size absorption unit could increase the initial cost of the unit significantly.

b) Solution Pump

Mechanical solution pump is needed for NH_3-H_2O unit to pump the working fluid from the absorber pressure to the generator pressure. The solution pump is driven by electric motor, thus, consuming extra electric power. Theoretical pump work of NH_3-H_2O unit is on the order of 0.05 HP/ton.

Water-Lithium bromide units could use a vapor-lift pump, and gravity return of solution from absorber to generator could be possible due to the small pressure difference between the high and low pressure sides of the unit. The theoretical pump work requirement of this unit is on the order of 0.0005 Hp/ton, which is about 1/100 of NH_3-H_2O unit. Commercially available H_2O-LiBr units, often employ mechanical solution pump. It is well known that the reliability of the solution pump is the key to the commercial success of the absorption units.

c) Direct Expansion Evaporator (D-X coil) versus Chilled Water Evaporator

Ammonia is considered flammable and toxic and is not recommended to be used in D-X coil, which has direct contact with the air to be conditioned. Therefore, separate chilled water loop has to be added, subsequently increasing the cooling system cost.

In H_2O-LiBr unit, the refrigerant is water and the D-X coil can be safely employed. In practice, however, the chilled water loop is also employed with the H_2O-LiBr unit since there is no sufficient pressure head available for the refrigerant-vapor to travel from evaporator to absorber especially when the distance between them is significant as in split systems.

The advantage of employing D-X coil to either NH_3-H_2O or H_2O-LiBr unit, thus may not be practiced or realized. The need of cold-side storage of chilled water in the absorption cooling system is a must to minimize the deterioration of system COP due to the cycling of the unit, especially when the cooling load is low. It means that the chilled water loop is also a must to the absorption unit.

d) Coefficient of Performance (COP)

As already indicated, the possible COP of the NH_3-H_2O unit ranges from 0.5 to 0.7 at the generator input temperature range of $250 \sim 350^{o}F$ with air-cooling of the absorber-condenser components.

The possible COP of the water-cooled H_2O-LiBr unit varies from 0.6 to 0.8 at the generator input temperature range of $170 \sim 250^{o}F$.

e) Other Features

Separation of NH_3 from NH_3-H_2O solution is a distillation process since both NH_3 and H_2O are volatile. NH_3-H_2O unit, thus, requires additional components such as analyzer and rectifier to further purify NH_3-rich vapor from the generator. In addition, a refrigerant heat exchanger (RHX) is usually employed to NH_3-H_2O unit in order to increase the unit COP. See Figure 1.1-1.

In the H_2O-LiBr unit, H_2O is volatile but LiBr is not. Therefore, the generator component can be a simple boiler.

5. What kind of absorption units are commercially available for solar application?

5.1 Arkla Absorption Chillers

Arkla Industries, Incorporated is the only manufacturer in the U.S.A. that provides H_2O-LiBr absorption units designed for solar applications. These are Solaire 36, Model WF36 and Solaire 300, Model WFB300 absorption chillers, which are the second generation solar units.

5.1.1 Solaire 36 Chiller (Model WF36)

It is nominally rated at 3 tons. With generator input water temperature between 170°F and 205°F, and with condenser inlet water temperature of 85°F, the unit can produce from 0.5 ton to 3.5 tons of cooling. The unit employs a small hermetically sealed stainless steel centrifugal pump to transfer solution from absorber to generator.

Specification and dimensional views of Solaire 36 is contained in Table 5.1-1 and Figure 5.1-1 respectively. Figures 5.1-2 and 5.1-3 represent the capacity and COP of the Arkla Solaire 36 as a function of generator inlet water temperature respectively. It is noted that the heat sink temperature, Tc, plays an important role on both the capacity and the COP of the unit. It is also seen in Figure 5.1-3 that the unit was optimized at the generator temperature of about 195°F based on the COP of the unit at the inlet condenser water temperature of 85°F.

The trade price (or builder's cost) of Solaire 36 is about $2880.00 (effective 5/1/77). The cost of the cooling tower is additional and is between $400.00 and $500.00.

Table 5.1-1
Specification of Arkla's Solaire 36[6]

DESIGN DELIVERED CAPACITY, Btu/h	36,000[1]
DESIGN DELIVERED CAPACITY, Tons I.M.E.	3.0[1]
ENERGY REQUIREMENTS	
Design Hot Water Input, Btu/h	50,000
Design Hot Water Inlet Temperature, °F	195
Design Hot Water Outlet Temperature, °F	185.9
Permissible Range of Inlet Temp.	170 to 205
Design Hot Water Flow, gpm	11.0
Pressure Drop, Feet of Water, at 11 gpm	9.8
Permissible Range of Flow, gpm	5 to 22
Pressure Drop, Feet of Water, at 22 gpm	29.9
Maximum Working Pressure, psig	100
Unit Water Volume, Gallons, Approx.	3.0
Electrical Voltage, 60 Hz, 1 Phase	115[2]
Maximum Wattage Draw	250
CHILLED WATER DATA	
Design Inlet Temperature, °F	55
Design Outlet Temperature, °F	45
Design Flow, gpm	7.2
Pressure Drop, Feet of Water, at 7.2 gpm	4.6
Premissible Range of Flow, gpm	4 to 13
Pressure Drop, Feet of Water, at 13 gpm	12.5
Maximum Working Pressure, psig	100
Unit Water Volume, Gallons, Approx.	1.5
CONDENSING WATER DATA	
Design Heat Rejection, Btu/h	86,000
Design Inlet Temperature, °F	85
Design Outlet Temperature, °F	99.3
Permissible Range of Inlet Temp.	75 to 90
Design Flow, gpm	12.0
Pressure Drop, Feet of Water, at 12 gpm	9.6
Permissible Range of Flow, gpm	9 to 25
Pressure Drop, Feet of Water, at 25 gpm	33.9
Maximum Working Pressure, psig	100
Unit Water Volume, Gallons, Approx.	3.0
FOR COOLING TOWER SELECTION	
Maximum Heat Rejection, Btu/h	106,000
Range, °F	14 to 17
Minimum Permissible Sump Temperature, °F	75[3]
SERVICE CONNECTIONS	
Hot Water Inlet and Outlet	1" FPT
Chilled Water Inlet and Outlet	1" FPT
Condensing Water Inlet and Outlet	1" FPT
PHYSICAL DATA, APPROXIMATES	
Operating Weight, Pounds	675[4]
Shipping Weight, Pounds	680[5]
Crated Size, Inches	36W, 34D, 75H

NOTES:
1. Capacity at design conditions. For capacities at other conditions, see Page 4.
2. Units equipped for operation on 230V-50Hz-1Ph available on special order.
3. Thermostatic switch to control tower fan MUST be used. Set to "cut out" at 75°F.
4. Includes circulating water weights.
5. Units as shipped contain Lithium Bromide charge.

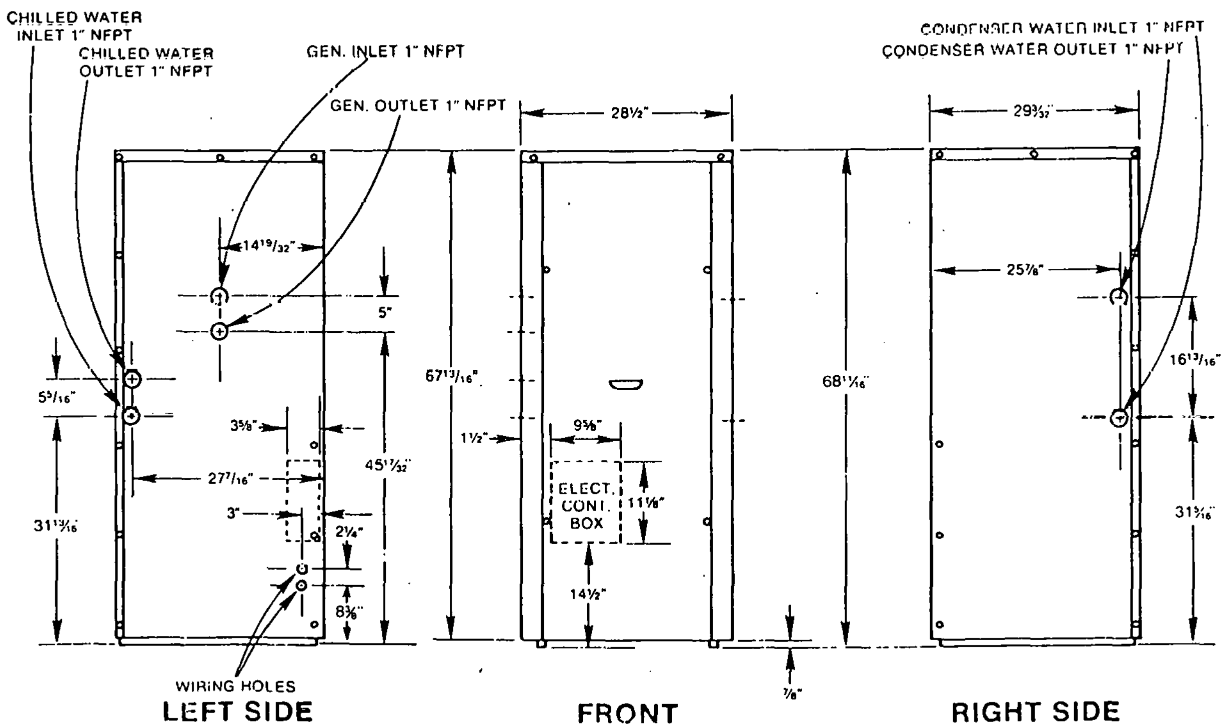


Fig. (5.1-1) Dimensional Views of Arkla's Solaire 36[6]

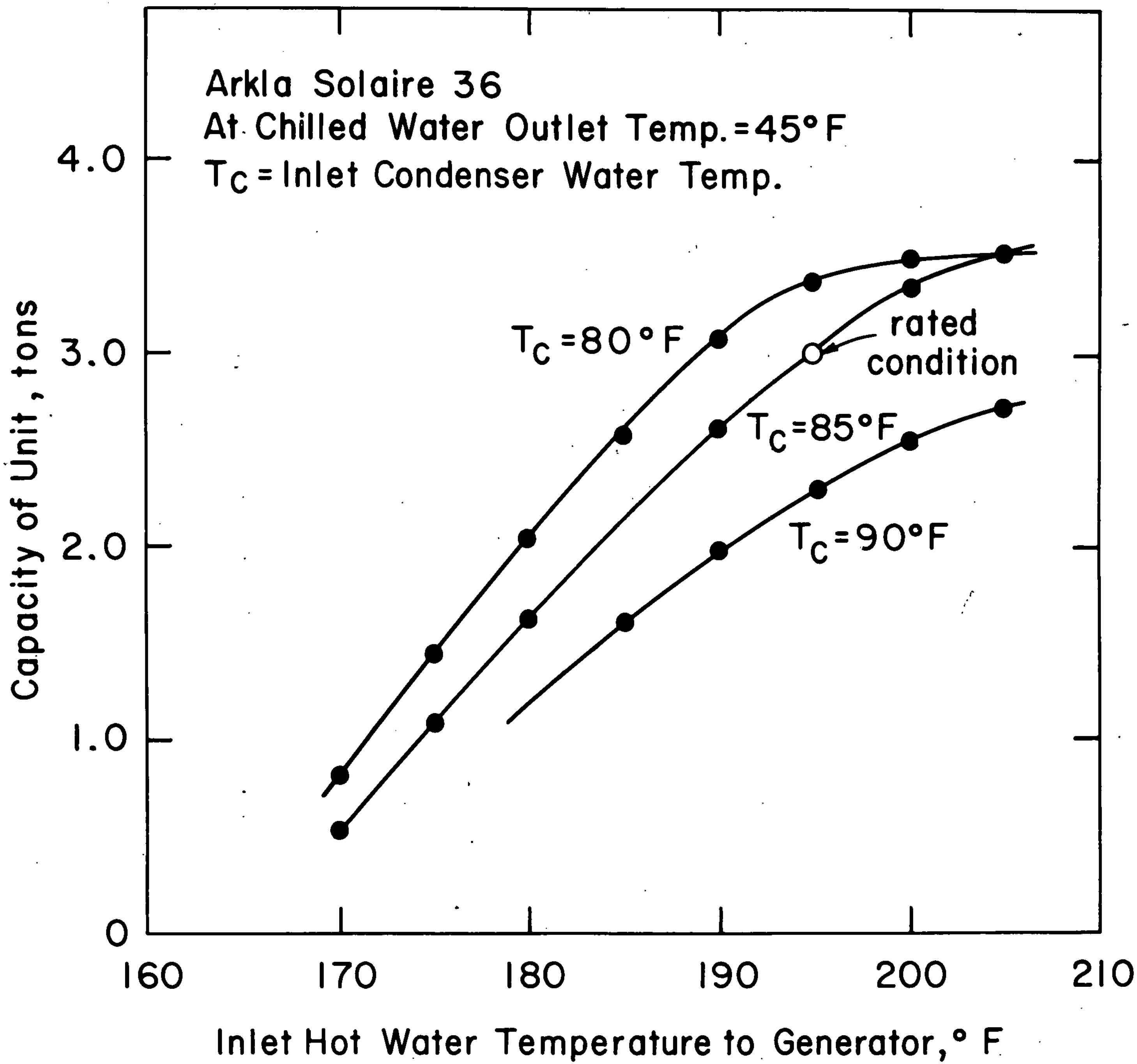


Fig. (5.1-2) Capacity of Arkla's Solaire 36 as a Function of Generator Inlet Water Temperature[6]

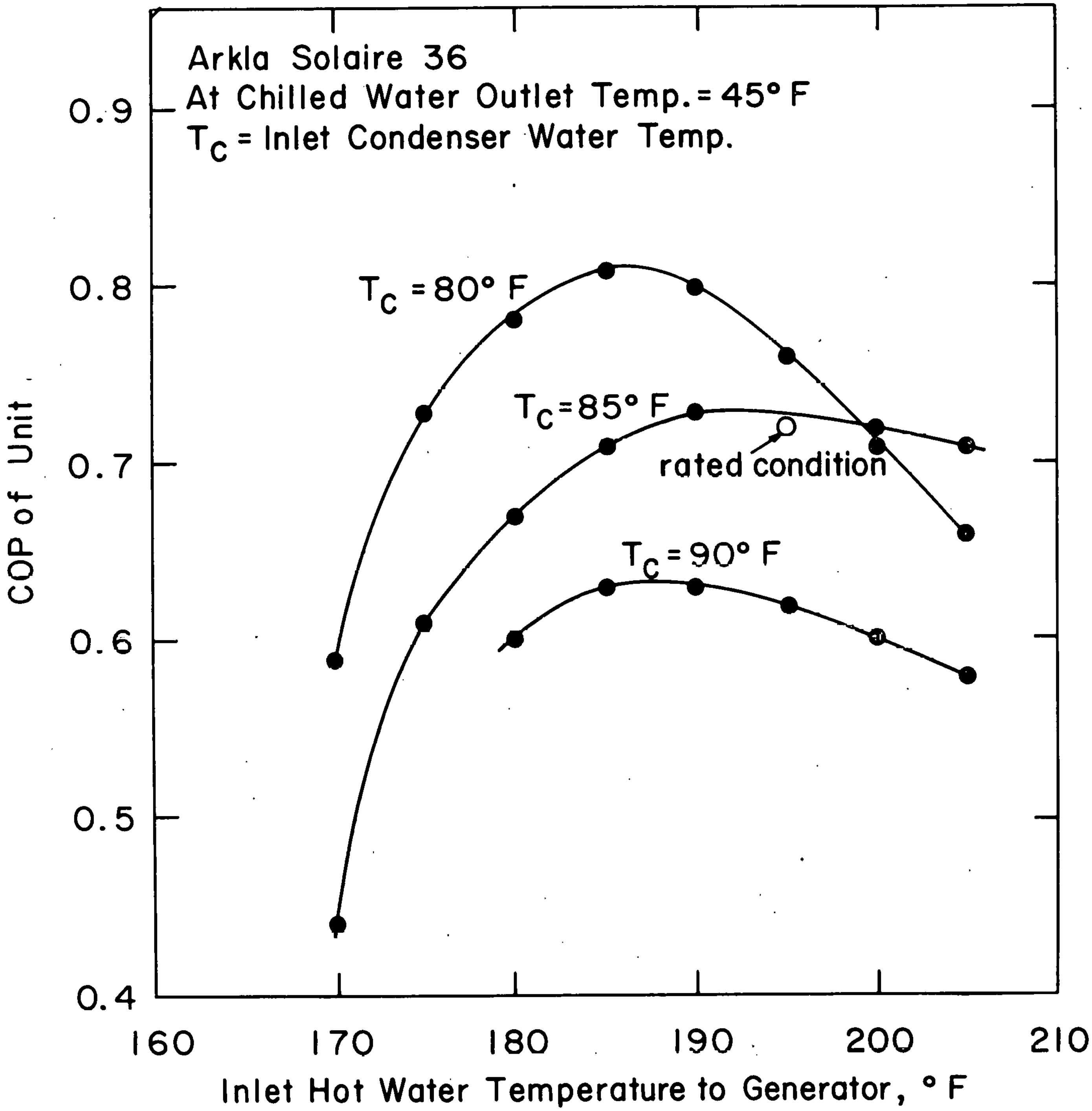


Fig. (5.1-3) COP of Arkla Solaire 36 as a Function of Generator Inlet Water Temperature[6]

5.1.2 Solaire 300 Chiller (Model WFB300)

The unit is nominally rated at 25 tons and produces 25.5 tons of cooling at the design generator inlet water temperature of 195°F and at the design condensing water temperature of 85°F. The unit uses magnetically driven centrifugal pump which is hermetically sealed.

The cooling capacity and the COP of the unit varies widely with firing water temperature between 160°F and 200°F as shown in Figures 5.1-4 and 5.1-5. Although the unit was designed at the generator inlet water temperature of 195°F, Figure 5.1-5 shows that the maximum COP of the unit occurs at the generator inlet water temperature of 180°F. Again, the advantages of using lower heat sink temperature (Tc) are shown in both Figures 5.1-4 and 5.1-5. However, the effect of heat sink temperature, Tc, on the COP of Solaire 300 is much less than that of Solaire 36 as can be compared in Figures 5.1-3 and 5.1-5.

Also shown in Figures 5.1-6 and 5.1-7 are the COP and the capacity of Solaire 300 versus the generator inlet water temperature with leaving chilled water temperature as parameter. Figure 5.1-6 shows that the cooling capacity of the unit improves as the leaving chilled water temperature increases. In Figure 5.1-7 significant difference in unit cooling COP is indicated at different chilled water temperatures. For example, the cooling COP of the unit is 0.69 at Tcw = 45°F and 0.57 at Tcw = 40°F when the generator inlet water temperature is 195°F in both cases. This corresponds to about 17% degradation in unit cooling COP due to the decrease in chilled water temperature by 5°F.

The possible use of the Solaire 300 as a nominal 15 ton unit was also indicated with about the same COP. For this operation the flow rate of condensing water, generator inlet water or chilled water is reduced by 40 %. Table 5.1-2 and Figure 5.1-8 are the specification and dimensional views of Solaire 300 respectively.

The trade price (or builder's cost) of Solaire 300 is about $15,200.00 (effective 5/1/77) and cooling tower costs about $3,100.00 or $125.00 per ton of cooling.

5.2 Other Absorption Chillers

Availability of absorption chillers of commercial and industrial sizes is limited to solar cooling experiments or demonstrations. Trane's single-effect H_2O-LiBr units could be used with minor modification of generator component and with adjustment of solution concentration. York's H_2O-LiBr units are also adaptable to solar applications with no hardware modification but with derated capacity at lower generator input temperature. These units were originally designed to operate at higher generator input temperature ($\approx 250^{\circ}F$) than that of the flat-plate solar collector. Therefore, the present chiller units should be optimized to be used with the flat-plate collectors, or the existing chillers could be used with the moderately concentrating solar collectors. Detailed discussion on Trane and York units is included in Sections 6.2 and 6.3.

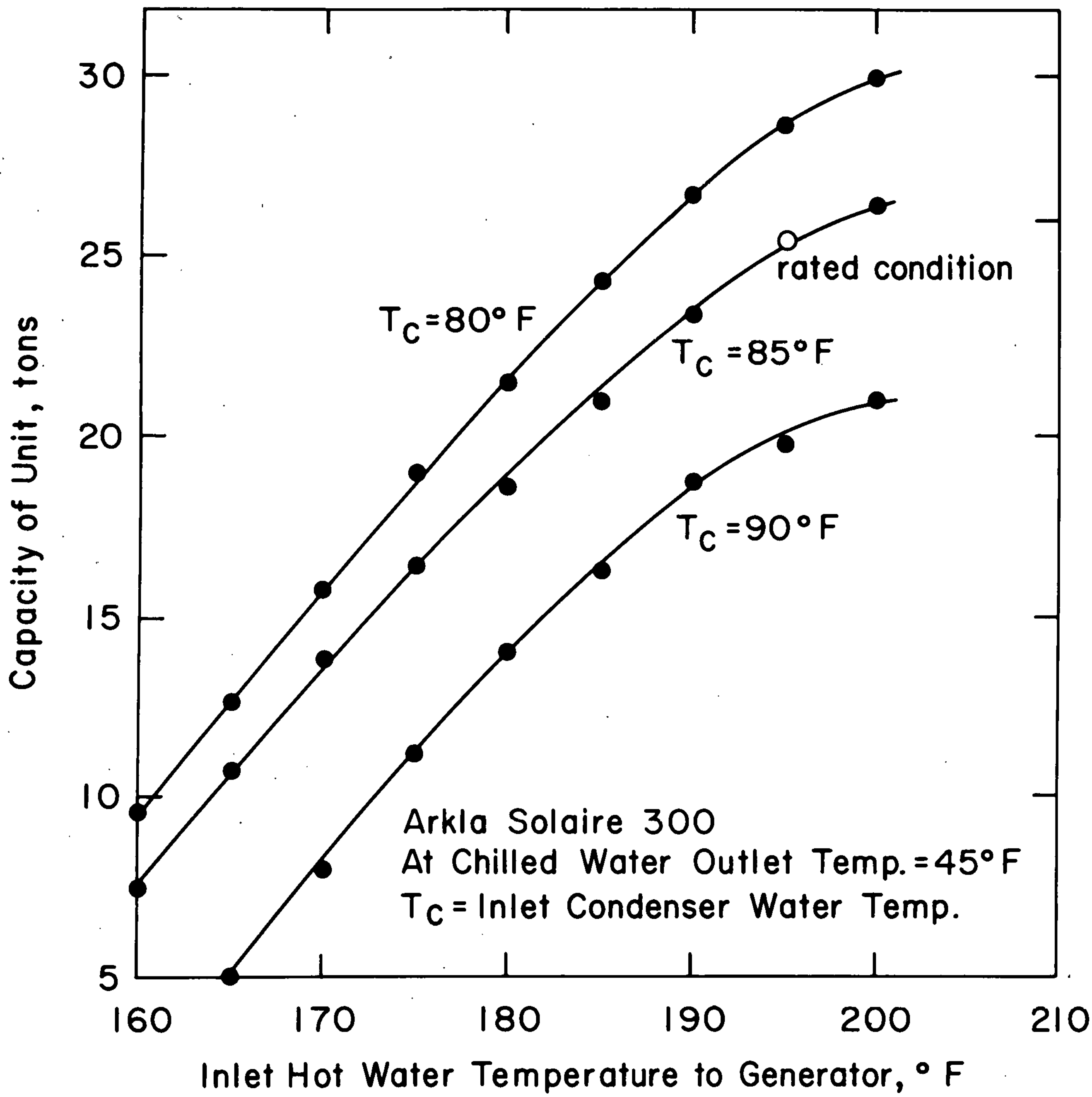


Fig. (5.1-4) Capacity of Arkla Solaire 300 as a Function of Generator Inlet Water Temperature[7]

