

Conceptual Design and Performance Analysis of Absorption Heat Pumps for Waste Heat Utilization

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INTRODUCTION

The U.S. industrial sector annually consumes about 3×10^{13} MJ of primary energy for heat and power.¹ It has been estimated that 50% of this energy is released to the environment as waste heat.² This heat may be classified according to temperature: high-temperature range, above 650°C; medium-temperature range, between 650 and 230°C; and low-temperature range, below 230°C.² In the low range, a large part of the heat rejected is at temperatures below 80°C; in fact, the energy rejected in the form of condensate, cooling water, and process water (at temperatures between 40 and 80°C) amounts to 2.85×10^{12} MJ/year, or roughly 3% of the U.S. annual energy consumption.³

Although the thermodynamic availability of this reject heat is low, its large amount justifies studying means of recovering it for useful purposes. It may be used directly in some applications for space heating or for preheating process streams.² In many instances, the temperature of the waste heat may not be high enough for its intended use, and it is necessary to upgrade its temperature. Several types of heat pumps may be used for this purpose.³ Vapor compression heat pumps, which require mechanical work for their operation, have been most commonly suggested for upgrading waste heat.^{2,4} Absorption heat pumps seem more attractive for this application because they rely on the waste heat itself as their source of power and do not require primary energy for their operation except for small auxiliary equipment.

The fundamentals of temperature-boosting absorption have been described in the literature⁵ where two alternative cycles are discussed most frequently.⁶ In one of them, high-temperature heat supply and low-temperature heat to be upgraded are the inputs to the heat pump, the useful output being at an intermediate temperature between the two.^{4,6,7} This cycle is of primary benefit when a high-temperature source, such as the combustion of natural gas, is available.⁸ In the other cycle, the heat input is all at the low temperature T_1 . A fraction of it is upgraded to a temperature T_2 greater than the input, and the remaining fraction is rejected to a sink at a temperature T_3 lower than the input. Because this cycle does not require high-grade heat for its operation, it is specially suited for working with low-temperature waste heat or with low-temperature solar energy.⁶ It is referred to as a waste-heat-actuated cycle.

Several waste-heat-actuated cycles for providing industrial heat have been implemented with ammonia-water solutions. A one-stage absorption heat pump, operating on 65°C hot water, 15°C sink, and 105°C output temperatures, has been reported in the literature although details of construction and operation were not disclosed.⁹ An absorption-resorption heat pump developed at the French Institute of Petroleum (the IFP Thermosorb Process) also has been described.¹⁰ In this system, high efficiencies are possible for modest temperature boosts (about 30°C) at heat input temperatures above 70°C. For waste heat temperatures below 70°C, it was found that efficiencies drop substantially. A thermodynamic analysis of a two-stage, open-cycle lithium bromide-water absorption heat pump has been presented.¹¹ This analysis

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shows that large temperature boosts (from 60 to 105°C) are possible at 25% efficiencies, provided that the cycle operates at high solution concentrations. The potential of absorption-desorption systems has also been studied for providing space heating.^{6,7,12-14} These applications generally call for lower delivery temperatures than industrial applications and will not be described here.

The purpose of the present study has been to explore the potential of the absorption cycle for recovering low-temperature waste heat (60°C) as a first step toward the construction of a working system to provide process heat. The system is considered with lithium chloride- (LiCl-) and lithium bromide- (LiBr-) water solutions as working fluids. The benefits of multistaging for achieving large temperature boosts are shown. The performance criteria are defined by means of parameters that are relevant to the operation of a waste-heat-actuated cycle. Components and overall system performance of a two-stage heat pump are calculated and the results discussed.

