

PERFORMANCE ANALYSIS OF STAGED ABSORPTION HEAT PUMPS: WATER-LITHIUM BROMIDE SYSTEMS

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ABSTRACT

Absorption heat pump studies conducted in recent years have made increasing use of different simulation techniques for predicting performance of advanced cycles. In complex, multi-stage systems, the calculations are quite lengthy, and in the published literature, a cycle performance is often characterized by no more than a single point. This study has employed a modular computer code designed to simulate absorption systems in a flexible form, making it possible to predict their performance in varying cycle configurations with different working fluids. The code has been employed to investigate the performance of various systems using the water-lithium bromide fluid pair. Complete performance maps under varying operating conditions have been generated for systems in single-stage and several double-stage configurations. The performance has been compared over the entire temperature domain applicable to the cycle. The thermodynamic potential of water-LiBr has been studied and compared to that of other absorption fluid pairs.

INTRODUCTION

Absorption heat pump studies conducted in recent years have made increasing use of different simulation techniques for predicting performance (Vliet et al. 1982; Grossman and Childs 1983; McLinden and Klein 1985; Perez-Blanco and Patterson 1986). The need for reliable simulation has become apparent with the new developments in advanced cycles and working fluid combinations during the past decade. Most of the simulation programs described in the literature have been system-specific, that is, written to simulate one particular system with a particular flow arrangement and working fluid. Their structure did not allow easy modification to model other systems on the same basis. In complex, multi-stage systems, the calculations can be quite lengthy; it is, therefore, not surprising that only a few comparative studies of advanced cycles have been carried out (Biermann 1984; Phillips 1984). In the published literature describing those studies, a cycle performance is often characterized by no more than a single point.

In order to facilitate the investigation and comparison of absorption systems in various cycle configurations, a computer code has been developed (Grossman

and Michelson 1985; Grossman et al. 1987) employing a flexible and modular simulation technique. The code is based on unit subroutines containing the governing equations for the system's components. Those are linked together by a main program according to the user's specifications to form the complete system. The equations are solved simultaneously, and fluid properties are taken from a property data base. The computer code has been described in detail in Grossman and Michelson (1985) and Grossman et al. (1987). A brief review of its main features is given in the following section.

Under the present study, the modular simulation code was used to investigate the performance of various systems employing the water-lithium bromide fluid pair. The approach has been to generate complete performance maps under varying operating conditions. Thus, the performance of systems in single-stage and several double-stage configurations could be compared not only at a single point but over the entire temperature domain applicable to the cycle. The results of these comparisons are discussed below. Another goal of the study was to evaluate the thermodynamic potential of water-lithium bromide as a working fluid for absorption systems in comparison with other absorption fluid pairs.

THE COMPUTER SIMULATION CODE

The objective of developing a flexible simulation code has led to the modular program structure, enabling the user to specify different cycle configurations, different working fluids, and, of course, different unit sizes and operating conditions. Recognizing that each absorption system consists of a number of standard components (e.g., absorber, evaporator, desorber, condenser, etc.), the following logic was developed for the program: Each basic component is simulated by a unit subroutine providing a mathematical expression of the physics of that component. Such a unit subroutine contains all the physical equations required to fully describe the unit's behavior, such as energy balance, conservation of mass for each material species, heat and mass transfer and thermodynamic equilibrium. The unit subroutine, when invoked, calls upon a property data base for the thermodynamic properties of the working fluids. This separation between the unit subroutines and property subroutines provides the flexibility for each unit to operate with differ-

ent working fluids. The main program calls the unit subroutines and links them together in a form corresponding to the user's specification of the system about to be simulated. Each call to a unit subroutine is equivalent to collecting all the equations associated with it, without attempting to solve them as yet. When the calls to all the unit subroutines have been completed and all the equations have been established, a mathematical solver routine is employed to solve the set of nonlinear equations simultaneously.

The recent version of the code, used in the present study, has been described in detail in Grossman et al. (1987). A brief description of the program structure and operation is given here. The code comprises four principal modules: a main program, a unit subroutine package, a property data base, and a solver package. The central module is the main program, which receives the user input, invokes the unit subroutines, activates the solver package, and produces the output. Properties of the working fluids, as required in the various equations, are retrieved by the unit subroutines from the property data base.

In preparing the input, the user must first prepare a flowsheet of the cycle, indicating all the units and all the state points. He then conveys to the code an "image" of the cycle to be simulated: the number and types of units contained in it, their interconnections and size or transfer characteristics, where applicable. The input must also contain the values of the parameters set fixed by the user, e.g., temperatures, flow rates, etc., at specific state points. The main program interprets the user input and creates a variable vector containing all the independent unknown quantities. It then calls the unit subroutines, thereby establishing the system's governing equations. All the equations and variables are normalized, to be on the same order of magnitude. The solver package is then activated, and the values of the unknowns calculated to a user-specified accuracy. Recognizing that the set of nonlinear equations may have more than one mathematical solution, a set of constraints is applied to direct the solver toward the physically valid solution. The output contains the temperature, enthalpy, flow rate, concentration, pressure, and vapor fraction at each of the cycle's state points, as well as the heat duty and transfer characteristics of each unit.