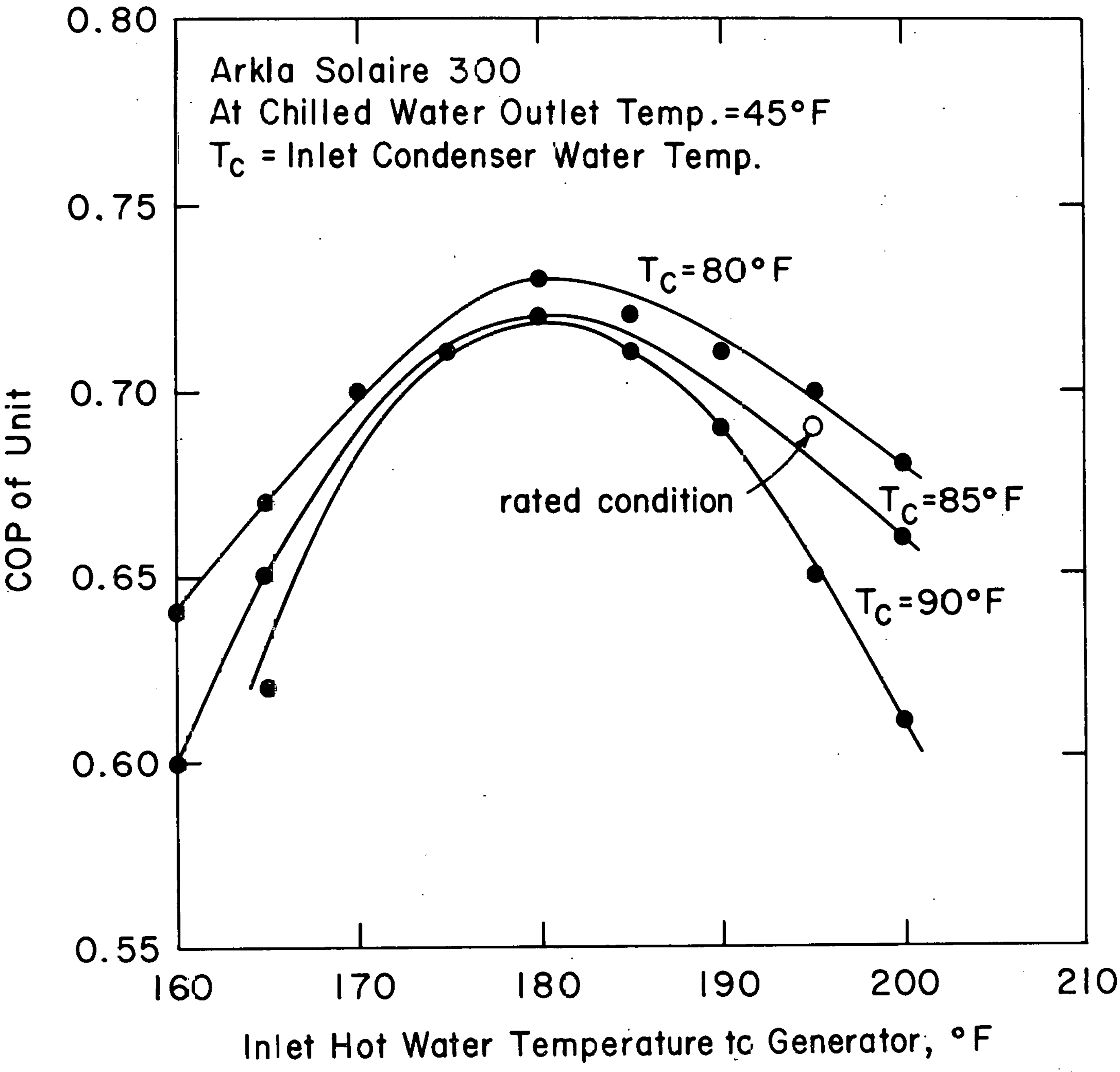
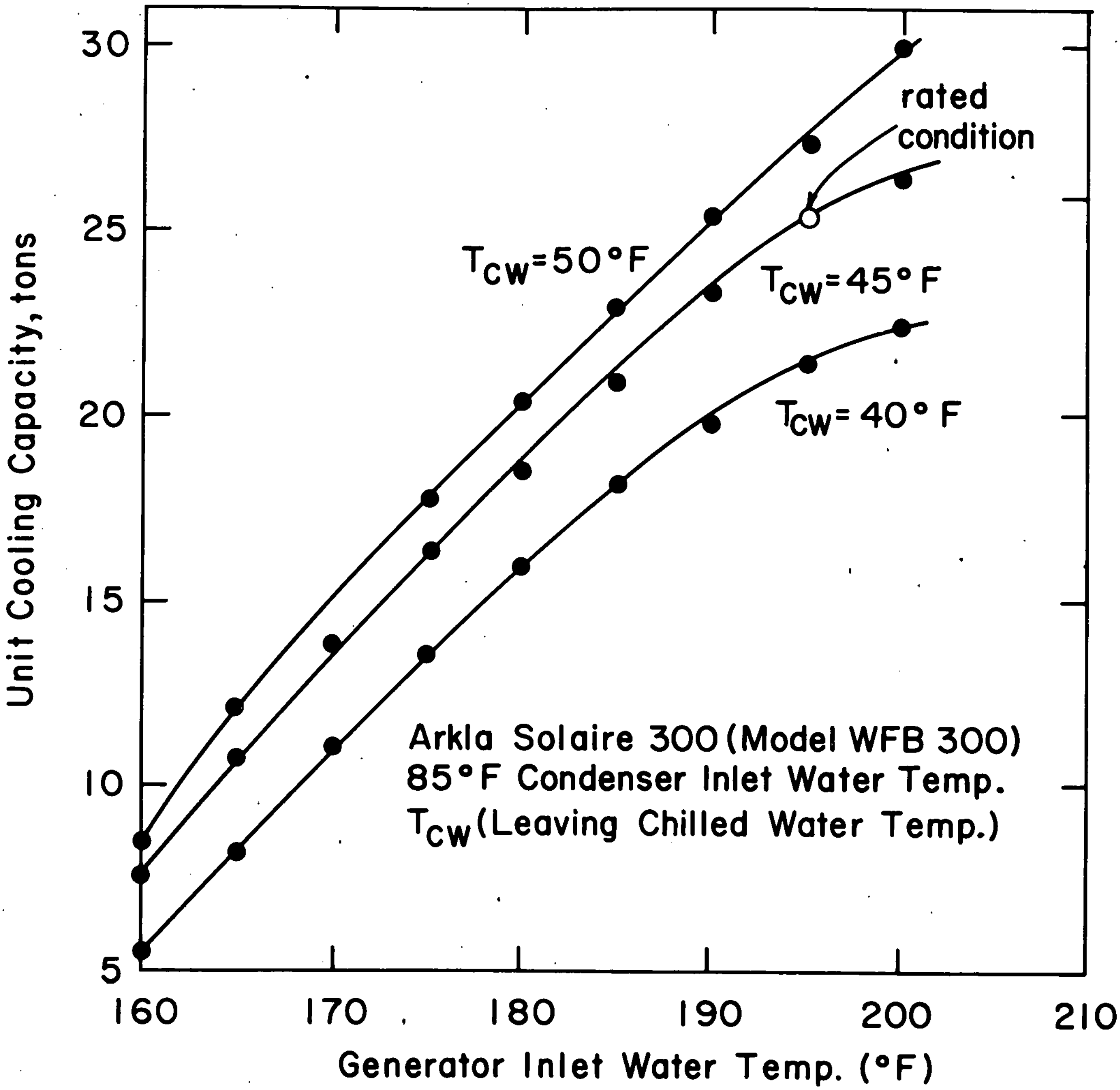


Fig. (5.1-5) COP of Arkla's Solaire 300 as a Function of Generator Inlet Water Temperature[7]



Fog. (5.1-6) Unit Cooling Capacity of Arkla's Solaire 300 versus Generator Inlet Water Temperature with Leaving Chilled Water Temperature as Parameter[7]

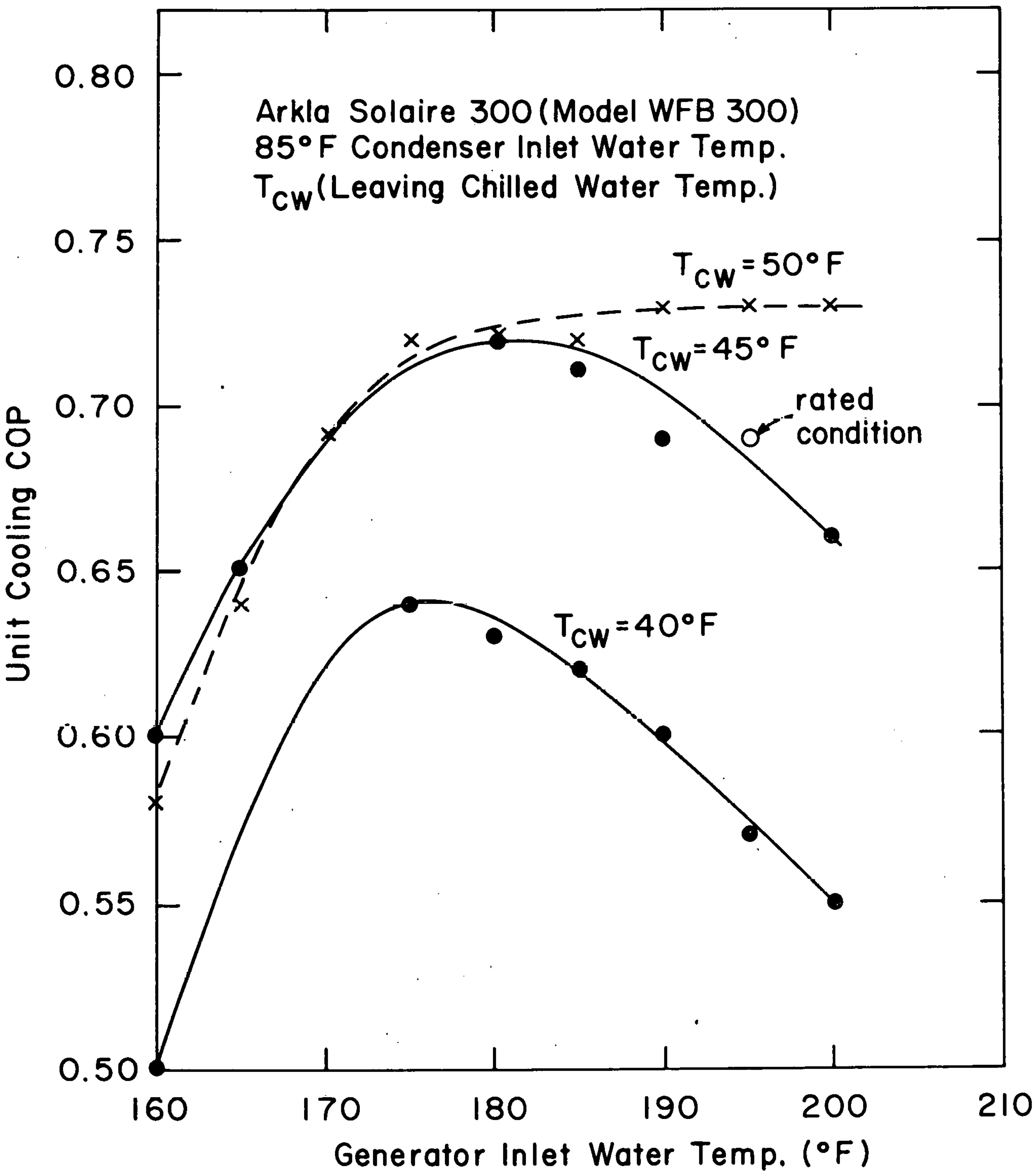


Fig. (5.1-7) Unit Cooling COP of Arkla's Solaire 300 versus Generator Inlet Water Temperature with Leaving Chilled Water Temperature as Parameter[7]

Table 5.1-2

Specification of Arkla's Solaire 300[7]

DESIGN DELIVERED CAPACITY, Btu/h	306,000[1]
DESIGN DELIVERED CAPACITY, Tons I.M.E.	25.5[1]

ENERGY REQUIREMENTS

Design Hot Water Input, Btu/h	447,000
Design Hot Water Inlet Temperature, °F	195
Design Hot Water Outlet Temperature, °F	184.8
Permissible Range of Inlet Temp.	160 to 200
Design Hot Water Flow, gpm	90
Pressure Drop, Feet of Water, at 90 gpm	20.7
Permissible Range of Flow, gpm	50 to 100
Pressure Drop, Feet of Water, at 100 gpm	25.6
Maximum Working Pressure, psig	100
Electrical Voltage, 60 Hz, 1 Phase	115[2]
Maximum Wattage Draw	150

CHILLED WATER DATA

Design Inlet Temperature, °F	55
Design Outlet Temperature, °F	45
Design Flow, gpm	60
Pressure Drop, Feet of Water, at 60 gpm	9.8
Permissible Range of Flow, gpm	30 to 100
Pressure Drop, Feet of Water, at 100 gpm	26.9
Maximum Working Pressure, psig	100
Unit Water Volume, Gallons, Approx.	12
Fouling Factor	.0005

CONDENSING WATER DATA

Design Heat Rejection, Btu/h	753,000
Design Inlet Temperature, °F	85
Design Outlet Temperature, °F	101.7
Permissible Range of Inlet Temp.	75 to 90
Design Flow, gpm	90
Pressure Drop, Feet of Water, at 90 gpm	22.9
Permissible Range of Flow, gpm	50 to 110
Pressure Drop, Feet of Water, at 110 gpm	33.5
Maximum Working Pressure, psig	100
Unit Water Volume, Gallons, Approx.	20
Fouling Factor	.001

FOR COOLING TOWER SELECTION

Maximum Heat Rejection, Btu/h	853,000
Range, °F	16 to 17
Minimum Permissible Sump Temperature, °F	75[3]

SERVICE CONNECTIONS

Hot Water Inlet and Outlet	2" FPT
Chilled Water Inlet and Outlet	2½" FPT
Condensing Water Inlet and Outlet	2½" FPT

PHYSICAL DATA, APPROXIMATES

Operating Weight, Pounds	3,420[4]
Shipping Weight, Pounds	3,145[5]
Crated Size, Inches	114 W, 45 D, 69 H

NOTES:
1. Capacity at design conditions. For capacities at other conditions, see Page 4.
2. Units equipped for operation on 230V-50Hz-1Ph available on special order.
3. Thermostatic switch to control tower fan MUST be used. Set to "cut out" at 75°F.
4. Includes circulating water weights.
5. Units as shipped contain Lithium Bromide charge.

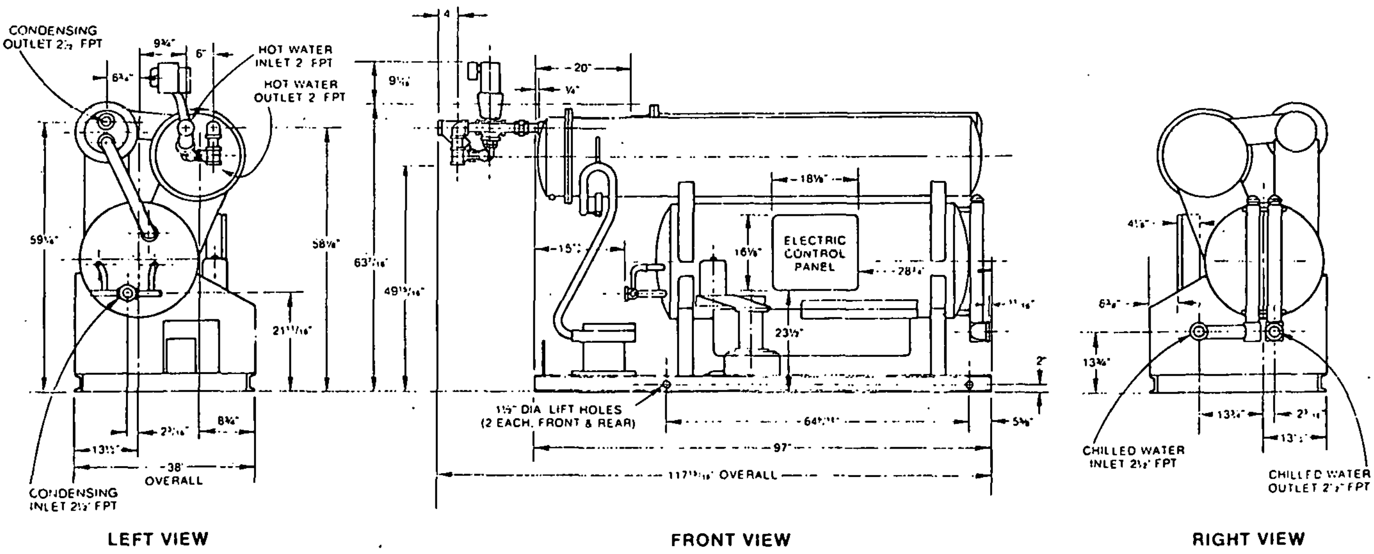


Fig. (5.1-8) Dimnesional Views of Arkla's Solaire 300[7]

6. What are the general trends of the absorption cooling industry toward the solar application at the present time?

6.1 Arkla Industries, Incorporated

Arkla is most actively involved in developing absorption chillers for the solar applications. Their present production line includes gas-fired NH_3-H_2O (aqua ammonia) units, gas-fired H_2O-LiBr units, steam-fired H_2O-LiBr units and hot water-fired H_2O-LiBr units.

6.1.1 Servel Gas-fired Water Chillers

The air-cooled NH_3-H_2O units are available in three sizes, 3 ton (Model ACB 36-00), 4 ton (Model ACB 48-00), and 5 ton (Model ACB 60-00). The rated COP of the above units are: 0.46 for 3 ton and 0.48 for 4 and 5 ton at the condenser entering air temperature of 95°F and at the chilled water leaving temperature of 45°F.

The water-cooled H_2O-LiBr unit combined with hydronic heater, is available in 25 ton capacity. The rated COP of the above unit (Model DF 300-600) is 0.5 at the condenser entering water temperature of 85°F and at the chilled water leaving temperature of 45°F.

6.1.2 Steam-fired Water Chiller

The water-cooled H_2O-LiBr unit of 25 ton capacity is available (Model SF).

6.1.3 Hot Water-fired Absorption Units for Solar Applications

A. First Generation Solar Units

a) Water-cooled H_2O-LiBr Unit (Solaire Model 501-WF)

The residential 3 ton, direct-expansion absorption air-conditioners were produced in limited quantity and were intended mainly for research and demonstration projects. Rated capacity of 3 ton could be obtained at the generator inlet water temperature of 210°F and at the condensing water inlet temperature of 85°F. Rated COP of the unit at the above conditions is 0.65.

b) Water-cooled H_2O-LiBr Unit (Solaire Model WF-300)

The 25 ton water chiller was commercially produced for comfort air conditioning and industrial process applications. The circulation of solution is accomplished by the thermal syphon principle. The capacity of the unit varies from 14 tons to 25 tons at the generator inlet water temperature between 190 ~ 245°F with condensing water inlet temperature of 85°F and with chilled water outlet temperature of 45°F. The rated capacity of 25 ton is obtained at the design generator inlet water temperature of 225°F. The COP of the unit is 0.69 at the rated conditions (i.e.: 225, 85 and 45°F). The number of installations of the first generation solar units (Model 501-WF and WF-300) exceeds 48 in the U.S.A. and 8 in foreign countries.

B. Second Generation Solar Units

a) Water-cooled H_2O-LiBr Unit (Solaire Model WF 36)

The residential 3 ton chiller is the newest model available in production quantities. The unit is designed for lower generator input temperature than that of the previous Model 501-WF. The design hot water temperature is 195°F and the unit requires a chilled water pump and a cooling tower. Detail performance data was previously presented in Section 5.1.1.

b) Water-cooled H_2O-LiBr Unit (Solaire Model WFB 300)

The new 25 ton chiller is also available in production quantities. The design input water temperature to generator was lowered from 225°F of previous Model WF-300 to 195°F. The unit requires chilled water pumpt and 85°F cooling water source. The detailed performance of the unit was previously presented in Section 5.1.2

c) Third Generation Solar Units

Arkla is presently developing residential size (3 ton) H_2O-LiBr absorption chillers which contain evaporative cooling package as an integral part of the unit. Separate cooling tower is no longer required since the absorber/condenser components are evaporatively cooled inside the unit. Other advantages of the third generation unit over the second generation unit are:

- Significant reduction in auxiliary power requirement (about 35% less)
- Less scaling on the heat transfer surface area of absorber/condenser components due to the increase in surface area
- Significant reduction in both the first cost and the installation cost (competitive to the present gas-fired units)

The design generator inlet water temperature will remain at 195°F, and the unit will employ mechanical solution pump. Prototype units would be available this Summer (1977) and production units, in the Spring of 1979.

d) Fourth Generation Solar Units

The development of air-cooled absorption chiller is under consideration. The hot water-fired unit would employ new working fluid, lithium bromide-lithium thiocyanate-water, developed by Institute of Gas Technology (IGT). IGT claims that the air cooling of absorber-condenser component is possible with the new fluid since it would allow operate absorber-condenser components at higher temperature without crystallizing LiBr.

However, the new system may require higher generator temperature than that which could be obtained from the flat plate collectors. Other potential problem could be the decomposition of the new working fluid due to the catalytic action of the metal impurities existing in the system (i.e., materials problem). Intensive R&D work is needed to solve the above problem.

6.2 The Trane Company

6.2.1 Single Effect Absorption Chiller (Model ABSC)

Trane manufactures water-cooled, H_2O-LiBr unit which uses either steam or hot water as an energy source. These units are not originally intended for solar applications, thus are not suitable for use with the flat-plate solar collectors. The nominal capacities of their units vary from 101 to 1660 tons of cooling.

6.2.2 Double-effect Absorption Chillers (Model ABTD)

Trane's double-effect, water-cooled, H_2O-LiBr chillers uses steam at pressures up to 150 psig (366°F) or hot water up to 400°F as an energy source. Nominal capacities of the units range from 385 to 1060 tons of cooling. The COP of the double-effect chiller is greatly improved over that of the single effect due to the decrease in energy input requirement by 30 ~ 40%. If the typical COP of 0.67 is assumed for a single effect unit, the COP of a double effect units could be in the range of 0.96 ~ 1.12.

6.2.3 Use of Single-effect Units for Solar Applications

In order to initiate the low temperature input applications of the single-effect absorption chillers, Trane made laboratory testing of their present units with no modification. The results of the tests are reproduced here in Figures 6.2-1, 6.2-2 and 6.2-3.

In Figure 6.2-1, the percent of nominal cooling capacity is plotted versus the generator inlet water temperature with the condenser inlet water temperature as a parameter. As previously seen in Figures 2.2-1, 5.1-2 and 5.1-4, the percent of nominal cooling capacity increases as the condenser inlet water temperature is decreased. Comparing Figures 2.2-1 and 6.2-1 (Tc = 85°F), the performance of the Trane unit (Model ABSC-02A) is less than the average performance of the non-residential absorption units.

