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Simulation and Performance Analysis of an Ammonia-Water Absorption Heat Pump Based on the Generator-Absorber Heat Exchange (GAX) Cycle

G. Grossman*
R. C. DeVault
F. A. Creswick**

*Technion-Israel Institute of Technology
**Martin Marietta Energy Systems, Inc.—Retired

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PAPER COVER SHEET

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by

G. Grossman, Sc.D., ASHRAE Member; R.C. DeVault; and F.A. Creswick, ASHRAE Member

G. Grossman is Professor, Faculty of Mechanical Engineering, Technion - Israel Institute of Technology, Haifa 32000, Israel;
R.C. DeVault is Research Staff Member, Energy Division, Oak Ridge National Laboratory, Oak Ridge, Tennessee 37831, USA;
F.A. Creswick is Group Leader, Energy Division, Oak Ridge National Laboratory, Oak Ridge, Tennessee 37831, USA.

ABSTRACT

A computer simulation has been conducted to investigate the performance of an absorption heat pump, based on the Generator-Absorber Heat Exchange (GAX) cycle employing ammonia-water as the working fluid pair. The particular feature of this cycle is the ability to recover heat from the absorber and employ it to partially heat the generator, thus improving the COP. In the present study, a detailed simulation has been conducted of one of the preferred configurations for the cycle. A modular computer code for flexible simulation of absorption systems (ABSIM) was employed. Performance parameters, including COP and capacity, were investigated as functions of different operating parameters over a wide range of conditions in both the cooling and heating mode. The effect of the ambient temperature, the rectifier performance, the flowrate in the GAX heat transfer loop and the refrigerant flow control were investigated. COP's on the order of 1.0 for cooling and 2.0 for heating have been calculated.

Key Words: Absorption, Heat Pump, Gas Fired, Generator, Simulation

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A computer simulation has been conducted to investigate the performance of an absorption heat pump, based on the Generator-Absorber Heat Exchange (GAX) cycle employing ammonia-water as the working fluid pair. The particular feature of this cycle is the ability to recover heat from the absorber and employ it to partially heat the generator, thus improving the COP. In the present study, a detailed simulation has been conducted of one of the preferred configurations for the cycle. A modular computer code for flexible simulation of absorption systems (ABSIM) was employed. Performance parameters, including COP and capacity, were investigated as functions of different operating parameters over a wide range of conditions in both the cooling and heating mode. The effect of the ambient temperature, the rectifier performance, the flowrate in the GAX heat transfer loop and the refrigerant flow control were investigated. COP's on the order of 1.0 for cooling and 2.0 for heating have been calculated.

INTRODUCTION

The possibility of internal heat recovery in certain absorption cycles with a wide solution concentration field was described at the beginning of the century by Altenkirch (1913), and forms the fundamental basis for the Generator-Absorber Heat Exchange (GAX) cycle. The particular feature of this cycle is a partial temperature overlap between the generator and absorber, associated with the wide concentration field, which leads to the possibility of recovering heat from the absorber and employing it to partially heat the generator, thus obtaining a high COP. Several theoretical studies have been conducted in recent years on the GAX cycle (Scharfe et al., 1986; McGahey and Christensen, 1993; Inoue et al., 1994) including some suggested improvements (Herold et al., 1991; Rane and Erickson, 1994). The GAX cycle employing ammonia-water has been recognized as a promising candidate for residential and light commercial gas-fired heat pumps. The cycle is particularly suitable for these applications because it can provide the functions of both heating and cooling in a compact size, with air or hydronic external heat sources/sinks. The ammonia-water working fluids constitute an attractive alternative to ozone-depleting CFC's employed in conventional vapor compression heat pumps.

A development program for "Advanced Absorption Cycle" heat pumps was initiated by the U.S. Department of Energy (DOE) in 1982 with a goal of a heating COP of 1.6 at 47°F (8.3°C) and a cooling COP of 0.7 at 95°F (35°C). As the result of a competitive procurement, three DOE-sponsored studies were launched to evaluate a variety of absorption cycles (Biermann, 1984; Phillips, 1990; Modahl and Hayes, 1992). Under Phase 1 of the Program, two of the studies recommended the GAX cycle as the preferred cycle for hardware development. In the ensuing decade, prototype hardware has demonstrated the thermodynamic principle of the GAX cycle as well as the potential for a first-generation commercial product. Currently, major GAX commercialization programs are underway in the U.S.A. and also, more recently, in Japan.

Several cycle configurations have been proposed for accomplishing the generator-absorber heat exchange . The two basic methods that have been published are those proposed by Phillips (1990), and by Modahl and Hayes (1992). The main difference between the two is in the design of the GAX absorber and desorber. The Phillips (1990) configuration employs counterflow heat and mass exchange between the liquid and vapor in both the GAX absorber and desorber, using an external liquid loop to transfer the heat from the former to the latter. The Modahl and Hayes (1992) configuration omits the external loop, and therefore employs co-flow heat and mass exchange in the GAX desorber, while maintaining counterflow heat and mass exchange in the GAX absorber. The inferior performance of the co-flow desorber is compensated for by an adiabatic exchange between the weak liquid stream entering the desorber and the vapor leaving it, to pick up some of the water contained in the latter before it goes to the rectifier.

Despite the laboratory hardware development efforts (best described as "cut and try"), relatively little systematic simulation work has been carried out on the potential performance of the GAX cycle. Because of the complexity of the cycle, detailed calculations were difficult. Estimates of the performance at design point conditions were performed under the DOE Phase 1 analysis of Advanced Cycles (Biermann, 1984; Phillips, 1990; Modahl and Hayes, 1992). Some system-specific calculations were conducted in-house by the hardware developers, involving physical equations in combination with their own empirical data. Recently, detailed simulation of the GAX cycle using ammonia-water has become more practical through the use of specially developed computer simulation tools such as ABSIM, which will be described further below. McGahey and Christensen (1993) have employed ABSIM to simulate a 10 RT heat pump based on the cycle configuration proposed by Phillips (1990). They concentrated mainly on optimizing the heat exchanger sizes in the cycle components, using a logarithmic mean temperature difference (LMTD) in conjunction with the overall heat transfer coefficient times area (UA) to characterize them. A detailed investigation using ABSIM was conducted by Hanna and Whitacre (1994) of a heat pump based on the cycle configuration according to Modahl and Hayes (1992). Theoretical studies using other simulation techniques have been conducted by Scharfe et al.(1986), Inoue et al.(1994), Herold et al.(1991) and Rane and Erickson (1994).

The purpose of the present work has been to simulate the GAX cycle and to investigate its performance in a systematic and realistic manner. Similar systematic simulation studies have been conducted in the past few years for other absorption cycles, including some rather complex ones (Grossman, Wilk and DeVault, 1994; Gommed and Grossman, 1990). These studies have demonstrated the importance to determine not only a design point performance, but to generate performance curves of COP, capacity and other parameters as functions of the operating conditions over their entire applicable range.

The modular computer code for ABsorption SIMulation (ABSIM) was employed to conduct this study. The code, developed specifically for flexible simulation of absorption cycles, has been described in detail by Grossman and Wilk (1992) and in a related report (Grossman, Gommed and Gadoth, 1987) containing a user's manual. The modular structure of the code makes it possible to simulate a variety of absorption systems in varying cycle configurations and with different working fluids. The code is based on unit subroutines containing the governing equations for the system's components and on property subroutines containing thermodynamic properties of the working fluids. The components are linked together by a main program, which calls the unit subroutines according to the user's specifications to form the complete cycle. When all the equations for the entire cycle have been established, a mathematical solver routine is employed to solve them simultaneously.

In conducting the present simulation, the same methodology employed in earlier studies of single-, double- and triple-effect cycles has been followed. Since the system's performance depends on many parameters, the approach has been to establish a design point for the cycle, then vary the relevant parameters around it. Thus, performance maps of COP and capacity as functions of the operating conditions can be generated.

DESCRIPTION OF THE CYCLE

The present simulation has been performed on the basis of the configuration proposed by Phillips (1990) . Figure 1 describes this cycle in terms of the components recognized by the ABSIM code. It consists of 14 units and 39 state points; numbers in a circle mark units and uncircled numbers indicate state points. Following is a brief review of this cycle diagram.

Liquid refrigerant at a subcooled state 29 expands through Valve 6 and enters the evaporator, simulated here as the combination of Desorber 1 and Mixer 9, to allow for incomplete evaporation of the refrigerant leaving at state 3. The refrigerant is not quite pure ammonia and contains about 0.5% water; this binary mixture causes a gliding evaporation temperature and does not evaporate completely in Desorber 1. The streams 27 and 28 show the amounts of liquid and vapor, respectively, leaving the evaporator. Their combined mixture (state point 3) passes through a precooler, simulated here as the combination of Desorber 10 and Mixer 14, again for the same reasons. The precooler of the condensate stream 15-29 causes part or all of the liquid that remains in the refrigerant stream 3 to evaporate; stream 30 is therefore generally saturated to an almost dry state, and under some conditions - reaches a dry saturated or even a superheated state. This stream enters the absorber consisting of three parts as shown, simulated here as the combination of Analysers 12 and 11 and Absorber 5. Unit 12 is externally cooled, Unit 11 is solution-cooled and Unit 5 is GAX-cooled. The weak solution leaving the bottom of the absorber at state 33 is pumped to the desorber, consisting also of three parts as shown, simulated here as the combination of Desorber 3 and Analysers 2 and 13. Unit 3 is externally heated, Unit 2 is solution-heated and Unit 13 is GAX-heated. An external liquid loop 4-5 (not completely shown), employing water as the heat transfer fluid, transfers heat between the GAX absorber (Unit 5) and desorber (Unit 13). The strong solution leaving the desorber at state 36 is returned to the absorber. The desorbed vapor at state 23 enters Rectifier 7 and is distilled to state 10, with the reflux at state 22 mixed with the weak solution in Mixer 8 before entering the desorber. The rectified vapor condenses in Condenser 4 and the condensate at state 15 is subcooled in the precooler (Unit 10) before expanding into the evaporator. This completes the cycle.