The code has been used successfully to simulate a variety of single-stage, double-stage, and dual-loop absorption systems operating in a heat pump or in a heat transformer mode, with the working fluids water-lithium bromide, ammonia-water, ammonia-water-lithium bromide, water-lithium chloride, methanol-lithium bromide-zinc bromide, and more. The model has been validated against experimental data (Grossman and Michelson 1985) and was found to be a very useful tool for evaluating new cycles and working fluids, investigating a system's behavior in off-design conditions, analyzing experimental data, and performing preliminary design optimization.

In the present study, the code was used to investigate the performance of several systems employing the water-lithium bromide fluid pair. Water-lithium bromide is

known as an effective absorption working fluid, due to favorable characteristics such as a large negative deviation from Raoult's law, a high latent heat refrigerant, and a nonvolatile absorbent. It is one of the few fluids actually employed in commercial applications and a great deal of practical experience has been gained in its use. While most of the current applications are based on the single-stage cycle, the last few years have seen increasing interest in the higher-COP double-effect systems, where several cycle configurations are possible. A systematic simulation and optimization study is in order.

SINGLE-STAGE SYSTEMS

Single-stage systems were analyzed first, as a basis for later comparison with more complex ones. The water-lithium bromide single-stage cycle has been studied extensively by equipment manufacturers; several studies have been published in the open literature, particularly on solar-powered systems (Wilbur and Mancini 1976; Grossman et al. 1979). The cycle's response to changes in operating temperatures is well known. Less well recognized are the effects of variations in internal flow rates and heat transfer area. Some of the single-stage cycle information generated by the present study may seem repetitious; however, it was thought best to conduct the simulation in a systematic manner in order to compare the results later with those of more elaborate systems on the same basis.

Figure 1 describes schematically a single-stage absorption chiller designed to operate with a nonvolatile absorbent such as water-lithium bromide.* The system has six sub-units (indicated by the circled numbers) and twenty state points (indicated by the uncircled numbers). Its performance under a given set of operating conditions depends, of course, on the design characteristics and particularly on the size of the heat transfer surfaces in its exchange units—the evaporator, absorber, condenser, and desorber. As a reference case for the simulation, a practical system was considered with economically reasonable, if not optimized, heat transfer areas. We selected a solar-powered lithium bromide-water chiller (Biermann 1978) that has been tested extensively. The main characteristics of this machine are listed in Table 1, including the externally imposed flow rates of hot, cooling, and chilled water; the absorbent circulation rate; the *UAs* (overall heat transfer coefficient times area), which characterize the heat transfer performance of the exchange units; and design point temperatures of the external fluids. With these values as input, the simulation code calculates the internal temperatures, flow rates, concentrations, and other operating parameters at all the system's state points, from which overall performance parameters can be derived. Calculated performance parameters are also given in Table 1 and are consistent with measured values (Biermann 1978).

The set of operating parameters in Table 1 has been selected as the design point. In conducting the simulation

*It is assumed that the reader is familiar with the operation of absorption systems. The single-stage cycle is described in detail in McLinden and Klein (1985), Biermann (1984), Wilbur and Mancini (1976), and Grossman et al. (1979).

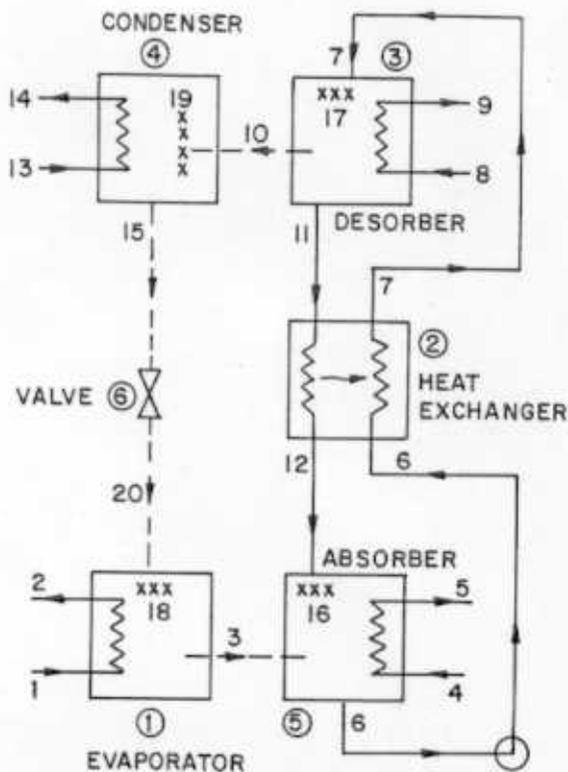


Figure 1 Schematic description of single-stage absorption chiller. Solid lines describe solution flow; broken lines describe refrigerant flow.

to generate the operating map for the system, one parameter was varied at the time while all others were kept constant. It was assumed that the values of UA for the exchange units remain constant while the temperatures and other parameters change. This is not strictly accurate; although the heat transfer areas (A) remain constant, the heat transfer coefficients (U) vary somewhat with the temperatures as well as with the loading conditions. However, this variation is relatively weak in most cases and the assumption of constant UA is a reasonably good approximation. Better fundamental understanding of the heat and mass transfer process in absorption and desorption would allow for taking the variation of UA with temperature into consideration.

Figure 2 describes the variation of COP and capacity of the system, with the temperature of the hot water input to the desorber, for several values of the cooling water inlet temperature. The chilled water outlet temperature is fixed at 45°F; all the flow rates and UA s are fixed at their design values. The capacity is presented in dimensionless form, normalized with respect to the design point value (41.8 kW). The design point is indicated in the diagram by a black dot. The Carnot COP corresponding to each set of temperature values is given by (Grossman 1981):

$$\text{Carnot COP} = \frac{T_L(T_H - T_C)}{T_H(T_C - T_L)} \quad (1)$$

where T_H , T_C , and T_L are the temperatures of the hot, cooling, and chilled water, respectively.

