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Open-Cycle Absorption Heat Pumps for Low-Grade Heat Utilization

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INTRODUCTION

For some time, the absorption heat pump has been considered as a possible scheme for upgrading low-temperature waste heat, which is available in very large quantities.[1] The heat pump, designed to provide a temperature boost typically from 60°C (140 F) to 120°C (250 F), thus makes low-grade heat useful for industrial and other applications. A recent study on the different means for waste heat utilization has shown the absorption heat pump to be one of the more promising among the other systems proposed for this purpose.[2] Unlike most other heat pumps, the absorption system relies on the waste heat itself as a source of power and does not require electricity or other primary energy for its operation (except for parasitics, which may be kept small).

Although the absorption cycle has been used extensively in cooling systems, relatively little work has been done on the industrial heat pump application. An experimental study of a lithium bromide absorption refrigerating and single-stage heat pump machine was carried out years ago at the Leningrad Technological Institute of the Refrigerating Industry. In the heat pump mode, which was fired exclusively with waste heat, temperature boosts of 30°C (54 F) were obtained with firing temperatures of 60°C (140 F) and cooling water at 1.4°C (34 F). A single-stage system employing an ammonia-water solution, operating with a 65°C (149 F) heat source and 15°C (59 F) sink, was built and experimented with.[3] An ammonia-water absorption-resorption heat pump developed at the French Institute of Petroleum achieved good efficiencies for modest temperature boosts [about 30°C (54 F)] at heat input temperatures above 70°C (158 F).[4] Thermodynamic and general feasibility analyses were done for other working materials and for different applications.[5,6]

In a study preceding the present one, the potential of absorption heat pumps was investigated for recovering waste heat at 60°C (140 F).[7] A complete system analysis at the conceptual design level was performed, and performance criteria were defined in terms of the operating parameters. Single- and double-stage systems were considered with lithium bromide-water (LiBr-water) and lithium chloride-water (LiCl-water) as the working materials. A preferred two-stage configuration for achieving a large temperature boost was identified, for which components and overall system performance were calculated.[5] The results have shown the effect of higher heat source or lower heat sink temperatures in increasing the temperature boost. They have also yielded the typical performance curves of the system, showing how a higher temperature boost may be traded off against a lower coefficient of performance (COP) for given operating conditions.

A typical characteristic of the heat pump operating with waste heat at 60°C (140 F) is that the heat source and heat sink temperatures are often close to each other. It was demonstrated in the earlier study that under these conditions, the capability of the system for temperature boosting deteriorates severely.[7] It is important under such conditions to minimize

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as far as possible any temperature differentials across heat exchangers within the system. This is where an open-cycle rather than a closed-cycle system may find an application. Specifically, there are two parts of the absorption heat pump where the open-cycle concept may be applied. One is the evaporator, where a flash chamber may be used in place of the common closed evaporator. The other is in the solution regeneration, where the common closed desorber/condenser is replaced by an open desorber, which utilizes air as a sink for the desorbed vapor. Both systems will be described in detail below.

This study considers the open-cycle absorption heat pump and its potential for recovering low-temperature waste heat in the absence of a much lower temperature heat sink. This would occur typically in warm, humid climates or where cooling water is not readily available. A cycle analysis is performed with suitable working materials. The performance of individual components and of the overall system is studied.

OPEN ABSORPTION HEAT PUMP CYCLES

The principles of absorption temperature boosting and the basic absorption heat pump cycle have been explained before, and it will be assumed that the reader is familiar with them.[2,7] These explanations will be repeated briefly here but only with reference to the open-cycle version of the heat pump.

Figure 1 describes a simple, one-stage absorption heat pump, similar to the more conventional closed-cycle system. Here, however, the evaporator is an open-type flash chamber, replacing the closed evaporator. The desorber is still of the closed type. In this sense, this system is only half open. The state points defined in this figure* are indicated on a LiBr-water equilibrium diagram in Fig. 2. Clean wastewater at temperature T_i (state 10) enters the heat pump and is split into two streams. One stream enters the flash chamber (state 1) where part of it is converted into vapor (state 8), and the rest leaves at a somewhat lower temperature (state 2). The vapor is absorbed in the absorber at a temperature (state 4) higher than T_i as determined by the equilibrium conditions of the solution (Fig. 2). The heat of absorption is transferred to the stream of wastewater entering at state 3 (temperature T_i) and serves to raise its temperature to T_f at state 9. To achieve that, a concentrated absorbent solution is supplied to the absorber at state 7, which leaves at a more dilute state 4. A recuperative heat exchanger is provided to transfer heat from the dilute solution, leaving the absorber to the concentrated one on its way in. Thus, the latter can be preheated and better efficiency is achieved.

The rest of the system is designed to reconcentrate the solution before it returns to the absorber. In Fig. 1, a closed desorber/condenser of the conventional type used in absorption chillers is shown. Waste heat at state 20 (temperature T_i) is applied through a heat exchanger to boil off the water from the dilute solution entering at state 5 and to concentrate it to state 6. The desorbed vapor is condensed in the condenser to which cooling water at state 23 (temperature T_s) is supplied. The vapor pressure in the desorber/condenser is determined by this cooling water, and the cooler it is, the higher the concentration achieved at state 6. The condensate (22) and the wastewater (2 and 21) may be returned to the process directly or via a cooling tower.

The thermodynamic diagram of Fig. 2 illustrates the quantitative effects of the factors influencing the magnitude of the temperature boost. (Clearly, it is desirable to have a solution concentration as large as possible.) These are determined partly by the waste heat temperature T_i, which should ideally be as high as possible, and partly by the heat sink temperature T_s, which should be as low as possible.

For a given waste heat temperature, the effect of the cooling water temperature on the available temperature boost is rather drastic. In Tab. A1, the mass flow, temperature, and concentrations at various locations are illustrated for a cooling water temperature of 8.3°C (47 F) and a firing temperature of 60°C (140 F). A temperature boost of the output stream of 16°C (28.8 F), from 60°C (140 F) to 75°C (168.8 F), is possible. If the cooling water temperature increases to 29.4°C (85 F) and the heat exchanger effectiveness is conserved, the temperature boost is reduced to 9.1C° (16.4F). The state points corresponding to the cases mentioned above are shown in Fig. 2 and identified with roman numerals I and II, respectively.

*A state point numbering system is used in a consistent manner for all the heat pump diagrams in this report.

The vapor pressure of the steam generated in the flash chamber is of great importance in determining the temperature boost too. For instance (Fig. 2), if the low concentration is 55% and if saturated steam at 60°C (140 F) and 20 kPa (2.9 psia) is available, the equilibrium temperature at the desorber exit is 97.8°C (208 F). If the steam temperature and pressure are 54.4°C (130 F) and 16.3 kPa (2.2 psia), respectively, then the equilibrium temperature will be 90°C (194 F). A flash chamber is more effective in providing high-pressure steam than is a closed evaporator, since no temperature drop across heat exchangers is needed to effect heat transfer. The temperature boost possible with a flash chamber is, therefore, larger than the one possible with a closed evaporator. In a flash chamber, the following advantages are realized over a closed evaporator:

1. there is a closer thermal equilibrium between the steam produced and the outgoing water,

2. the temperature-boosting effect is enhanced, and

3. metallic heat exchanger surfaces are unnecessary.