It is important to note that in this configuration, the liquid and the vapor streams in all three parts of the absorber and desorber exchange heat and mass in counterflow, which is the most efficient way of carrying out the respective processes. Also, the transfer of heat to or from each of the external streams is done in counterflow to the internal flow of the liquid.

THE DESIGN CONDITION

As a first step toward conducting the simulation, a design condition has been selected for the system described in Figure 1. Table 1 summarizes the values of the various parameters specified, and a brief explanation follows.

The design condition has been selected to represent a gas-fired domestic heat pump, delivering a cooling capacity of 3.0 RT (600 BTU/min or 10.54 kW) under typical cooling conditions. Normally, in order to define such a system to the simulator, one would have to specify the following parameters: The heat and mass transfer characteristics of all the exchange units, typically in terms of their overall heat transfer coefficient times area (UA's) and deviations from equilibrium (DEV's); The inlet temperature and flowrate in each of the external loops supplying chilled water, cooling water and hot flue gases; and the internal flowrates controlled by the designer through various pumps and valves. When testing the system for off-design behavior, the variation of these fixed parameters under the control method should be taken into account. All other parameters at the various state points are unknown variables.

Specifying the heat transfer characteristics of exchange units in terms of UA's and LMTD's is rather problematic in ammonia-water systems, and may lead to erroneous results. The problem is related to the equation:

$$Q = UA*LMTD \qquad (1)$$

which is strictly valid only when the streams involved in the heat transfer either have a constant heat capacity or are at a fixed temperature such as in phase change of a pure substance (Kays and London, 1964). These conditions are not satisfied in most ammonia-water systems and Equation (1) may be used only as an approximation, which is reasonably good in some units but quite inaccurate in others. The non-linearity of the temperature-enthalpy relation for ammonia-water is more pronounced for a mixture containing predominantly one substance, and less for one with an even content of both substances. Therefore, using Equation (1) may lead to a large error particularly in the components employing the refrigerant (evaporator, condenser, precooler); the error would be much smaller in the absorber and desorber. Because of this, it was chosen to avoid specifying UA's wherever possible. In all the units involving external heat exchange loops, we exercised the option provided by ABSIM to omit those loops (along with the associated heat transfer equations) and instead specify outlet temperatures of the internal working fluids. Accordingly, outlet temperatures of the refrigerant or solution have been specified in the evaporator, condenser, rectifier, externally cooled absorber and externally heated desorber. The UA's in the solution-cooled absorber (11) and the solution-heated desorber (2) were calculated to give zero subcool or superheat of the solution inlet into the absorber (s.p. 6) and desorber (s.p. 21). Finally, the UA's in the GAX absorber and desorber were sized to give a 9.0°F (5.0°C) LMTD. It was found that in these units the conditions approximately support Equation (1). A particularly problematic component is the precooler (10), where the behavior deviates strongly from that expressed by Equation (1) due to the composition of the refrigerant evaporating on its low-pressure side. The precooler is a very important component to compensate for imperfect rectification, yet small and relatively inexpensive to build. Therefore, it is reasonable to assume it sufficiently large to evaporate all the liquid left over from the evaporator, and even to superheat it, when conditions permit, i.e. when the condensate temperature at state point 15 is higher than the dry saturation temperature of the refrigerant leaving the evaporator. Accordingly, and in the absence of sufficient data on the properties of superheated ammonia-water mixtures, the outlet condition from the precooler at state point 30 was specified as dry saturated (vapor quality of 1.0) when possible; otherwise, T(30)=T(15) or T(3)=T(29) were specified, depending on the location of the pinch.

TABLE 1

Characteristic Parameters at Design Point for GAX Cycle

Temperatures:

Refrigerant outlet from Evaporator (s.p. 3)	50°F (10°C)
Refrigerant outlet from Condenser (s.p. 15):	108°F (42°C)
Refrigerant outlet from Rectifier (s.p. 10):	149°F (65°C)
Weak solution outlet from Absorber (s.p. 33):	108°F (42°C)
Strong solution outlet from Desorber (s.p. 36):	383°F (195°C)

Mass Flow Rates:

GAX coupling water loop, (s.p. 4-5)	9.0 lbs/min (4.1 kg/min)

Weak solution (s.p 33) - pumping controlled to accommodate fixed heat input in Desorber 3.

Refrigerant (s.p. 15) - flow controlled by Thermostatic Expansion Valve 6 to keep a fixed 9.0°F (5.0°C) difference across the evaporator.

Heat Quantities:

Desorber firing rate:	575.0 Btu/min (10.1 kW)

Heat Transfer Characteristics (UA):

Solution-cooled Absorber (Unit 11):	8.492 Btu/min.°F (269 W/°C)
GAX Absorber (Unit 5):	39.64 Btu/min. °F (1254 W/°C)
Solution-heated Desorber (Unit 2):	2.287 Btu/min. °F (72 W/°C)
GAX Desorber (Unit 13):	39.64 Btu/min. °F (1254 W/°C)

Precooler (Unit 10) - UA sufficient to keep refrigerant at outlet (s.p. 30) dry saturated.

Other Cycle Characteristics:

Refrigerant at Condenser outlet (s.p. 15) subcooled by 9.0°F (5.0°C); All other liquid and vapor outlets from exchange units - in equilibrium.

The outlet temperatures specified for the internal working fluids in the different units are typical under cooling conditions: 50°F (=10°C) for the refrigerant vapor outlet from the evaporator; 108°F (=42°C) for the weak solution outlet from the absorber and the refrigerant liquid outlet from the condenser; and 149°F (=65°C) for the refrigerant vapor outlet from the rectifier. The strong solution outlet temperature from the desorber has been specified as 383°F (=195°C), to maintain some safety margin on the concentration and avoid reaching the state of pure water.

The deviation from equilibrium at the outlets of the exchange units, characterizing mass transfer, has been assumed zero for all the units except the condenser, where a 9.0°F (5.0°C) subcool is a reasonable assumption.

Two internal flowrates are controlled in the GAX cycle: The pumped weak solution (s.p. 33-34), and the throttled refrigerant (s.p. 15-29-20). For the weak solution, the pumping rate has been set to accommodate a fixed firing rate (external heat input) in Desorber 3 of 575 BTU/min (10.1 kW) (which yields a cooling capacity of 3.0 RT at the design point). The refrigerant flowrate is controlled by a thermostatic expansion valve set to keep a fixed temperature difference of 9.0°F (5.0°C) across the evaporator (between state points 20 and 3). A somewhat arbitrary, yet to be optimized value of 9.0 lbs/min (4.1 kg/min) has been selected for the water flowrate in the GAX heat transfer loop at the design point.

Applying the ABSIM code to simulate the design condition yields the temperature, flowrate, concentration, pressure and vapor fraction at all 39 state points and the heat duty and transfer characteristics at all 14 units. The resulting cooling COP is 1.0443.

RESULTS OF SIMULATION

In order to investigate the system's performance under off-design conditions, it is necessary to establish a method by which to control certain operating parameters. Unlike in lithium bromide-water chillers which normally operate with a fixed pumping rate of the weak solution and a very simple refrigerant expansion device, here it is necessary to control both the solution and refrigerant flows, because of the high pressures involved. The ABSIM code makes it possible to trade some variables for fixed parameters and vice versa - a desirable feature for certain types of simulations, particularly for checking control strategies. As mentioned earlier, in the basic control scheme selected in this study for the GAX heat pump, the pumping rate of the weak solution is set to accommodate a fixed firing rate (external heat input) in the desorber; the refrigerant flowrate is controlled by a thermostatic expansion valve set to keep a fixed temperature difference across the evaporator; the water flowrate in the GAX heat transfer loop is maintained fixed. Other control schemes have been tried under the simulation, which proved not as efficient. They will be discussed briefly later.

Using the basic control scheme, the effect of the ambient temperature on the cooling performance was investigated first. The heat rejection temperature at the absorber and condenser (T(15)=T(33), set to 108°F (42°C) at the design point) was varied while keeping all other parameters fixed at their design condition value (Table 1). Figure 2 describes the cooling COP and capacity as functions of this temperature. The COP is defined here as the ratio of heat duties between the evaporator (1) and the externally heated desorber (3). The heat rejection temperature was varied over the range from 90°F to 130°F (32°C to 54°C)which represents a wide range of ambient temperatures under cooling conditions. It is evident that both the COP and capacity decrease with the increasing ambient temperature, as expected, and the decrease is almost linear. Figure 3 describes the flowrates of the weak and strong solution and of the refrigerant (F(33), F(36) and F(10), respectively). The flow of refrigerant decreases with the increasing ambient temperature at a rate corresponding to the decrease in capacity; the weak solution (pumped) flowrate decreases and the strong solution (return) flowrate increases so as to close the gap between them, this gap corresponding exactly to the flow of the refrigerant. The concentration gap between the weak and strong solution follows the same trend. Note that at 90°F (32°C) the strong solution is almost pure water under the given conditions. Figure 4 shows the concentrations of the refrigerant before and after rectification. It is evident that as the ambient temperature increases, the water content at the desorber exit increases significantly. The rectifier is able to cope with it and produce a distilled refrigerant stream containing less than 1% water over the entire range of heat rejection temperatures. It can be seen that the degree of purity improves slightly with the increasing ambient temperature. This is achieved at the cost of an increasing heat duty in the rectifier, as will be shown later, which contributes to the decrease in COP with increasing ambient temperature.