TABLE 1
Characteristic Parameters at Design Point for Single-Stage LiBr-H₂O Absorption Chiller

Heat Transfer Characteristics (UA):	
Absorber:	193.0 Btu/min · °F
Desorber:	268.0 Btu/min · °F
Condenser:	565.0 Btu/min · °F
Evaporator:	377.0 Btu/min · °F
Recuperative Heat Exchanger:	64.0 Btu/min · °F
Mass Flow rate:	
Absorber (cooling water)	483.0 lbs/min
Desorber (hot water)	416.0 lbs/min
Condenser (cooling water)	391.0 lbs/min
Evaporator (chilled water)	300.0 lbs/min
Weak Solution	60.0 lbs/min
Temperatures:	
Hot water inlet (state point 8):	180°F
Cooling water inlet (s.p. 4 and 13):	85°F
Chilled water outlet (s.p. 2):	45°F
Calculated Parameters:	
Cooling capacity:	41.8 kW
Coefficient of performance (COP):	0.7148

It is evident from Figure 2 that the COP increases with the hot water temperature and decreases with the cooling water temperature, as may be expected from the increase in the Carnot COP under the same temperature variations. However, the absorption system COP is always below that of Carnot and does not exactly follow the Carnot trend. As T_H increases, the absorption COP tends to level off, rather than continue to increase, and with a further increase in T_H , even drops somewhat. Also, for

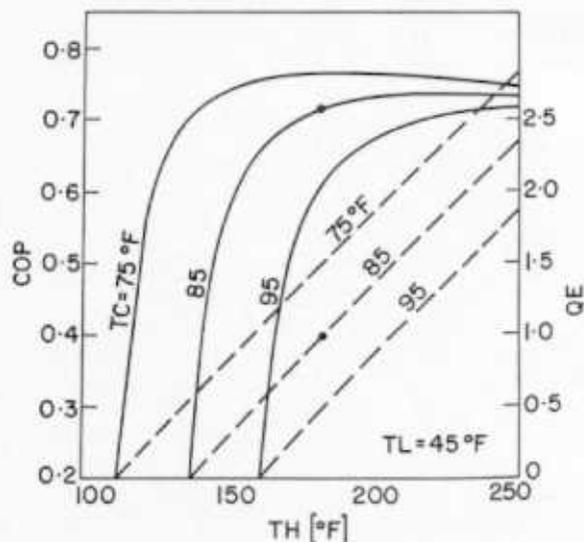


Figure 2 Coefficient of performance (COP, solid lines) and normalized cooling capacity (QE, broken lines) for single-stage system as functions of the desorber hot water inlet temperature (T_H) for different values of absorber and condenser cooling water inlet temperature (T_C). All other parameters are fixed at the design value (Table 1).

each value of the cooling water temperature, there is a minimum value of T_H where the COP vanishes and below which the absorption system does not perform at all. The irreversibilities in an absorption system that make its COP lower than Carnot are due to three main factors: imperfect heat and mass transfer in the system's units, mixing losses, and circulation losses. Mixing losses are associated with the evaporation of the refrigerant in the desorber from a concentrated solution, which requires a greater amount of heat than that required to evaporate it in a pure state. Also, the refrigerant vapor leaves the desorber superheated, as the temperature required for desorption is higher than the evaporation temperature of the pure refrigerant under the same pressure. The superheat energy spent in the generator constitutes a thermodynamic loss; it leads to extra cooling requirement in the condenser. The mixing losses increase with the absorbent concentration. Circulation losses are associated with the imperfect heat transfer in the recuperative heat exchanger, which requires preheating of the solution stream entering the desorber and precooling of the one entering the absorber. The energy associated with this is, again, a thermodynamic loss. Circulation losses increase with the solution flow rate.

When operating the absorption system at a low hot water temperature, circulation losses dominate. It is easy to see from an equilibrium diagram of the working fluid pair that the low (weak) solution concentration is determined by T_L and T_C whereas the high (strong) concentration is determined by T_H and T_C . At low T_H , the two concentrations are close to each other and the amount of refrigerant circulating in the system, which is proportional to the difference in concentrations, is very low. Thus, the system does little useful cooling; in the limit when the two concentrations approach each other, the COP and the capacity both vanish. This determines the minimum operating temperature. As the hot water temperature increases, so does the difference in concentrations, the amount of refrigerant, and hence the capacity. The relative importance of circulation losses diminishes. When operating at full design capacity, most of the heat supplied to the desorber goes to vaporize the refrigerant. Since this heat is fairly insensitive to the desorber temperature and is proportional to the amount of refrigerant evaporated, the ratio of evaporator to desorber heat tends to be constant. This explains the "leveling off" of the COP. Further increase in T_H will lead to greater mixing losses and a decline in COP.

Additional simulation runs have shown a qualitatively similar behavior of the COP and capacity when keeping the cooling water temperature constant and changing the chilled water temperature. The higher the T_L , the higher the COP.