The principal drawback of a flash chamber with a small temperature drop is the need to provide both large interfacial areas for evaporation and long liquid residence times. This may produce a somewhat cumbersome design. Also, if large amounts of air are dissolved in the wastewater, it is necessary to install a continuous purge for the removal of this air. This may increase the parasitic power required for cycle operation.

From the above considerations, it appears that temperature boosts between 20°C (36 F) in winter and 10°C (18 F) in summer are possible with the cycle outlined above when it is intended to recover and upgrade heat at 60°C (140 F). As shown in another study, it may be desirable in many industrial sectors to provide process heat year-round at temperatures close to 100°C (212 F) or above.[2] In order to enhance heat recovery, it appears desirable to explore absorption systems that can, in principle, achieve larger temperature boosts. Thus, the COP is considered of secondary importance, since the waste heat at low temperature levels is not only free but, in most cases, entails an additional cost of disposal.

As mentioned before, the temperature boost available from a waste-heat-fired cycle may be amplified by reducing the temperature of the cooling water. This is equivalent to reducing the vapor pressure in the desorber/ condenser shell. There is, however, another configuration to reduce the vapor pressure against which the solution must desorb. The fact that the water vapor pressure in atmospheric air is usually low is taken advantage of in this configuration. The solution is regenerated with waste heat by evaporating the water into a stream of ambient air.[8,9] This concept of open desorption offers some advantages and drawbacks that will be discussed below.

Another possible way to increase the temperature boost is through a two-stage cycle. Although several ways exist to build the system of Fig. 1 in two stages, a complete discussion of those possibilities is beyond the scope of this paper. A preferred two-stage configuration has been identified,[7] which is illustrated in Fig. 3. In this configuration, there are two absorber/evaporator stages and one desorber serving both. The two stages operate between the same high and low concentrations, and the temperature-boosted stream from the first stage is boosted further in the second. This preferred two-stage configuration has less hardware and operates further from crystallization than other staging configurations designed to provide the same temperature boost.

Figure 3 shows the two concepts (open desorber and multistaging) referred to above, combined in a single system. The systems of Figs. 1 and 3 are similar, with the latter having an additional evaporator/absorber and recuperator. The temperature-boosted stream from the first stage (state 9) is split into two streams entering the second stage. At state 11, one stream is flashed to provide vapor at state 18, which is absorbed in the second-stage absorber to boost the temperature of state 13. The second evaporator outlet (state 12), which contains useful heat at a temperature higher than T_1, is returned to the first stage. The dilute and concentrated solution streams of both absorbers are combined in the single desorber.

The open desorber uses ambient air as a sink for the vapor desorbed from the solution. Waste heat at temperature T_1 is applied through a heat exchanger to boil off the water from the dilute solution (state 25) in much the same way as in the closed desorber. The desorbed vapor is taken up by a stream of air entering at state 22 and leaving at state 27. Because the air comes in contact with the hot solution, it picks up some heat along with the vapor.

An air recuperator is therefore provided, as shown, to recover heat from the exiting air stream and use it to preheat the inlet air. This recuperator tends to increase the COP. (State points and mass flows obtained from the calculations described below for this system are illustrated in Tab. A2.)

The concept of open desorption offers the following advantages over closed desorption:

1. Lower waste heat temperature may be used when cold enough cooling water is not available.

2. Because no cooling water is needed, cooling tower equipment, piping, and parasitic power are eliminated.

3. No condensing surfaces are needed, which eliminates the metallic area of the condenser and also its auxiliaries (controls, pumps, purges, etc.).

This concept has two main drawbacks:

1. As the solution exchanges mass with the air, heat is also transferred to it. In order to preserve high levels of thermal efficiency, it is necessary to install an air-to-air heat recuperator. This device may recover both the sensible and part of the latent heat transferred to the air, but it increases capital costs and the parasitic power necessary to handle large air volumes.

2. Exposing a LiCl- or LiBr-water solution to the air causes oxygen to dissolve into the solution. This may cause corrosion problems, which call for the use of corrosion-resistant materials and/or of corrosion inhibitors.

The open-cycle heat pump may take on different configurations, as indicated before. This study will focus on the two-stage cycle with flash evaporators and an open desorber shown in Fig. 3. As outlined above, large temperature boosts may be obtained with this configuration. The wastewater temperature will be taken equal to 60°C (140°F) unless otherwise indicated. Other relevant parameters will be varied to calculate performance for changing conditions.

WORKING FLUIDS SELECTION

The properties required of the working materials for absorption systems have been discussed extensively in the literature, mainly for cooling systems.[10,11] The particular requirements for a heat pump application also have been described.[7] In open-cycle systems, the choice of the refrigerant as water is obvious. It therefore remains to select a proper absorbent, keeping in mind its continuous contact with air in the desorber.

The important characteristics of the absorbent may be summarized as follows:

1. The equilibrium temperature of the solution in contact with vapor at a saturation temperature equal to that of the waste heat should be as high as possible to provide for a maximum temperature boost. Conversely, the solution equilibrium temperature should be as low as possible in contact with vapor at the sink conditions for ease of desorption. Generally, for a real material, these two conflicting requirements must be compromised.

2. The heat of dilution (which together with the water's latent heat of condensation forms the heat of absorption) should be as high as possible at the absorber conditions and as low as possible, or negative, in the desorber. Again, for a real material, there is a conflict between these two conditions.

3. The water content of the solution at the operating concentration should be as high as possible to avoid circulation of large solution quantities, which results in excessive internal heat losses.

4. The properties of the absorbent-water solution that affect heat and mass transfer, such as viscosity, diffusion coefficient, thermal conductivity, etc., should be favorable.

5. The absorbent should be nonvolatile — an absolute must in open systems.

6. The saturation concentration of the absorbent/refrigerant solution should be as high as possible and beyond the range of operating concentrations of the system at the given temperature.

7. The working fluids should be chemically stable, nonflammable, nontoxic, and noncorrosive to common materials of construction. These requirements are more difficult to satisfy in open systems than in closed ones due to the contact with air.

Open-cycle absorption systems built for cooling purposes have employed various absorbents including lithium bromide, lithium chloride, calcium chloride, ethylene glycol, and several solid absorbents of water.[12] Of the many candidate materials, two emerge as being most suitable for the heat pump system. These are the lithium bromide and lithium chloride absorbents; both are nonvolatile and satisfy most of the above criteria. Another possible candidate is calcium chloride, which, although an inferior absorbent as compared to the other two, is less expensive and less corrosive than the lithium salts.

The present study will consider LiBr-water and LiCl-water as working materials for the open-cycle heat pump system. Figure 2 shows a typical equilibrium diagram for the LiBr-water combination, based on data from more than 15 sources compiled recently by McNeely.[13] Similar data for LiCl-water have been provided.[14,15] Enthalpy data were also obtained from those references.

THE OPEN DESORBER

In an open desorber, a two-phase, three-component heat and mass transfer process occurs. A simplified model (Fig. 4) was formulated to calculate the performance of this component. In this model, the dilute solution (state 25) enters the desorber, where water evaporates from it in direct contact with the air, which flows countercurrent to the solution. Waste heat is added in counterflow to the solution in order to keep both its temperature and vapor pressure high so that desorption can proceed. If heat was not added during this process, the solution temperature and vapor pressure would decrease rapidly until equilibrium with the air vapor pressure would be reached. At this point, all mass exchange ceases. The situation idealized in this model may be achieved in practice in a packed tower with heat-exchanging coils immersed in the packing or in a falling film reactor with counterflow of air.