THE ABSORPTION HEAT PUMP CYCLE

Several versions of the absorption heat pump cycle may be used to achieve different degrees of temperature boosting under various operating conditions. To illustrate the principle of the cycle, a schematic of a simple, one-stage, closed-cycle absorption heat pump is shown in Fig. 1. The state points defined in this figure* are indicated on a LiBr-water property diagram in Fig. 2. In this cycle (Fig. 1), steam is generated in an evaporator (pts. 1 to 8) by applying waste heat at temperature T_1 as a heat source. The steam is absorbed in the absorber by a concentrated LiBr-water solution, releasing useful heat at a temperature higher than T_1 (pts. 7 to 4). This heat is transferred to a water stream (state 3) and boosts its temperature from T_1 to T_2 . The diluted solution (pt. 4) passes through a recuperative heat exchanger into the desorber, where it is reconcentrated by evaporating water from it through the application of more waste heat at temperature T_1 . The solution leaving the desorber (pt. 26) returns to the absorber via the recuperative heat exchanger (pts. 6 to 7). The steam leaving the desorber is condensed (pts. 27 to 22) and the condensate returned to the evaporator (pt. 1).

Figure 2 describes a typical thermodynamic equilibrium diagram for the working fluid in the absorption heat pump. The particular material combination shown here is LiBr-water. The diagram describes the vapor pressure of pure water and the corresponding saturation temperature, as a function of the LiBr-water solution temperature, in equilibrium with it for different solution concentrations. For a given concentration, the rise in vapor saturation temperature with the solution temperature is almost linear. Both are proportional to the logarithm of the vapor pressure. For a given solution temperature, the higher the concentration, the lower the equilibrium vapor pressure. Conversely, for a given vapor pressure or saturation temperature, the higher the concentration, the higher the solution temperature. The cycle points of Fig. 1, indicated on the equilibrium diagram, illustrate the quantitative effects of the factors influencing the magnitude of the temperature boost. Clearly, it is desirable to have solution concentrations as large as possible. These are determined partly by the waste heat temperature T_1 , which ideally should be as high as possible, and partly by the heat sink temperature T_2 , which should be as low as possible. Once the concentrations have been fixed, the next important factor is the temperature of the steam generated in the evaporator, which is a function of T_1 .

Generally, the temperature boosts needed for industrial applications are too large to be achieved in one stage. In the example described above (Fig. 2), if saturated steam at 60°C can be obtained from the waste heat and the desorption step proceeds also at 60°C with condensation at 30°C, temperature boosts on the order of 30°C may be obtained with high and low LiBr-water solution concentrations of 54% and 52%, respectively. Larger temperature boosts may be achieved by means of a two-stage cycle, as illustrated in Fig. 3. Here, flash chambers are used in place of the closed-cycle evaporator described in Fig. 1. In this cycle, wastewater (pt. 1) enters the first-stage evaporator, where some of it is evaporated and the rest leaves at a somewhat lower temperature (pt. 2). The steam produced (pt. 8) is absorbed by a LiBr solution, releasing heat (pts. 7 to 4). The dilute solution (pt. 4) returns to the desorber via a recuperative heat exchanger. The heat released in this first stage is used to heat the stream of water (pts. 3 to 9), which is divided into two fractions. One is flashed in the

*A state point numbering system is used in a consistent manner for all the heat pump diagrams in this paper. The numbers in later diagrams will correspond to those in Fig. 1.

second-stage evaporator (pts. 11 to 12), where it generates more vapor (state 18). The other fraction is heated further in the second-stage absorber by the heat of absorption (pts. 13 to 19) and released as hot process water. The weak solution from the second stage (pt. 14) is returned to the desorber via a recuperative heat exchanger. In the desorber, the solution is concentrated using waste heat as a heat source (pts. 25 to 26), and the evaporated water is condensed (pt. 22). The strong solution is pumped back to the absorbers via the recuperative heat exchangers.

The state points corresponding to Fig. 3 are shown on a LiBr-water equilibrium diagram in Fig. 4 for high and low solution concentrations of 54% and 52%. It is assumed that steam at 60°C is available from the waste heat and that the solution temperature at the desorber outlet is also 60°C. For the purpose of illustrating the increased temperature boost effect, a steam temperature of 83°C was assumed in the second-stage absorber. Temperature boosts on the order of 50°C may be obtained with the two-stage cycle.