Figure 3 describes the COP and capacity as functions of the weak solution flow rate (state points 6 and 7) for different flow rates of the chilled water in the evaporator (state points 1 and 2). Both flow rates are presented in dimensionless form, normalized with respect to their design values (60.0 and 300.0 lbs/min, respectively). Increasing the chilled water flow rate while keeping all other parameters constant lowers the evaporator temperature,

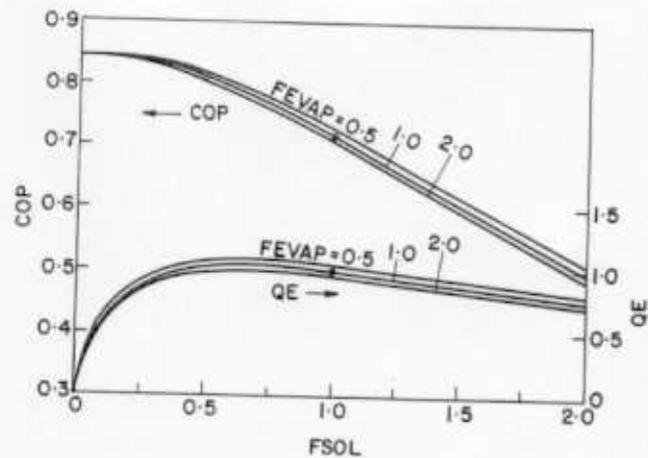


Figure 3 Coefficient of performance (COP) and normalized cooling capacity (QE) for single-stage system as functions of the normalized flow rates of chilled water (FEVAP) and weak solution (FSOL). All other parameters are fixed at the design value (Table 1).

which lowers the capacity and COP, as explained earlier regarding T_L . The dependence is, however, relatively weak, as can be seen from Figure 3. Reducing the solution flow rate, with other parameters constant, reduces the amount of refrigerant circulating through the system and hence the capacity. Increasing the solution flow rate, therefore, leads to an increase in capacity up to a certain point; beyond that, the associated increase in circulation losses, manifested in the steady decline of the COP, causes a reduction in capacity as well.

Figure 4 describes the COP and capacity as functions of the normalized solution flow rate, for the entire spectrum of UA values for the recuperative heat ex-

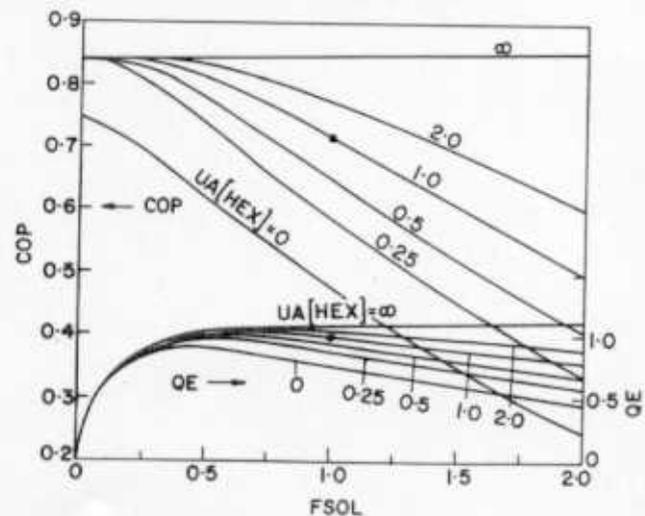


Figure 4 Coefficient of performance (COP) and normalized cooling capacity (QE) for single-stage system as functions of the normalized flow rate of the weak solution (FSOL) for different values of the normalized UA of the recuperative heat exchanger. All other parameters are fixed at the design value (Table 1).

changer. The UA is normalized with respect to its design value (64.0 Btu/min · °F) and varies from zero to infinity. It is evident that increasing the recuperation effectiveness leads to reduced circulation losses and an increase in both COP and capacity. Additional simulation runs have shown COP and capacity improvements due to an increase in UA of other units; however, only the recuperative heat exchanger has a direct effect on circulation losses.

Increasing the heat transfer areas in all of the system's units to infinity, while economically impractical, makes it possible to investigate the thermodynamic potential of the working fluid pair. The flow rates no longer matter, since with infinite UA s the temperatures at all the internal state points equal those of the external inputs, and circulation losses are reduced to zero. All the irreversibilities in the system are associated with mixing. Figure 5 describes this condition of perfect heat transfer, showing the COP of the absorption system and the corresponding Carnot COP as functions of the operating temperatures. Here, as before, there exists a minimum hot water temperature below which the system does not work. Slightly above this minimum, the system's COP approaches that of Carnot. Further increase in the temperature leads to a decrease in COP, due to increasing mixing losses. A comparison of Figures 5 and 2 shows that under the design temperatures, the water-lithium bromide fluid pair can theoretically yield a COP of about 0.85, which is quite close to the Carnot limit; in a practical machine, however, the COP is typically only about 0.7.