In order to calculate performance, the following assumptions were made:

1. The solution vapor pressure is maintained constant during desorption.

2. The resistance to mass transfer is concentrated in the air phase and is negligible in the solution. This is true, in practice, for thin well-mixed films.*

3. The turbulent transport of water vapor and the heat transfer in the air phase are related by a Lewis number equal to one.

Under these assumptions, it is possible to derive a relation between the air temperatures and humidities at the inlet and outlet of the desorber (see appendix). Then, the following equations may be written to describe the process in the open desorber system:

Salt mass conservation:

$$m_{25}C_L = m_{26}C_H \ . \tag{1}$$

Water mass balance:

$$m_{23}(W_{27} - W_{22})/(1 + W_{23}) = m_{25} - m_{26} \ . \tag{2}$$

Mass exchange effectiveness:

$$\epsilon_m = (W_{27} - W_{22})/(W_1 - W_{22}) \ . \tag{3}$$

*In a desorber where the solution is not thin, both a temperature and concentration gradient exist across the film. The solution of the problem under those conditions becomes considerably more difficult. The performance results of the present solution should be regarded as an upper limit for an actual open desorber.

Solution exit temperature:

$$T_{26} = T_{20} - \Delta T \ . \tag{4}$$

Equilibrium conditions:

$$W_i = f_1(T_{26}, C_H) \ ; \quad T_{25} = f_2(C_L, W_i) \ . \tag{5), (6}$$

From the Lewis analogy:

$$\frac{dT}{dW} = (W_i - W)/(T_i - T) \ . \tag{7}$$

From a mass balance:

$$\frac{dC}{dW} = \frac{M_{22}}{(1 + W_{22})M_{25}} \frac{C^2}{C_L} \ . \tag{8}$$

Local equilibrium condition:

$$T_i = f_2(C, W_i) \ . \tag{9}$$

Air recuperator effectiveness:

$$T_{27} = T_{23} + (T_{22} - T_{23})/\varepsilon_r \ . \tag{10}$$

From a heat balance around the recuperator (assuming no change in specific heat due to humidity pickup and neglecting air mass changes):

$$T_{24} = T_{27} - T_{22} + T_{23} \ . \tag{11}$$

Overall heat balance:

$$\Delta H = m_{23}(h_{24} - h_{23})/(1 + W_{23}) + m_{26}h_{26} - m_{25}h_{25} \ . \tag{12}$$

In the above equations, m stands for mass flow, C for salt concentration (C_H for high concentration, C_L for low concentration), W for humidity ratio, ε for effectiveness, T for temperature, and h for specific enthalpy. The dependence of the equilibrium solution vapor pressure on temperature and concentration is denoted by f_1. The equilibrium temperature of the solution with concentration C_L and vapor pressure corresponding to W_i is given by f_2. The symbol T_i denotes the solution interface temperature and W_i the humidity ratio corresponding to the solution interface vapor pressure. ΔT denotes the temperature difference between the solution and the wastewater at the bottom of the desorber and ΔH the heat delivered to the solution and to the air in the desorber.

Equations 7 and 8 have the initial conditions:

$$T = T_{22} \text{ and } C = C_H \quad \text{at } W = W_{22} \tag{13}$$

and they may be integrated, varying W as an independent variable from $W = W_{22}$ to $W = W_{27}$, to yield the value of T_{27}.

Equations 1 through 12 include 26 unknowns (W is an independent variable, taking arbitrary values in the interval $W_{22} \leq W \leq W_{27}$). By choosing m_{25} equal to unity, the high and low concentrations C_H and C_L, the initial air conditions W_{23} and T_{23}, the heat and mass transfer effectiveness, the waste heat temperature T_{20}, and the approach temperature ΔT, the number of unknowns is reduced to 17. Taking into account the additional relations

$$W_{22} = W_{23} \ , \tag{14}$$

$$h_{24} = f(W_{24}, T_{24}) \ , \tag{15}$$

$$h_{23} = f(W_{23}, T_{23}) \ , \tag{16}$$

$$h_{26} = f(C_L, T_{26}) \ , \tag{17}$$

$$h_{25} = f(C_H, T_{25}) , \tag{18}$$

the system may be solved, since it exhibits 17 equations Eqs 1-12 and 14-18 with 17 unknowns.

A computer code was developed to effect this solution. Differential Eqs 7 and 8, together with Eq 10, were solved in the following way. A value of T_{22} within a specified interval was adopted. Equations 7 and 8 were then solved using a fifth-order Runge-Kutta method with the initial conditions of Eq 13, to yield a value for T_{27}. If this value for T_{27} was within 1% of the value of T_{27} given by Eq 10, the program kept on solving the remaining equations. Otherwise, a new value of T_{22} was selected by means of a combination of a binominal and Newton-Raphson technique, until convergence was achieved.

The desorber performance using a LiBr-water solution is illustrated in Fig. 5. The air-to-solution mass ratio (m_{23}/m_{25}) is shown for several inlet air humidity ratios, as a function of the desorber approach temperature ΔT, for a concentration change from 52 to 54%. It is seen that as the approach temperature increases, the solution vapor pressure decreases, calling for larger mass ratios in order to complete desorption. The heat delivered to the desorber varies with the recuperator effectiveness and with the approach temperature as depicted in Fig. 5b, for an inlet air humidity ratio of 0.008 (curve indicated by a solid line in 5a). From the above, it is possible to conclude that if it is desired to minimize the desorber heat load, it is necessary to provide a high effectivness recuperator. For example, for an effectiveness of 0.7, the approach temperature and inlet humidity ratio determine primarily the amount of air to be circulated but do not greatly influence the heat load. Minimizing the desorber heat load may not be the objective in those cases where disposing of the waste heat is the primary objective. In those instances, a small recuperator may then be used.

The same calculations were repeated for LiCl with similar results.

OVERALL CYCLE PERFORMANCE

In order to calculate the overall cycle performance, the mathematical equations describing the staged absorbers and recuperators must be formulated. For this purpose, it is assumed that (1) the weak solution leaving the absorbers is in vapor pressure equilibrium with the vapor coming from the evaporators, (2) the water and the steam leaving the evaporator are in thermal equilibrium, and (3) pressure drops in the system are negligible.