The advantage of a flash chamber in place of a closed evaporator is clearly in eliminating the approach temperature required in a heat exchanger. To illustrate this, let us assume that hot wastewater is available at a temperature of 60°C. When this water is used as a heat source in a closed evaporator or boiler for producing steam, a temperature difference between the minimum water temperature and the steam temperature must exist in order to induce heat transfer. Thus, if the water leaves the boiler at 54°C and a temperature drop of 4°C is assumed, then steam at only 50°C may be generated. The temperature boosting effect of the cycle is thus reduced from 30°C to approximately 21°C, as indicated in Fig. 2. If, instead of a closed evaporator, a flash chamber is employed, a closer thermal equilibrium between the steam and the water leaving the chamber may be obtained. If the water is flashed from 60 to 54°C, a mass of steam at pt. 8 equal to 1% of the water mass would be produced. Thus, the water leaves the flash and the closed evaporators at the same temperature (54°C), but the steam temperature is 54°C in the former and only 50°C in the latter. The temperature boost possible with a flash chamber is, therefore, larger than the one possible with a closed evaporator.

The two-stage cycle described in Fig. 3 is a preferred configuration out of several other multistage concepts that could be used to produce similar temperature boosts. Another possibility is to have the first stage in the configuration described in Fig. 1 and use the temperature-boosted stream out of the first-stage absorber as a heat source for a second desorber. This desorber would then be able to produce a more concentrated solution than the first one, powered by the original waste heat. The more concentrated solution could be used in a second-stage absorber to give a higher temperature boost. This concept and others have several disadvantages compared to the preferred configuration of Fig. 3. A detailed discussion of the various multistage concepts is beyond the scope of the present work.

This paper will focus on the two-stage cycle with flash evaporators in the preferred configuration described in Fig. 3. As outlined above, the largest temperature boosts may be obtained with these configurations. The wastewater temperature will be taken equal to 60°C in all cases, and other relevant parameters will be varied in order to calculate performance for changing conditions.

SYSTEM PERFORMANCE CRITERIA

The performance of the absorption heat pump is characterized by several parameters that should be considered in the design of an efficient working system. These performance parameters depend on two given operating conditions: the temperature of the source of heat to be upgraded (T_s) and the heat rejection environment as determined by the cooling medium. For these two given conditions, the design variables of the system, such as the absorbent concentrations and different flow rates, must be selected to optimize the performance parameters.

The first and perhaps most important performance characteristic of the heat pump is the output temperature (T_f) that the system can deliver with given heat source and heat rejection temperatures. Clearly, a large temperature boost is desirable. However, for each cycle configuration with given working materials and number of stages, T_f has a theoretical upper limit, as will be shown. Trying to operate close to this limit results in very large heat exchange surfaces and high flow rates. The latter are associated with excessive pumping power requirements. A practical choice of T_f must consider these factors.

Another important performance criterion is the overall coefficient of performance (COP) of the system, defined as the ratio between the useful heat output and the energy input. The latter consists of two parts: input from the heat source and parasitic power. In systems where the second part is negligibly small, the overall COP is essentially equal to the thermal COP. Using the notation of Fig. 3,

$$(\text{COP})_{\text{th}} = \frac{m_{19}(T_f - T_1)}{m_2(T_1 - T_2) + m_{20}(T_1 - T_{21}) + m_{22}(T_1 - T_{22})} \quad (1)$$

In this equation, T stands for the temperature at the point indicated, and m stands for the mass flow rate. For comparison, it would be of interest to calculate the COP that may be obtained under the same operating conditions with the ideal Carnot cycle. Assuming a reversible heat engine operating in place of the desorber/condenser between two heat reservoirs at temperatures T_1 and T_s and supplying work to a reversible heat pump operating in place of the absorber/evaporator between two heat reservoirs at T_f and T_1 , the following expression is obtained:

$$(\text{COP})_{\text{max}} = \frac{T_f(T_1 - T_s)}{T_1(T_f - T_s)} \quad (2)$$

When $T_1 = 333 \text{ K}$ (60°C), $T_f = 383 \text{ K}$ (110°C), and $T_s = 300 \text{ K}$ (27°C), the maximum COP that results from Eq 2 is equal to 0.46.