In order to compare the performance of water-lithium bromide with other absorption fluids of a similar type, another working pair with a nonvolatile absorbent, methanol-zinc bromide-lithium bromide, was selected. This material has been used in experimental absorption systems, and its properties are included in the simulation code data base (Grossman et al. 1987). A comparison with the more common ammonia-water would be unfair to the latter, since it contains a volatile absorbent that

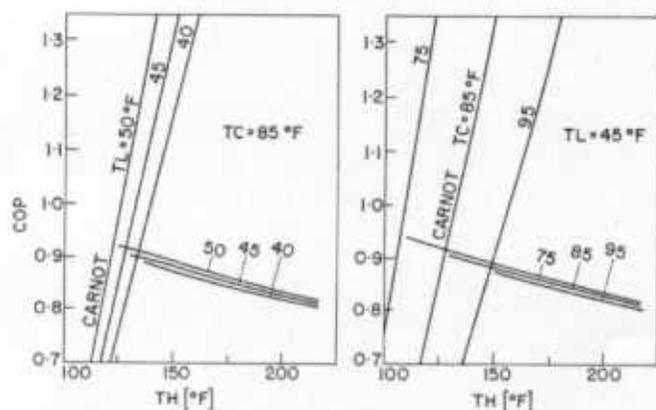


Figure 5 Coefficient of performance (COP) for single-stage system LiBr-H₂O with perfect heat transfer in all units and corresponding Carnot COP, as functions of hot water temperature: (a) Different chilled water temperatures with cooling water fixed at design value; (b) Different cooling water temperatures with chilled water fixed at design value.

requires a different system design. Ammonia-water has many advantages as a working fluid for absorption systems but under the same operating temperatures has a COP consistently lower than lithium bromide-water, due in part to rectification losses. To make the comparison, a system with CH₃OH-ZnBr₂-LiBr was simulated, assuming the design characteristics of the H₂O-LiBr chiller as in Table 1. Figure 6 describes the COP of this system as a function of the hot water inlet temperature, for different cooling water temperatures, with the chilled water outlet temperature fixed at the design value. The COP is compared to that of the H₂O-LiBr system from Figure 2. The pair, methanol-zinc bromide-lithium bromide, shows the same qualitative behavior as water-lithium bromide; quantitatively, it gives an inferior COP over the temperature range desirable for operation, including a sharper drop with T_H beyond the maximum value, which indicates greater mixing losses associated with CH₃OH-ZnBr₂-LiBr. For the low range of values of T_H , where the COP with both materials is low, CH₃OH-ZnBr₂-LiBr gives a slightly better COP than H₂O-LiBr, which indicates lower circulation losses.

DOUBLE-STAGE SYSTEMS

With the growing interest in advanced absorption cycles during the past few years, various methods have been proposed for staged operation (Biermann 1984; Phillips 1984). The most common double-stage systems are based on the so-called double-effect cycle. The primary purpose of using those systems is to obtain higher COP values than possible with a single-stage system,

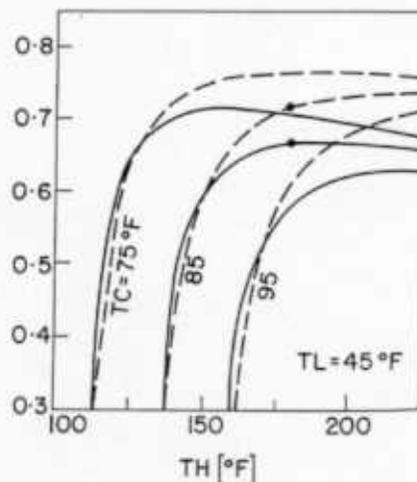


Figure 6 Coefficient of performance (COP) for single-stage ZnBr₂-LiBr-CH₃OH system (solid lines) compared to that of LiBr-H₂O system (dashed lines), as functions of operating temperatures. Both systems are assumed to have the design UA s and flow rates of the SAM15 chiller (Figure 1).

when a high-temperature heat source is available. As was shown earlier, a single-stage system is not suited to utilize a heat source at a temperature higher than a certain point; unlike other heat-activated heat pumps that follow the Carnot trend, the absorption COP levels off with the increase in desorber temperature due to increased mixing losses. The principle of double-effect systems is to utilize the heat rejected from the condenser to power an additional desorber, thereby approximately doubling the amount of refrigerant extracted out of solution with no extra heat spent. A double-effect system thus contains two desorbers and two condensers, serving one evaporator/absorber combination. The high temperature condenser is coupled with the low-temperature desorber.

Three methods of connecting the units are possible with double-effect systems, as shown schematically in the following figures (Biermann 1984): (1) A series flow arrangement, with the entire amount of solution flowing from the absorber to the low-temperature desorber, from there to the high-temperature desorber and back to the absorber (Figure 7); this system will be referred to as Series Flow Type I. (2) A series flow arrangement where the entire amount of solution from the absorber flows first to the high-temperature desorber, then to the low-temperature desorber and back to the absorber (Figure 8); this system will be referred to as Series Flow Type II. (3) A parallel flow arrangement where the solution leaving the absorber is split between the two desorbers (Figure 9). There has been a debate among absorption professionals as to which system is preferable. One purpose of the present simulation has been to resolve this question.

Earlier work on double-effect systems was conducted by several investigators. Vliet et al. (1982) carried out a thorough investigation of a series flow double-effect H₂O-LiBr chiller for solar application. Using a system-specific simulation code, they calculated the performance under varying temperatures, flow rates, and other relevant parameters. No comparison was attempted, however, with systems based on the other flow arrangements. A similar study, relatively limited in the number and range of parameters, was conducted by Chinnappa and Martin (1976) for a series flow ammonia-water chiller. A comparison of the three flow-type systems was done by Biermann (1984) but only at some reference design points. Fallek (1986) described and explained in a qualitative fashion the advantages of the parallel flow over the series flow systems. A quantitative comparison and optimization seems in order and may be conducted using the modular simulation code.