For the nomenclature indicated in Fig. 3 and for the first stage evaporator/absorber, the following equations may be written:

1. In the evaporator:

Mass balance:

$$m_8 = m_1 - m_2 . \tag{19}$$

Energy balance:

$$m_8 h_8 = m_1 c_p T_1 - m_2 c_p T_2 ; \tag{20}$$

$$T_2 = T_8 . \tag{21}$$

2. In the absorber:

Equilibrium:

$$T_4 = f(T_8, C_L) . \tag{22}$$

Salt mass conservation:

$$m_4 C_L = m_7 C_H . \tag{23}$$

Mass balance:

$$m_8 = m_4 - m_7 . \tag{24}$$

Energy balance:

$$m_3 c_p (T_9 - T_3) = m_7 h_7 + m_8 h_8 - m_4 h_4 \; ; \tag{25}$$

$$m_3 = m_9 \; . \tag{26}$$

Absorber effectiveness:

$$\varepsilon_a = (T_9 - T_3)/(T_4 - T_3) \; . \tag{27}$$

3. In the recuperator:

$$m_4 = m_5 \; ; \tag{28}$$

$$m_6 = m_7 \; . \tag{29}$$

Recuperator effectiveness:

$$\varepsilon_r = (T_7 - T_6)/(T_4 - T_6) \; . \tag{30}$$

Heat balance:

$$m_4 h_4 - m_5 h_5 = m_7 h_7 - m_6 h_6 \; . \tag{31}$$

Here, m, h, and T indicate a mass flow rate, enthalpy, and temperature of the working material at the different states, respectively, and c_p is the specific heat of water. In these calculations, 1% of the incoming wastewater is flashed in the flash chamber in both stages. The above system may be solved once the high and low concentrations, the heat exchanger effectiveness, the waste heat temperature, the desorber approach temperature, and the mass of delivered hot water are chosen. A similar set of equations may be written for the second stage.

Equations 1 through 31 define the performance of the system. A computer code of these equations was written to calculate cycle performance for varying conditions. This code can calculate cycle performance for either closed or open desorbers and for LiBr- or LiCl-water solutions.

As mentioned before, for a given waste heat temperature, the temperature boost that may be achieved with an absorption cycle depends on the vapor pressure against which the solution must desorb. In a closed cycle, this vapor pressure is determined by the condensation temperature, which is in turn determined by the cooling water temperature. In an open cycle, this vapor pressure is determined by the water vapor pressure in the ambient air. If the cooling water for a closed-cycle machine is obtained from a cooling tower, then the condenser pressure is also determined by the air vapor pressure. Therefore, in this case, the conditions of atmospheric air set the magnitude of the temperature boost. Since in both closed and open cycles air must be circulated to complete the desorption step, it is of interest to compare the unit of heat output of the heat pump per kilogram of air circulated.

This comparison is shown in Fig. 6 for a two-stage absorption cycle with air at a dry-bulb temperature of 26.6°C (80 F) and a wet-bulb temperature of 19.7°C (67 F). The temperature boost and the amount of air circulated per unit of heat output are shown versus the desorber approach temperature for both a closed and an open system. A counterflow, induced-draft cooling tower was assumed for providing the cooling water to the closed system with performance calculated from Ref 16. The cooling tower performance is defined by the approach to the wet-bulb temperature, the hot water temperature, and the amount of water sprayed on the tower per unit time and unit of frontal area, also known as the water concentration. A wet-bulb approach of 2.7°C (5 F) was selected, with an air-to-water mass ratio of 1.1. Under these conditions, increasing the hot water temperature increases the cooling tower capacity per unit of frontal area, but the water concentration must decrease correspondingly in order to maintain the 2.7°C (5 F) approach. Thus, increasing cooling tower capacity from 28 kJ/m^2 (2.5 Btu/ft^2) to 47.3 kJ/m^2 (4.2 Btu/ft^2) calls for less water circulation and less frontal area, but means higher condensing temperatures. It is seen in Fig. 6 from the curves for the open cycle and for the closed cycle with a cooling tower of 28 kJ/m^2 (2.5 Btu/ft^2) that for roughly similar temperature boosts and heat output capacities, it is necessary to circulate about 40% more air in the cooling tower than in the open desorber. This suggests that an increase in the pressure drop across the open desorber may be tolerated without increasing the fan power over that

required by a closed-cycle heat pump. Increasing the capacity of the cooling tower from 28 kJ/m^2 (2.5 Btu/ft^2) to 47.3 kJ/m^2 (4.2 Btu/ft^2) greatly decreases the temperature boost, since the condensing temperature is increased. From this comparison, it would appear that open cycles may find applications when cooling water is not readily available.

Figure 6 was shown for relatively small concentration changes (2%) in the desorber. As this concentration change is increased, the temperature boost decreases, and the COP increases. Figure 7 illustrates this effect, both for LiBr- and for LiCl-water solutions as working fluids, for several source temperatures. As the temperature boost is reduced with increasing concentration changes, the circulation losses in the intermediate recuperators are reduced, and the COP increases. This effect predominates until the heat that can be transferred from the dilute to the strong solution in the recuperator is decreased for large concentration changes. At that point, the COP curves tend to level off. It is also possible to conclude that a slightly greater temperature boost is possible with LiBr than with LiCl aqueous solutions. When the firing temperatures are large [for example, 71.1°C (160 F) and 82.2°C (180 F)], LiCl-water solutions cannot be utilized to a maximum advantage, since they crystallize. These solutions are, therefore, only suitable for low firing temperatures.

Changes in the atmospheric air conditions greatly influence the output temperatures. Figure 8 displays this temperature boost for LiBr and LiCl solutions versus the wet-bulb temperature. Low wet-bulb temperatures mean reduced water vapor pressures in the air and increased brine concentrations in the cycle, which bring about large temperature boosts. For large wet-bulb temperatures, the temperature boosts decrease. To preserve their magnitude, it becomes necessary to increase the amount of air circulated for a given concentration change. Increasing the amount of air circulated decreases the water vapor pressure in the air at the desorber outlet, thus allowing for larger operating brine concentrations. The parasitic power for air circulation is, however, increased too.

CONCLUSIONS

The interest in open absorption cycles stems from the need to utilize low-temperature residual heat. The thermodynamic analysis of a two-stage open cycle with suitable working fluids shows that temperature boosts on the order of 45°C are possible with heat exchanger effectiveness of 0.75 and COPs on the order of 0.20.

The open desorber is the critical component of an open-cycle system. To maximize the temperature boost (i.e., the maximum cycle concentration), the open desorber must operate at relatively small approach temperatures, typically, 5-10°C (9-18 F). The scope of this study does not allow the calculation of the heat/mass exchange areas required in the desorber. However, it is reasonable to conclude that open cycles will offer a potential for heat recovery only if the open desorber, designed for small approach temperatures, is economically feasible. Therefore, future efforts should focus on defining the optimum configuration and on understanding and predicting the performance of open desorbers.

The practicality of the whole system may certainly be questioned on three bases: economics, the lack of operating experience on systems of this nature, and the need for circulating large volumes of air to complete the regeneration process. In this study, it has been shown that the amount of air circulated in an open system is equivalent to the amount of air circulated in a cooling tower of conventional design supplying cooling water to a closed system. The economics and practicality remain open to question, and only further work may shed light on these matters.

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APPENDIX

The heat and mass transfer processes taking place in the open desorber are analyzed here. As stated before, it is assumed that the solution vapor pressure is maintained constant during desorption and that the resistance to mass transfer is concentrated on the air side of the air-solution interface.

The differential of water mass dm_w exchanged per unit time between the solution surface with a vapor pressure P_s corresponding to a humidity ratio W_i and an airstream with humidity ratio W may be written as*:

$$dm_w = h_D \cdot (W_i - W)\, da_i \,, \tag{a1}$$

$$\text{with} \quad W_i = 0.622 P_s/(P - P_s) \,,$$

where h_D is the mass transfer coefficient, da_i is the differential of interface area, and P is the total air pressure. The change of humidity ratio dW for the airstream of mass flux m_a is

$$m_a\, dW = dm_w \,. \tag{a2}$$

From Eqs a1 and a2 one obtains

$$m_a\, dW = h_D \cdot (W_i - W)\, da_i \,. \tag{a3}$$

*ASHRAE Handbook, 1977 Fundamentals Volume, Chap. 3, American Society of Heating, Refrigerating and Air-Conditioning Engineers, New York, 1977.