The parasitic power consumers in the system include circulation pumps, air fans, purge systems, etc. At given operating temperatures, the power required is usually proportional to the third power of the amount of fluids circulated. Increasing these flows can often help raise the thermal COP at the cost of increasing the parasitic power requirements.

A key consideration in the design of the heat pump system is the price of the heat to be upgraded. If this heat is abundantly available and free, the system should be designed so as to minimize the parasitic power requirements — at the possible cost of a low thermal COP. A typical example of this case is waste heat, which is free but often entails an additional cost for disposal. With a source of heat such as solar or geothermal, the thermal COP is of greater importance.

The last, but not least, performance criterion of the system is its initial cost. As in many similar cases, good thermal performance calls for the temperature differentials in the heat exchangers to be kept to a minimum. The designer must weigh this consideration against the cost of the heat exchangers that are a major part of the overall system's cost.

WORKING FLUIDS SELECTION

The properties required of refrigerant/absorbent combinations for use in absorption cooling systems have been discussed extensively in the literature.¹⁵⁻¹⁷ Many of those requirements are equally applicable to a heat pump system of the type discussed in the present work. The most important characteristics may be summarized as follows:

1. The equilibrium temperature of the absorbent/refrigerant solution in equilibrium with the vapor should be as high as possible at the absorber conditions and as low as possible at the desorber conditions. This provides for a maximum temperature boost and ease of regeneration.
2. The solubility of the refrigerant in the absorbent must be as high as possible at the absorber temperature and pressure and as low as possible at the corresponding desorber conditions. In other words, the concentration difference between the strong and weak absorbent as determined by the conditions in the desorber and absorber, respectively, should be as large as possible. If this requirement is not met, large amounts of absorbent solution must be circulated between the absorber and desorber in order to transfer the necessary quantity of refrigerant, thus causing excessive internal heat losses.

3. The heat of absorption/desorption should be as high as possible at the absorber conditions and as low as possible at the desorber conditions. This heat is the sum of the latent heat of vaporization/condensation of the refrigerant and of the heat of dilution. The latter should, therefore, be as high as possible in the absorber and low or negative in the desorber.
4. The properties of the absorbent/refrigerant solution that affect heat and mass transfer, such as viscosity, diffusion coefficient, thermal conductivity, etc., should be favorable.
5. The absorbent should be nonvolatile, or considerably less volatile than the refrigerant, to prevent the transfer of absorbent into the condenser.
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 non-corrosive to common materials of construction and available at a reasonable cost.

Of the many fluid pairs considered and experimented with for absorption systems, only a few were found practical, having satisfied most of the above criteria. Water-ammonia and LiBr-water have been used extensively in closed-cycle cooling systems, each with its own merits and disadvantages. In the former pair, ammonia is toxic and the absorbent (water) is volatile, requiring a rectifier between the desorber and condenser. The LiBr-water combination is corrosive to metals, and the salt crystallizes out of solution at high concentrations. These problems have been dealt with effectively in closed-cycle cooling systems, and both fluid pairs seem suitable for absorption heat pumps. The LiBr-water pair is clearly more promising, since in the water-ammonia combination, the working pressures are high even in cooling applications and become much higher at the heat pump range of working temperatures, to the point where construction may become impractical. The LiCl-water solution has properties similar to those of LiBr-water and has been used particularly in open systems.¹⁸ Other working fluid combinations have been tested in experimental closed-cycle absorption setups, but it appears that none has shown a definite advantage justifying its use in place of the above three.¹⁷

The present study will consider LiBr-water and LiCl-water as working materials for the heat pump system. The intention is not to rule out other combinations but rather to evaluate the potential of the cycle with what seem to be the most suitable fluid pairs at present. Figures 2 and 4 show a typical equilibrium diagram for the LiBr-water combination, based on data from more than 15 sources compiled recently by McNeely.¹⁹ Similar data for LiCl-water has been provided.^{20,21} Enthalpy data was also obtained from those references.