A reference case for the double-effect systems simulation was created by adding to the single-stage system (Figure 1 and Table 1) another desorber, condenser, and heat exchanger with the same UA values as in the original machine. The design hot water temperature was raised to 260°F. All other parameters remained the same as in the single-stage case. The design characteristics of the double-effect systems, operating with H₂O-LiBr, are summarized in Table 2. For the sake of a fair comparison, the same design values were employed in simulating each of the three flow-type systems.

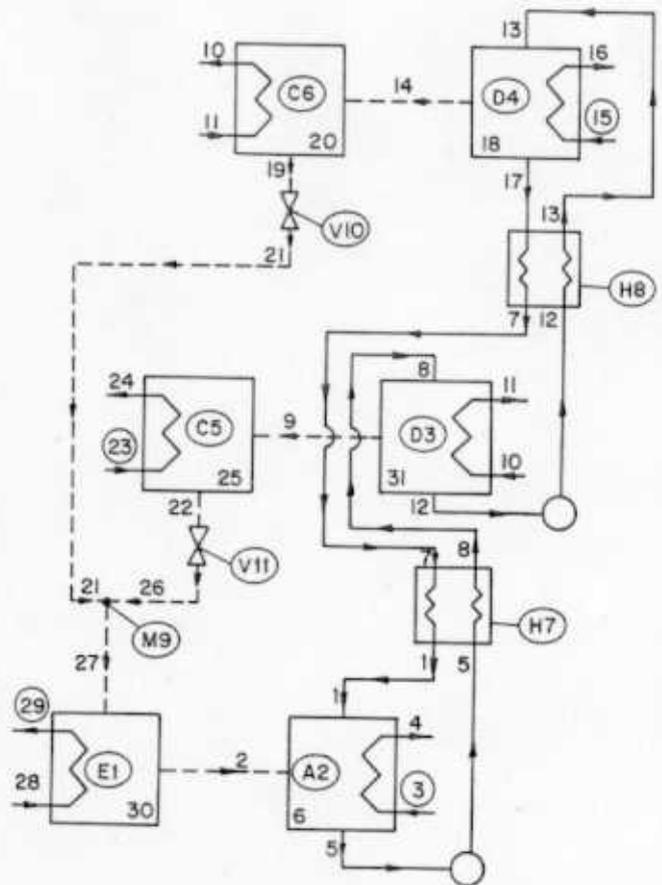


Figure 7 Schematic description of double-effect-absorption chiller, Series Flow Type I. Solid lines describe solution flow; broken lines describe refrigerant flow.

Figure 10 describes the COP of the double-effect Series Flow Type I system (Figure 7) in comparison with that of the single-stage system (Figure 1) for parametric values per Tables 2 and 1, respectively. The COP is presented as a function of the hot water inlet temperature, for different cooling water inlet temperatures. The design point for each system is indicated by a black dot. The COP of the ideal Carnot cycle operating under the same conditions is included for comparison. It is evident that both systems exhibit the same qualitative behavior; the double-effect system has a higher COP but requires a higher minimum hot water temperature to begin operating. Both systems' COP is closest to Carnot in the "knee" of the curve and levels off as the hot water temperature increases. The single-stage system gives best results in the temperature range 150-220°F. Above that, from the COP viewpoint, it is worth while to switch to the double-effect system.

The cooling capacity of the double-effect systems follows the same trend as that of the single-stage one (Figure 2), increasing almost linearly with desorber temperature. Calculated values of capacity for the three flow-type systems at the design point are given in Table 2. The capacity curves of all three double-effect systems

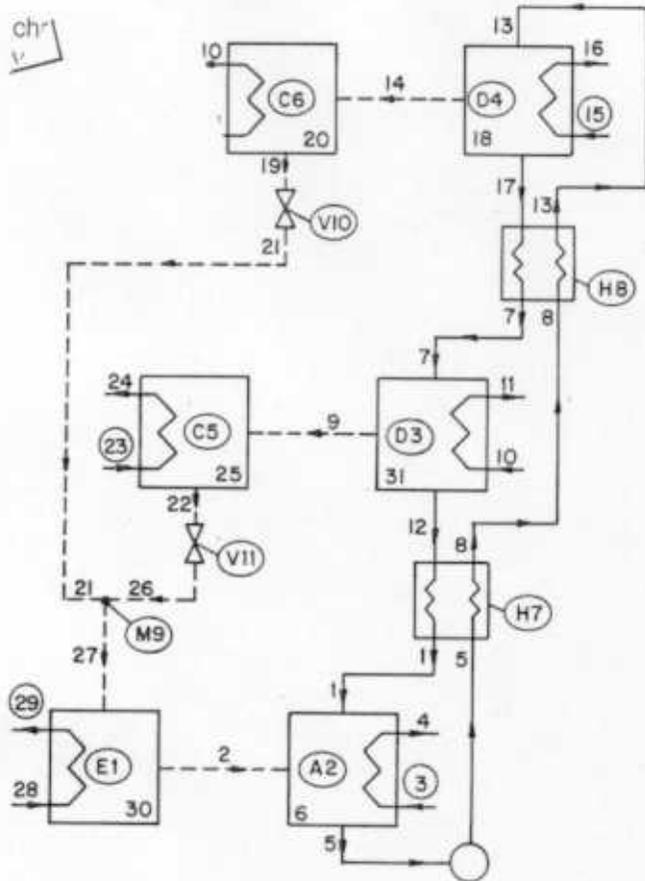


Figure 8 Schematic description of double-effect absorption chiller, Series Flow Type II. Solid lines describe solution flow; broken lines describe refrigerant flow.