Similarly, a thermal balance for the air phase yields

$$m_a \, dh_a = h(T_i - T) \, da_i + m_a h_i \, dW \, , \tag{a4}$$

where h_a is the enthalpy of humid air (dependent both on its temperature and humidity), T the air temperature, h is the heat transfer coefficient from the interface to the air, T_i the interface temperature, and h_i the enthalpy of the water vapor evolved from the solution interface. The above enthalpies may be expressed as follows:

$$h_a = C_{pa}T + W(h_{fg_o} + C_{pw}T) \, , \tag{a5}$$

$$h_i = h_{fg_o} + C_{pw}T_i \, , \tag{a6}$$

where C_{pa} and C_{pw} are the specific heats of dry air and steam, respectively, and h_{fg_o} is the latent heat of vaporization of water at 0°C (32°F). Combining Eqs a3 and a4 and using Eqs a5 and a6 one obtains

$$\frac{dT}{T_i - T} = \frac{h}{C_p h_D} \frac{1}{(W_i - W)} + \frac{C_{pw}}{C_p} dW \, , \tag{a7}$$

where $C_p = C_{pa} + WC_{ps}$ is the specific heat of humid air.

The ratio $(h/h_D C_p)^{3/2}$ is the Lewis number, which for turbulent transport of water vapor in air is given by the following*:

$$\left(\frac{h}{h_D C_p}\right)^{3/2} = \frac{D}{\alpha} \, , \tag{a8}$$

where $\alpha = k/\rho C_p$, with k = thermal conductivity of air, C_p = specific heat of air, ρ = air density, and D = diffusivity of water vapor in air. As shown in the ASHRAE Handbook,* the Lewis number is close to one. Since C_{pw}/C_p is of order unity and $(W_i - W)$ is generally small ($\sim 10^{-2}$), Eq a7 may be approximated as

$$dT/dW = [(T_i - T)/(W_i - W)] \, . \tag{a9}$$

Integration of this equation yields the air temperature change along the desorber:

$$T_{27} - T_{22} = \int_{W_{22}}^{W_{27}} [(T_i - T)/(W_i - W)] \, dW \, . \tag{a10}$$

*Threlkeld, J. L., Thermal Environmental Engineering, Prentice Hall, Englewood Cliffs, NJ, 1973.

TABLE A1

State Points Corresponding to the Single-Stage Closed Cycle Shown in Fig. 1 for a LiBr-water Solution

Firing temperature = 60°C (140 F); cooling water = 8.3°C (47 F); effectiveness of heat exchangers = 0.75

Point	Temperature		Mass flow		Concentration	pressure	
	°C	F	kg/s	lb/s	%	kPa	psia
1	60.0	140.0	3.5	7.7			
2	53.7	128.7	2.5	5.5			
3	60.0	140.0	1.0	2.2			
4	90.1	194.2	0.60	1.32	55		
5	66.4	151.5	0.60	1.32	55		
6	53.7	128.7	0.58	1.31	59		
7	81.2	178.2	0.58	1.31	59		
8	53.7	128.7	0.025	0.06		15	2.18
9	82.6	180.7	1.0	2.2			
20	60.0	140.0	5.2	11.5			
21	55.8	132.4	5.2	11.5			
22	15.0	59.0	0.025	0.06			
23	8.3	46.9	2.9	6.4			
24	13.3	55.9	2.9	6.4			
27	15.0	59.0	0.025	0.06		1.65	0.24

TABLE A2

State Points Corresponding to the Single-Stage Closed Cycle Shown in Fig. 1

Firing temperature = 60°C (140 F); air = 23°C (73.4 F) and 15.6°C (60 F) wet-bulb; effectiveness of heat exchangers 0.75; desorber approach temperature 9°C (16.2 F)

Point	Temperature		Mass flow		Concentration	pressure	
	°C	F	kg/s	lb/s	%	kPa	psia
1	60.0	140.0	12.6	27.8			
2	53.7	128.7	12.5	27.6			
3	74.1	165.4	6.7	14.8			
4	86.1	187.0	3.5	7.7	53.0		
5	61.4	142.5	3.5	7.7	53.0		
6	51.0	123.8	3.4	7.5	55.0		
7	77.3	171.1	3.4	7.5	55.0		
8	53.7	128.7	0.126	0.28		15	2.2
9	83.1	181.6	6.7	14.8			
10	60.0	140.0	1.06	2.3			
11	83.1	181.6	5.8	12.8			
12	76.7	170.1	5.7	12.6			
13	83.1	181.6	1.0	2.2			
14	112.3	234.1	1.6	3.5	53.0		
15	69.1	156.2	1.6	3.5	53.0		
16	51.0	123.8	1.5	3.3	55.0		
17	97.0	206.6	1.5	3.3	55.0		
18	76.9	170.4	0.057	0.13		41	5.9
19	105.0	221.0	1.0	2.2			
20	60.0	140.0	24.7	54.5			
21	54.5	130.1	24.7	54.5			
22	40.6	105.1	40.2	88.6			
23	23.0	73.4	40.2	88.6			
24	28.8	83.8	40.3	88.8			
25	63.8	146.8	5.1	11.2			
26	51.0	122.0	4.9	10.8			
27	46.9	115.5	40.3	88.8			