Figure 6.2-2 shows the effect of varying the leaving chilled water temperature on the nominal cooling capacity of the unit. As the leaving chilled water temperature increases, the percent nominal cooling capacity also increases. Note that the effect of increasing the leaving chilled water temperature increases as the generator inlet water temperature increases for this particular unit. Compare with Figure 5.1-6

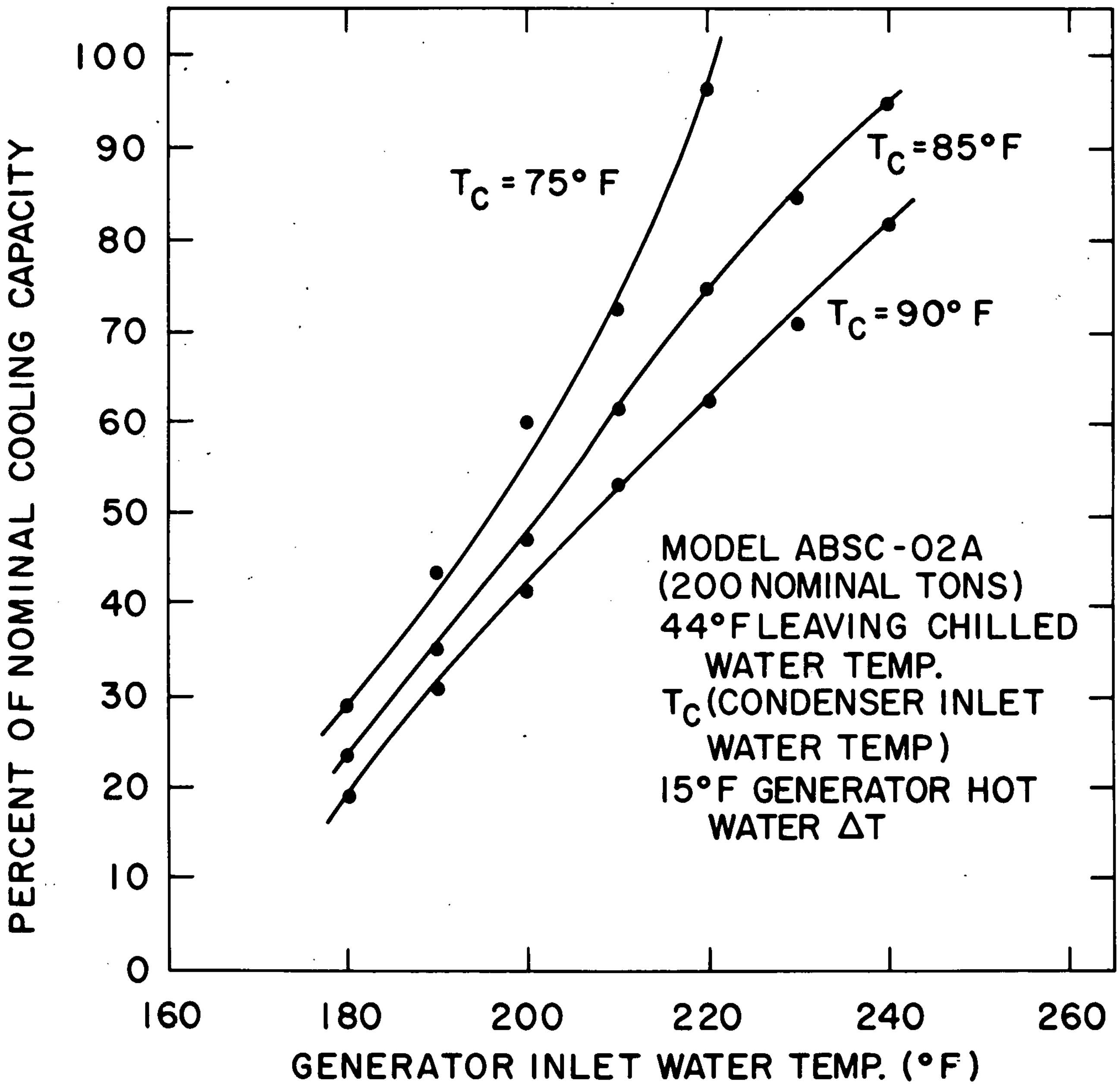


Fig. (6.2-1) Percent Nominal Cooling Capacity of Nominal 200 Ton Trane Absorption Unit versus Generator Inlet Water Temperature with Condenser Inlet Water Temperature as Parameter[8]

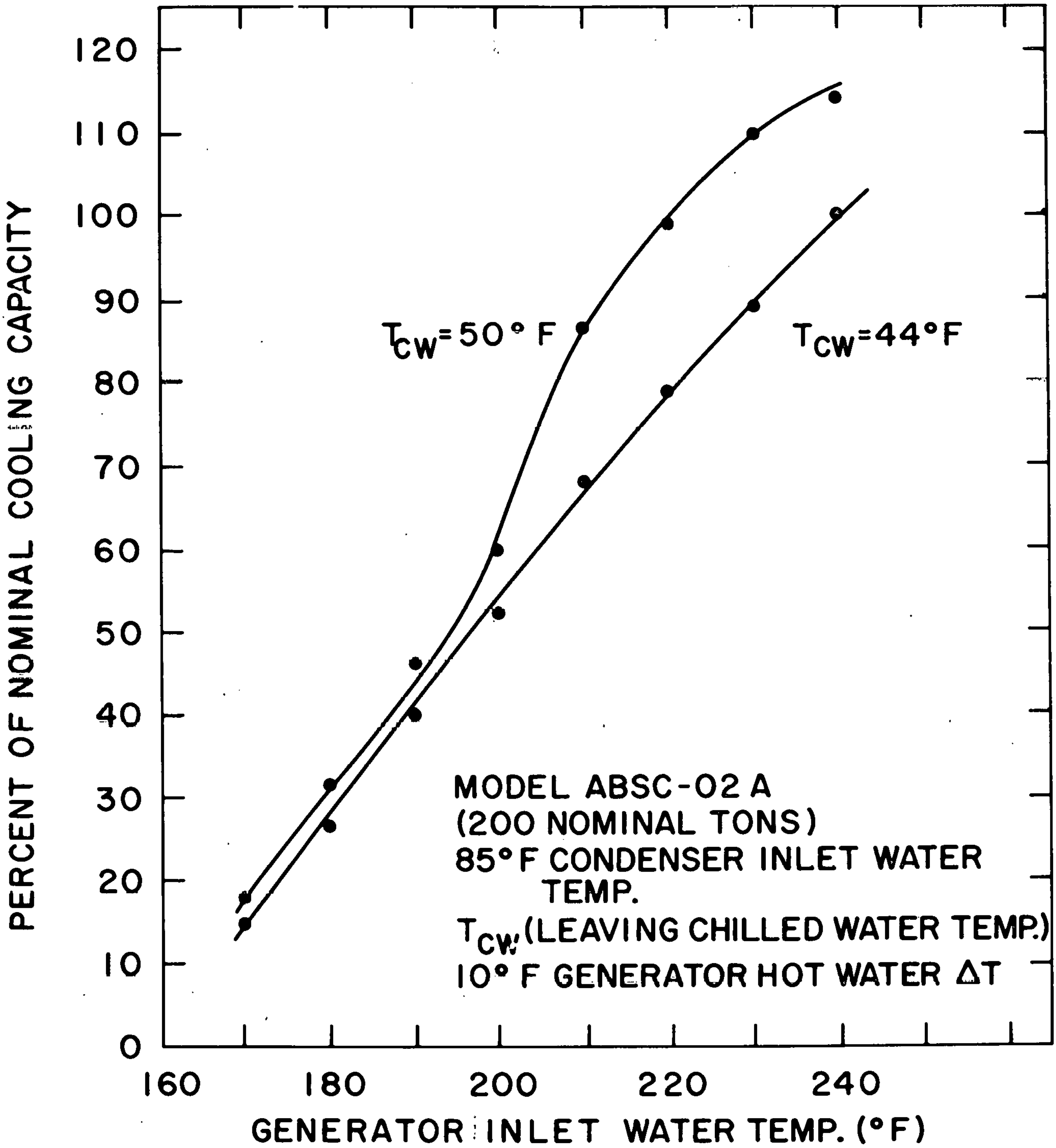


Fig. (6.2-2) Percent Nominal Cooling Capacity of Nominal 200 Ton Trane Absorption Unit versus Generator Inlet Water Temperature with Leaving Chilled Water Temperature as Parameter[8]

As the mean hot water temperature in the generator is increased, the available cooling capacity of the unit also increases due to the following reasons. The amount of heat transferred to the generator (Qg) can be expressed as:

$$Qg = UoAo\ \Delta Tm \qquad \text{Eq. 6.2-1}$$

where Uo = over-all heat transfer coefficient

Ao = over-all heat transfer area

ΔTm = mean temperature difference between hot water and generator solution

In equation 6.2-1, ΔTm increases as the mean hot water temperature in the generator is increased or the generator hot water temperature difference (ΔTgw) is decreased. In order to decrease ΔTgw, the hot water flow rate to the generator must be increased. In doing so, the value of Uo may also be increased. Then the cooling capacity of the unit increases since the value of Qg of Eq. 6.2-1 is increased.

Figure 6.2-3 illustrates the effect of varying the generator hot water temperature difference on the nominal cooling capacity of the unit. For instance, when the generator inlet water temperature of 200^{o}F is assumed, the percent of nominal cooling capacity is 42% at ΔTgw of 15^{o}F or 50% at ΔTgw of 5^{o}F. Thus the percent of nominal cooling capacity of the unit is improved by 19% with the decrease in ΔTgw.

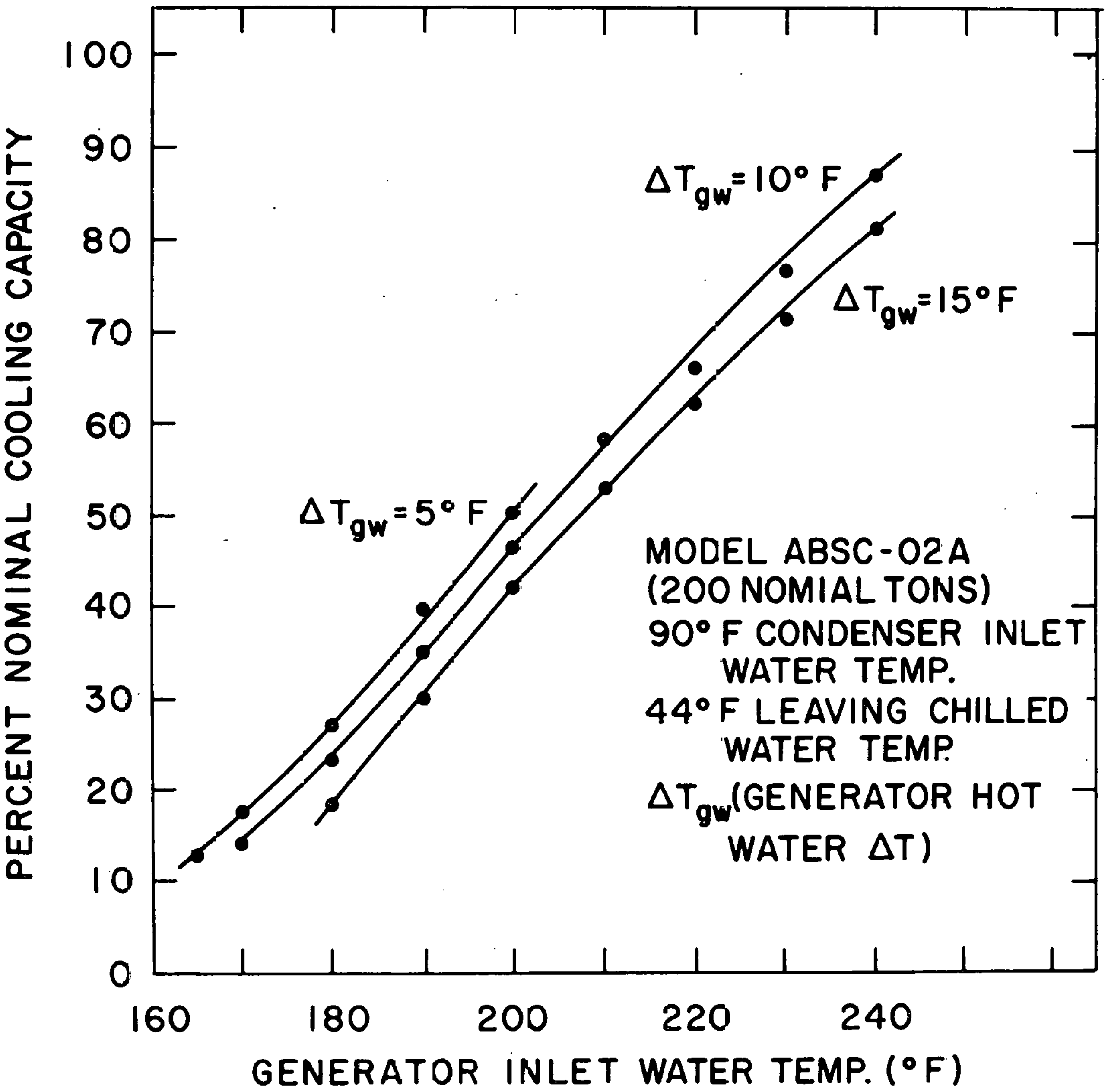


Fig. (6.2-3) Percent Nominal Cooling Capacity of Nominal 200 Ton Trane Absorption Unit versus Generator Inlet Water Temperature with Generator Hot Water Temperature Difference as Parameter[8]

As mentioned earlier, the need of higher hot water flow rate requires more pump work or a higher gpm pump.

There is, however, another way of improving unit cooling capacity without employing a new higher gpm pump. It involves minor modification of generator or mere increase in generator heat transfer area.

For solar application, Trane said that their single-effect H_2O-LiBr units can be used with minor modification of generator component with adjustment of solution concentration. Then the performance of the modified unit should be better than that of Figures 6.2-1, 6.2-2 or 6.2-3 and probably approach that of Figure 2.2-1.

Six Trane absorption units have been sold for a number of solar projects including one at NASA Systems Engineering Building in Langley Air Force Base. Nominal cooling capacities of these units vary from 129 to 294 original tons.

6.3 York Division of Borg-Warner Corporation

York had some experience on large, custom built, water-cooled, NH_3-H_2O units during the period of 1930 to 1960. At the present time, York manufactures complete line of Model ES absorption liquid chillers, which are, of course, water-cooled, H_2O-LiBr units. The absorption chillers come in 21 sizes from 110 to 1377 nominal tons of cooling, and all 21 sizes were said to be adaptable to solar applications with no modification but with much derated capacity at lower generator input temperature. The chillers are applicable to solar installations requiring 40 tons or more of full load capacity. Minimum inlet hot water temperature to the generator is about 170^{o}F.

Although the actual performance data of the York unit at low generator inlet water temperatures is not available, the capacity of the unit reduces about 1/3 of the nominal rating when the generator heat source temperature falls below 190°F. (Refer to Figure 2.2-1 in Section 2.2 for the comparison). The COP of the unit, at the derated conditions, however, remains about the same as that of the same unit at nominal ratings ($\approx$ 0.70).

York claims that their generator has 20% more heat transfer surface area than that of others and the generator inlet water temperature can be lower by 10°F than that of others for the same capacity of the unit.

Two of York units have been installed to school buildings. One of them is the installation of nominal 150 ton, water-cooled, H_2O-LiBr chiller at Timonium Elementary School in Maryland. With no unit modification, its derated capacity is about 50 ton at the generator inlet water temperature of 180 ~ 187°F, with the COP of about 0.65. The original 150 ton unit was rated at generator inlet steam temperature of 237°F, and the rated COP was 0.69. The performance of the installed unit was said to be poor (0.4 ~ 0.68) due to the low cooling load condition (6 ~ 9 tons) of the building.

Presently York does not have any plans to develope optimum chillers of large sizes for low temperature solar applications using York's own fund. York, however, may be willing to proceed when funds are available from outside source.

6.4 Carrier Corporation (including Bryant Division)

Carrier used to be a manufacturer of large gas-fired NH_3-H_2O absorption air conditioners for commercial applications until 1972. Presently they offer 15 standard models of water-cooled, H_2O-LiBr absorption chillers.

These units (Carrier 16JB series) use either low pressure steam or hot water to produce cooling capacity in the 70 to 815 ton range.

Although Carrier is interested in the solar area, they do not have any plans to modify their present absorption units for solar application. They think that the solar application of present absorption units is premature and has no immediate future due to the poor performance and high cost of the present absorption units in solar applications.

More recently, it is heard that Carrier is being funded by Bonniville power to develop a 15-ton H_2O-LiBr absorption chiller at the design generator temperature of about 180°F.

7. How does the absorption cooling unit perform when it is installed in an actual solar cooling system?

The performance of the absorption cooling unit tested under steady state conditions is quite different from the actual performance of the unit when employed as an integral part of the solar cooling system. It is, therefore, very important to study the unit performance operating in an actual solar cooling system. It is also noted that the cooling performance of chiller depends strongly upon the kind of adapted cooling system as well as the geographical location.

7.1 Colorado State University (CSU)

To demonstrate the feasibility of utilizing solar energy in space conditioning and domestic hot water heating of residential buildings, CSU built three test houses, two of which employ absorption cooling units as solar cooling subsystems. CSU Solar House I is claimed to be the first heated and cooled residential-size building in the world, and was put into operation in August, 1974.

The results obtained from the tests of the CSU Solar House I by the Colorado State University, will be summarized here since the data is readily available in reasonable detail from References 32~35.

7.1.1 General Information

A) House Specifications

3 bedroom residential house

Total floor area: 3000 square feet

Floor area, main level: 1500 square feet

Floor area, lower level: 1500 square feet

Roof pitch: 45° from horizontal

Design heating load: 17,600 Btu/degree day (55,000 Btuh at $-10^{\circ}F$)

Design cooling load: 26,000 Btu/degree day (36,000 Btuh at $95^{\circ}F$)

Insulation: Ceiling 5 inches fiberglass batt

Walls 3 1/2 inches fiberglass batt

B) Test Location

CSU, Fort Collins, Colorado

Latitude: $41^{\circ}N$

Longitude: $105^{\circ}F$

Altitude: 5200 feet

C) Test Period

June 1, 1974 through August 31, 1974

D) Solar Heating/Cooling System

- Solar Collection - double glazed aluminum flat plate collector (liquid type), 768 ft^2
- Thermal Storage - 1,130 gallon steel tank
- Auxiliary Subsystems - gas-fired hot water heater for domestic hot water, and gas-fired hot water boiler for space conditioning
- Cooling Unit - modified ARKLA-Servel 3-ton, Lithium bromide absorption air conditioner
- Cooling Tower - forced draft type, 96,000 Btuh capacity

7.1.2 Cooling Subsystem

A) Cooling Unit - gas-fired ARKLA-Servel 3.5 ton, LiBr absorption air conditioner was modified as follows:

- The gas-fired generator section was replaced by the hot-water powered generator section
- The LiBr concentration was adjusted from 54 ~ 57% to 51 ~ 54% for effective performance at the climate conditions of the test site. For example; the generator temperature could be lowered by 6^{o}F if the lower cooling water temperature (say 65^{o}F) would be available.
- Absorber bypass valve was installed to divert approximately 50% of the cooling water directly to the condenser when the cooling water temperature falls below 65^{o}F. It will give approximately 8% increase in cooling capacity of the design generator temperature.

B) Cooling Unit Specifications (modified)

Refrigeration capacity: 36,000 Btuh (3 tons)

Chilled water outlet temperature - 45^{o}F

Generator inlet temperature (range) - 175 ~ 202^{o}F

Generator inlet temperature (design) - 188^{o}F

Generator flow rate - 11 gpm

Cooling water inlet temperature - 75^{o}F

Cooling water flow rate - 10 gpm

C) Cooling Unit Performance

The cooling COP of the original gas-fired 3.5 ton unit was 0.53 at the condenser inlet water temperature of 75^{o}F and at the return air temperature of 80^{o}F DB (67^{o}F WB). If the generator efficiency of 75% is assumed for the gas-fired unit, the revised COP for the 100% efficient generator (or equivalent to water fired generator) would be 0.71. The use of the modified unit at lower inlet generator temperature may degrade the above COP slightly.

Figure 7.1-1 shows the measured cooling capacity as a function of the generator inlet temperature. At the rated cooling capacity of 36,000 Btuh, the effect of cooling water temperature on the generator inlet temperature is shown in the table below.

At the rated cooling capacity of 36,000 Btuh	
Cooling Water Temperature (^{o}F)	Generator Inlet Temperature (^{o}F)
63.5	180
65.0	181
75.0 (design)	187

Note that the lower generator inlet temperature would produce the same cooling capacity if the temperature of the cooling water is available at a lower level.

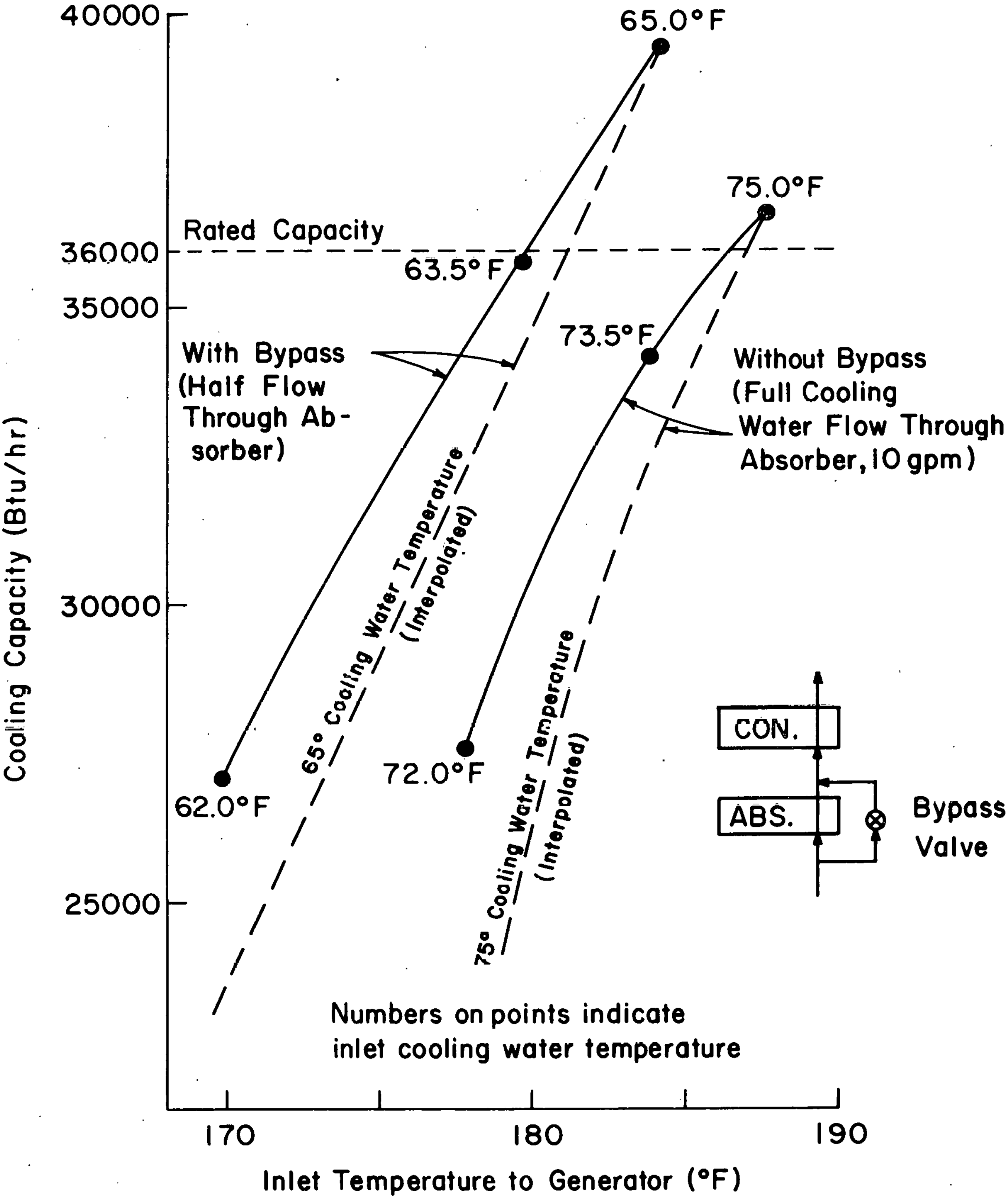


Fig. (7.1-1) Measured Performance of Modified Arkla 3-ton Lithium Bromide Absorption Chiller[35]

7.1.3 Simplified Schematic Diagram of Cooling System

Shown in Figure 7.1-2 is the simplified shcematic diagram of the cooling system. The lower portion of the diagram shows the water cooling of the absorber and condenser components, and the hot water input (solar/auxiliary) to the generator. The cooling unit has an external direct expansion coil as an evaporator with which direct heat exchange takes place between the evaporating refrigerant and the warm room air.

7.1.4 Performance of the Cooling System

A) COP of Cooling Unit when on Auxiliary Heat

The average COP of the cooling unit during the period of August 11 to 31, 1974 was 0.48. When the unit was in the gas-fired mode, it was usually over-fired (54,000 Btuh) since the latent heat load did not exist. The COP of the unit was thus reduced. Had there been a cold storage, the COP of the unit would have not reduced.

B) COP of Cooling Unit when on Solar Heat

During the same period as in A), the average COP of the unit was reported to be 0.70 when the unit was operated by the solar heat. The reasons for the substantially higher value of COP were explained as follows:

The heat input rate to the generator was decreased from 54,000 Btuh to 38,000 Btuh when the solar heat is supplied from the storage. The reduced heat input rate was capable of providing 31,000 Btuh of combined sensible and latent cooling. Since the required heat removal rate (mostly sensible) of Solar House I was about 27,000 Btuh, even the reduced heat input rate was capable of providing more cooling capacity than was actually required. The COP of 0.71 (27,000 Btuh/38,000 Btuh) was calculated at the above conditions, which is in good agreement with the daily average solar COP of 0.70.