Figure 5 describes the heat duty under the cooling mode at some key components of the system as functions of the heat rejection temperature. As the ambient temperature increases, the external heat input into the desorber (3) remains fixed at the design value of 575 BTU/min (10.1 kW), and the heat taken up by the evaporator (1), representing the cooling capacity, decreases as seen before. A marked decrease is observed in the heat exchanged between the two GAX components, absorber (5) and analyser (13). This is a major contributor to the decrease in COP, and is manifested also by a decrease in the concentration gap between the weak and strong solutions which leads to a narrower temperature overlap between the absorber and desorber. It is this temperature overlap which makes GAX possible. A somewhat smaller contribution to the decreasing COP comes from the rectifier, where the heat duty increases significantly with the ambient temperature, as discussed earlier. Relatively small changes are observed in the heat duties of the solution-cooled absorber (11) and solution-heated desorber (2). The amount of heat transferred in the precooler is one of the smallest, and decreases with the increase in heat rejection temperature. It should be recalled that in conducting the simulation, the precooler size was specified large enough to keep the refrigerant exiting at state point 30 dry saturated whenever possible; this was in fact found possible over most part of the heat rejection temperature range (except at 90°F-100°F (32°C-38°C) where the condensate temperature T(15) is too low for that; refer to the discussion in the previous section). In fact, with a precooler sized to transfer the heat under the more severe conditions, the refrigerant is likely to exit superheated at the higher heat rejection temperatures.

A cooling capacity decreasing with increasing ambient temperature is not desirable, because the building cooling load follows just the opposite trend. It was therefore attempted to test, under the simulation, other control methods which may help correct this behavior. Three changes to the basic (design) control scheme were attempted: 1. Using a thermostatic expansion valve (TXV) to control the refrigerant flowrate as in the design control scheme, but with a different fixed value of the temperature difference across the evaporator. 2. Replacing the TXV with a control valve which keeps the low pressure constant. 3. Replacing the TXV with a control valve which keeps the refrigerant flowrate constant. Figure 6 describes the cooling COP obtained under these conditions as a function of the heat rejection temperature. Note that the capacity is exactly proportional to COP since the firing rate (external heat input) is constant. In comparing the results of the changes with the original control scheme it is evident that the TXV with the a smaller delta T across the evaporator (4.5°F (2.5°C) instead of 9.0°F (5.0°C)) improves the COP over the entire range of ambient temperatures. This would also result in a larger capacity, and would require larger heat exchangers. Controlling the refrigerant flow to maintain the low pressure constant yields results almost identical with the basic scheme; this is therefore a possible alternative to the TXV under cooling conditions. Keeping the refrigerant flow constant originally seemed promising, since it helps maintain an almost fixed COP and capacity under varying ambient conditions. However, when doing so the low pressure rises with increasing heat rejection temperature and the temperature glide in the evaporator is reduced until at some temperature one can no longer operate.

The performance of the cycle under the heating mode was investigated next. Employing the basic control scheme, the evaporator outlet temperature (T(3)=T(27)=T(28), set to 50°F (10°C) at the design point) was varied while keeping all other parameters fixed at their design condition value (Table 1). Figure 7 describes the heating COP and capacity as functions of this temperature. The heating capacity is defined by the sum of the heat quantities rejected from the absorber (12), condenser (4) and rectifier (7) and the heating COP is the ratio between this capacity and the heat supplied to the externally heated desorber (3). The evaporator outlet temperature was varied over the range from 20°F to 85°F (-6.7°C to 29.4°C) which represents a wide range of ambient temperatures under heating conditions (as will be shown later, the cycle described in Figure 1 cannot operate at lower ambient temperatures due to the loss of GAX capability, and one must resort to a different cycle). It is evident from Figure 7 that both the COP and capacity increase almost linearly with increasing ambient temperature, as may be expected. Figure 8 describes the flowrates of the weak and strong solution and of the refrigerant (F(33), F(36) and F(10), respectively). The flow of refrigerant increases with the increasing ambient temperature at a rate corresponding to the increase in capacity; the weak solution (pumped) flowrate increases and the strong solution (return) flowrate decreases so as to open the gap between them, this gap corresponding exactly to the flow of the refrigerant. The concentration gap between the weak and strong solution follows the same trend. Figure 9 shows the concentrations of the refrigerant before and after rectification. It is evident that as the ambient temperature decreases, the water content at the desorber exit increases significantly. This refrigerant stream is distilled by the rectifier to a practically constant concentration of 99.57% ammonia, over the entire range of heat rejection temperatures. This concentration is determined by the high pressure and the rectifier outlet temperature, both remaining constant.

Figure 10 describes the heat duty under the heating mode at some key components of the system as functions of the evaporator outlet temperature. As the ambient temperature decreases, a marked decrease is observed in the heat exchanged between the two GAX components, absorber (5) and analyser (13). As observed also under the cooling mode, this is a major contributor to the decrease in COP, and is manifested also by a decrease in the concentration gap between the weak and strong solutions which leads to a narrower temperature overlap between the absorber and desorber. At the low limit of the range selected for the evaporator outlet temperatures (20°F), it may be seen from Figure 10 that GAX capability is lost almost completely. To operate at still lower ambient temperatures, it would be advantageous to switch to a liquid heat exchange (LHE) cycle, with an absorber comprising an externally-cooled and a solution-cooled part, a desorber comprising an externally-heated and a solution-heated part, and a recuperative heat exchanger for the weak and strong solution streams flowing between them. Figure 10 also indicates a significant decrease in the heat duty of the rectifier with the increase in ambient temperature. As in the cooling mode, relatively small variations are observed in the heat duty of the solution-heated desorber (2), while there is a decline in the heat duty of the solution-cooled absorber (11). The amount of heat transferred in the precooler remains one of the smallest in the cycle.

In the analysis so far, an equilibrium condition has been assumed for both the liquid and the vapor streams at the exit of each unit (except for the condenser). It is known that a deviation from equilibrium in a unit where there is an exchange of heat and mass, is characteristic of imperfect mass transfer and generally hurts performance. It is relatively easy by proper design to keep the liquid close to equilibrium. The vapor, however, has poor heat transfer characteristics and is therefore unlikely to reach equilibrium at its outlet. Consider, for example, the solution-heated desorber (analyser 2) in Figure 1. The inlet and outlet temperatures of the liquid solution in this unit under the design condition are 243.4°F (s.p.35) and 270.9°F (s.p.37), respectively. Vapor generated from the liquid in this desorber under extremely poor heat transfer between liquid and vapor, would be at approximately the average between these two temperatures, that is at 257.1°F (125.0°C). This vapor would exit at s.p. 17 at this temperature rather than in equilibrium with stream 35, that is - superheated by 13.7°F (7.6°C). A similar condition would exist in desorber 3, analyser 13 and rectifier 7. The reverse situation, that is - a subcooled vapor condition, would prevail at the outlets of the absorber, in analysers 11 and 12. Figure 11 describes the cooling COP as a function of the heat rejection temperature as in Figure 1, for two conditions: In one, the deviation from equilibrium is zero everywhere except for the condenser, as set in the design condition (Table 1); in the other, a deviation from equilibrium of +9.0°F (+5.0°C) has been specified for the vapor at the outlet from units 2,3 7 and 13, and -9.0°F (-5.0°C) at the outlet from units 11 and 12. All other parameters remain the same. The deterioration in performance due to the equilibrium deviation is evident.

The effect of the water flowrate in the GAX heat transfer loop was investigated next. Recall that this flowrate has been set rather arbitrarily to 9.0 lbs/min (4.1 kg/min) at the design condition, and maintained fixed throughout the former runs. Here it was varied over a wide range of values while keeping all the other parameters fixed at the design point values. Figure 12 describes the cooling COP as a function of this flowrate, showing an optimum at about 6 lbs/min. The COP decreases gently for larger flowrates and rather sharply for smaller flowrates. The reason for the optimum stems from heat transfer considerations and may be understood by following the schematic temperature diagram in Figure 12. It shows a Lorenz-type plot of the solution temperatures in the GAX absorber and desorber, with the heat transfer loop between them. The slope of this loop in the plot depends on the flowrate in it: The larger the flowrate, the gentler the slope (smaller difference between T(5) and T(4)) and vice versa. Too large or too small a flowrate would create a pinch at either end; best conditions for heat transfer would be achieved with a flowrate yielding a slope appropriately in the middle. The optimum flowrate is expected to vary under different operating conditions.

Additional simulation runs were performed to study the behavior of the rectifier. The rectifier outlet temperature T(10) was varied between two limits - the desorber outlet temperature and the condensing temperature. As expected, the purity of ammonia in the refrigerant stream increases as T(10) decreases, but a higher heat of rectification is required. Thus, by raising the rectifier temperature a small increase in COP may be gained, but the water content in the refrigerant increases which leads to a large temperature glide in the evaporation. It may be of advantage to study the trade-offs in rectification further.

SUMMARY AND CONCLUSIONS

Performance simulation was conducted for the GAX cycle employing ammonia-water in the configuration proposed by Phillips (1990), to serve as a gas-fired heat pump for domestic and light commercial applications. A design point was established as a reference condition, for a system to deliver 3 RT (10.54 kW) of cooling under normal summer conditions. The simulation was performed over a wide range of ambient conditions under both heating and cooling modes; the COP and capacity were calculated along with the internal flows and concentrations of the solution and refrigerant. Several methods of controlling the flows were considered. The influence of some of the design parameters was investigated, such as the flowrate in the GAX heat transfer loop. Additional investigation is required on the control method, including the flowrates of solution and refrigerant and perhaps the firing rate. The optimum flowrate in the GAX heat transfer loop under different operating conditions should be explored. For operating in the heating mode at very low ambient temperatures, a switch is necessary from the GAX to a different cycle such as LHE, and the performance of this cycle under those conditions should be studied.

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REFERENCES

E. Altenkirch, 1913: "Reversible Absorptionsmaschinen", Zeitschrift fur die gesamte Kalte-Industrie, XX. Jahrgang, Heft 1: pp.1-9, Heft 6: pp.114-119, Heft 8: pp.150-161.

W.J. Biermann, 1984: "Advanced Absorption heat Pump Cycles". ORNL/Sub/81-17499/1, Oak Ridge National Laboratory.

K. Gommed and G. Grossman, 1990: " Performance Analysis of Staged Absorption Heat Pumps: Water-Lithium Bromide Systems." ASHRAE Transactions, vol.96, part 1, pp. 1590-1598.

G. Grossman, K. Gommed and D. Gadoth, 1991: "A computer Model for Simulation of Absorption Systems in Flexible and Modular Form". ORNL/Sub/90-89673, Oak Ridge National Laboratory.