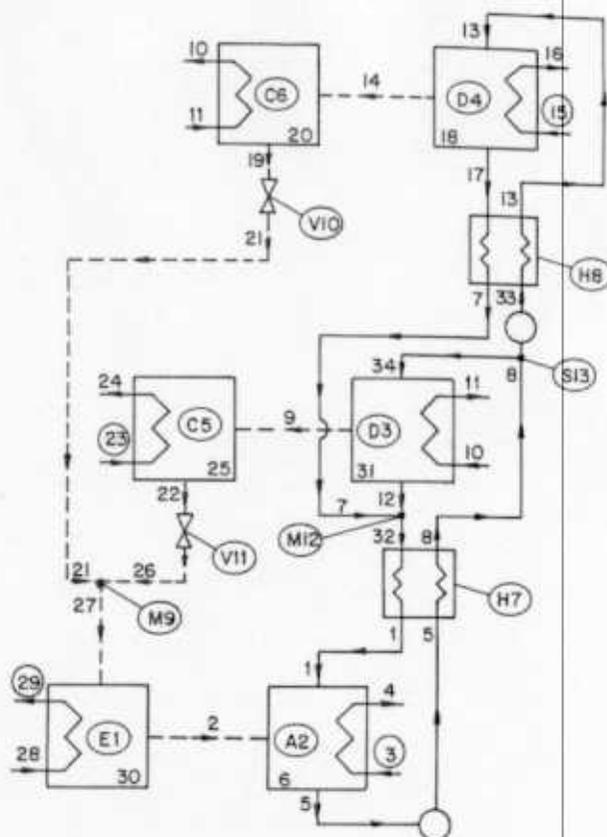


Figure 9 Schematic description of double-effect absorption chiller, Parallel Flow. Solid lines describe solution flow; broken lines describe refrigerant flow.

are quite close to each other over the entire operating temperature range.

Figure 11 compares the COP of the three flow-type systems as influenced by the hot and cooling water temperatures. The chilled water outlet temperature and all other parameters are fixed at the design value. The difference between the two series type systems is small; a considerably improved COP (about 12% higher) is seen with the parallel flow system. The main reason for this is that in the series flow systems, the entire amount of solution passes through both desorbers, whereas in the parallel flow system, each desorber receives only the amount of solution it must regenerate. This reduces circulation losses considerably.

The behavior response of the double-effect systems to changes in flow rates and UAs of the components is qualitatively similar to that of the single-stage system, but values differ. Normalizing the capacity, flow rates, and UAs with respect to their design values, as in Figures 3 and 4, yields capacity curves quite close to those of Figures 3 and 4.

In the simulation of the parallel flow type, double-effect system, under the design condition, we assumed the solution to be distributed at equal flow rates to both desorbers. An equal distribution of solution is not neces-

sarily optimal. Figure 12 shows the COP and normalized capacity of the parallel flow system as functions of the solution distribution fraction (FSF) defined as the fraction of the total solution flow rate going through the high-temperature desorber. The two extreme cases occur when either the high- or the low-temperature desorber does not receive any flow (FSF = 0 and FSF = 1, respectively). As evident from Figure 12, under these two cases both the COP and capacity tend to zero. The optimum value of FSF for maximum capacity under the present values of design parameters is approximately 0.5; however, as can be seen, the capacity curve is quite flat and insensitive to the distribution fraction for most of the FSF range. The optimum value of FSF for maximum COP is approximately 0.25, indicating best results when most of the solution is distributed to the low-temperature desorber, where mixing losses are lower. It appears that under the present design, operating with FSF around 0.3 would yield a close to optimum COP without sacrificing capacity.

CONCLUSION

The modular computer simulation code has been employed to simulate the performance of single-stage and three different double-stage configurations of absorption systems using water-lithium bromide. Of the

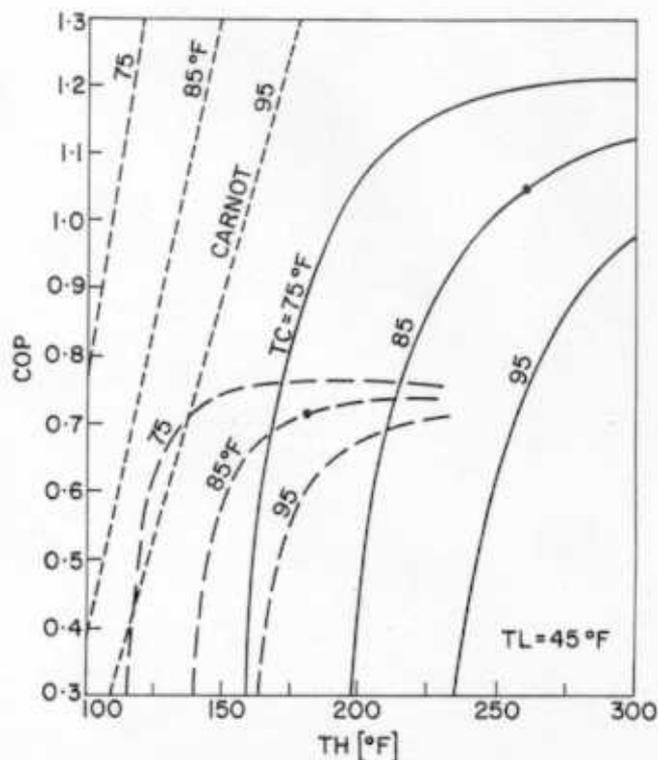


Figure 10 Coefficient of performance for double-effect, Series Flow Type I system (solid lines) and single-stage system (dashed lines) as functions of operating temperatures. Carnot COP for the same operating conditions is given by the dotted lines.

latter, the parallel flow configuration yields better results than the two series flow systems.