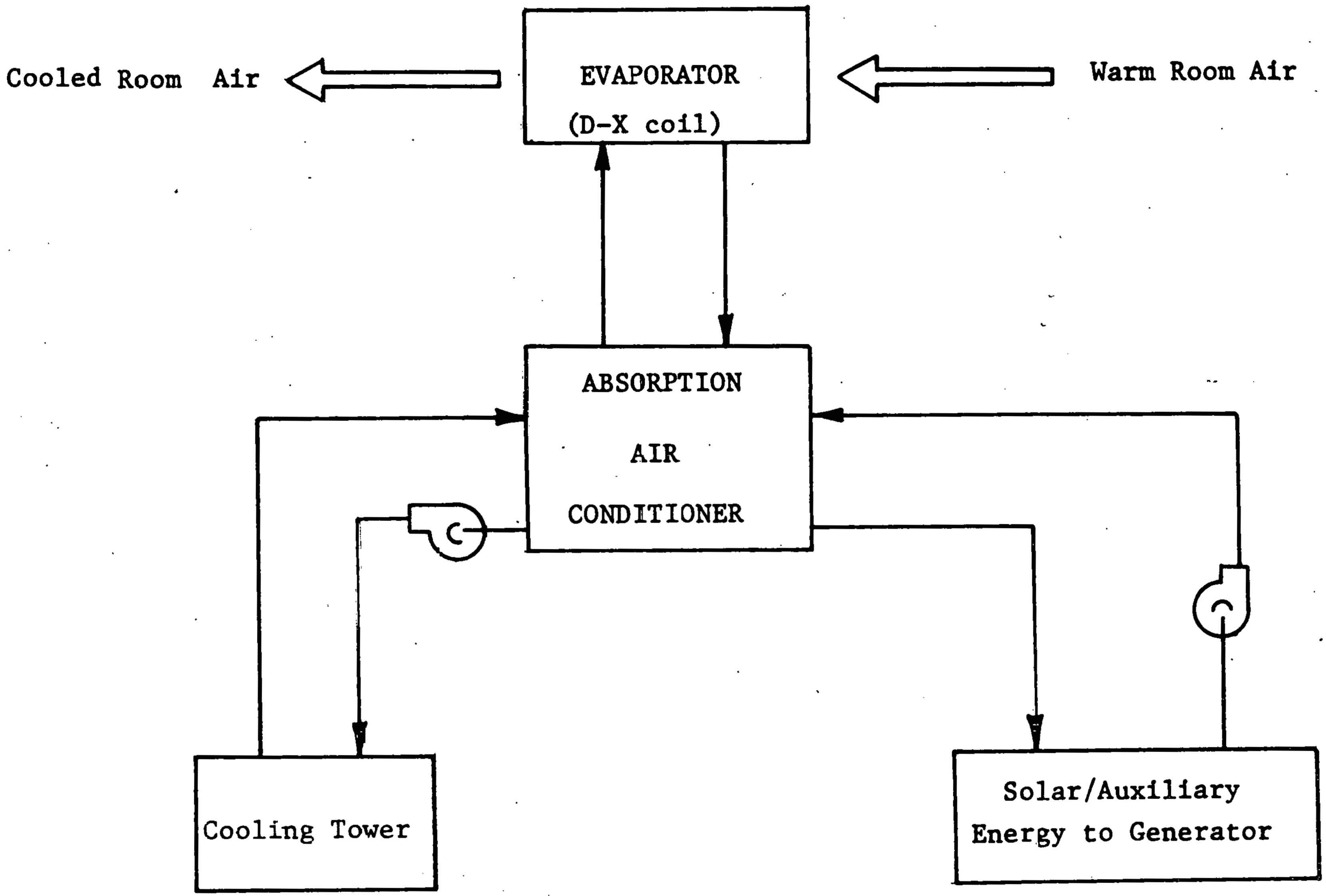


Fig. (7.1-2) Simplified Schematic Diagram of CSU Solar Cooling System[35]

The reduction of heat input to the generator, thus, resulted in a significant COP increase of the cooling unit. The average operating temperature of the storage was reported lower when the heat was supplied by solar than that by auxiliary.

C) Summary of Cooling System Performance

The results of the cooling system performance during the Summer of 1974, are summarized in Table 7.1-1, which is self-explanatory.

Average COP of the cooling unit, based on both solar and auxiliary (gas) heat inputs, was approximately 0.60.

From Column (8), it is seen that the percent cooling by solar energy was about 31% during the three-month period. That is the actual value obtained from the tests of the CSU Solar House I.

Had there been no heat losses from the heat storage and from other solar equipments, the percent cooling by solar energy would jump from 31% to 55% during the same period. See the numbers in Column (9).

During the period of 7/10 - 8/7, the cooling unit was not allowed to run at night when the cooling demand was low, thus eliminating much of the cycling of the unit. In this case, the percent cooling by solar energy could increase up to 81% when no heat losses from the solar equipment is again assumed. The estimated value of 81% may still be conservative since the improvement of system control (such as control Mode II and III during the period of 8/11 - 8/31) would result in better performance of the solar cooling system.

7.1.5 Effects of Control System Modifications on Cooling System Performance

During the month of August, 1974, three different modes of control systems were employed and the results were reflected as follows.

TABLE (7.1-1) MONTHLY AVERAGES OF COOLING VALUES OF CSU SOLAR COOLING SYSTEM (1974)[35]

Column	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
							% Cooling by Solar		
Month	"Total"[a] Heat to Cooling Unit (MBtu/day)	Total Cooling Load (MBtu/day)	COP of Cooling Unit	Imposed Cooling Load[b] (MBtu/day)	"Revised"[c] Cooling Load (MBtu/day)	Cooling by Solar Only (MBtu/day)	Solar Heat to Cooling Unit ÷ Column (1)	Column (6) ÷ Column (2)	Column (6) ÷ Column (5)
June	0.295	0.173	0.59	0.080	0.093	0.032	12.3	18.6	34.6
July	0.360	0.239	0.66	0.088	0.151	0.095	34.3	39.7	62.9
August	0.408	0.218	0.54	0.083	0.135	0.069	28.2	31.4	50.8
Season	0.354	0.210	0.59	0.084	0.126	0.065	25.9	31.0	51.6
7/10 - 8/7[d]	-	0.181	-	0.081	0.101	0.082	-	-	81.1

(a) Total = Solar + Auxiliary

(b) Added cooling load caused by heat losses from Solar equipment to buildings interior

(c) Revised = Without imposed load caused by heat losses

(d) Special period during which cooling unit was not allowed to run at night in order to eliminate much of the unit cycling

A) Control System Modification

a) August 1, 1974 - August 10, 1974

A two-stage control was utilized for selecting solar or auxiliary to meet the cooling load. Cooling was provided by solar whenever the building temperature rose above 72°F and the storage temperature exceeded 180°F. If the cooling demand could not be met, and the building temperature rose above 74°F, auxiliary heat was, then, used to meet the full cooling load even though the storage temperature exceeded the minimum generator inlet temperature.

b) August 10, 1974 - August 20, 1974

The previous control system was modified to provide solar cooling whenever the storage temperature exceeds the minimum generator inlet temperature of 180°F.

c) August 20, 1974 - August 31, 1974

The new modification is the lowering of the minimum generator supply water temperature from 180°F to 171°F. This is possible due to the availability of low cooling water temperature.

B) Results of Control System Modifications

The table below shows the results reflected by the change in control modes. By going from Control Mode I to Mode III, the useful heat delivered to the thermal storage was increased by 81%, while 111% increase in solar cooling was also achieved.

Control Mode	Time Span	Qu* (Btu X 10^{-6})	Cooling** (Btu X 10^{-6}) Aux.(Gas)	Solar	Solar Cooling (%)
I	8/1 - 8/10	2.03	0.96	0.35	27
II	8/11 - 8/20	2.68	1.37	0.71	34
III	8/21 - 8/31	3.68	0.90	1.13	56
	TOTAL	8.39	3.23	2.18	40

* Useful heat delivered to thermal storage

** Heat delivered to generator times COP

7.1.6 Effects of Unit Cycling on Cooling System Performance

A) During Days of High Cooling Demand

The cooling unit starts in the morning and runs nearly continuously at the COP which closely corresponds to the design value.

B) During Days of Low Cooling Demand

The unit runs intermittently and the number of cycling can be as many as 40 times in a 24-hour period. Warming up period of the unit could be as long as 15 minutes. The generator cools off in about 10 minutes after the unit is off. Assuming that the unit runs only 10 minutes continuously, the COP would decrease in half due to the heat losses during one stop-start period.

In September, 1974, when the cooling load was down to only 5,230 Btuh (seasonal average, 8,570 Btuh), the COP fell to 0.37 (seasonal average, 0.59), which is a drastic 37% decrease.

Figure 7.1-3 is an excellent example of the cooling performance of the unit during the period of low cooling demand. From the figure, significant reduction in COP can be easily visualized during the unsteady state period caused by the cycling of the unit. On this particular day, the number of cycling was 39 and the average COP of the unit was below 0.4.

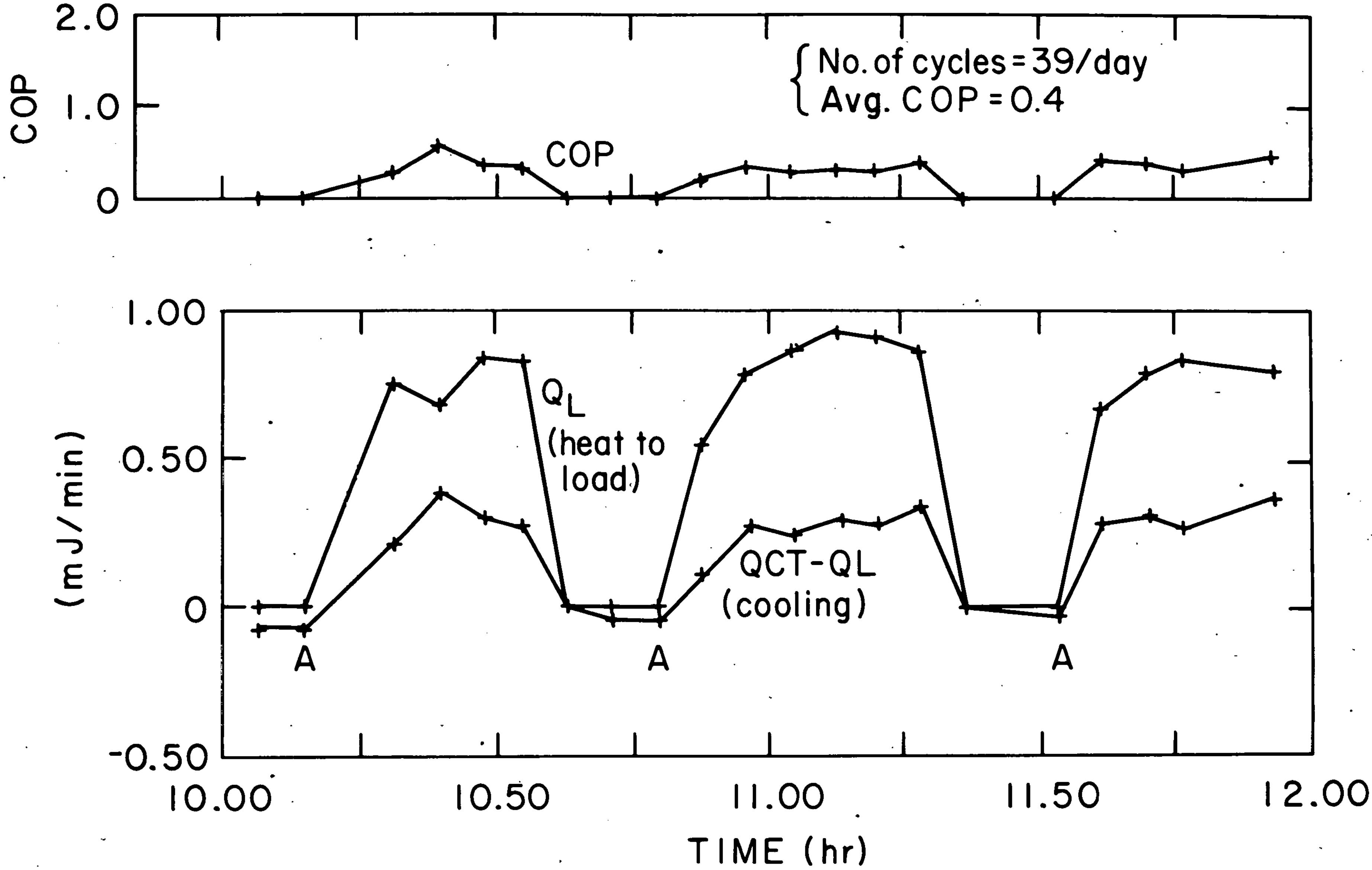


Fig. (7.1-3) Coefficient of Performance and Cooling Rate versus Time of Start-up for Absorption Cooling Unit at CSU Solar House I[35]

C) Addition of Cold-Side Storage

The use of cold storage can significantly reduce the ineffectiveness resulting from cycling of the cooling unit since the coolness produced by the continuous running of the unit can be stored in the cold storage and can be used later.

The effect of adapting cold storage on the cooling performance will be included in the later sections (CSU Solar House III).

7.1.7 Effects of Heat Losses on Cooling System Performance

A) Heat Loss from Storage Tank

Heat loss from the thermal storage which was located indoors, not only increased the cooling load but also reduced the amount of cooling provided by the absorption unit. Approximately 21 Btuh/^{o}F of thermal loss from the storage tank was measured. It amounted to the average heat loss rate of 1,780 Btuh during the month of August, 1974, which corresponds to 7 - 8 % of the sensible heat, 21,600 - 26,900 Btuh, removed by the cooling unit.

B) Other Losses

Heat losses, from the solar preheat tank of hot water system (≈ 500 Btuh), the surfaces of the auxiliary hot water tank, pumps, piping, and heat exchangers, contribute a significant factor to the cooling load in August.

C) Total Heat Losses from Solar Equipment

Approximate total heat loss of 2.48 X 10^{6} Btu was reported during the summer of 1974. It corresponds to 46% of the total cooling load of 5.41 X 10^{6} Btu during the same period.

D) Improvements

Assuming no such heat losses, solar heat would have provided 74.7% of the cooling load requirements. Better insulation of solar equipment and the outdoor venting of excess heat from the system components, would significantly improve the cooling system performance. With improved insulation, the heat loss from the thermal storage was cut in half.

7.1.8 Effect of Low Humidity on Unit's Cooling Performance

In the area of Fort Collins where CSU Solar House I (or III) is located, low humidity prevails during the cooling season. Since the performance data were taken during the periods of low latent heat load, the COP of the cooling unit was substantially reduced.

To illustrate the point, the following example was given. Under the normal conditions, sensible cooling of 27,000 Btuh and latent heat removal of 10,000 Btuh can be obtained at the generator heat input of 54,000 Btuh. This corresponds to COP of 0.69. Assuming no latent heat load, however, the COP is reduced from 0.69 to 0.50. Therefore, the cooling unit at CSU Solar House I should have the COP close to 0.50 if it had been run at design operating conditions.

7.1.9 Maintenance of Cooling Unit

During the period of July 1, 1974 to August 31, 1975, there was only one service call on the cooling unit. The problem was a slow leak of the system. Vacumm pumping of the system simply solved the problem.

7.2 University of Florida (UF)

Much of the R & D work concerning the use of NH_3-H_2O absorption units for solar cooling applications has been generated at the University of Florida. Following the technically successful operation of an intermittent NH_3-H_2O absorption units at low generator input temperatures (140 - 180°F) from the flat plate collector in 1958,[9] University of Florida designed and built a 3.5 ton continuous NH_3-H_2O absorption units for the solar applications in 1962. Absorber/condenser components of the absorption chiller were water-cooled and the hot water input to the generator was supplied from the flat plate collector.

The performance of the installed system is shown in Figure 7.2-1. The absorption unit was powered by the water supply temperature from 140 to 200^{o}F with usual operation at about 175^{o}F. At the steady state operation of the unit for 1.5 hours, the unit produced 2.4 tons of cooling capacity with the COP of about 0.57 .

Although the technical feasibility of operating the water-cooled NH_3-H_2O absorption chiller at the flat plate collector temperature was demonstrated, the overall performance of the non-optimized unit was far from the satisfactory level.

In 1974, UF adapted an intermittent NH_3-H_2O absorption chiller to its Solar Test House. They said that the intermittent unit was chosen because of its simplicity in design and operation. However, it requires a complex control system. The simplified schematic diagram of the UF cooling system is shown in Figure 7.2-2.

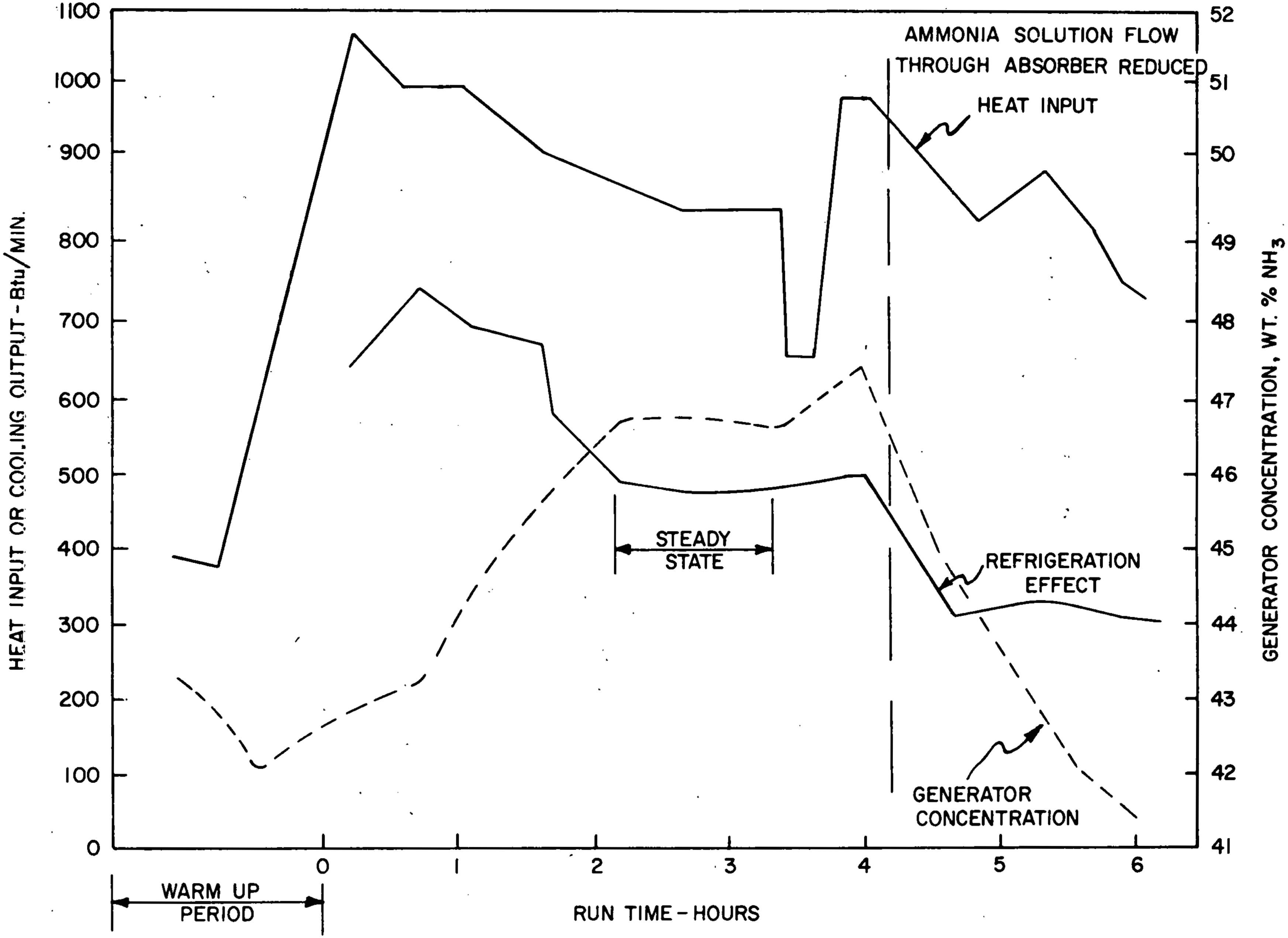


Fig. (7.2-1) Performance of University of Florida's 3 ton Continuous NH_3-H_2O Absorption Chiller[10]

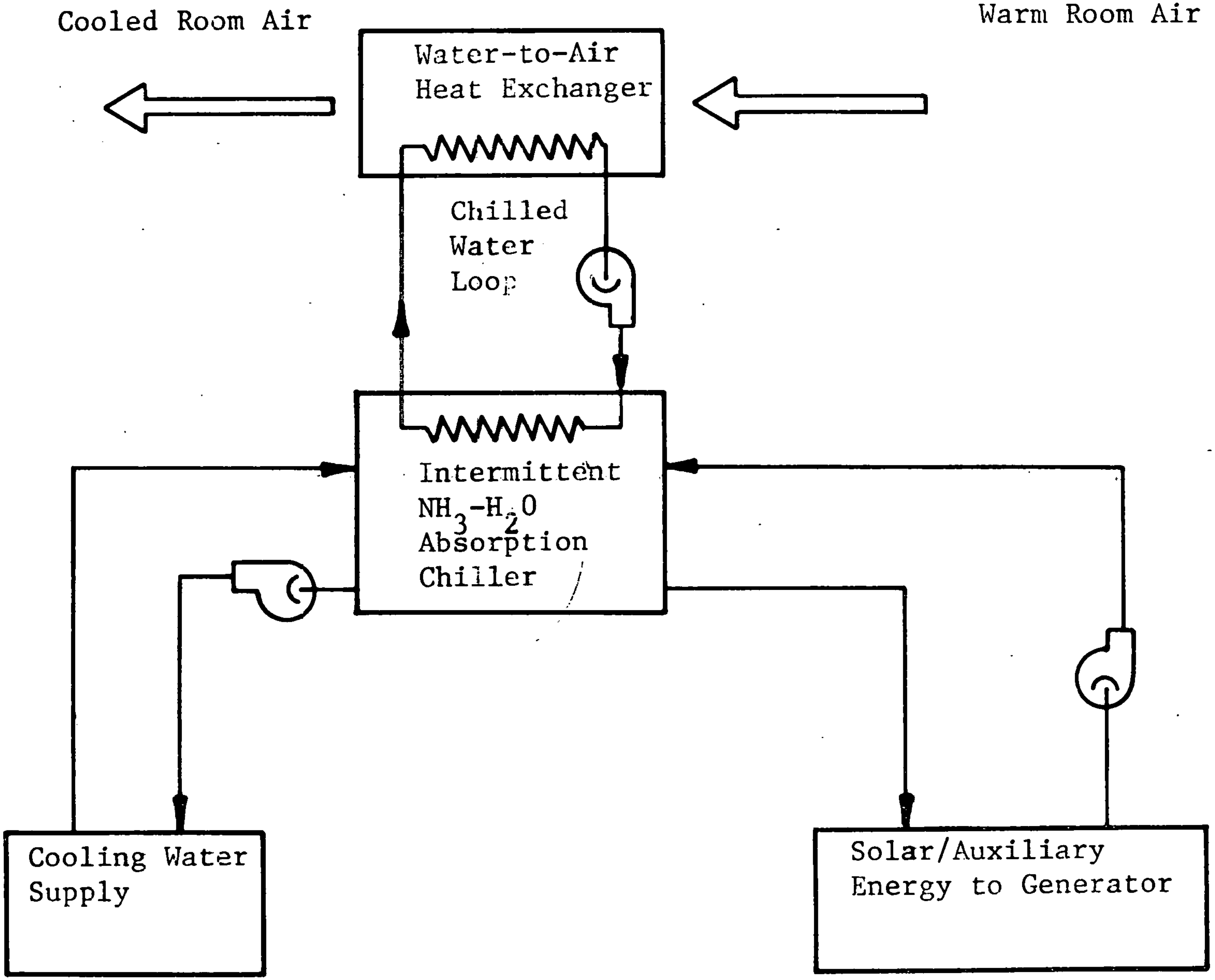


Fig. (7.2-2) Simplified Schematic Diagram of University of Florida's Absorption Cooling System[11]

University of Florida's intermittent system (3 ton) was designed to provide enough cooling for 3 - 6 hours of operation for a single generation cycle of 45 minutes. The intermittent cooling unit was tested under the actual living conditions at the University of Florida's Solar Research Residence. Although the detailed results of the tests are not available, it was reported that the COP as high as 0.57 was obtained[20].

The design of new continuous NH_3-H_2O absorption unit was started in the Spring of 1976 and now has been completed. The unit was water-cooled and was designed to be operated at the generator inlet water temperature less than 180°F. UF is now seeking for funding to build the prototype unit to be tested at the UF Solar Test House. Then the performance comparison will be made between the intermittent and the continuous NH_3-H_2O units and the complete assessment of the two different systems will be evaluated. It was said that the energy efficiency ratio (EER) and the initial cost of the continuous absorption unit are more favorable than those of intermittent type units.

7.3 University of Wisconsin

In 1959, Chung et al[30] indicated the use of hot-side and cold-side energy storages as well as the possible storage of liquid refrigerant in the absorption cooling system.

Technological feasibility of solar cooling with a H_2O-LiBr absorption air conditioner (3-ton, Arkla DUCS-2, steam-fired) was demonstrated experimentally by the same group[31] in 1962. The results of their experiments are presented in Figures 7.3-1 and 7.3-2, which are self explanatory. Figure 7.3-1 is the performance of the solar cooling system on a clear day, and Figure 7.3-2 on a cloudy day.