G. Grossman and M. Wilk, 1992:"Advanced Modular Simulation of Absorption Systems". Proceedings, the ASME Winter Annual Meeting, Anaheim, California, November 8-13, vol. AES-28, pp. 15-26.

G. Grossman, M. Wilk and R. C. DeVault, 1994:"Simulation and Performance Analysis of Triple-Effect Absorption Cycles". ASHRAE paper No. 3781, presented at the ASHRAE Semiannual Meeting, New Orleans, Louisiana, January 23-26.

W.T. Hanna and G.R. Whitacre, 1994: "GAX Absorber Development: GAX Configuration Study", ORNL Subcontract 86X-SL266C to Battelle Memorial Institute, Task 1 Report (to be published).

K.E. Herold, X. He, D.C. Erickson and M.V. Rane, 1991: " The Branched GAX Absorption Heat Pump Cycle". Proceedings, the Absorption Heat Pump Conference, Tokyo, Japan, September 30 - October 2, pp. 127-132.

N. Inoue, H. Iizuka, Y. Ninomiya, K.I. Watanabe and T. Aoki, 1994:" COP Evaluation for Advanced Ammonia-Based Absorption Cycles". Proceedings, the International Absorption Heat Pump Conference, New Orleans, Louisiana, January 19-21, ASME AES-Vol.31, pp.1-6.

W.M. Kays and A.L. London, 1964: "Compact Heat Exchangers", McGraw-Hill Book Co.

K.R. McGahey and R.N. Christensen, 1993: "The Modeling and Optimization of a Generator Absorber Heat Exchange (GAX) Heat Pump Using the ORNL Absorption System Simulation". Proceedings, the ASME Winter Annual Meeting, New Orleans, Louisiana, Nov 28-Dec 3, Vol. AES-29, pp. 57-63.

R.J. Modahl and F. C. Hayes, 1992: "Development and Proof-Testing of Advanced Absorption Refrigeration Cycle Concepts". ORNL/Sub/86-17498/1, Oak Ridge National Laboratory.

B.A. Phillips, 1990: "Development of a High-Efficiency Gas-Fired Absorption Heat Pump for Residential and Small Commercial Applications: Phase 1 Final Report: Analysis of Advanced Cycles and Selection of the Preferred Cycle". ORNL/Sub/86-24610/1, Oak Ridge National Laboratory.

M.V. Rane and D.C. Erickson, 1994: "Advanced Absorption Cycle: Vapor Exchange GAX". Proceedings, the International Absorption Heat Pump Conference, New Orleans, Louisiana, January 19-21, ASME AES-Vol.31, pp. 25-32.

J. Scharfe, F. Ziegler and R. Radermacher, 1986: "Analysis of Advantages and Limitations of Absorber-Generator Heat Exchange", International Journal of Refrigeration, Vol. 9, pp. 326-333.

LIST OF FIGURES

Figure 1: Schematic description of the GAX cycle according to the configuration proposed by Phillips (1990): The evaporator is represented by Desorber (1) and Mixer (9); the precooler is represented by Desorber (10) and Mixer (14); The absorber consists of Analysers (12) and (11) and of Absorber (5); the generator consists of Analysers (2) and (13) and of Desorber (3).

Figure 2: Cooling COP and capacity as functions of the heat rejection temperature. Conversion factor: kW = 0.01757 Btu/min.

Figure 3: Variation in mass flowrate of the refrigerant, weak and strong solution with the heat rejection temperature under the cooling mode. Conversion factor: kg/min = 0.454 lb/min.

Figure 4: Variation in refrigerant concentration before and after rectification with the heat rejection temperature under the cooling mode

Figure 5: Heat duty of various components as a function of the heat rejection temperature under the cooling mode. Conversion factor: kW = 0.01757 Btu/min.

Figure 6: Cooling COP as a function of the heat rejection temperature for different refrigerant flow control schemes.

Figure 7: Heating COP and capacity as functions of the evaporator outlet temperature. Conversion factor: kW = 0.01757 Btu/min.

Figure 8: Variation in mass flowrate of the refrigerant, weak and strong solution with the evaporator outlet temperature under the heating mode. Conversion factor: kg/min = 0.454 lb/min.

Figure 9: Variation in refrigerant concentration before and after rectification with the evaporator outlet temperature under the heating mode

Figure 10: Heat duty of various components as a function of the evaporator outlet temperature under the heating mode. Conversion factor: kW = 0.01757 Btu/min.

Figure 11: Effect of deviation from equilibrium on the COP under the cooling mode. 9.0°F = 5.0°C

Figure 12: Cooling COP as a function of the flowrate in the GAX heat transfer loop. Conversion factor: kg/min = 0.454 lb/min. A qualitative description of the temperature variation in the GAX absorber, desorber and heat transfer loop is shown.

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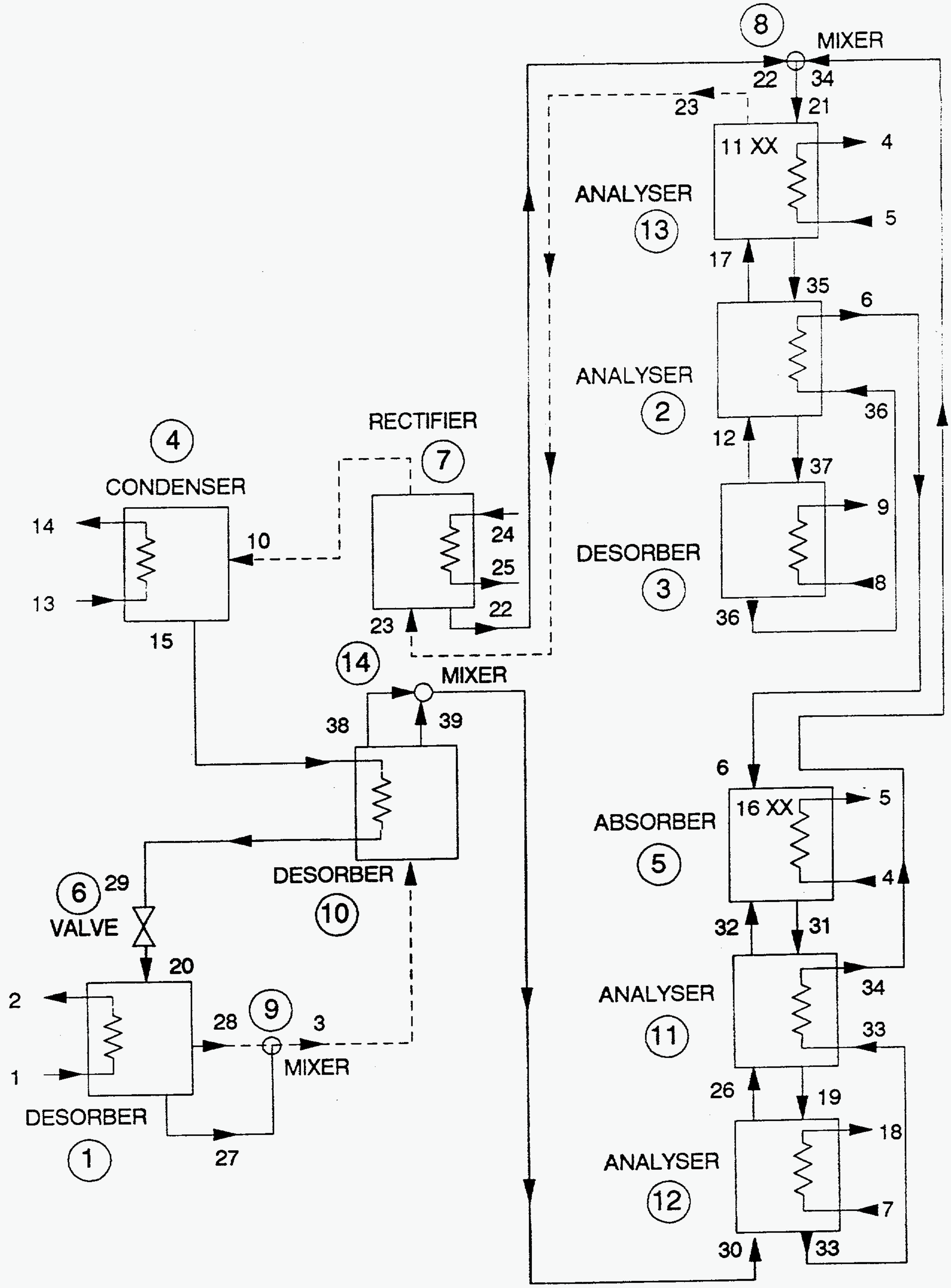