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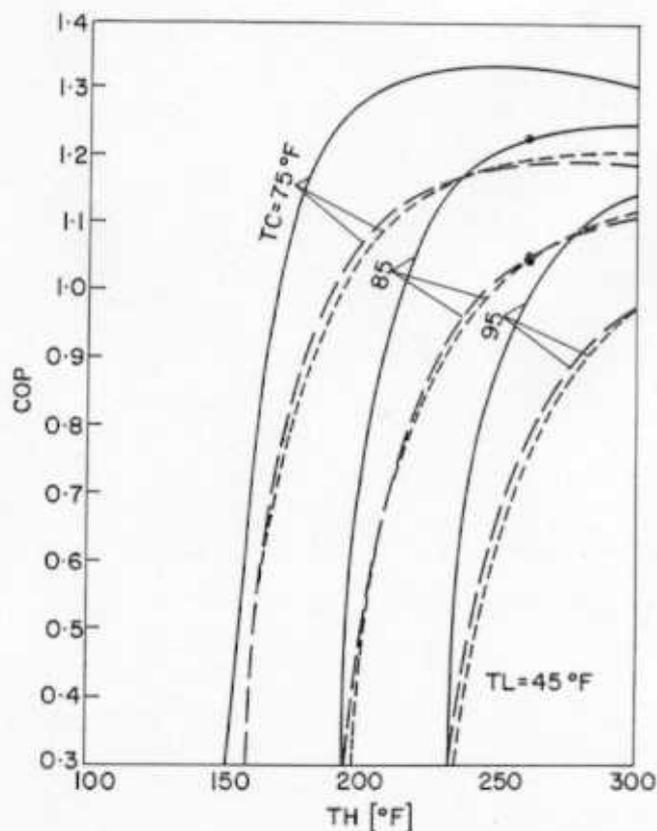


Figure 11 Coefficient of performance of double-effect, parallel flow type system (solid lines), Series Flow Type I (dotted lines), and Series Flow Type II (dashed lines) as functions of the operating temperatures.

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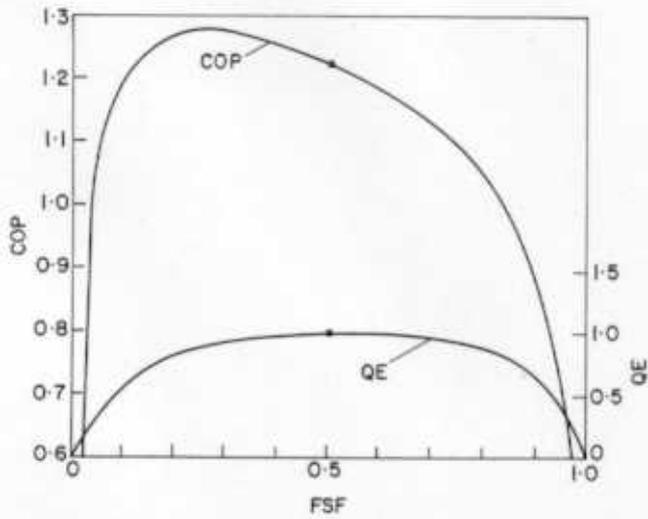


Figure 12 Coefficient of performance and normalized capacity of double-effect, parallel flow type system as functions of the solution distribution fraction between the high and low temperature desorbers.

TABLE 2
Characteristic Parameters at Design Point for Double-Effect H₂O-LiBr Absorption Chillers

Heat Transfer Characteristics (UA):

Absorber:	193.0 Btu/min · °F
High-Temperature Desorber:	268.0 Btu/min · °F
Low-Temperature Desorber:	268.0 Btu/min · °F
High-Temperature Condenser:	565.0 Btu/min · °F
Low-Temperature Condenser:	565.0 Btu/min · °F
Evaporator:	377.0 Btu/min · °F
High-Temperature Heat Exchanger:	64.0 Btu/min · °F
Low-Temperature Heat Exchanger:	64.0 Btu/min · °F

Mass Flow rate:

Absorber (cooling water)	483.0 lbs/min
High-Temperature Desorber (hot water)	416.0 lbs/min
Low-Temperature Desorber (internal transfer loop)	
High-Temperature Condenser (internal transfer loop)	
Low-Temperature Condenser (cooling water)	391.0 lbs/min
Evaporator (chilled water)	300.0 lbs/min
Weak Solution (total, from absorber)	60.0 lbs/min
(in Parallel Flow system - 30.0 lbs/min to each desorber)	

Temperatures:

Hot water inlet (state point 15):	260 °F
Cooling water inlet (state points 3 and 23):	85 °F
Chilled water outlet (state point 29):	45 °F

Calculated Parameters:	Series Flow	Series Flow	Parallel
	Type I	Type II	Flow
Cooling Capacity (kW):	41.0	42.0	42.0
Coefficient of Performance:	1.046	1.054	1.225