The COP of the H_2O-LiBr unit in the above experiments was about the same ($\approx$ 0.6 with hot water supply temperature above 180^{o}F) as that predicted in their previous study[30]. The experimental values of the flat-plate collector efficiency, however, were 24 to 37% as compared to the values (38 to 52%) used in the previous study. As a result, the overall, steady state COP of the experimental system (= Air conditioner COP X collector efficiency) using a flat-plate collector of 25% efficiency, was about 0.15 compared to the predicted value of about 0.30 in their previous study (with collector efficiency of 0.5).

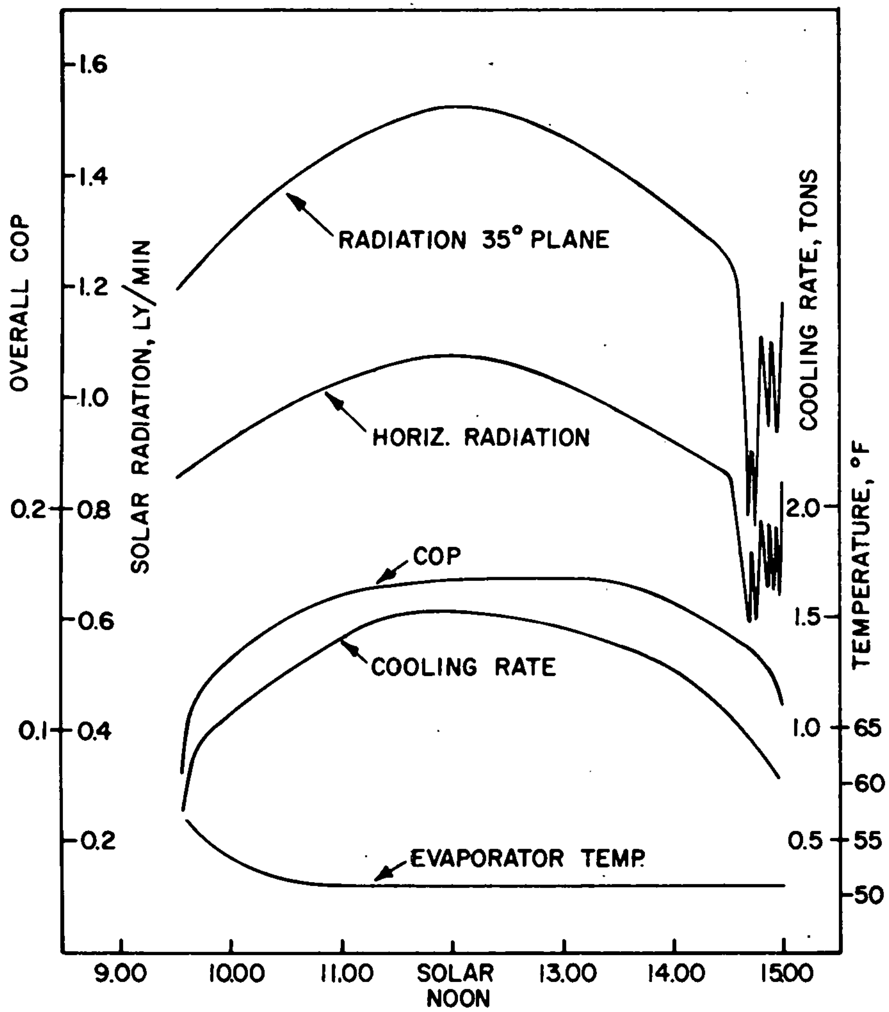


Fig. (7.3-1) Performance of Solar Absorption Cooling System on a Clear Day at Univeristy of Wisconsin[31]

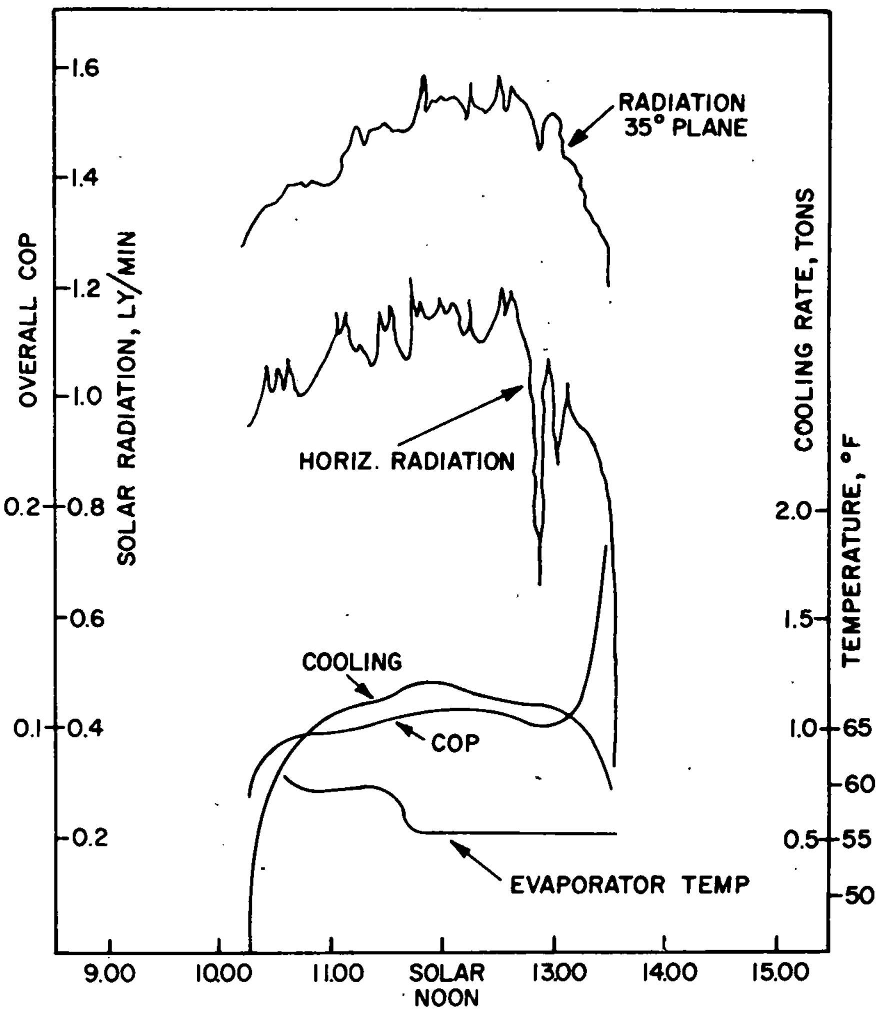


Fig. (7.3-2) Performance of Solar Absorption Cooling System on a Cloudy Day at University of Wisconsin[31]

7.4 The Ohio State University (OSU)

OSU has a solar house which was claimed to be the first in Ohio. The solar house which is well insulated, is a one-story building with about 2200 ft^2 of conditioned living space. The OSU Solar House employs 720 ft^2 flat plate collector, 1200 gallon water storage tank (well insulated) and a fan-coil unit for direct heating and uses modified Arkla 3-ton H_2O-LiBr absorption unit which is fired by hot water from the storage tank, for space cooling. As a back-up system for the period of no sunshine, a solar-assisted heat pump (modified 4 ton G. E. Weathertron split unit) is employed for both heating and cooling. An auxiliary electric duct heater is also installed for the worst weather conditions.

Previously OSU Solar House employed two 20,000 gallon water storage tanks which did not properly provide high enough temperature to power the modified Arkla absorption unit (similar to Solaire Model 501-WF). In order to operate the absorption unit properly, approximately 190^{o}F hot water from the storage was required. Difficulties in obtaining sufficiently high temperatures in the storage tanks were found to be due to the excessive heat loss, poor collector performance as well as large size of the storage tanks.

The cooling test of the OSU Solar House is again planned during the Summer of 1977. They will use smaller storage tank (one 1200 gallon size) with improved insulation and a new flat plate collector, which would be double glazed PPG collector with selective coating.

8. What can be done to reduce or eliminate the limitations of absorption cooling?

8.1 Water-cooling limitation of Absorber/Condenser Components

8.1.1 H_2O-LiBr Absorption Cycle

The major drawback of the H_2O-LiBr unit could be the requirement of water-cooling. As mentioned earlier in Section 4.1, addition of cooling tower could increase the initial cost and create maintenance problem especially when the unit is of residential size. It was also mentioned in Sections 2.2 and 4.1 that the H_2O-LiBr unit can <u>not</u> be air-cooled due to the crystallization problem.

In 1968, however, new refrigerant-absorbent solution for possible use in an air-cooled absorption unit was introduced[12]. The working fluid comprises 3 components: water as a refrigerant. lithium bromide/lithium thiocyanate mixture (LiBr/LiSCN) as the absorbent. The addition of the third component, lithium thiocyanate, to the H_2O-LiBr pair does prevent crystallization at temperatures prevalent in an air-cooled system. The working fluid was said to have low heat capacity, low viscosity and low heat of dilution. It is also non-corrosive, non-toxic and thermally stable. The molar concentration ratio of LiBr to LiSCN may range from 0.25 to 4.00.

Although the possible air-cooling with the new working fluid is indicated, the fully air-cooled unit still requires a generator temperature of about 245^{o}F, which is somewhat higher than that from the present flat plate solar collectors[15]. As mentioned in Section 6.1-3, Arkla/IGT is planning to employ this working fluid in developing their fourth generation air-cooled solar absorption unit. The material's problem (previously mentioned in Section 6.1-3), however, should be solved first.

8.1.2 NH_3-H_2O Absorption Cycle

In Section 4.1a, it was said that the operation of NH_3-H_2O absorption unit at the generator input temperature below about 250^oF would require water-cooling. In other words, NH_3-H_2O absorption unit requires water-cooling when powered by hot water from the flat-plate solar collectors.

Recently an experimental study of NH_3-H_2O absorption chiller at low generator temperatures (say 175 ~ 210^oF) was performed and possible operating conditions for an air-cooled NH_3-H_2O unit were indicated[13,14]. Arkla gas-fired 5 ton NH_3-H_2O absorption unit (Model ACB-60) was modified and used for the study. The evaporator temperature, Te, was expressed as[14],

$$Te = fn\ (Tc,\ Ta,\ Tg,\ \Delta Pa,\ \&\ \Delta x) \qquad \text{Eq. 8.1-1}$$

where Tc = condensing temperature

Ta = absorbing temperature

Tg = generating temperature

ΔPa = pressure drop across absorber

Δx = concentration difference between weak and strong solutions

The relationship of Eq. 8.1-1 was presented and is shown here as Figure 8.1-1. Here, Δx of 0.03 corresponds to the maximum relative circulation ratio of about 16. The relative circulation ratio is defined as the mass flow rate of weak solution (NH_3-poor solution) per unit mass flow rate of refrigerant (NH_3), or;

$$RCR\ (\text{relative circulation ratio}) = \frac{1-Zs}{Zs\ -\ Zw} \qquad \text{Eq. 8.1-2}$$

where Zw = refrigerant concentration in weal solution

Zs = refrigerant concentration in strong solution.

In Figure 8.1-1, the possible operating zone for an air-cooled unit is indicated by the dotted line. In other words, for $181 \leq Tg \leq 195^{o}F$, $43 \leq Te \leq 47^{o}F$ or $51 \leq Tcw$ (chilled water) $\leq 55^{o}F$ and $104 \leq Tc$ (or Ta) $\leq 111^{o}F$ or $9 \leq \Delta Tc$ (= Tc - 95) $\leq 16^{o}F$. In order to operate the unit at the above conditions, the following remedies have been suggested[14].

a) Remedies for high evaporating temperature

- Heat transfer surfaces of evaporator and/or chilled water fan-coil unit be increased
- Horse power requirement of chilled water pump may be increased

b) Remedies for small ΔTc

- Heat transfer area of absorber-condenser components be increased (or doubled)
- Cooling air flow rate for absorber-condenser components be increased (or doubled). This will increase the fan power requirement from 0.35 HP to 0.70 HP for 3 ton cooling capacity.

c) Remedies for high relative circulation ratio

- Power requirement of solution pump must be increased to 0.70 HP (assuming the pump efficiency of 40%) for 3 ton cooling capacity when the relative circulation ratio is 16 (compared with about 3 for conventional gas-fired NH_3-H_2O unit).

For 3 ton cooling unit, the total power requirement of fan and solution pump thus amounts to about 1.4 HP, which is equivalent to about 10% of the total cooling capacity (or 30% if based on fossil fuel consumption). Conventional gas-fired NH_3-H_2O unit employs fan/solution pump motor of 3/4 HP.

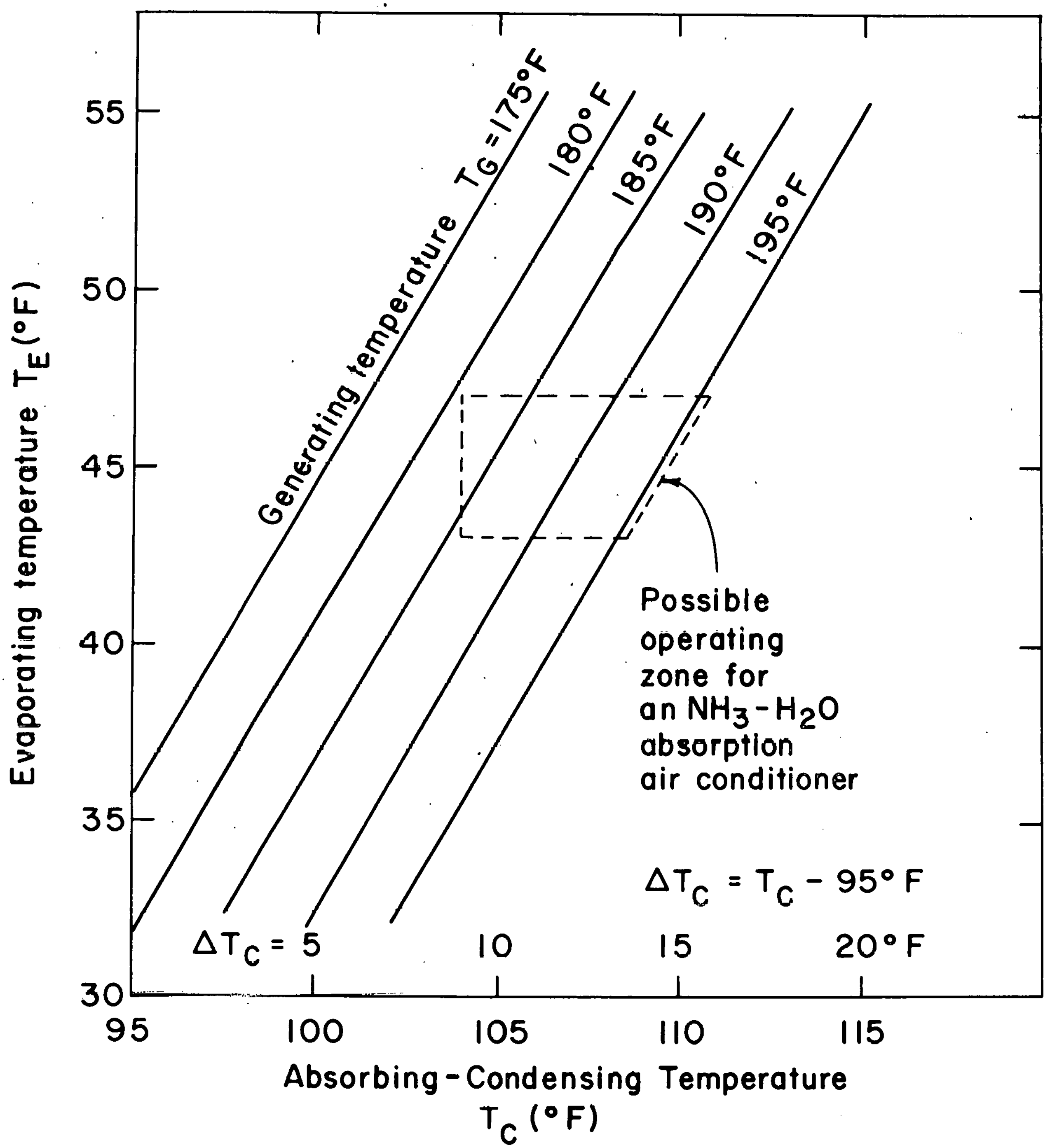


Fig. (8.1-1) T_E as a Function of T_G and T_A=T_C for a Pressure Drop Across the Absorber of 3 psi and a Solution Concentration Difference of 0.03[14]

The total power consumption equivalent to 35 Btuh of cooling per watt of electrical power input was also compared with about 8 Btuh per watt for a mechanical vapor compression unit.

The COP of 0.65 could be obtained for the modified NH_3-H_2O unit at $\Delta x = 0.03$ and at the solution heat exchanger effectiveness of 90%.

In summary, technical feasibility of using air-cooled NH_3-H_2O absorption unit at low generator temperature (corresponding to flat-plate solar collector) was demonstrated. Practicality of the proposed unit, however, needs to be proven.

8.2 COP Limitation

In general, the COP of the single effect absorption unit using conventional working fluid is limited and is always less than 1.0 (see Section 3.2). Some of the high efficiency absorption cycles which have not been mentioned before, will be discussed.

8.2.1 Double-effect Absorption Cycle

If the generator inlet temperature of steam or hot water is available at $350^{\circ}F \sim 400^{\circ}F$, double effect absorption cycle can be employed and the significant improvement in COP can result. Typically COP of the water-cooled, double-effect, H_2O-LiBr unit is improved to 0.99 from the 0.67 of single effect unit, and also the COP of the double effect unit is no longer limited to 1.0.

Figure 8.2-1 shows the schematic diagram of the Trane, double-effect, H_2O-LiBr unit[16]. The pressure of the first stage generator and condenser should be sufficiently high that the refrigerant condensing temperature of the first effect is to be higher than the second stage generator temperature.

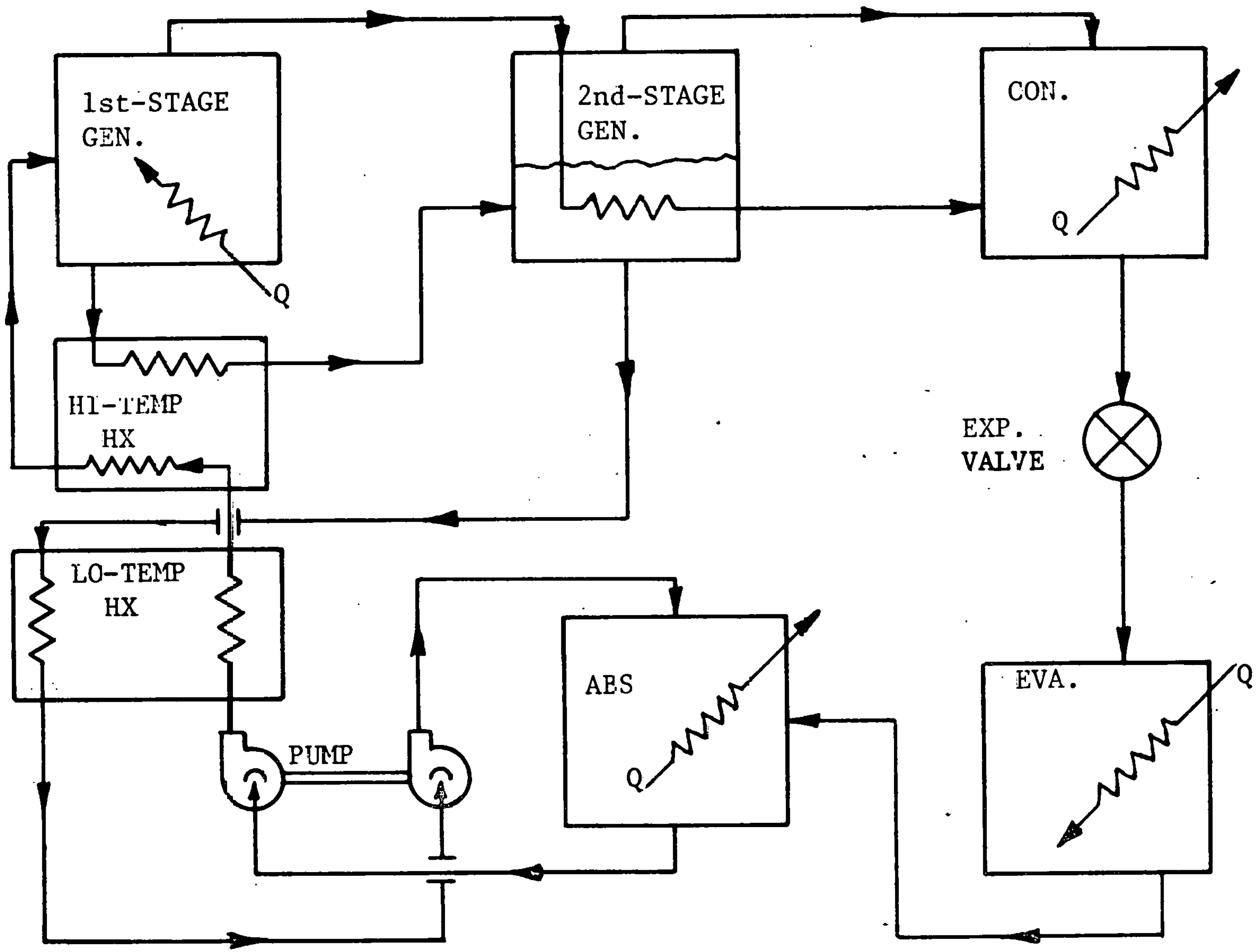


Fig. (8.2-1) Schematic Diagram of Trane, Double-Effect, H_2O-LiBr Absorption Cycle (Water cooled)[16]

Starting from the top left corner of the figure, steam or hot water is circulated into the first-stage concentrator to boil out refrigerant (water) from solution (H_2O-LiBr). Water vapor from the first-stage concentrator is then condensed in the second-stage concentrator, thus giving up the heat to the partially concentrated H_2O-LiBr solution from the first-stage concentrator.

The partially concentrated H_2O-LiBr solution from the first-stage concentrator is cooled in the high temperature heat exchanger by the returning diluted solution from the absorber and is passed to the second stage concentrator, where the solution again boils due to the heat gain from the condensing refrigerant from the first-stage concentrator. Refrigerant vapor generated from the second-stage concentrator passes to the condenser and is liquified by the cooling water circulated. Now the total amount of refrigerant or water accumulated in the condenser is the sum of the refrigerant liquid originated from first stage concentrator and the condensed refrigerant vapor from the second-stage concentrator.

The refrigerant liquid from the condenser is forced through the orifice (acts as an expansion valve) into the evaporator causing cooling effect as it evaporates. In the absorber, the refrigerant vapor from the evaporator is combined with the concentrated H_2O-LiBr solution from the second-stage concentrator.

Heat of absorption is carried away by the cooling water. Resulting diluted solution in the absorber is then pumped back to the first-stage concentrator via low temperature and high temperature heat exchangers. Heat content of the diluted solution to the first-stage concentrator increases as it passes through the above heat exchangers, thus improving the COP of the cycle.

Since the refrigerant vapor from the first-stage concentrator condenses in the second-stage concentrator by rejecting heat of condensation to the partially concentrated solution from the first-stage concentrator, the heat per ton of refrigeration rejected to the cooling water is reduced by approximately 20% as compared to the single-effect unit.

Concentrating type solar collector is a must to operate double-effect, H_2O-LiBr absorption unit. The capacities of the double-effect, industrial absorption units usually come in large sizes (400 - 1100 tons).

The double effect operation with air-cooled NH_3-H_2O unit is <u>not</u> practical due to the facts that the first effect pressure is too excessive (> 1100 psia), the pumping power requirement increases (7 times that of the single-effect cycle), and the second refrigeration effect reduces to 1/3 or less of that in the first effect[17].

8.2.2 Absorption Cycle Using Working Fluid that has Positive Deviation from Raoult's Law

As indicated in Section 3.2, the COP of a single-effect absorption cycle can theoretically break the 1.0 barrier if the employed working fluid has positive deviation from Raoult's Law, or $\Delta Hdil < 0$. See Eq. 3.2-2. It also states that the COP of the new fluid system depends strongly on the magnitude of the heat of dilution or $\{(\Delta Hdil)(Rr + 1)\}$ of the positive deviation fluid pair, which is usually 10 $\sim$ 20% of the heat of vaporization (ΔHv).

Fluid combinations that have positive deviation from Raoult's Law exist, but none of them has been identified adequate for a practical use. Recently vigorous R&D plans have been formulated to find or create such fluid pair in a systematic approach by IGT. Application of such new working fluids in a variety of absorption cycles would create a new horizon in solar application of the absorption technology.

8.2.3 Combination Absorption-Resorption Cycle (CAR Cycle)

The schematic diagram of CAR Cycle is shown in Figure 8.2-2. As seen in the diagram, this is a combination of two cycles. The left side of the double dotted line represents the Standard Cycle (Figure 1.1-1 or Figure 1.2-1) and the right, the Simple Resorption Cycle. Therefore, this cycle works just the same as those two separate cycles do.