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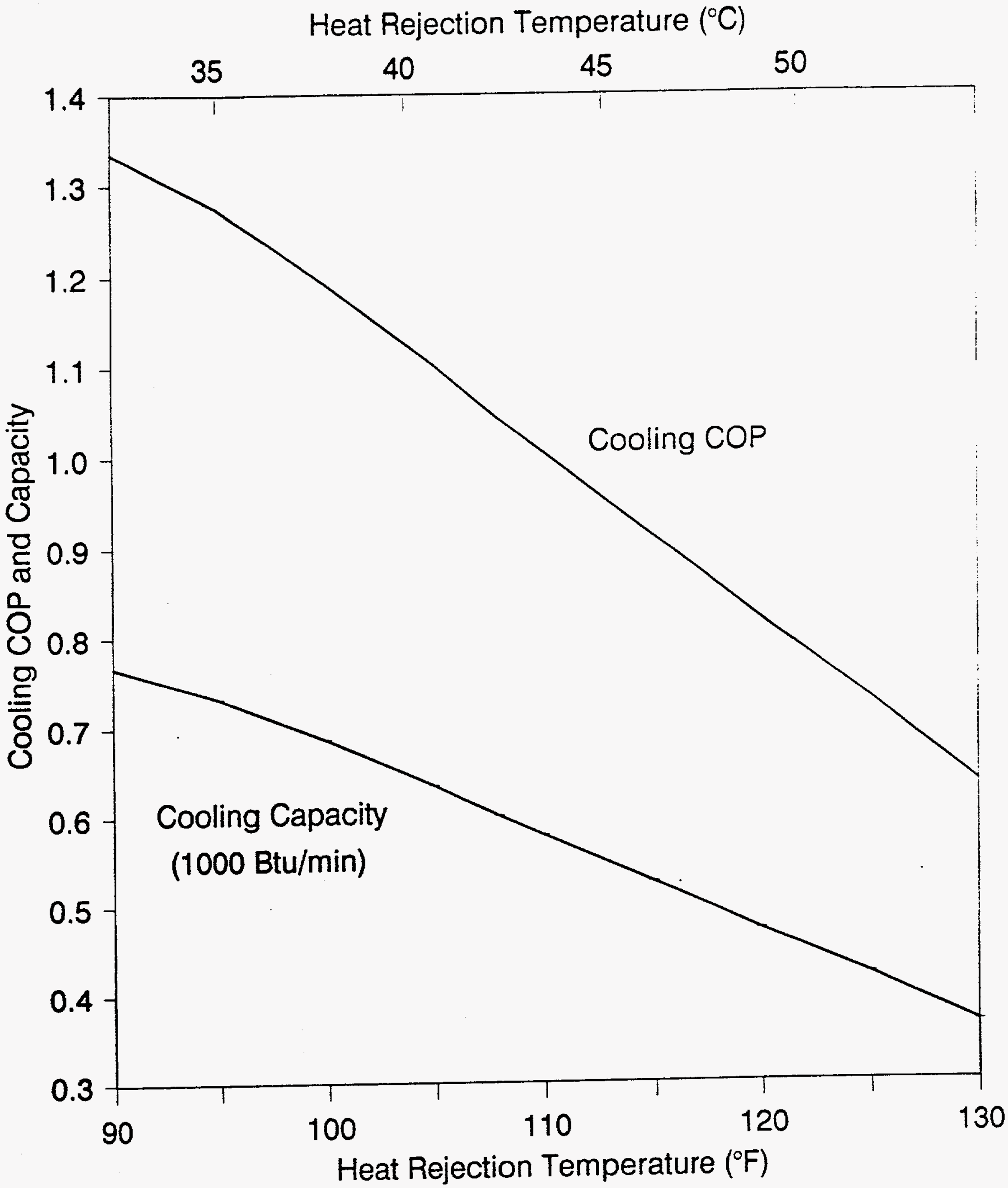

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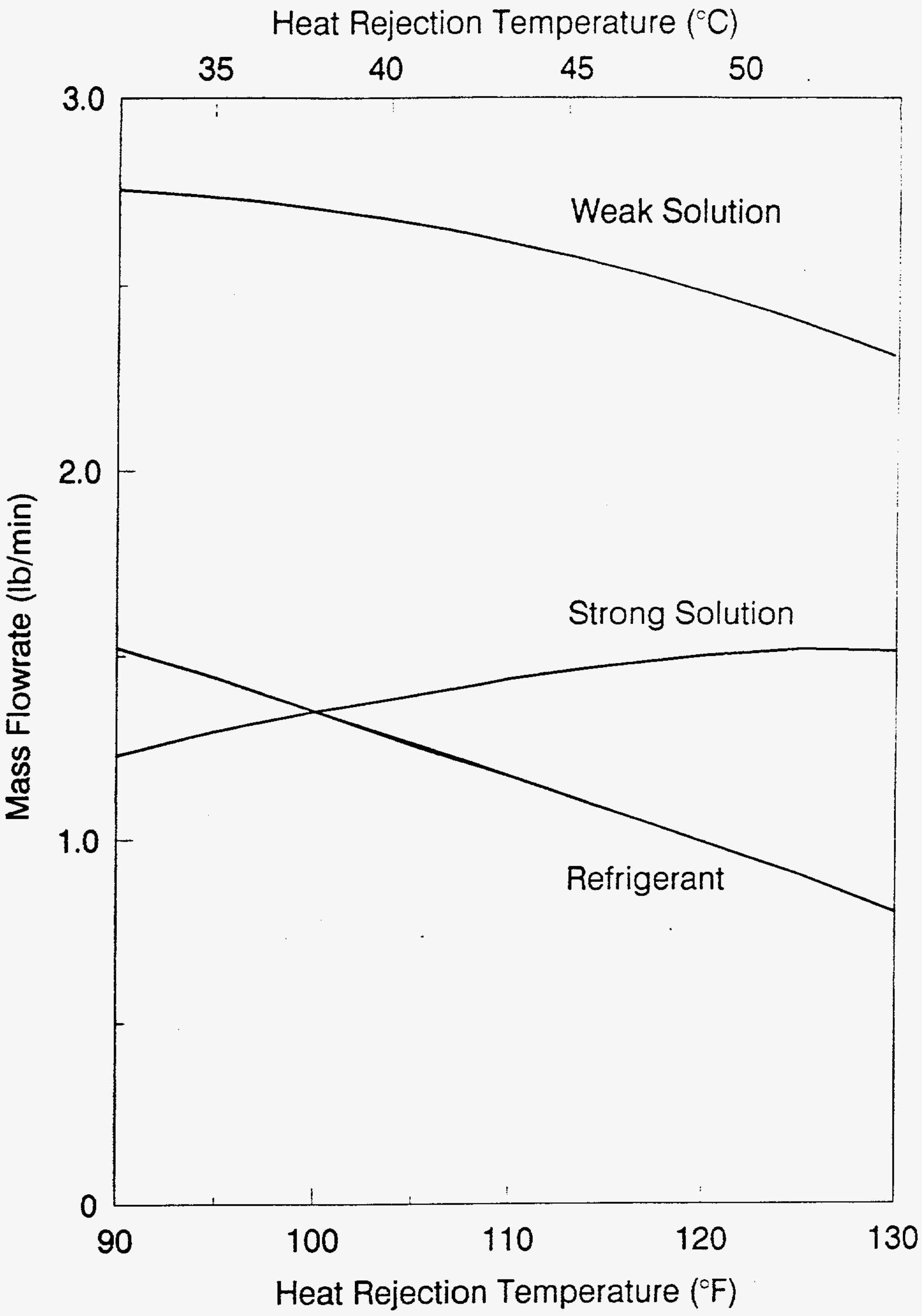

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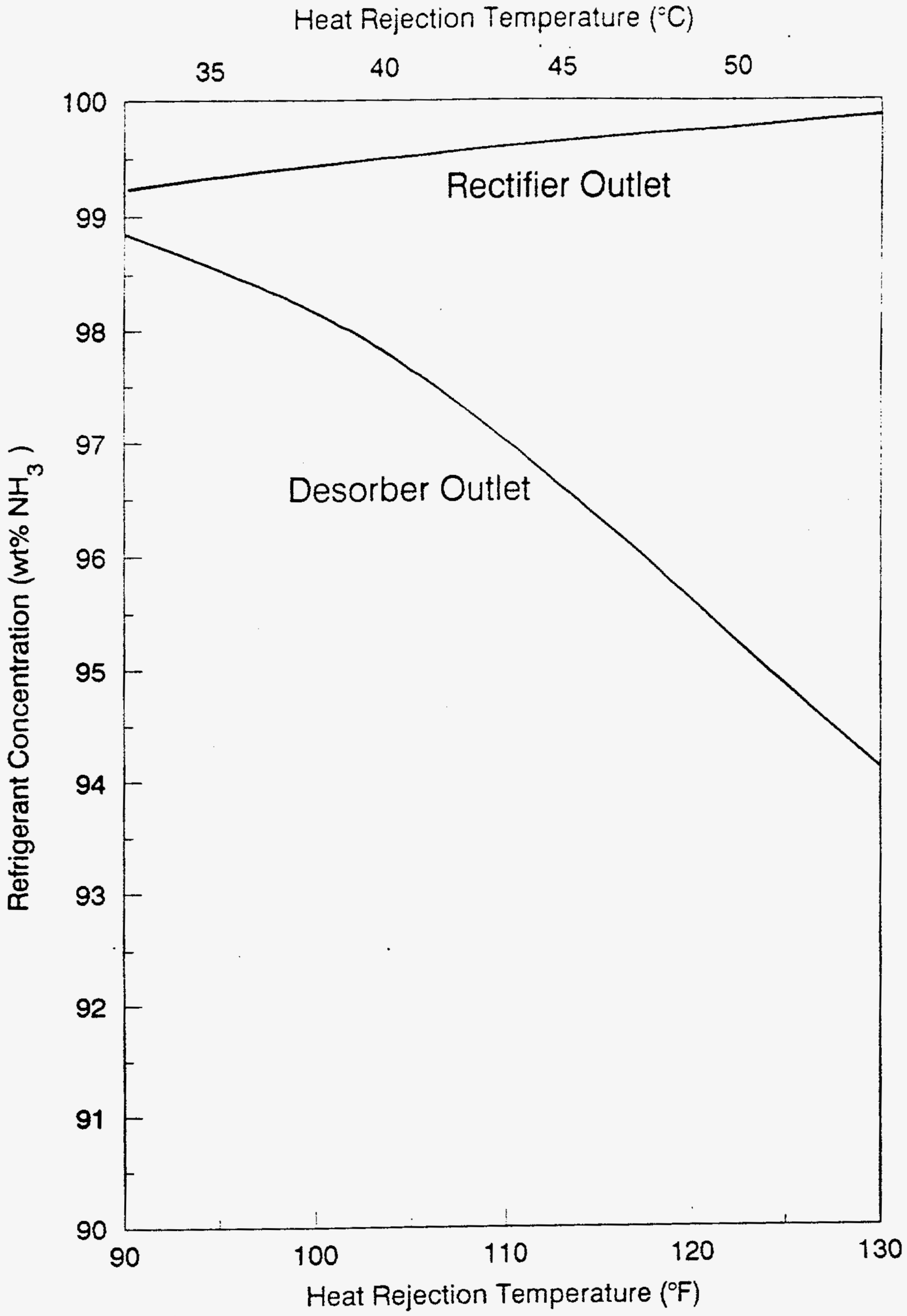