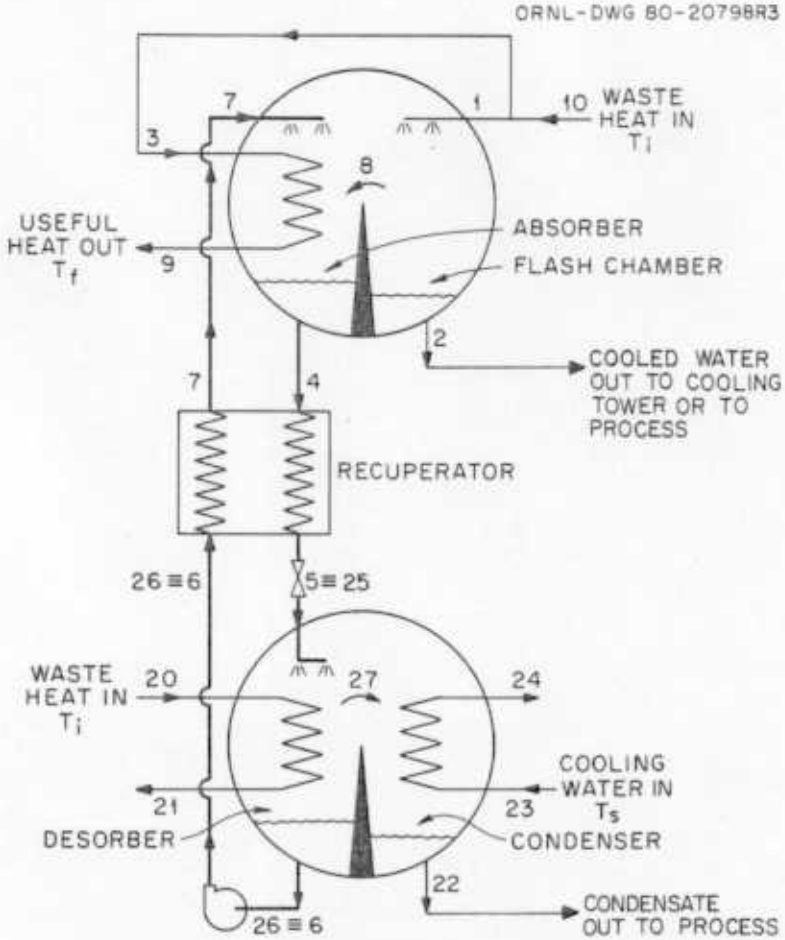

Figure 1. Schematic of a single-stage absorption heat pump

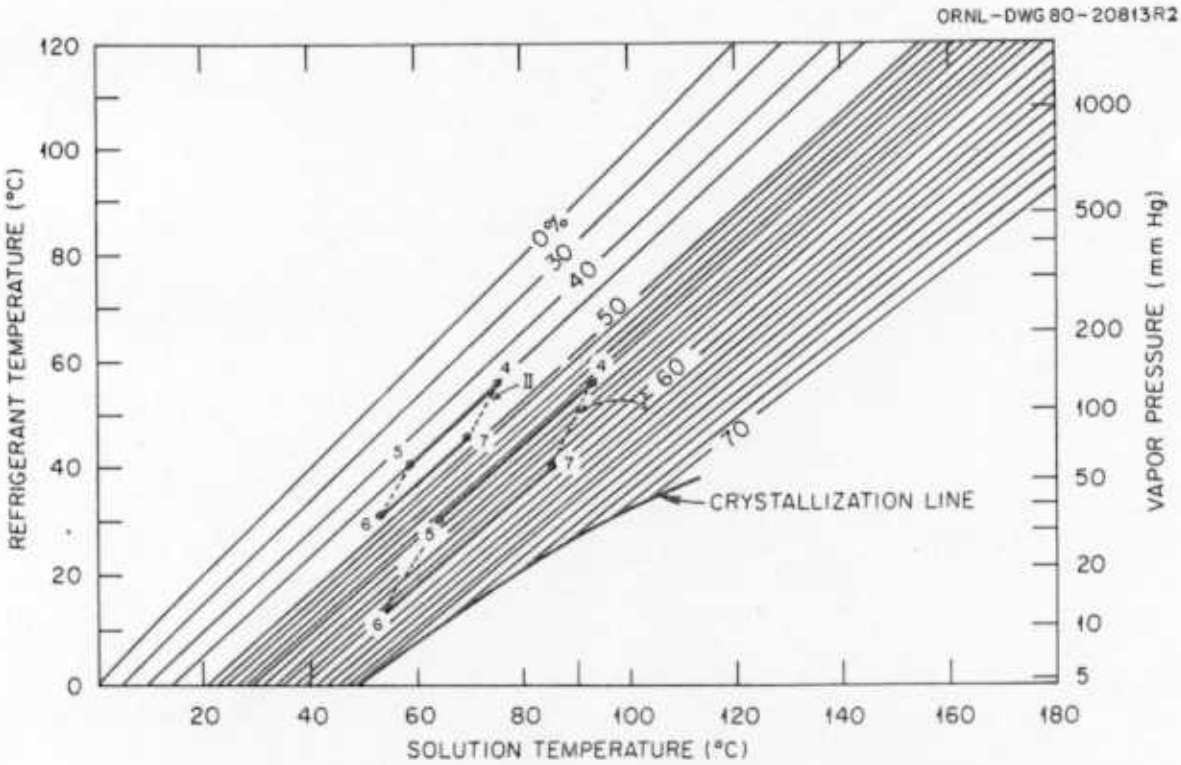

Figure 2. Equilibrium diagram for an LiBr-water solution, showing two heat pump cycles fired by the same temperature, but with different cooling water temperatures