I. Maiuri (England) first proposed an original concept of the Simple Resorption Cycle in 1936. It works as follows.

It is assumed here that the cycle employs NH_3-H_2O as a working fluid. Nearly pure ammonia vapor from the evaporator 1 passes to the resorber, where it is absorbed by the special solution pumped from the evaporator 2. Resulting ammonia-rich solution from the resorber is cooled in solution heat exchanger 2 before entering the expansion valve. In evaporator 2, a refrigerating effect is produced by both the heat of vaporization of NH_3 and the heat of dissociation of NH_3 from its solution.

Evaporated pure ammonia vapor passes to the absorber and it is absorbed by the ammonia-poor solution from the generator. Separated solution or special solution in the evaporator 2 is sent back to the resorber by pump 2 via solution heat exchanger 2.

Here, the special solution can be any combination of ammonia and other solvent. For this analysis, aqua ammonia solution which is still considered to be the best would be the special solution. The rest of the cycle works the same as the Standard Cycle. Note that evaporator 1, the resorber and solution heat exchanger 2 are under the intermediate pressure. The cycle needs two solution pumps and has two separate evaporators for the double refrigeration effect.

The COP of such a unit is about two times that of the standard cycle or 0.8 - 1.0 at about 300°F heat source temperature when the unit is water cooled[18]. This cycle has particular advantage that the two evaporating temperatures in evaporator 1 and 2 can be made different. For instance, Evaporator 1 can be used for air conditioning and Evaporator 2 for refrigeration.

The possibility of using air-cooled, CAR, NH_3-H_2O unit at slightly higher generator temperature (≈ 330°F) has also been indicated[17]. In this cycle, sub-atmospheric conditions exist as in H_2O-LiBr cycles. It is, however, obvious that the unit is not well suited for small tonnage applications due to the added initial cost of construction.

8.2.4 Generator-Absorber Heat Exchange Cycle

The cycle concept has been presented recently by Philips[17] and the schematic diagram of the cycle is shown in Figure 8.2-3.

The proposed cycle is the same as the standard NH_3-H_2O Cycle (Figure 1.1-1) with the following exceptions:

a) Solution heat exchanger (SHX) no longer exists.

b) Absorber heat exchanger (AHX) is added as an integral part of the absorber. Refrigerant-rich solution from the absorber is pumped through the absorber heat exchanger and the absorption heat is transferred to the solution, thus increasing the heat content of the solution to the distillation column.

c) Generator-absorber heat exchanger (GAX) is added between ANA/GEN and absorber (Conceptually shown in Figure 8.2-3). When generator temperature is sufficiently high (>> 300°F), the peak absorption temperature could be higher than the feed solution (refrigerant-rich solution from absorber to analyzer) temperature. Then the absorption heat at the hot end of the absorber can be transferred to the cool end of the distillation column via GAX.

As explained in (b) and (c) above, significant portion of the absorption heat could be utilized in the distillation process, thus the external heat input to the generator could be reduced and the significant improvement of COP may result. It was reported [17] that, for an air-cooled NH_3-J_2O unit, COP of 1.0 or more could be realized if all absorption heat at temperatures higher than the feed solution temperature by 20°F, is used to heat the cool

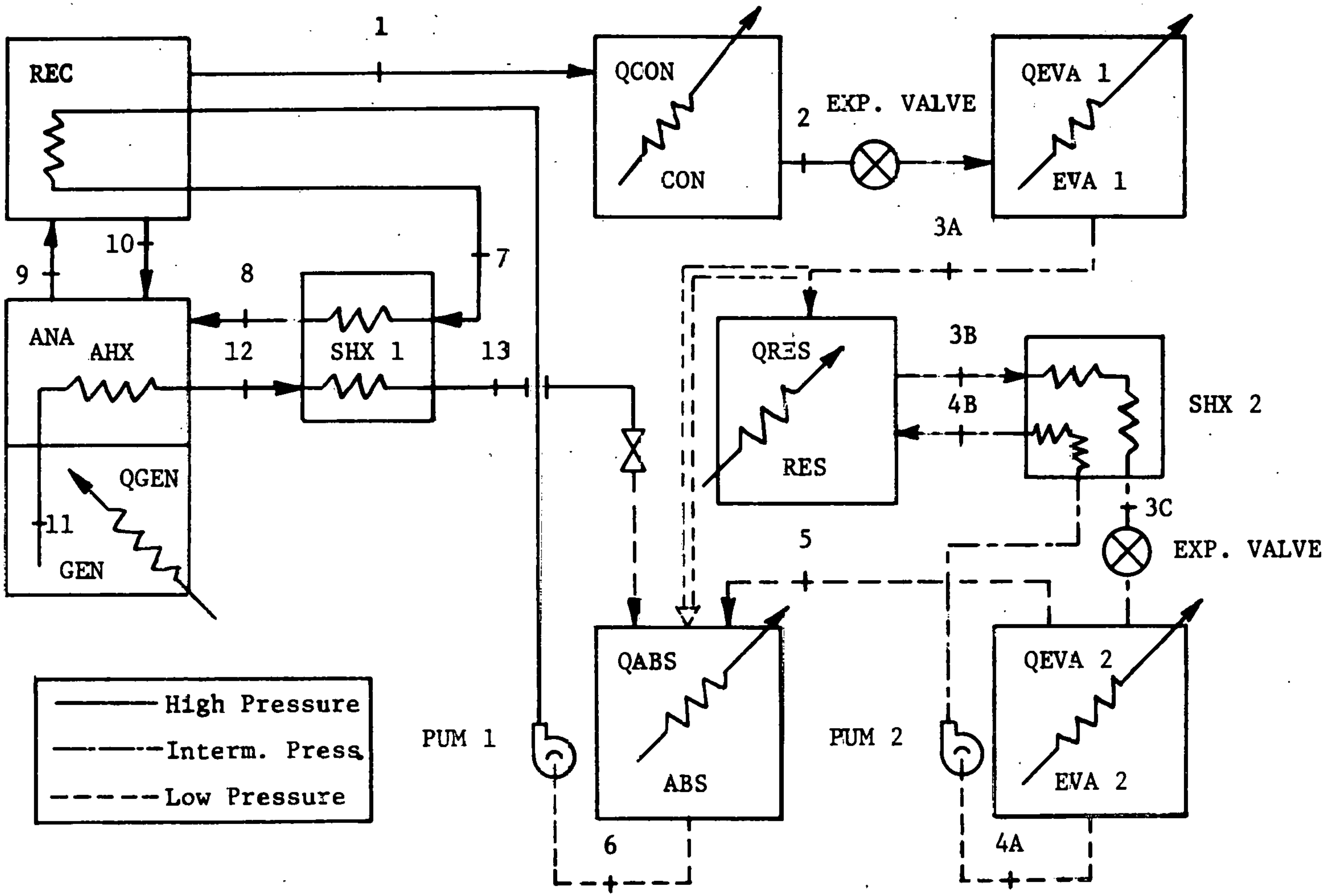


Fig. (8.2-2) Combination Absorption - Resorption Cycle[19] end of the distillation column.

No actual unit, using this concept, exists and the concept of GAX may generate difficult hardware problems.

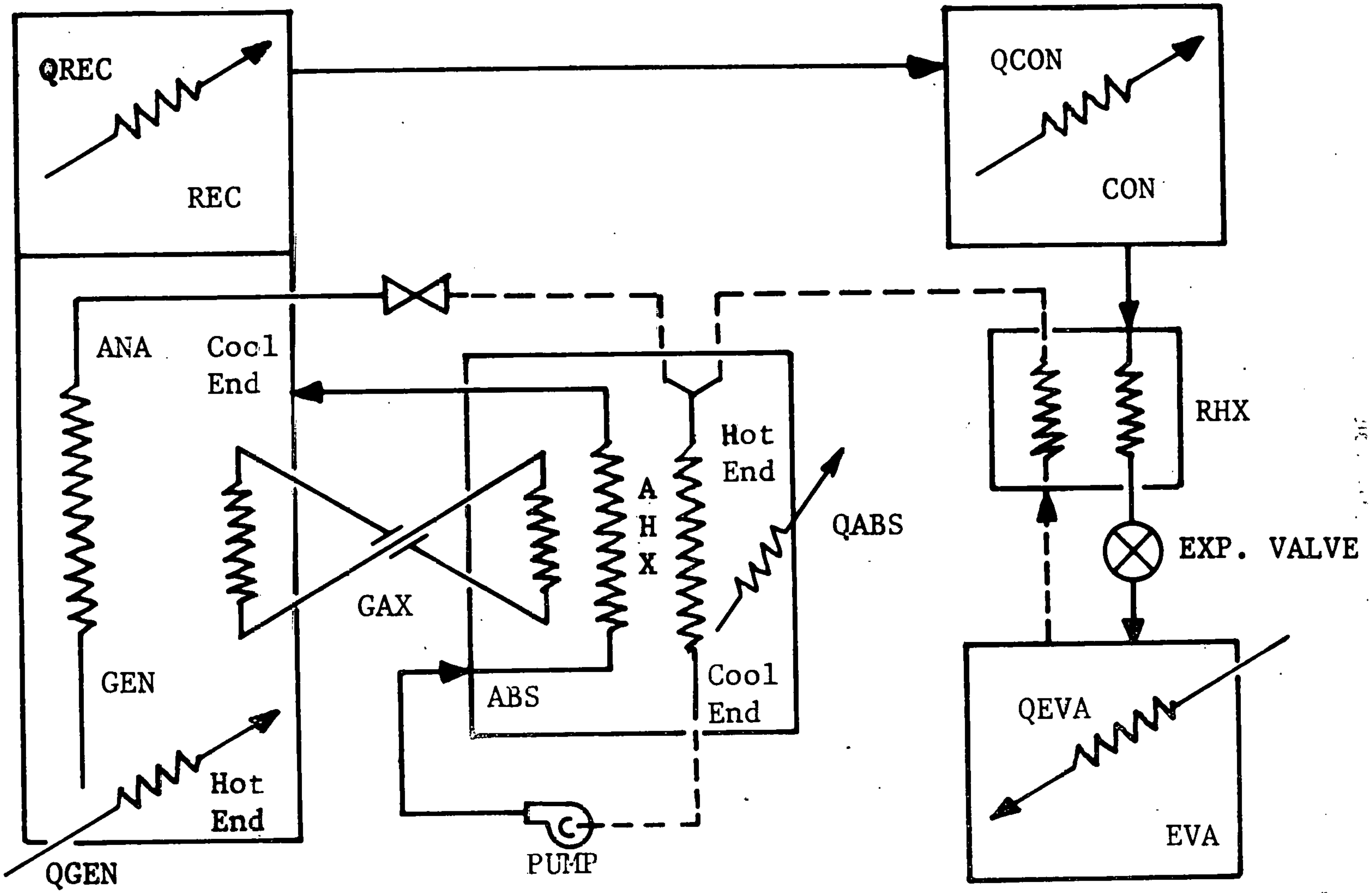


Fig. (8.2-3) Schematic Diagram of Generator - Absorber Heat Exchange Cycle[17]

8.2.5 Miscellaneous Hybrid Cycles

Proposed recently are the various types of hybrid cycles, which are the combinations of electric vapor compressor with the various types of absorption cycles. Significant improvement on chiller performance may result from the application of the hybrid cycle, but the major drawback seems to be the increase in the initial cost of construction and the added problems associated with the vapor compressor. The detailed description of the hybrid cycles are omitted here since the information is premature and also is proprietary in nature.

8.3 Degradation of Absorption System COP due to Frequent Cycling of Absorption Unit at Reduced Cooling Load

When the cooling load demand is low (late Spring or early Autumn), frequent cycling (on-off) of the absorption unit results. Energy loss from the generator component during frequent transient periods, contributes to the significant degradation of the system COP. See section 7.1.6 for detailed discussion.

8.3.1 Addition of Cold-Side Storage

The use of cold storage(s) could significantly reduce the unit cycling. The degradation of system COP due to cycling could be minimized since the coolness produced by the continuous running of the unit can be stored and can be used later.

This important subject should be discussed in detail and is summarized here from Reference 36, "Cooling Subsystem Design in CSU Solar House III".

A) General Information

Solar House III is generally the same as Solar House I except that:

- it uses high performance collector (evacuated tubular, liquid type) for more efficient collection of solar energy

- it employs cold storages
- improved absorption chiller (Yazaki) is used
- the system is being constructed and will be used for the cooling test during the Summer of 1977

B) Cooling Subsystem

a) Cooling Unit Specifications

Table 8.3-1 contains the specifications of Yazaki absorption chiller.

b) Performance of Cooling Unit

Figures 8.3-1 and 8.3-2 show the COP and cooling capacity as a function of the generator inlet temperature at the cooling water temperature of 75°F. Note that the COP is 0.68 at the design generator temperature of 190°F (88°C) when the cooling water temperature is 75°F. Due to the lower cooling water temperature available in the Fort Collins area, it is 13% increase in COP over the design value of 0.60 at the cooling water temperature of 85°F.

C) Simplified Schematic Diagram of Modified Cooling System

Figure 8.3-3 shows the simplified schematic diagram of the modified cooling system employing cold storages. The use of two, variable-level cold storage tanks of identical size provides the required stratifications for the near optimum operation of the chiller at close design conditions (inlet chilled water temp. of 14 - 15°C and outlet chilled water temperature of 8 - 9°C).

Compared to Figure 7.1-2 in Section 7.1, the modified system of Figure 8.3-3 requires additional components such as a fan/coil unit and two water pumps as well as two cold storages. A refrigerant-to-water evaporator is now an integral part of the cooling unit.

Table 8.3-1

Specification of Yazaki's H_2O-LiBr Absorption Chiller[36]

(Model No. WFIC-600S)

Refrigeration Capacity	6000 Kcal/hr (25.1 MJ/hr)	26,500 Btu/hr (2.2 tons)
Chiller water outlet temperature (design)	9°C	48°F
Chiller water outlet temperature (minimum)	8°C	46.5°F
Chilled water inlet temperature minus outlet temperature	5°C	9°F
Chilled water flow rate (design)	1200 ℓ/hr	5.3 gpm
Chilled water flow rate (maximum)	2040 ℓ/hr	9.0 gpm
Generator input	10,000 Kcal/hr (42.9 MJ/hr)	44,160 Btu/hr
Coefficient of performance (COP)	0.6	0.6
Generator inlet temperature (design)	88°C	190°F
Generator inlet temperature (range*)	75-100°C	167-212°F
Generator inlet temperature (range**)	70-100°C	158-212°F
Generator inlet temperature minus outlet temperature	6°C	10.8°F
Generator flow rate	1668 ℓ/hr	7.4 gpm
Heat rejection capacity	16,000 Kcal/hr (67.0 MJ/hr)	70,660 Btu/hr (5.9 tons)
Cooling water inlet temperature (design)	29.5°C	85°F
Cooling water inlet temperature (range)	24-32°C	75-90°F
Cooling water outlet temperature minus inlet temperature	5°C	9°F
Cooling water flow rate	3200 ℓ/hr	14.1 gpm
External dimensions	0.53X0.625X 1.75 m	1.74X2.05X 5.74 ft
Weight	160 Kg	352.8 lbs

* Cooling water temperature 30°C

** Cooling water temperature of 20-30°C

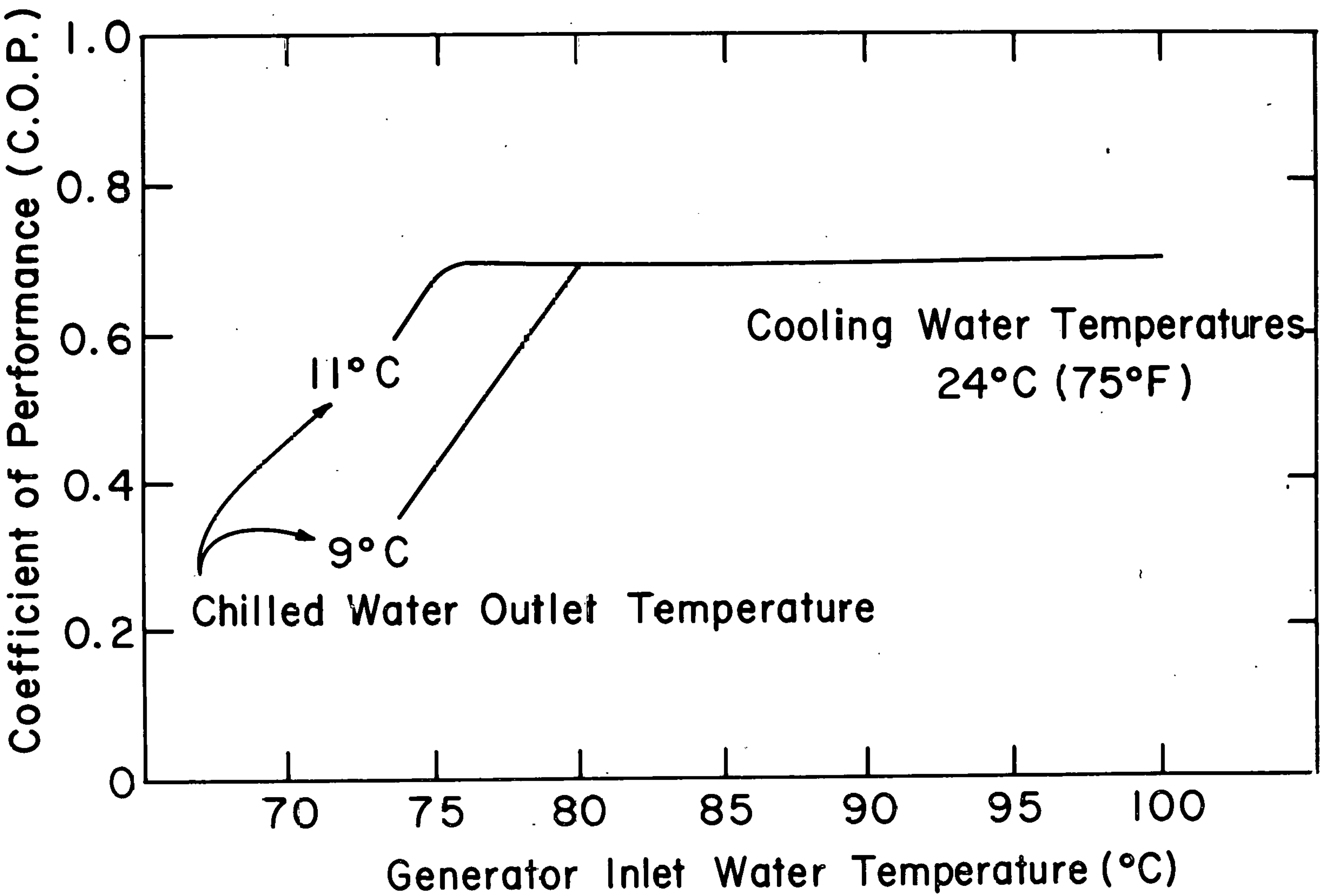


Fig. (8.3-1) COP of Yazaki's H_2O-LiBr Absorption Chiller as a Function of Generator Inlet Water Temperature[36] (Model WFC-600S)

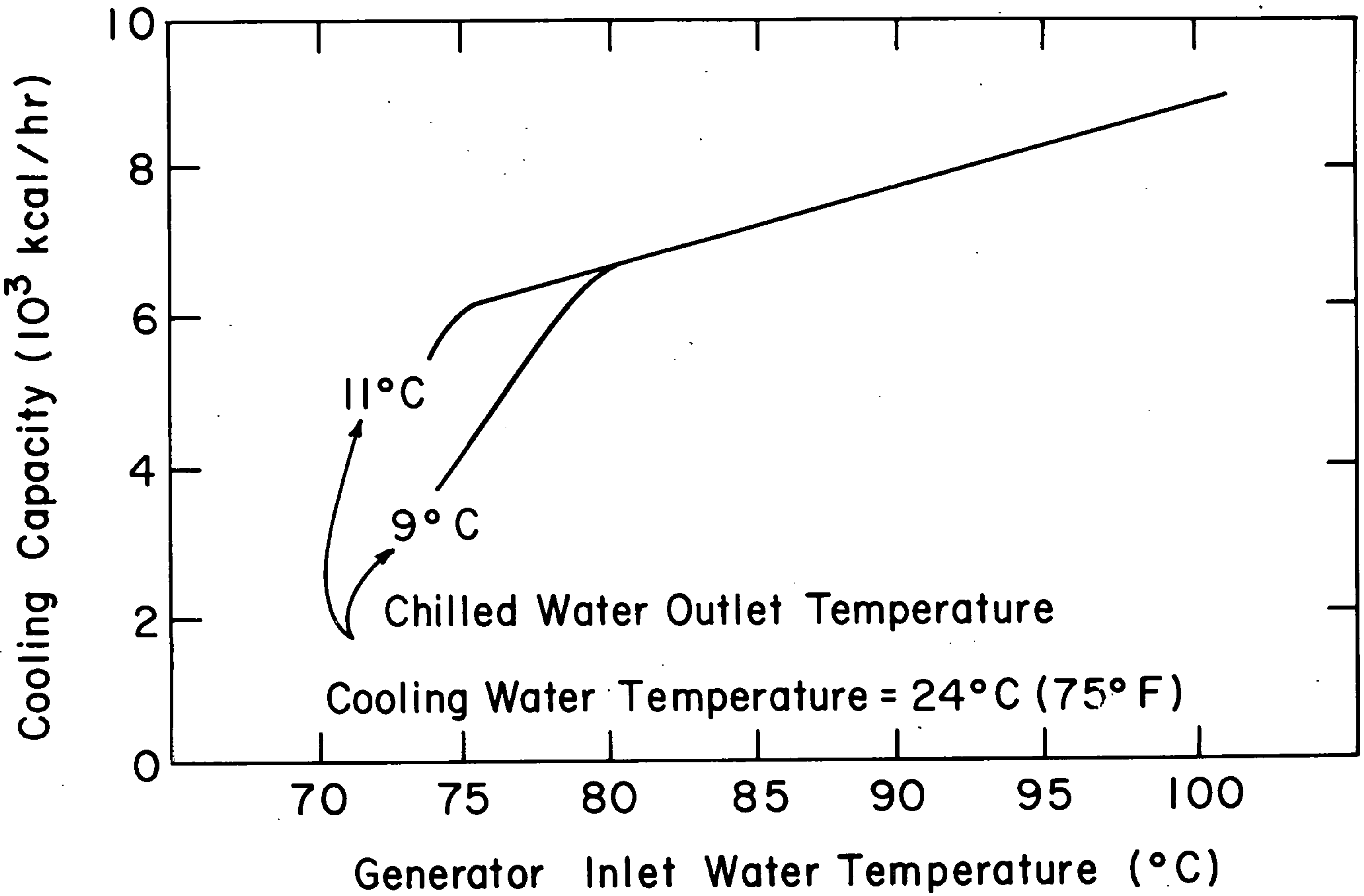


Fig. (8.3-2) Capacity of Yazaki's H_2O-LiBr Absorption Chiller as a Function of Generator Inlet Water Temperature[36] (Model WFC-600S)

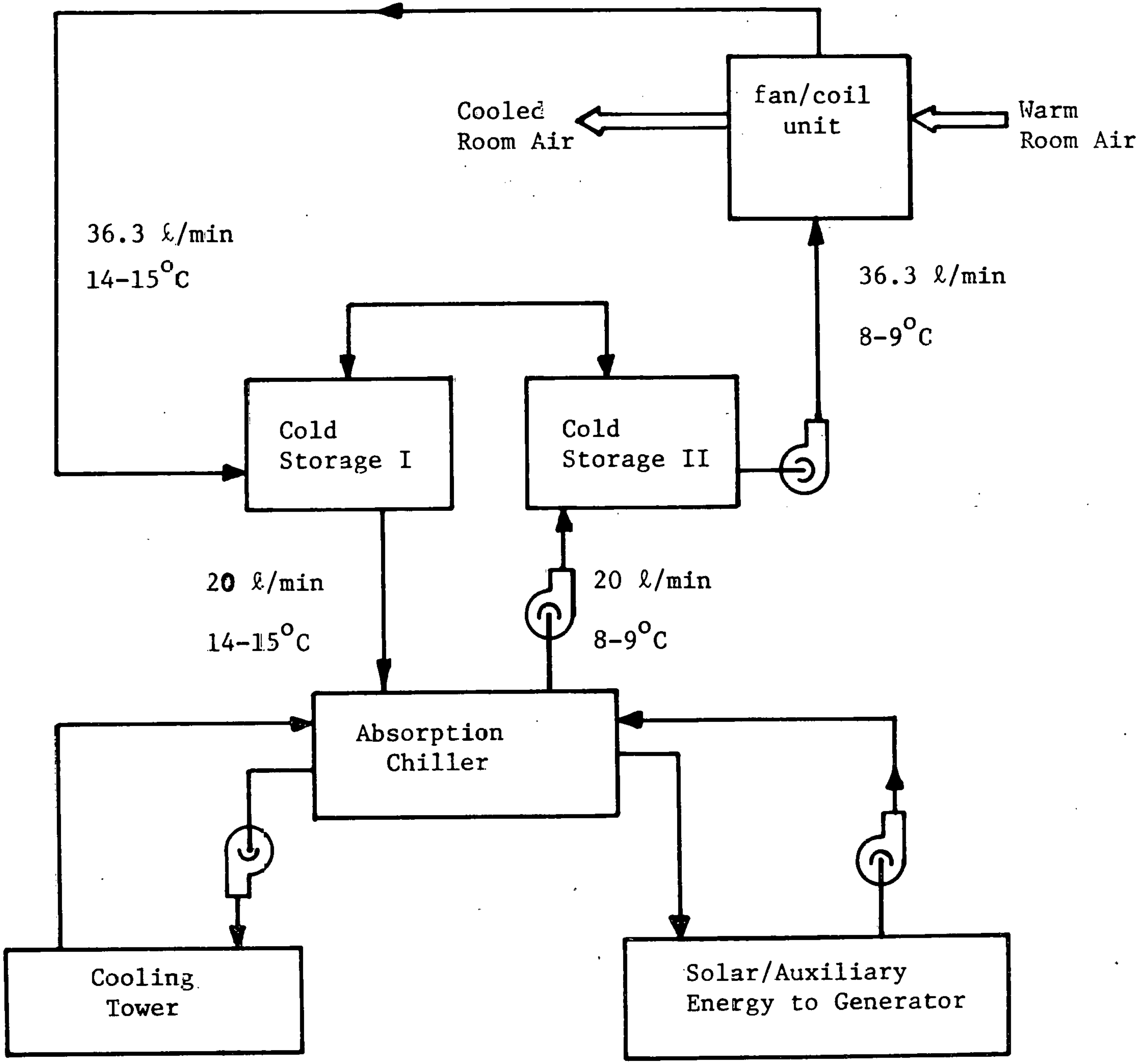


Fig. (8.3-3) Simplified Schematic Diagram of Modified Cooling System at CSU Solar House III[36]

D) Effect of Adding Cold Storage(s) on Unit's Cooling Performance

As noted in Section 7.1.6, significant improvement of the seasonal average COP of the cooling subsystem can be resulted by employing cold storage due to the elimination of short-term cycling of the absorption cooling unit during the period of low cooling demand.