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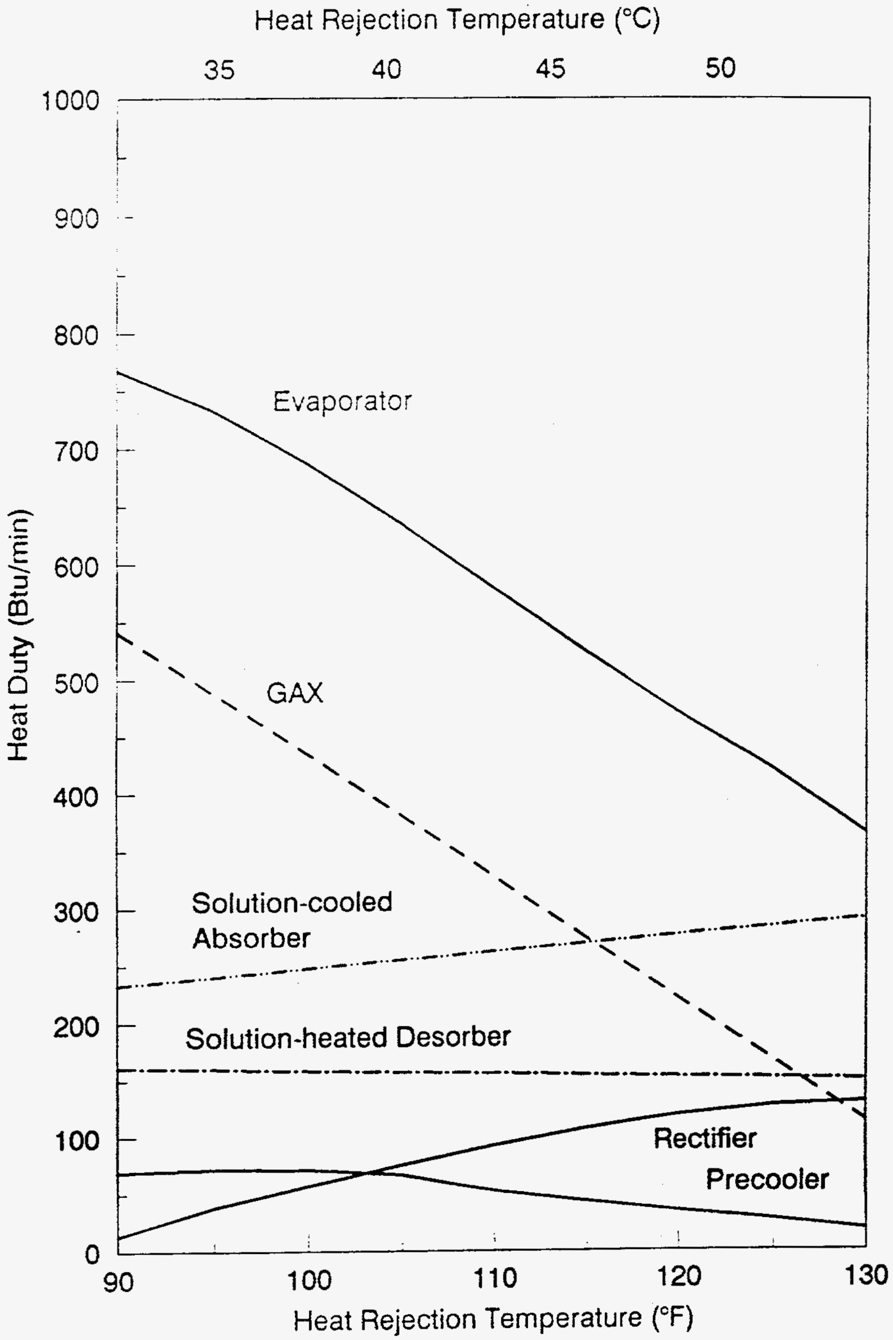

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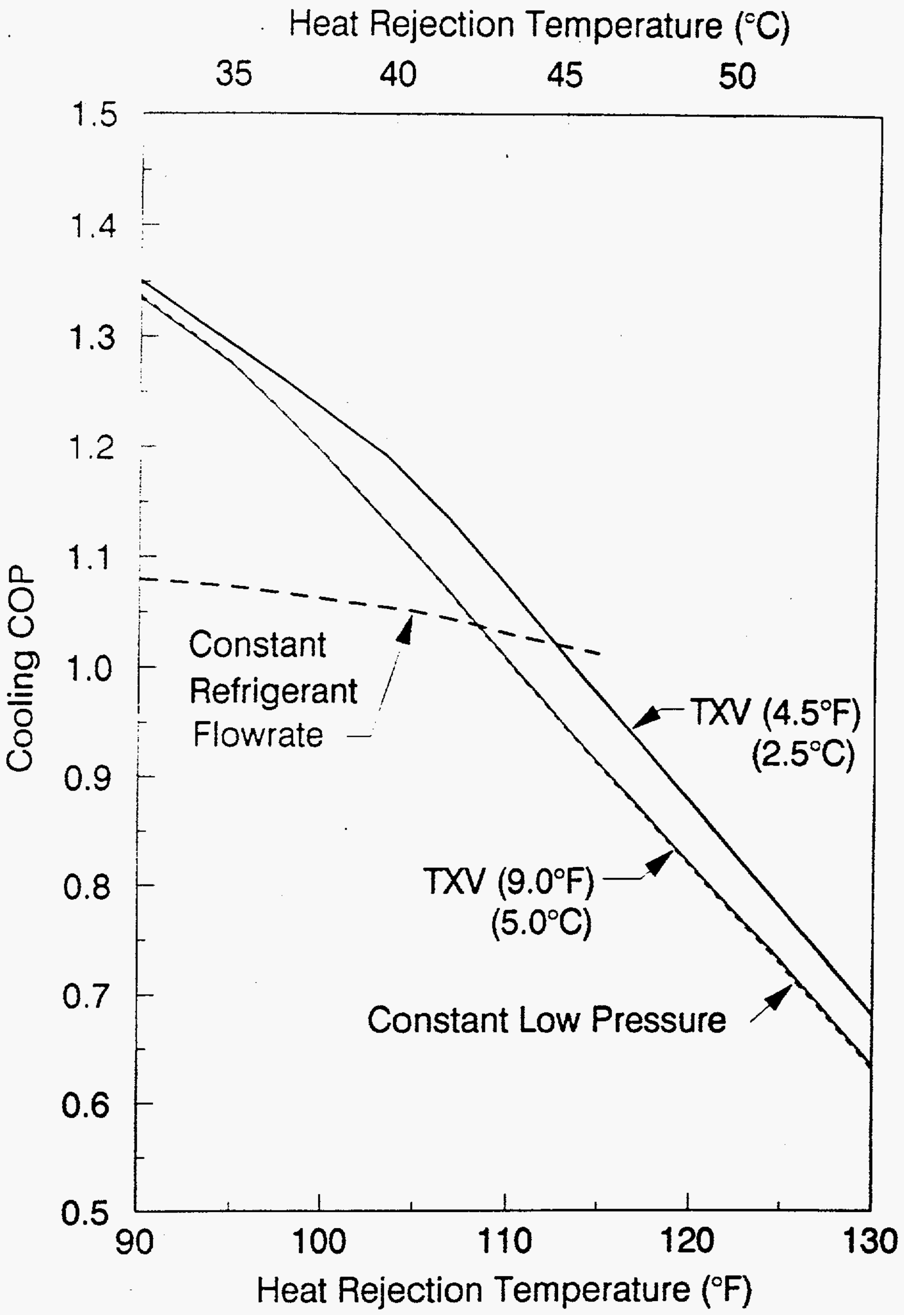

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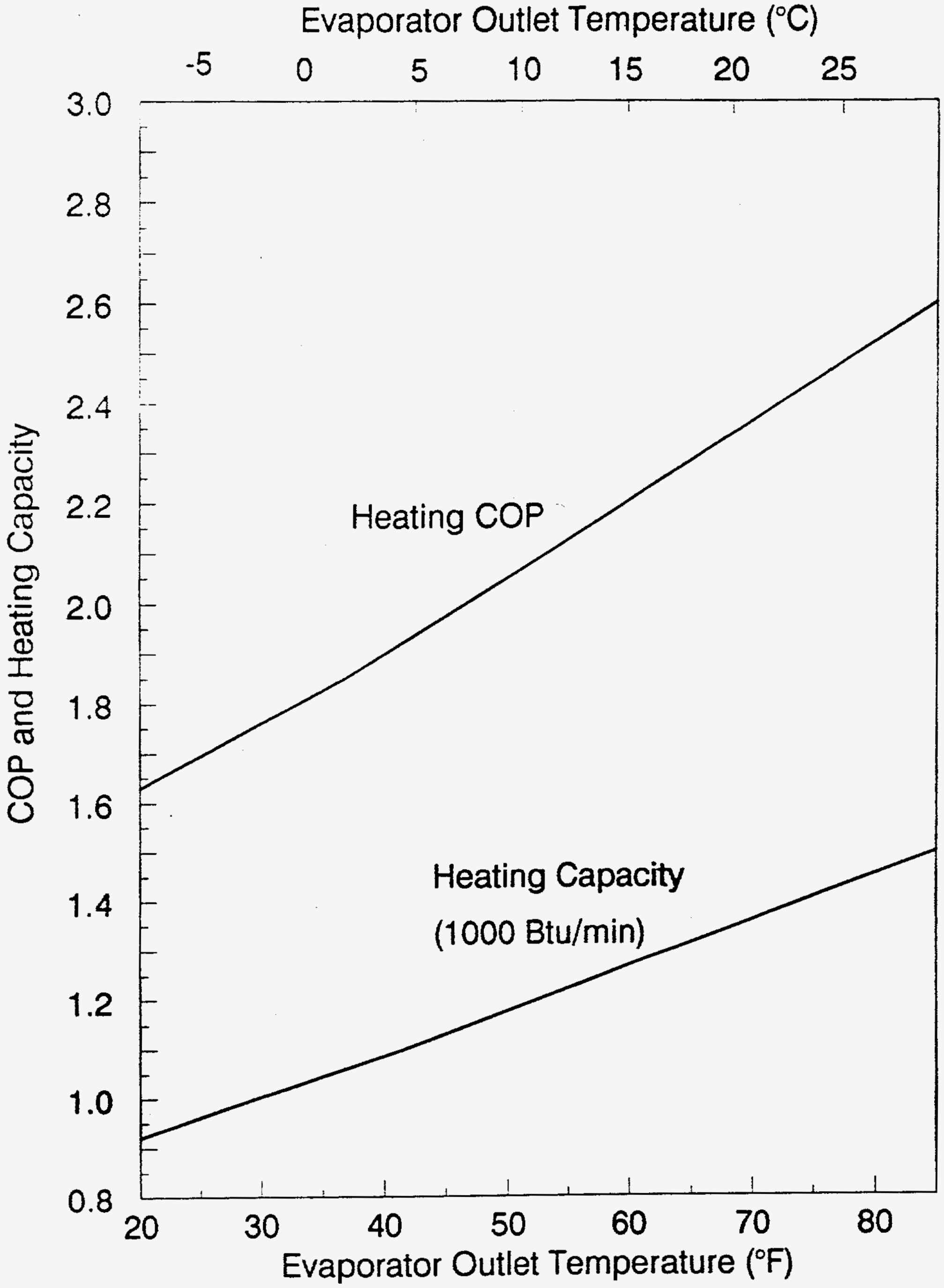