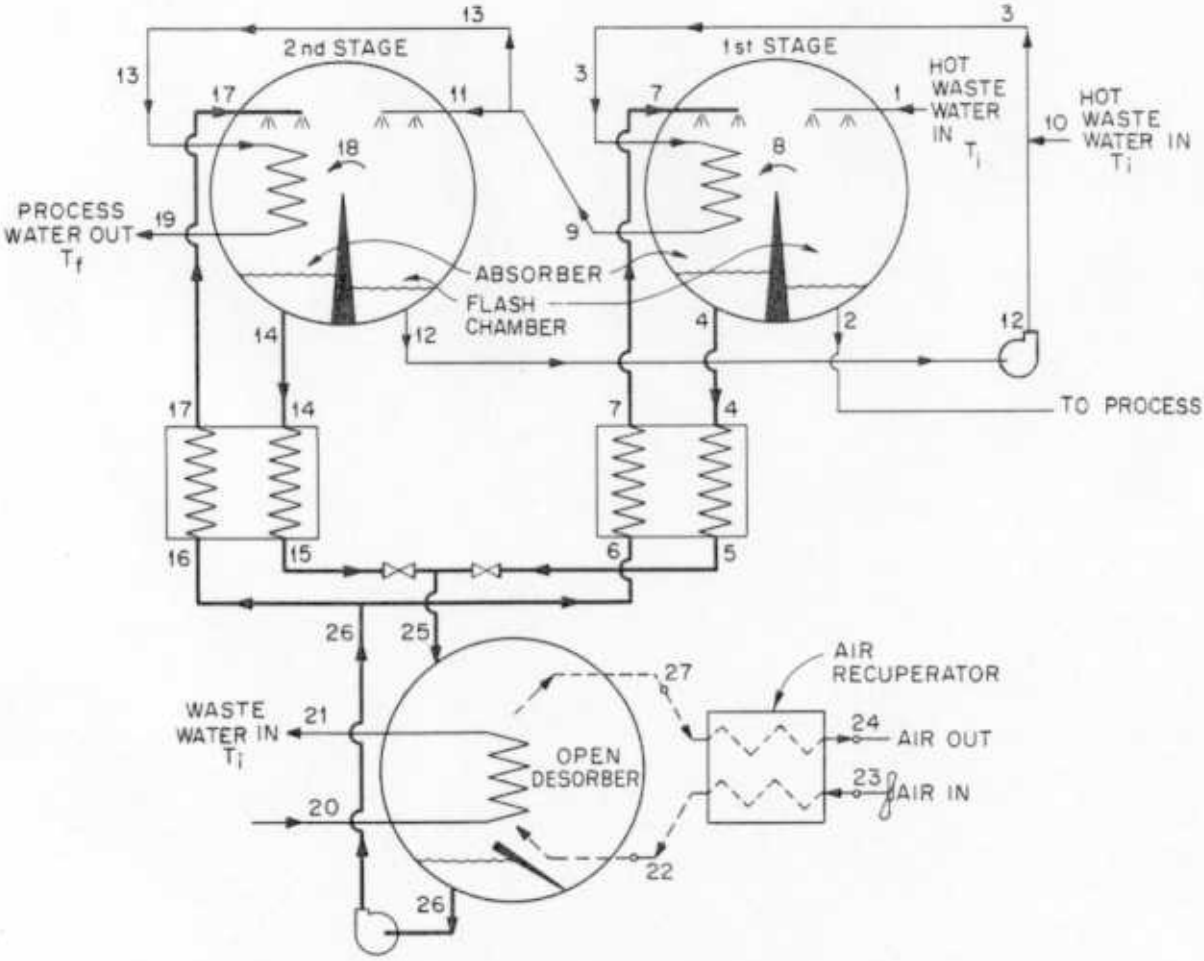

Figure 3. Schematic of the two-stage absorption heat pump

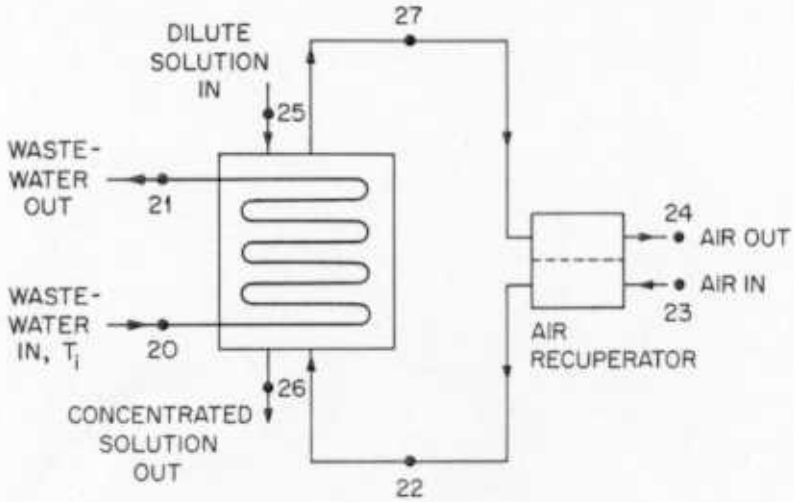

Figure 4. Schematic of an open desorber, showing the air-to-air recuperator

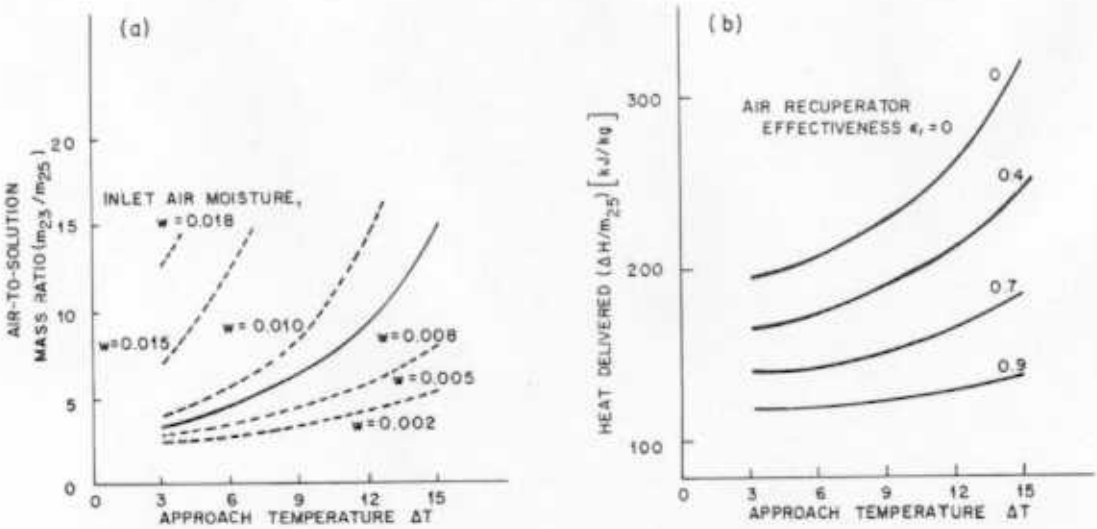

Figure 5. Performance map for an open desorber with LiBr-water for C_L = 52% and C_H = 54%

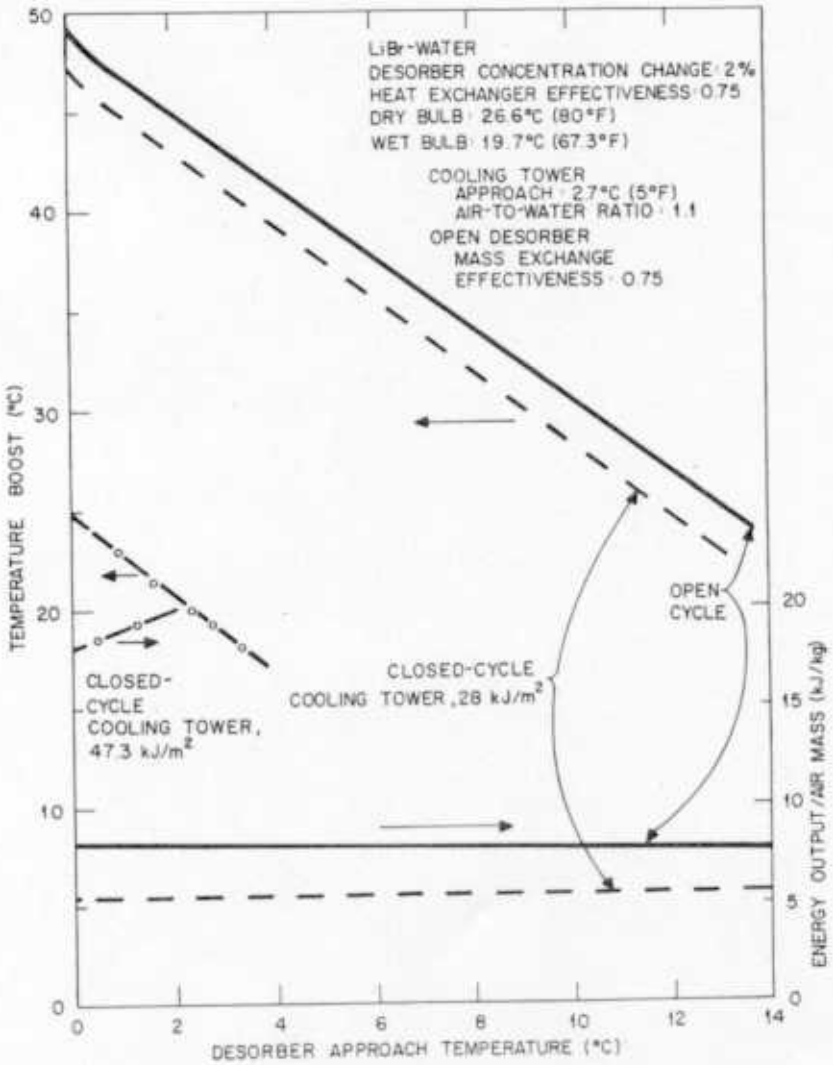

Figure 6. Temperature boost and power output per unit mass of air circulated for an open and a closed cycle