In addition, the use of cold storage would allow the cooling capacity of the unit to be smaller without decreasing the performance of the subsystem to meet the same cooling load. The cooling unit does not have to be sized to meet the peak cooling load, but the daily steady state cooling capacity of the unit should be at least large enough to match the total daily cooling load.

Figure 8.3-4 is the theoretical curve of average COP of the cooling unit versus the cooling unit's operating period. Note that the significant reduction in COP results as the unit's operating period decreases. For example, when a cooling unit has the steady state or the maximum COP of 0.6 at its design conditions, then the average COP of the same unit is:

$\approx$ 0.45 if unit's operating period is 30 minutes

$\approx$ 0.52 if unit's operating period is 1 hour

$\approx$ 0.57 if unit's operating period is 2 hours, etc.

In Figure 8.3-5, plotted are the curves (I and II) of cold storage thermal capacity versus cooling unit's maximum cooling capacity, based on two different load characteristics as follows:

Curve I - CSU Solar House I (526 MJ/day)

Curve II- Hypothetical Commercial Building (526 MJ/day)

Although the total daily cooling load of the above two buildings are the same, significant difference exists between the curves as shown in Fig. 8.3-5.

Figure 8.3-5, also shows the same relationship with the unit's operating period as a parameter. For instance, if 25 MJ/hr cooling unit is used for Solar

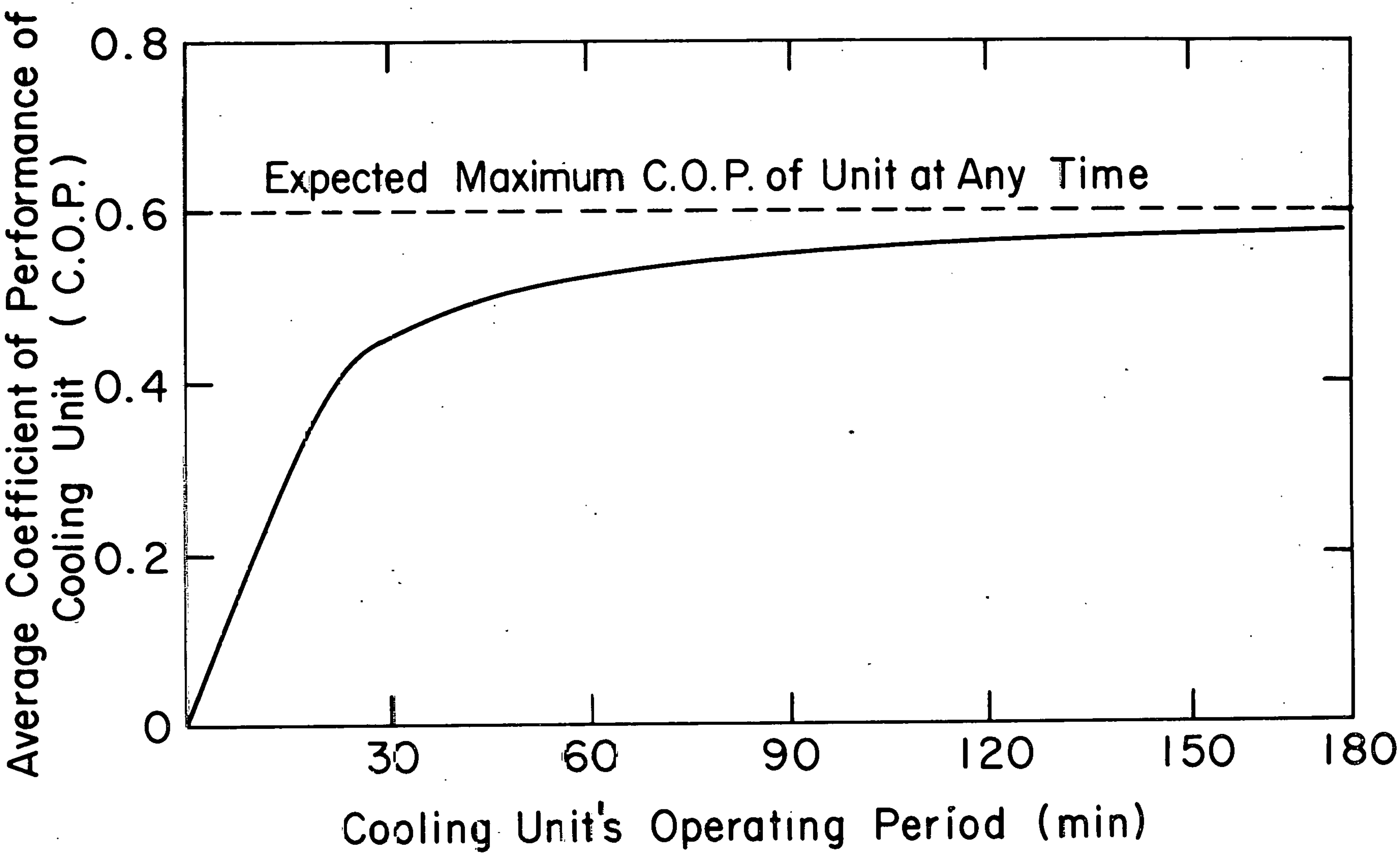


Fig. (8.3-4) Average COP of Cooling Unit versus Unit's Operating Period[36]

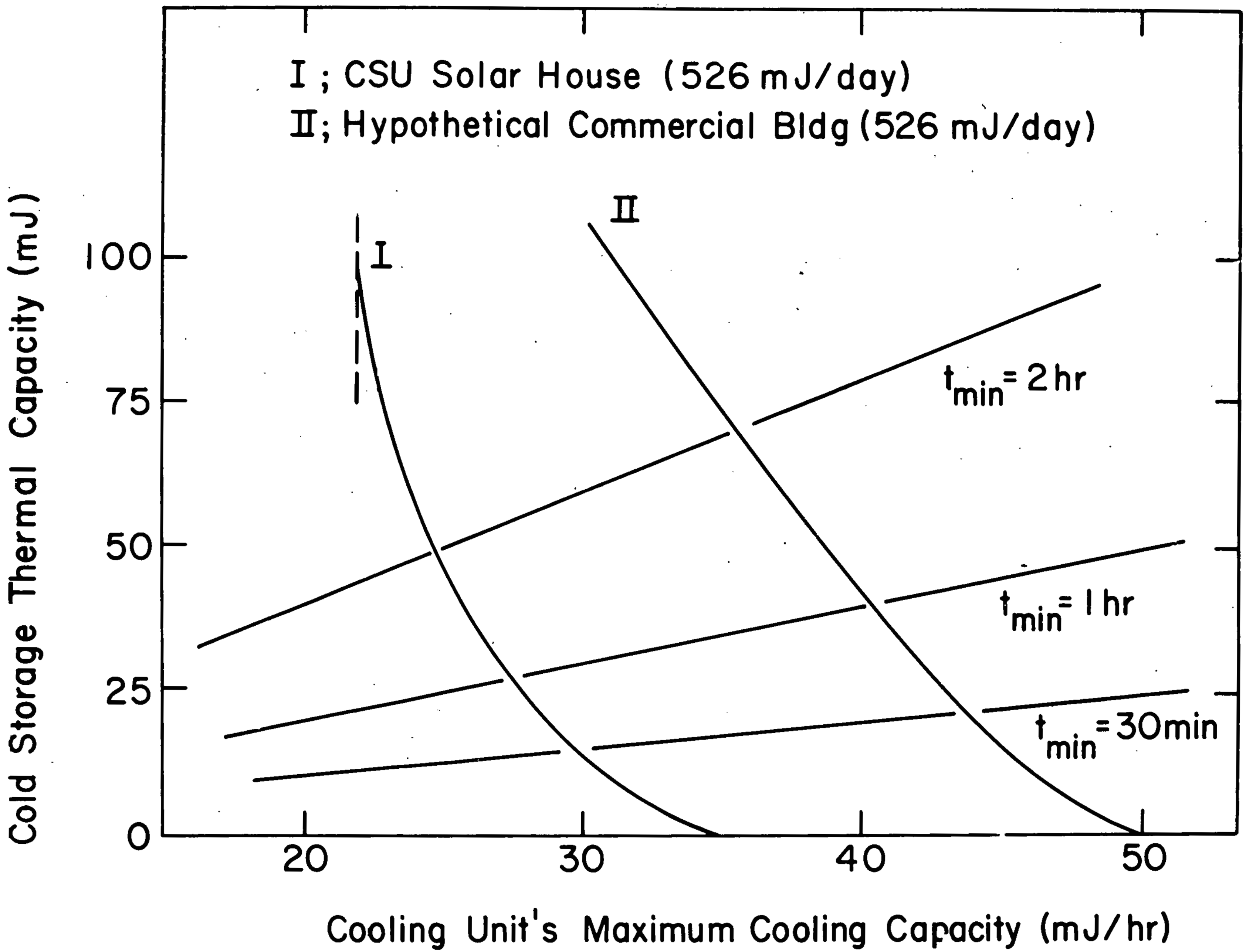


Fig. (8.3-5) Cooling Unit's Cycling Effects on Cold Storage Thermal Capacity and Unit's Maximum Cooling Capacity[36]

House III (or I), the thermal capacity of the cold storage would be 50 MJ and the minimum operating period of the unit would be 2 hours. This would provide cooling unit's COP in excess of 0.57 on a seasonal basis as readily seen in Fig. 8.3-4.

The following table summarizes the effect of the minimum operating period for CSU Solar House III. Larger cooling unit permits smaller cold storage. Minimum COP, however, decreases due to the decrease in minimum operating period.

TABLE

For CSU Solar House III

Min. operating period of Cooling Unit (hr)	Cold Storage Capacity (MJ)	Min. COP of Cooling Unit	Cooling Unit's Capacity (MJ/hr)
0.5	15	0.45	30
1.0	28	0.52	28
2.0	50	0.57	25

The actual storage capacity chosen for the Solar House III was 1900 ℓ each and the Yazaki chiller (rated cooling capacity of 25 MJ/hr), is expected to perform with the minimum COP of 0.57.

8.3.2 Method of Internal Energy Storage

Storage of solar energy has been usually shown as an external component to the absorption unit, i.e., a hot-side storage or a cold-side storage. Recently there have been several suggestions of storing solar energy within the absorption cycle[21,22,23].

Figure 8.3-6 shows the schematic diagram of a simple absorption cycle with three internal energy storages, i.e., liquid refrigerant storage, strong solution storage and weak solution storage.

During the period of insufficient supply of solar energy, required cooling load could be met by vaporizing the stored refrigerant in the evaporator. Refrigerant vapor from the evaporator is then absorbed into the stored weak solution to generate strong solution which can also be stored.

When the sufficient solar energy is available, the stored strong solution is fed to the generator and is heated to produce refrigerant vapor and the weak solution. The refrigerant vapor is liquefied in the condenser before storing.

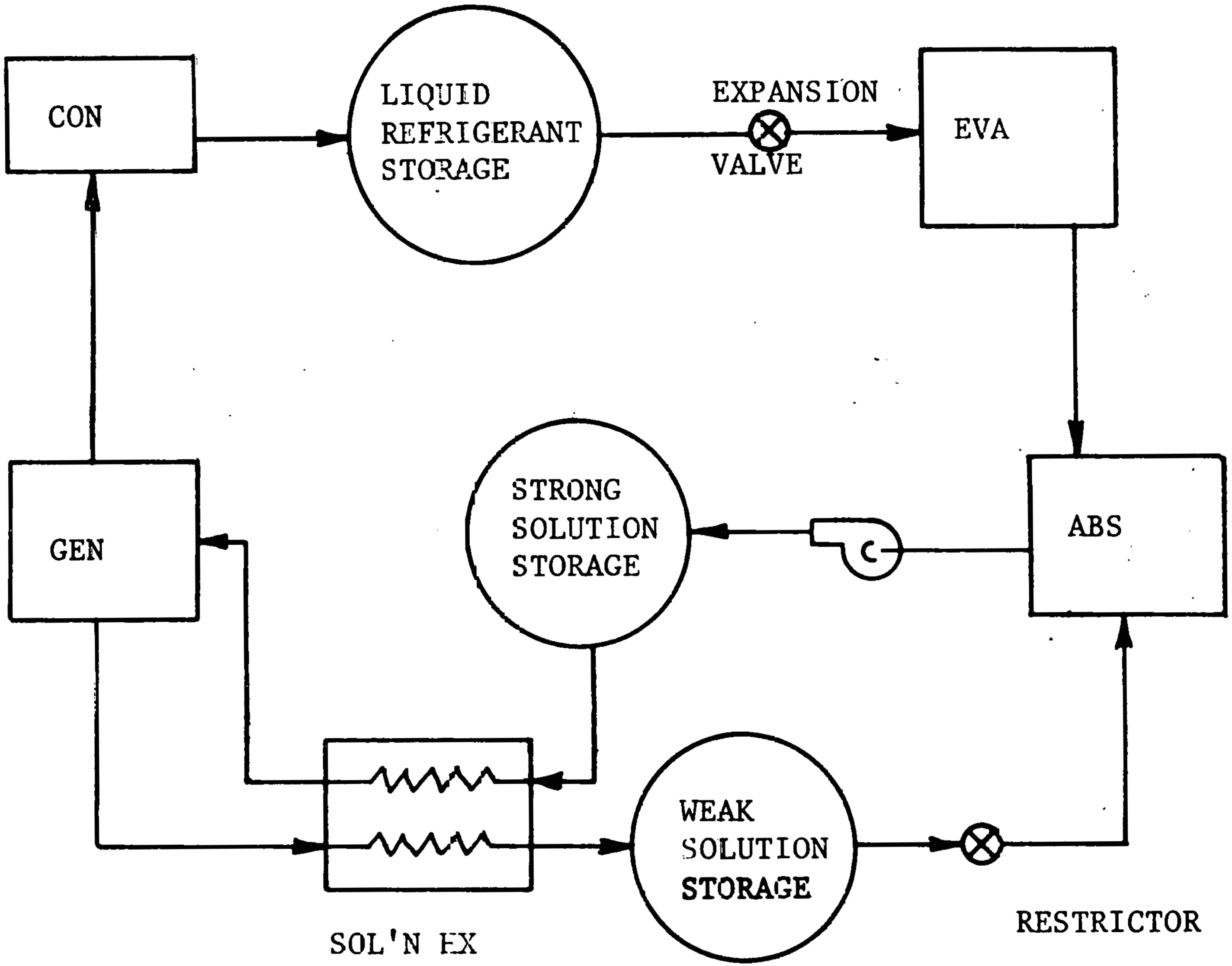


Fig. (8.3-6) Schematic Diagram of Simple Absorption Cycle with Internal Energy Storages[23]

Major advantages for adopting internal energy storages are as follows:

a) The cooling unit is more adaptive to solar energy supply, environmental conditions (heat sink temperature) and especially the cooling load. Thus, the system performance degradation due to cycling loss be significantly reduced or eliminated.

b) Component sizes can be made smaller since each component does not have to be designed at its maximum capacity. In addition, the storage volume is also smaller since the stored energy is of latent type compared to that of sensible type.

c) Thermal losses of the storages are small since they are kept at near ambient temperature.

d) An expensive auxiliary heat supply system can be minimized.

The disadvantages are:

a) It needs larger volume of working fluids as well as additional equipments; mainly storage tanks with insulations, thus significant increase in initial cost, especially for small capacity units, may result.

b) It requires complex system controls, which must be adaptive to the solar energy input, the environment, the building load requirements, etc.

It was said that the optimum tank volume ratios should generally be equal to 40:20:1 (strong solution : weak solution : refrigerant)[23]. The water/lithium bromide pair was reported to be very attractive due to the low storage volume and low storage pressure[21]. The ammonia/water pair is also attractive due to the low storage volume, but has some disadvantage of storing fluids at high pressure, thus increasing the cost of the storage tanks.

8.3.3 Insulation of Absorption Chiller

Other possible way of reducing heat loss due to frequent unit cycling is proper insulation of the chiller components, mainly, the distillation column (analyzer and generator). Study on this subject still needs to be performed and the results may turn out to be helpful.

9. Can solar absorption unit be used as a heat pump?

First of all, the answer to the above question is "yes"!

Efficient use of the solar collector all year around is the key to the success of the economically viable solar heating/cooling system, since the cost of the collector component covers the significant portion of the total system cost. Solar powered (& solar-assisted) absorption heat pump is, thus believed to be the best candidate since it offers excellent capabilities of both heating and cooling functions all year around.

9.1 Absorption Heat Pump Cycle

The standard absorption cycle, Fig. 1.1-1, is redrawn as a heat pump in Fig. 9.1-1. The principles of operation is the same as explained in Section 1.1.

The cycle can provide heating, cooling or simultaneous heating/cooling. When in the heating mode, heat dissipated from absorber-condenser components can be utilized while the outside air provides heat to the evaporator. Operating under cooling mode, heat from absorber-condenser components is dissipated to the outside air and the room air is cooled by providing its heat to the evaporator. When all three heat exchangers (absorber, condenser, and evaporator) are connected to the different zones of a building, simultaneous heating/cooling can be obtained, i.e., heat dissipated from absorber and/or condenser components is used for the zone to be heated. At the same time, evaporator can extract heat from the zone to be cooled.

Connection between the heat pump components and the building fan/coil unit for each mode of heat pump operation is illustrated in the table below. For example, in the heating mode of operation, evaporator component should be connected to the outdoor fan/coil unit and the absorber/condenser components, to the indoor fan/coil unit.

TABLE

Connection Between Heat Pump Components and Building Fan/Coil Unit

	Heating Mode			Cooling Mode			Heating/Cooling Mode		
	ABS	CON	EVA	ABS	CON	EVA	ABS	CON	EVA
Outdoor f/c Unit			X	X	X				
Indoor f/c Unit	X	X				X	X	X	X

Solar powered absorption heat pump could also be solar-assisted in the heating mode of operation. Improved heating COP of the solar-assisted unit could be expected due to the boost in heat source temperature by solar energy.

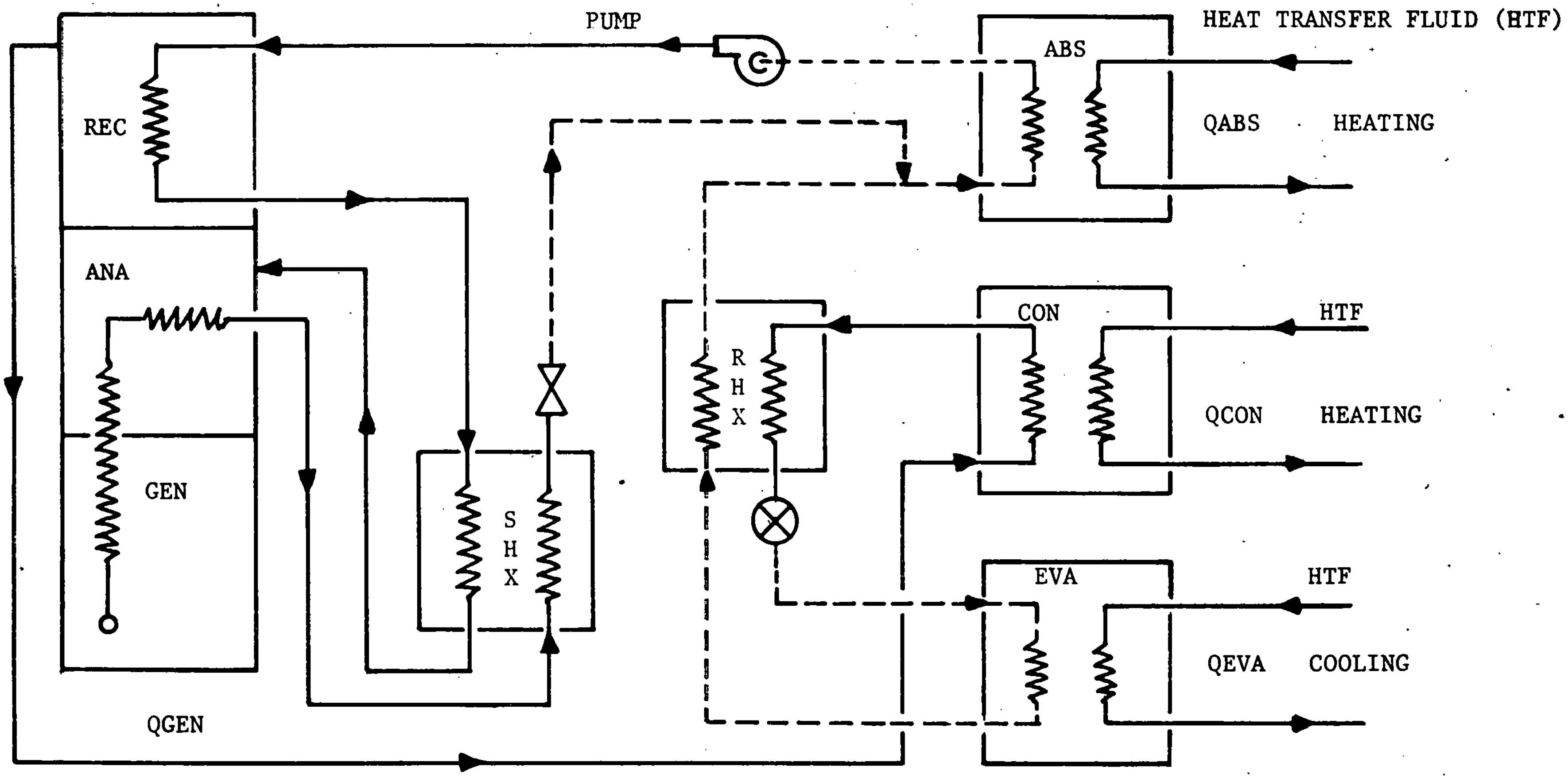


Fig. (9.1-1) Schematic Diagram of Standard NH_3-H_2O Cycle as a Heat Pump

9.2 Performance of Absorption Heat Pump

The cooling COP of the Carnot absorption heat pump cycle is the same as Eq. 3.1-1 in Section 3.1 or,

$$COP\ (Carnot),c = \left(\frac{Tgen - Tsink}{Tgen}\right)\left(\frac{Tevap}{Tsink - Tevap}\right) \qquad Eq. 3.1\text{-}1$$

The heating COP of Carnot absorption heat pump cycle can be expressed as follows:

$$COP\ (Carnot),_h = \left(\frac{Tsink}{Tgen}\right) + \left(\frac{Tgen - Tsink}{Tgen}\right)\left(\frac{Tsink}{Tsink - Tevap}\right) \qquad Eq.\ 9.2\text{-}1$$

In the above equation, the first term represents the contribution from the heat sink of the heat engine cycle and the second term is the contribution from the heat sink of the combined heat engine/mechanical refrigeration cycle.