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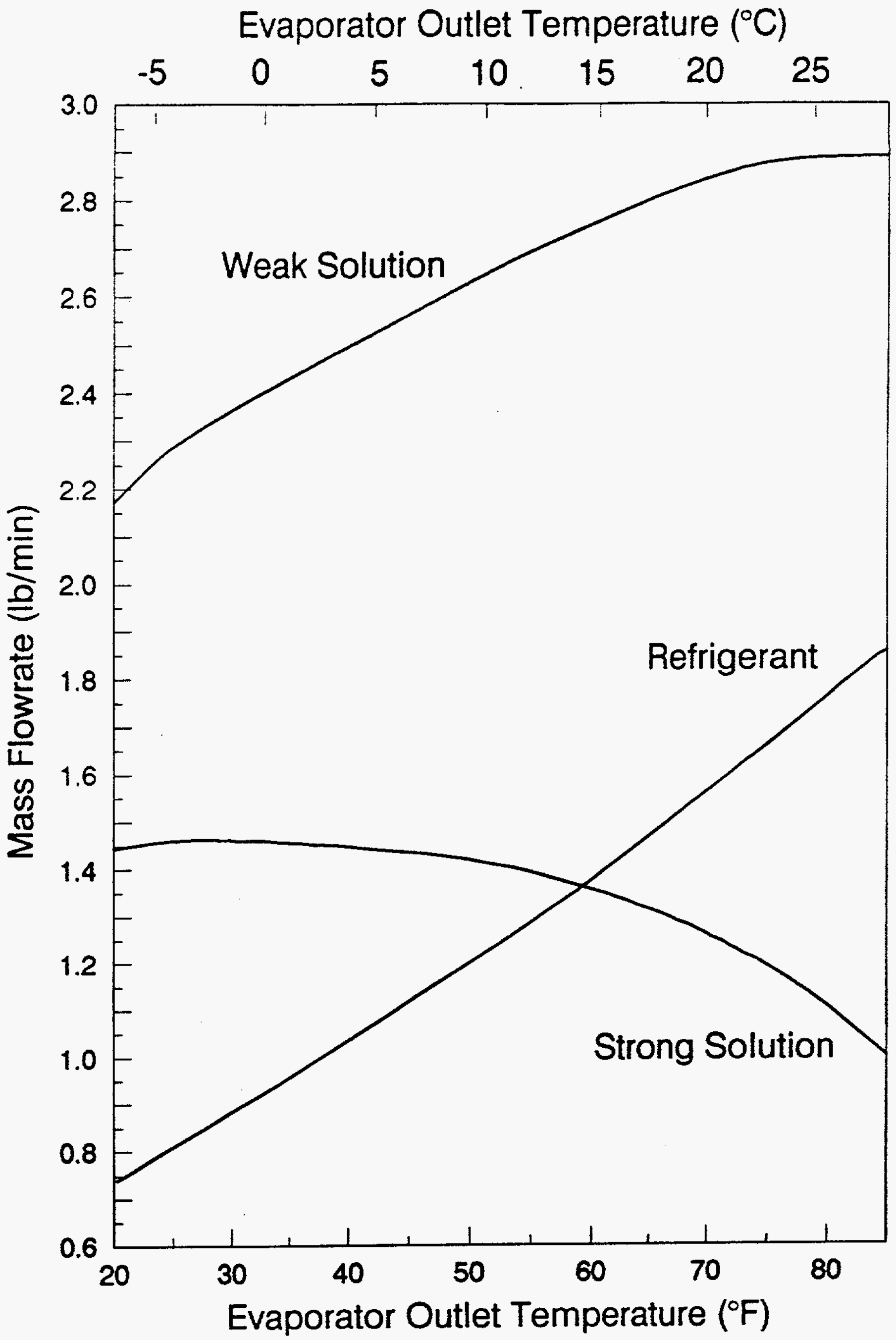

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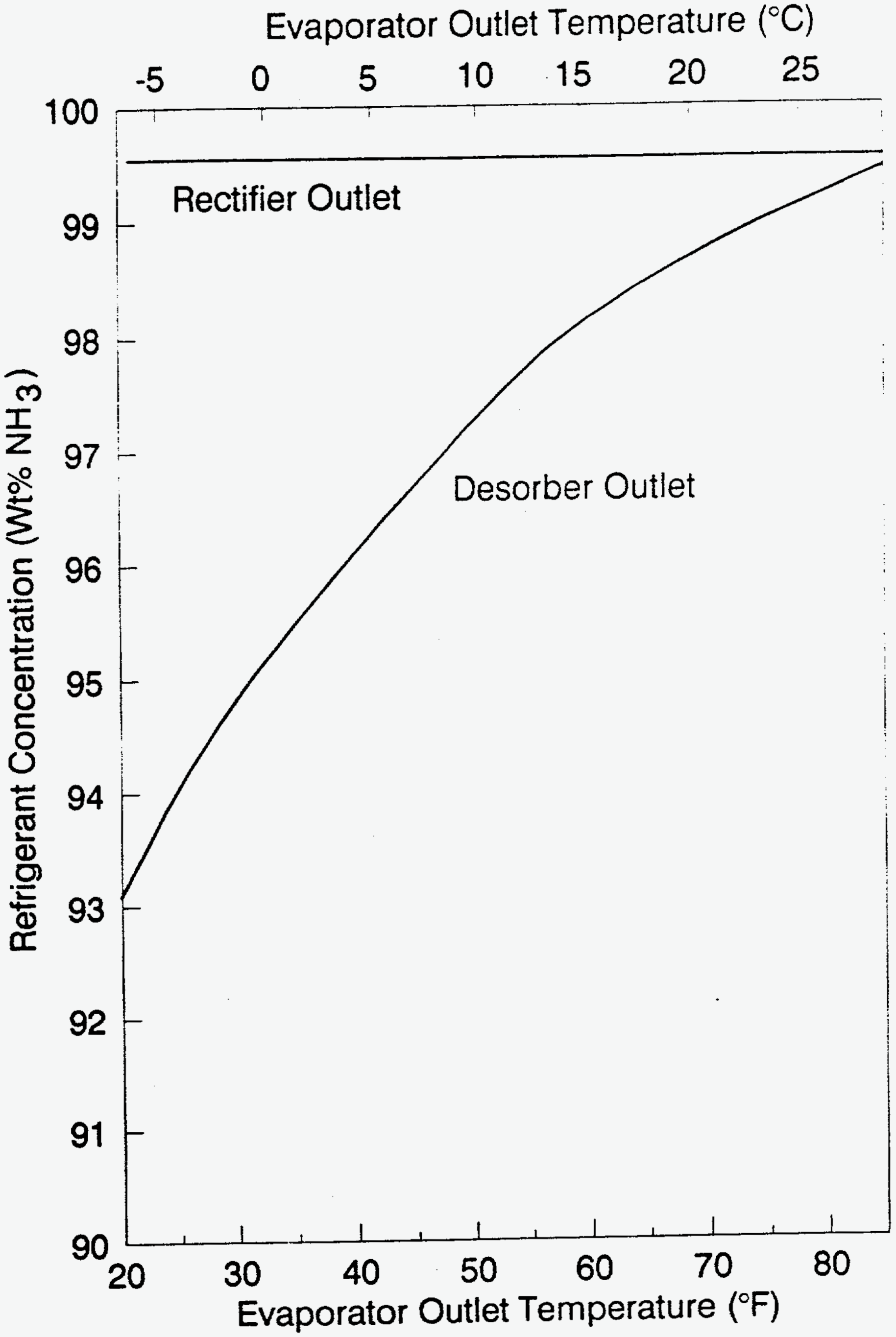