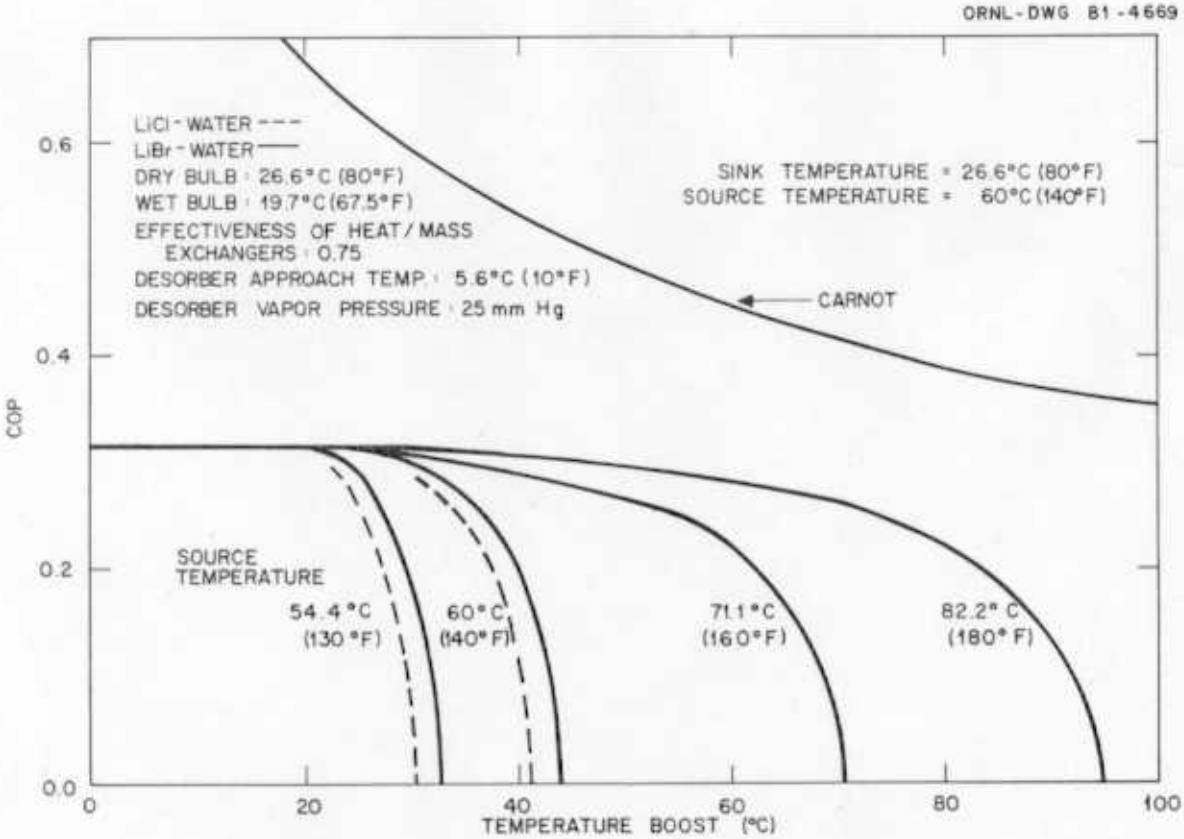

Figure 7. Coefficient of performance versus temperature boost for the open cycle

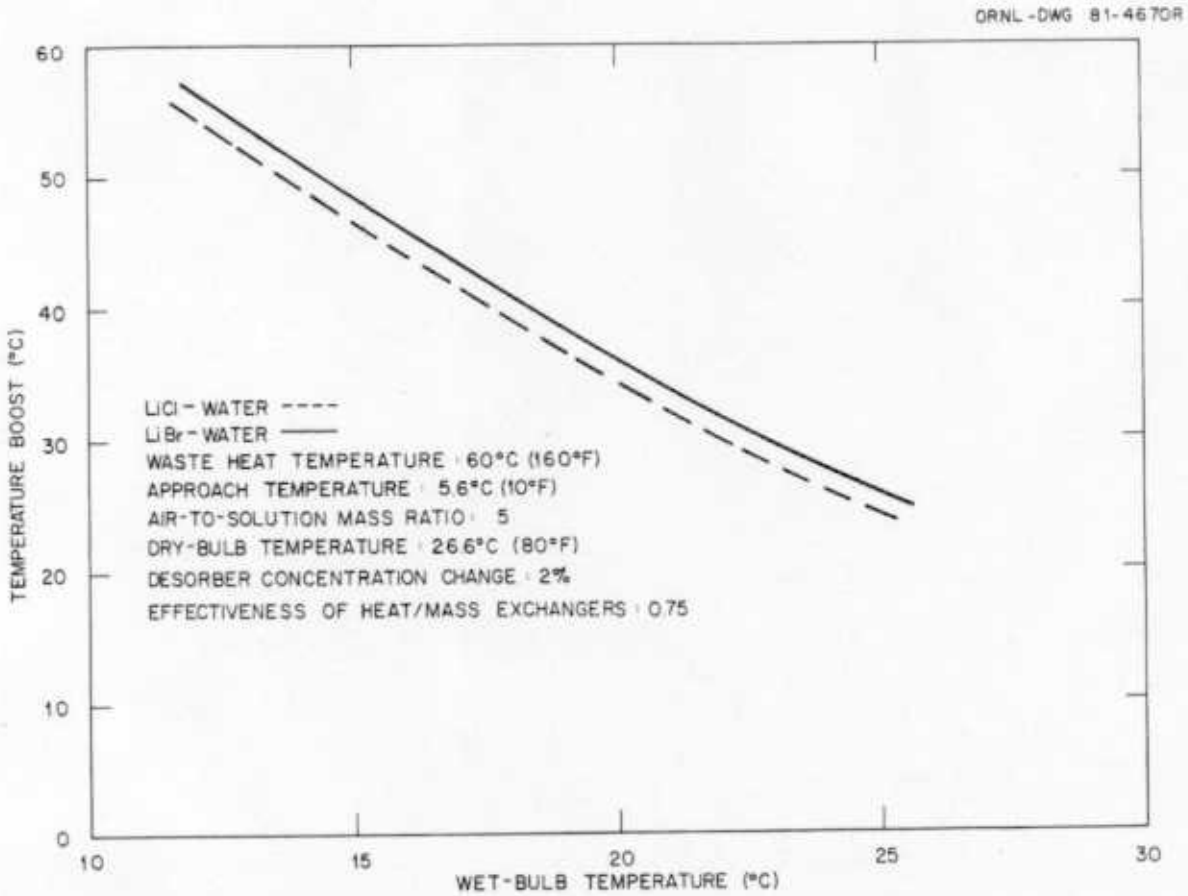

Figure 8. Effect of wet-bulb temperature on temperature boost, for constant air-to-solution mass ratio

DISCUSSION

G. VLIET, Prof., Mech. Eng., Univ. of Texas, Austin: What do the authors consider to be the magnitude of the parasite power required to maintain the continuous purge? Since air can only be reduced and not eliminated, what are the estimated effects on absorption and corrosion due to the air present in the system?

H. PEREZ-BLANCO: The magnitude of the parasitic power required for a continuous purge depends on the amount of air dissolved in the solution during desorption, on the fraction of this air released on the absorber, and on the partial vapor pressure of air maintained in the absorber by the continuous purge. An estimate based on Henry's low indicates that in the open desorber, about 8.4×10^{-6} kilograms of air per kilogram of solution will be dissolved in the solution. For an absorber vapor pressure of 120 mm Hg and an air mol fraction of 10^{-2}, the compression work necessary to extract the air (assuming that all of it is released) is approximately 0.5 kilojoules per kilogram of solution. Naturally, there is a loss of steam associated with this purge, but the steam is basically free.

For a solution concentration change of 4%, roughly 100 kilojoules of useful heat are released in the absorber. Thus, the parasitic work is indeed small as compared to the heat load.