Combining Eq. 3.1-1 and Eq. 9.2-1, the following equation results:

$$COP\ (Carnot),_h = 1.0 + COP\ (Carnot),_c \qquad Eq.\ 9.2\text{-}2$$

The above fact can also be illustrated for non-ideal cycles such as NH_3-H_2O or H_2O-LiBr cycles. Recalling,

$$COP,c = \frac{Qevap}{Qgen} \qquad Eq.\ 3.2\text{-}1$$

And, by definition, $COP,_h = \dfrac{Qcon + Qabs}{Qgen}$ Eq. 9.2-3

Since $Qcon \approx Qevap$ and $Qabs \approx Qgen$ (see table 1.1-2), Eq. 9.2-3 becomes,

$$COP,_h = \frac{Qevap + Qgen}{Qgen}$$

Combining the above equation with Eq. 3.2-1,

$$COP,_h \approx 1.0 + COP,_c \qquad Eq.\ 9.2\text{-}4$$

Since Eq. 9.2-4 is identical to Eq. 9.2-2, it could be applicable for both ideal and non-ideal cycles.

If cooling COP of about 0.7 is assumed for a typical absorption unit, the heating COP of about 1.7 could be expected for the same unit.

The ratio of heating to cooling capacity is estimated to be about 2.8 when the working fluid of NH_3-H_2O is employed to the cycle shown in Figure 9.1-1. In other words, the heating capacity of an absorption heat pump would be about 100,000 Btuh if the same unit has 3 ton cooling capacity.

9.3 Working Fluids for Absorption Heat Pump

Working fluids for absorption heat pump needs additional property requirements other than those for the conventional absorption air-conditioners. One of such requirements is the non-freezing property of the working refrigerant. Thus, the NH_3-H_2O pair may be a good candidate, while H_2O-LiBr may be not. However, the toxic nature of ammonia refrigerant does not allow direct heat exchange take place between the refrigerant and the room air, thus requiring an additional heat exchange loop in absorber, condenser, and evaporator component separately.

At the present time, no solar absorption heat pump is commercially available. Development of gas-fired absorption heat pumps, however, is in progress and such heat pumps may be available in the near future.

Several other refrigerant pairs for the absorption heat pump have been recently proposed. York Division of Borg-Warner Corporation suggested LZM system (Methanol in LiBr and ZnBr), and the new fluid pair suggested by Allied Chemical Corporation is Genetron 21 in ETFE (high boiling organic ether). The claimed benefits of adopting these new refrigerant pairs for the solar absorption heat pump are listed as follows:

a) Heat source temperature to the generator can be moderately low ($\leq 200^{o}F$) or can be matched to the flat plate solar collector temperature.

b) Direct expansion evaporator coil (D-X Coil) is a possibility due to non-toxic nature of the refrigerant.

c) Problem of either the freezing of the refrigerant nor the cyrstallization exists at the operating conditions.

d) Absorber-condenser components could be air-cooled,

e) Materials of construction could be inexpensive.

f) Predicted COP of the absorption heat pump using the proposed fluid pairs is at least comparable to that of H_2O-LiBr absorption system.

10. What are the present U. S. Government (ERDA) activities in developing the solar-powered absorption cooling technology?

Figure 10.0-1[24] shows the current SHACOB (Solar Heating and Cooling of Buildings) activities of the Division of Solar Energy in ERDA. The activities mainly consists of the demonstration program and the technology program.

10.1 Demonstration Program

As shown in Figure 10.0-1, this program has three elements; commercial demonstrations, development in support of demonstration, and residential demonstrations. Participants in each element are also indicated in the figure.

The milestone charts[24] of the 5-year demonstration program are shown in Figures 10.1-1 and 10.1-2. For example, as of November, 1976, Cycle 1 of the integrated system projects is in near completion and Cycle 2 has been initiated for HUD Residential Demonstration Program. For Commercial Demonstration Program, Cycle 1 has been completed and Cycle 2 is in much progress.

Table 10.1-1 indicates the number of projects in each application as shown through July, 1976. The early cycle demonstration projects were emphasized on the application of solar energy to space heating and domestic hot water. As the cycle progresses, more combined heating and cooling systems are expected. From Table 10.1-1, it is also seen that;

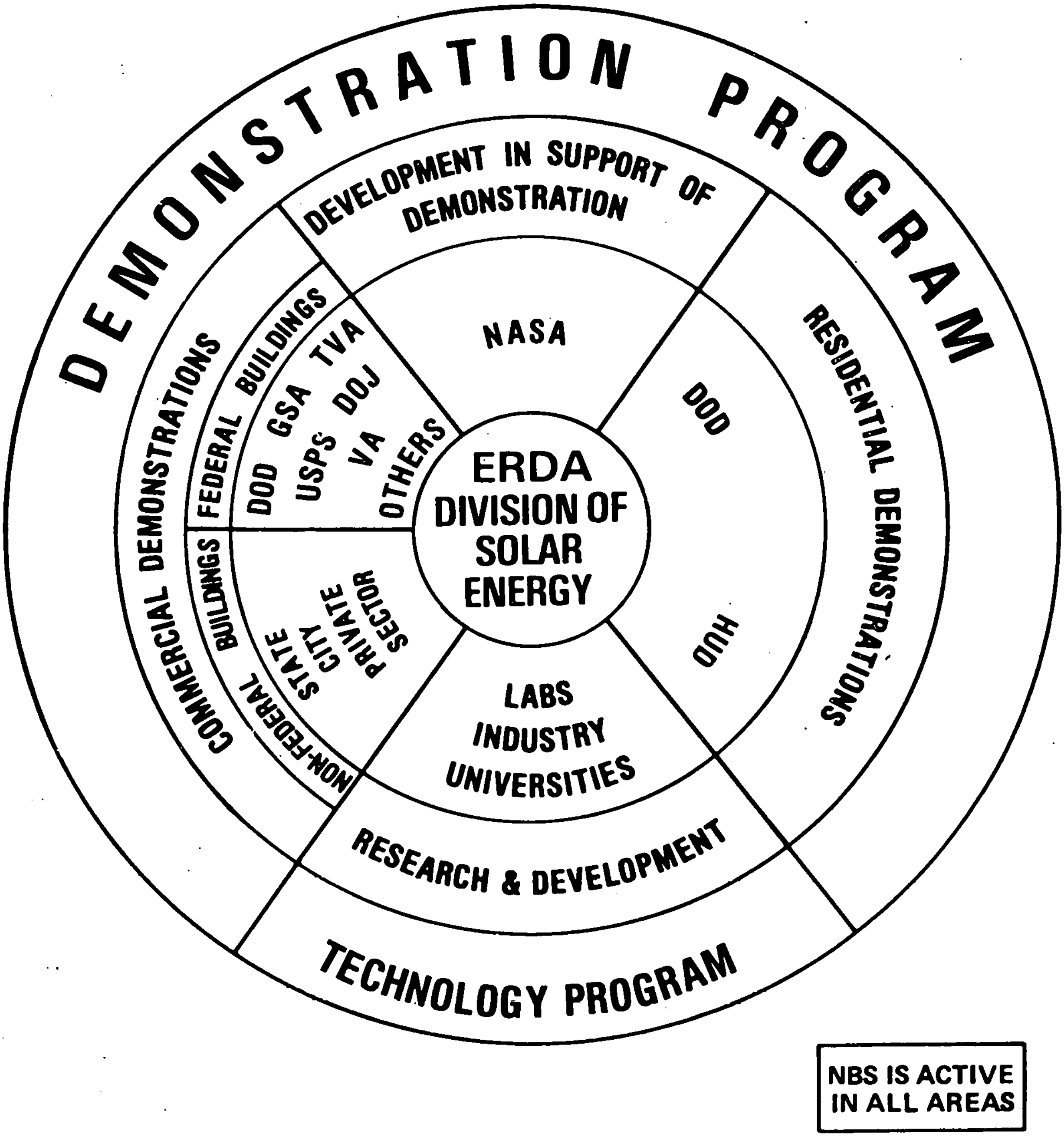
a) Most of the cooling systems employed solar-power absorption units.

b) Applications of the chiller units were dominant in commercial buildings.

Table 10.1-2 lists the Demonstration Projects that have been employed absorption units in their cooling systems. Arkla was the primary supplier of the absorption prototype units for the cooling demonstration in both residential and commercial sizes.

The majority of the demonstration projects (except residential/non federal buildings) are to be instrumented so that both the technical evaluation and the economic assessment of the solar energy systems could be made. The available information from these projects in near future would be very useful in implementing the current absorption technology further in solar applications. Data collection and evaluation of the early cycles are in progress.

Fig. (10.0-1) Program Participation in National Program for Solar Heating and Cooling of Buildings[24]



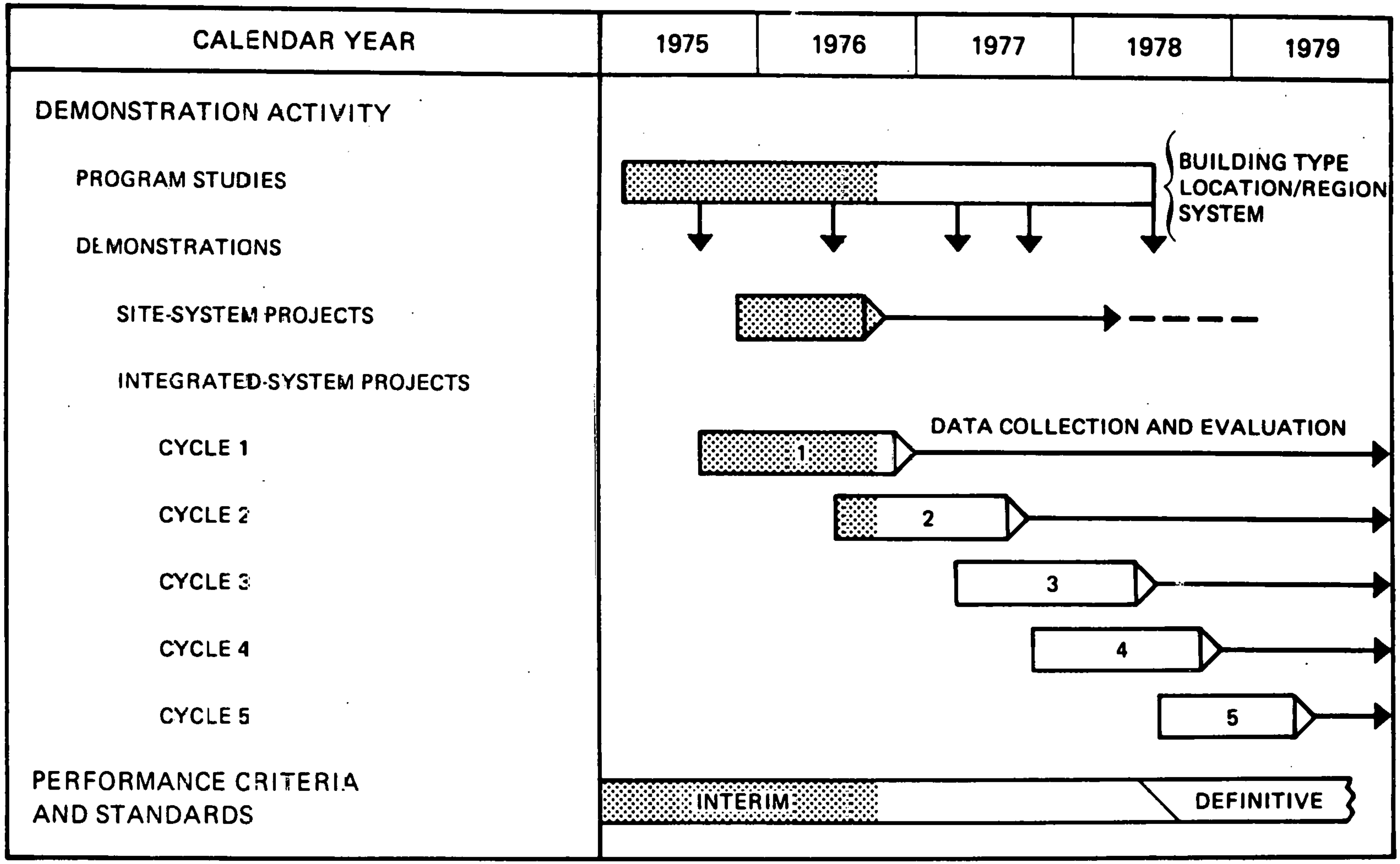


Fig. (10.1-1) HUD Residential Demonstration Program[24]

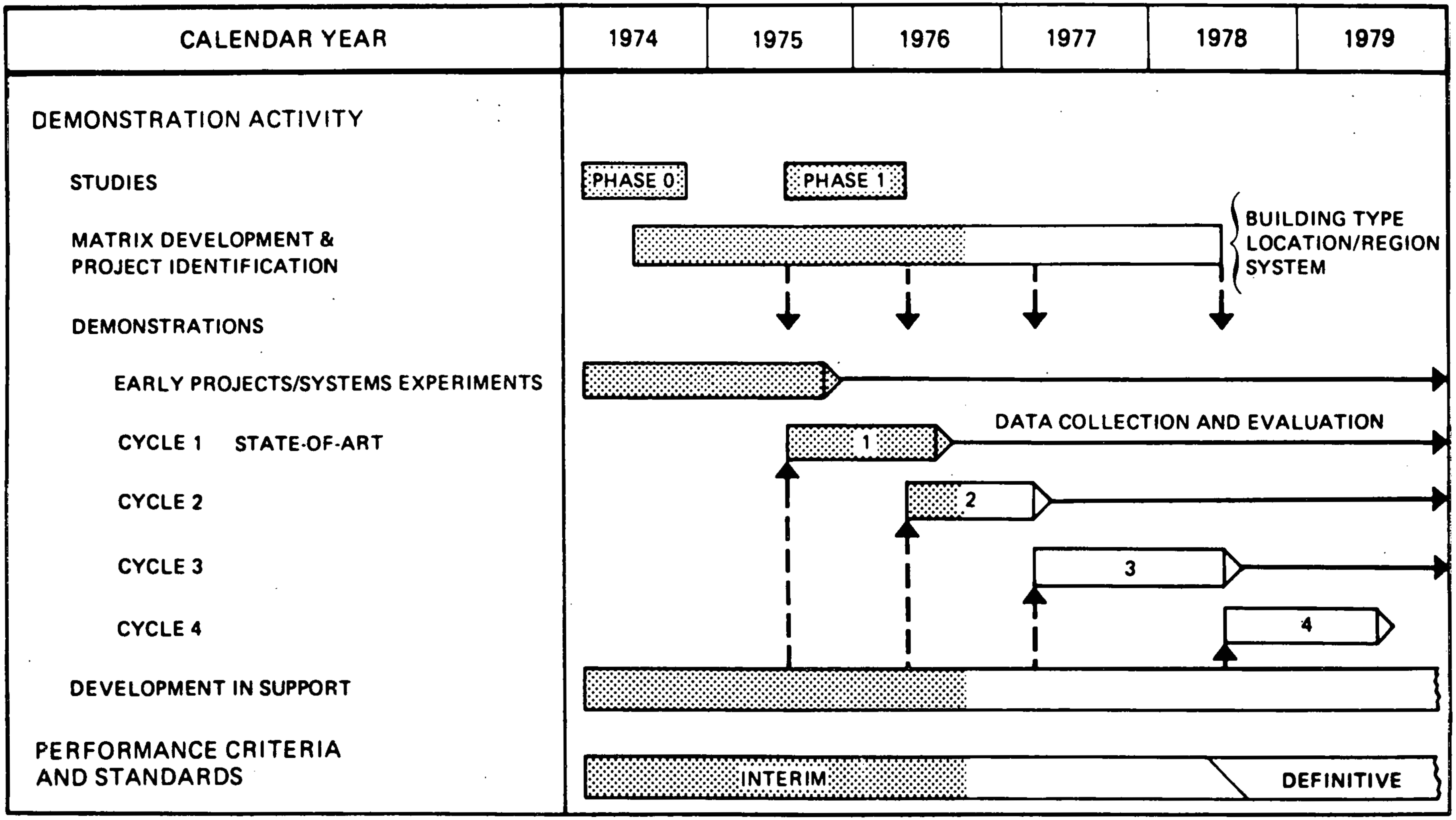


Fig. (10.1-2) Commercial Demonstration Program Plan of ERDA[24]

Table 10.1-1

Number of Solar Projects in Each Application in ERDA's Demonstration Program[25]

Application	Total No. of Projects	Number of Projects (as of July, 1976)						
		Cooling*					Heating	Hot Water
		ABS	EHP	SEHP	PASS	UNK		
Residential (Federal Bldg.)	15	-	-	-	-	-	15	15
Residential (Non-Fed. Bldg.)	46	3	2	3	4	-	39	42
Commercial (Federal Building)	9	5	-	-	0	3	9	8
Commercial (Non-Federal Bldg.)	55	18	-	-	1	1	42	40

* ABS = Absorption Chiller

EHP = Electric Heat Pump

SEHP = Solar-assisted Electric Heat Pump

PASS = Passive System

UNK = Unknown

Table 10.1-2

ERDA's Demonstration Projects Employing Absorption Cooling Chiller (as of July, 1976)[25]

Location	Owner/Builder	Building Type	Solar Application*
Montevallo, AL	Alabama Power Co.	Office	H, C, & HW
Irvine, CA	Irvine Unified School District	Elementary School	
Santa Clara, CA	City of Santa Clara	Community Center	H, C, & HW
Coral Gables, FL	Dade County	School	H, C, & HW
Nassau County, FL	Florida Dept. of General Services	Welcome Station	H & C
Atlanta, GA	City of Atlanta	School	H, C, & HW
Shenandoah, GA	Town of Shenandoah	Community Center	H, C, & HW
Cockeysville, MD	Baltimore County	School	H & C
Timonium, MD	Baltimore County	School	H & C
Minneapolis, MN	Univ. of Minnesota	Bookstore	H, C, & HW
Basking Ridge, NJ	Somerset County Park Commission	Auditorium/ Classroom	H, C, & HW
Lumberton, NJ	RKL Controls, Inc.	Manufacturing	H, C, & HW
Las Cruces, NM	New Mexico State University	Office/Laboratory	H, C, & HW
Canovanas, PR	Commonwealth of Puerto Rico	Industrial Factory	C & HW
Keystone, SD	South Dakota School of Mines	Visitor's Center	H & C
St. Thomas, VI	American Motor Inns, Inc.	Hotel	C
Richland, WA	Development Dynamics Co.	Office	H, C, & HW
Charles Town, WV	Jefferson County Schools	School	H & C
Boulder, CO	U.S. Postal Service	Post Office	H, C, & HW
Manchester, NH	Gen. Services Admin.	Office	H, C, & HW
Kirtland AFB, NM	Dept. of Defense	Exch. Shopping Ctr.	H, C, & HW
Ft. Hood, TX	Dept. of Defense	Office/Classroom	H & C
Randolph AFB, TX	Dept. of Defense	Exch. Shopping Ctr.	H, C, & HW
Denver, CO	Perl-Mack Enter.	Single Family Detached	H, C, & HW
Shenandoah, GA	Peachtree Homes Inc.	Single Family Detached	H, C, & HW
San Antonio, TX	San Antonio Ranch, Limited	Single Family Detached	H & C

* H = Heating
C = Cooling
HW = Hot Water

10.2 R & D Program

Solar heating and cooling R & D Branch in the Solar Energy Division of ERDA has developed vigorous national R & D program plan, which is well documented in References 26 and 27.

Summarized in Table 10.2-1 are the ERDA's current R & D projects in solar absorption air conditioning technology. Brief description and important publication(s) of each project are also included in the table.

Recently, the Solar R & D Branch placed an emphasis on the directed R & D activities as opposed to the unsolicited proposal mode of operations in the past. As a result, five PRDA's (Program Research and Development Announcements) and five RFP's (Requests For Proposals) have been issued for solar heating and cooling research and development.

Among the above PRDA's/RFP's, one RFP, which is closely related to the development of solar absorption cooling technology, is described below;

"Solar Activated Cooling Projects for Solar Heating and Cooling Applications" (RFP EG-77-R-03-1439, expected release in April 1977)

a) Analysis of Advanced Conceptual Designs for Single Family-Size Absorption Chillers

b) Development of Single Family Size Absorption Chillers for Use in a Solar Heating and Cooling System

c) Development of High Temperature Solar Powered Chillers for Use in a Solar Heating and Cooling System

d) Development of Solar Desiccant Dehumidifier for Use in a Solar Heating and Cooling System

Table 10.2-1

Solar Heating and Cooling R&D Projects of ERDA in Solar Absorption Cooling Technology (FY76 & 77)[27]

Contractor	Title	Description	Publications
Richard Merrick ARKLA Industries, Inc	Engineering, Design, Construction and Testing of a Salt Water Absorption Cooling Unit Optimized for Use with a Solar Collector Heat Source, Phase II	Development of a 3-ton, Evaporatively cooled H_2O-LiBr chiller suitable for low temperature solar applications.	- Engineering, Design, Construction and Testing of a Salt Water Absorption Cooling Unit Optimized for use with a Solar Collector Heat Source, Final Report, Report No. COO/4095-7612.
Redfield Allen Maryland, University of	Performance Studies of Solar Absorption Air Conditioning Systems, Phase II	Model simulation of absorption cycle and system performance	- Optimization Studies of Solar Absorption Air Conditioning Systems, Final Progress Report, Report No. NSF/RANN/SE/GI-39117/PR/76/2. - Reference 29
Henry Curran Hittman Associates, Inc.	Assessment of Solar Powered Cooling of Buildings	Analysis of Solar Absorption Cooling	- Assessment of Solar-Powered Cooling of Buildings, Final Report, Report No. NSF/RANN/75-012.
Michael Wahlig Lawrence Berkeley Lab.	Solar Heating and Cooling of Buildings	Development of a 3-ton, Air-Cooled NH_3-H_2O Absorption Air Conditioner for Low Temperature Solar Applications	- References 13 and 14
T. S. Zawacki Institute of Gas Tech.	Development of New Fluids for Solar Absorption Cooling	Identification of fluid systems (Exhibiting positive deviations from Raoult's Law) most suitable for solar powered absorption chillers	
Francis DeWinter Altas Corporation	Workshop on the Use of Solar Energy for Cooling Buildings	Solar Cooling Workshop (Includes Absorption Cooling Technology)	- The Use of Solar Energy for the Cooling of Buildings, Final Report, SAN/1122-76/2

The objectives of Tasks (a) and (b) of the above RFP are repeated here since they are clearly associated with the absorption cooling technology.

The objective of Task (a) is "to develop and analyze new refrigerant pairs, or additives to presently used absorbent/refrigerant pairs, or other cycle modifications that can potentially improve the performance or reduce the cost of an absorption chiller in the size range of 3 tons. The absorption chiller using new absorbent/refrigerant pairs, or additives, or cycle modifications would be expected to operate efficiently with fluid supply temperatures of 160° to 230° providing 45° chilled water at full load with ambient conditions of 95°F DB and 75°F WB[28]".

The objective of Task (b) is "to procure the design, fabrication, test and analysis of a single-family-size residential solar-powered absorption chiller having improved performance and reduced cost. This chiller shall be designed for efficient operation with fluid supply temperatures of 160° to 230°. At full cooling load it must be able to provide 45° chilled water with an ambient conditions of 95°F DB and 75°F WB in accordance with applicable ARI Standards[28]".

Task (a) is the software development of the advanced absorption technology, and Task (b) covers the hardware development. Both Tasks (a) and (b) emphasize the development of the absorption cooling units for residential applications.

In addition to the above mentioned efforts, the Solar R & D Branch is also planning accelerated cooling program in support of the Demonstration Program (see Section 10.1). The prime objective of the accelerated cooling program is to expeditiously develop solar cooling systems to be used for Cycle 5 of the Demonstration Program within the allowable time frame.

Three elements of the proposed accelerated cooling program are the developments of absorption chiller systems, a heat engine/vapor compression chiller system and a heat pump system. Arkla is expected to carry out the developments of 3 ton and 25 ton absorption chiller systems. An evaporatively cooled absorption chiller which is currently developed, would be used in the 3 ton chiller system, and the 25 ton chiller system would employ Solaire 300 chiller with a cooling tower.

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