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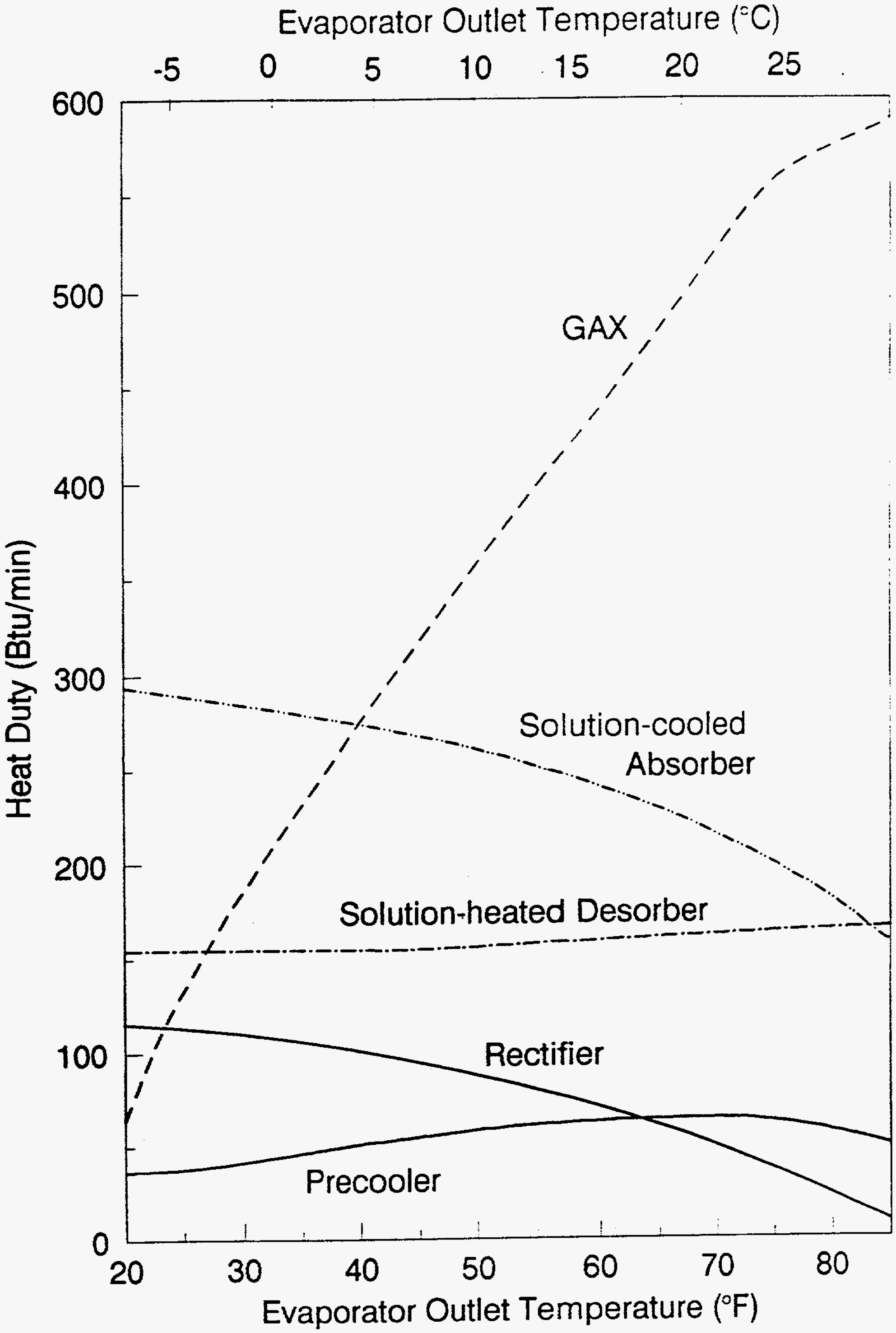

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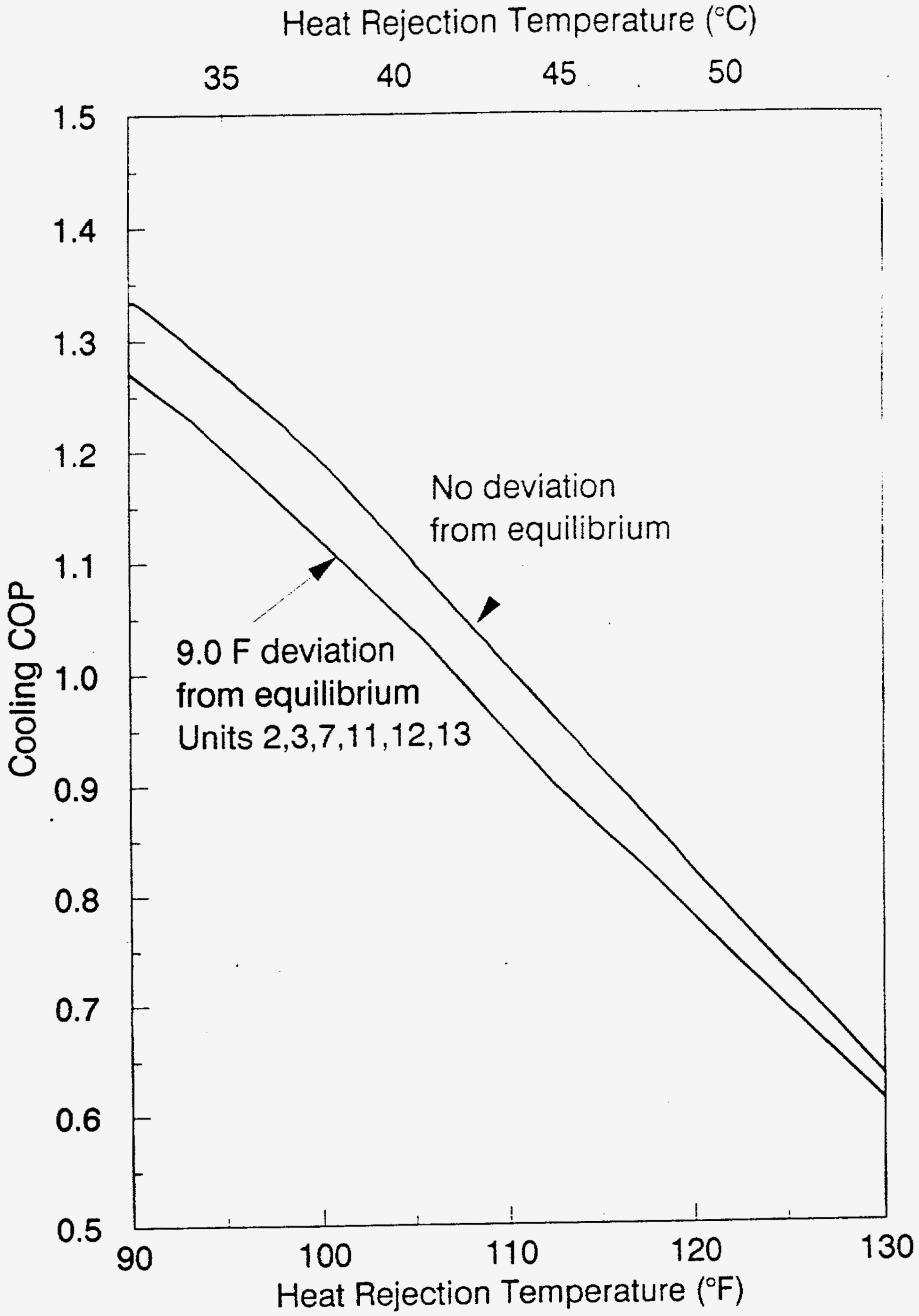

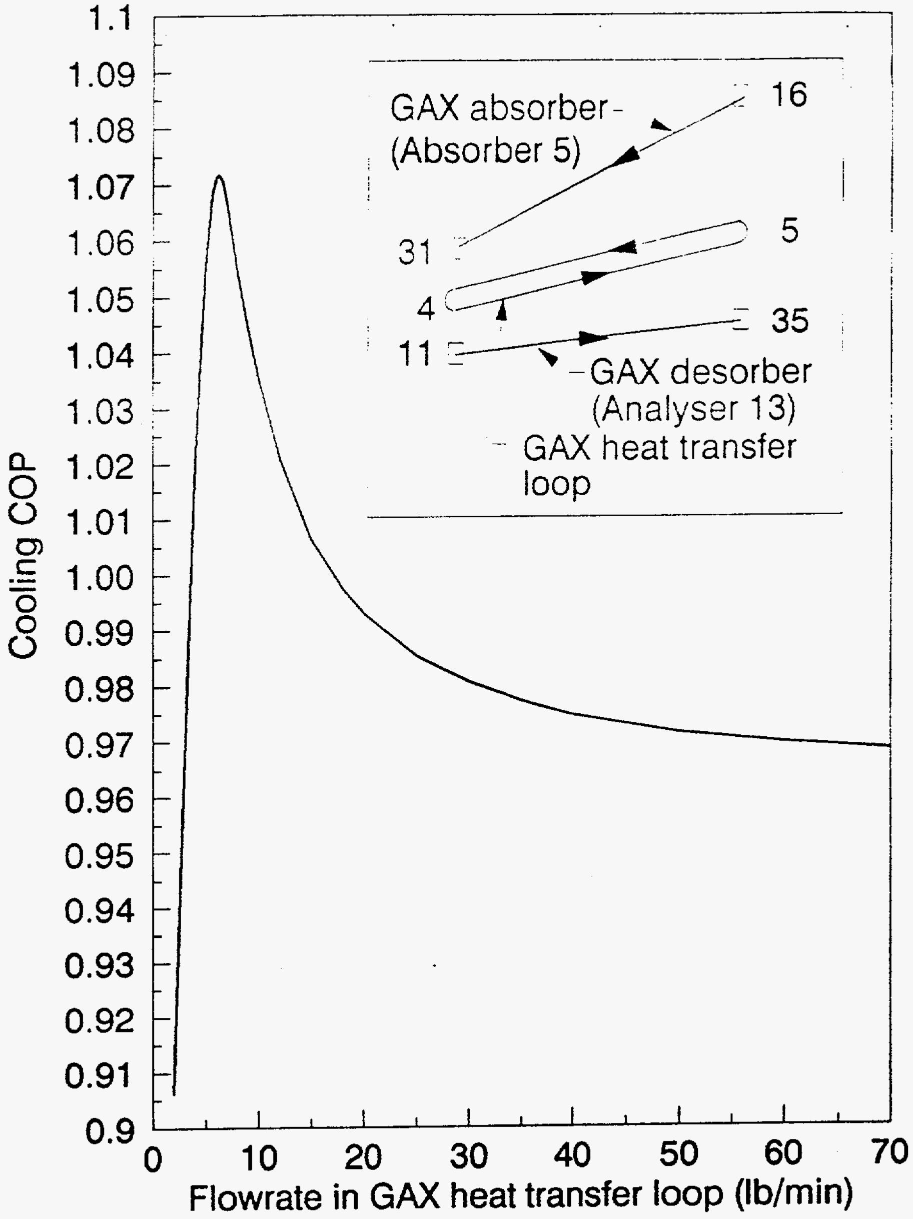ORNL-DWG 94-2056