ABSORBER/EVAPORATOR SYSTEM PERFORMANCE

The part of the heat pump where the actual temperature boosting takes place is the absorber/evaporator unit. This unit consists of an absorber and a flash chamber used for an evaporator, both packaged in the same shell and served by a recuperative heat exchanger (see Fig. 3 for description and notation).

In the following analysis, the high and low absorbent concentrations (C_H and C_L , respectively) are preselected. The former is determined by the conditions in the desorber; the latter is a variable for performance optimization, the effect of which will be evaluated and discussed. The temperature of the strong solution entering the recuperator (state 6) is also fixed by the desorber and will be referred to as T_6 . The properties at all other states may be calculated as functions of the above parameters² and of the temperature of the heat source to be upgraded (T_s). Heat exchanger effectivenesses ϵ_a and ϵ_r are assumed for the absorber and recuperative heat exchanger. The effectiveness is^a defined as the ratio of the actual heat transferred to the thermodynamically limited, maximum-possible heat transfer rate.²² Two basic assumptions are made: (1) the weak solution leaving the absorber (state 4) is in vapor pressure equilibrium with the vapor coming from the evaporator (state 8) and (2) pressure drops in the system are negligible.

For the given concentrations and temperatures, the following equations may be written to describe the process in the absorber/evaporator system:

1. in the evaporator:

$$\text{mass balance: } m_8 = m_1 - m_2, \quad (3)$$

$$\text{energy balance: } m_8 h_8 = m_1 c_p T_1 - m_2 c_p T_2, \quad (4)$$

$$T_2 = T_8; \quad (5)$$

2. in the absorber:

$$\text{equilibrium: } T_4 = f(T_8, C_L), \quad (6)$$

$$\text{salt mass conservation: } m_4 C_L = m_7 C_H, \quad (7)$$

$$\text{mass balance: } m_8 = m_4 - m_7, \quad (8)$$

$$\text{energy balance: } m_3 c_p (T_9 - T_3) = m_7 h_7 + m_8 h_8 - m_4 h_4, \quad (9)$$

$$m_3 = m_9; \quad (10)$$

$$\text{absorber effectiveness: } \epsilon_a = (T_9 - T_3)/(T_4 - T_3), \quad (11)$$

3. in the recuperator:

$$m_4 = m_5, \quad (12)$$

$$m_6 = m_7, \quad (13)$$

$$\text{recuperator effectiveness: } \epsilon_r = (T_7 - T_6)/(T_4 - T_6), \quad (14)$$

$$\text{heat balance: } m_4 h_4 - m_5 h_5 = m_7 h_7 - m_6 h_6. \quad (15)$$

Here, m , h , and T indicate the mass flow rate, enthalpy, and temperature of the working material at the different states, respectively, and c is the specific heat of water. The enthalpies of the solution at the different states may be expressed in terms of the temperature and concentration; h_8 is the enthalpy of dry saturated steam at temperature T_8 . With the inlet flow rates m_1 and m_3 and inlet temperatures T_1 , T_3 , and T_6 given, the above set of equations may be solved for the temperatures and flow rates at all nine states.

A system of equations similar to Eqs 3-15 may be written for the second stage of the absorber/evaporator system.* In combining the two stages, the following additional conditions are in effect:

$$T_{11} = T_9, \quad (16)$$

$$T_{13} = T_9, \quad (17)$$

$$m_{11} + m_{13} = m_9, \quad (18)$$

$$m_3 = m_{10} + m_{12}, \quad (19)$$

$$m_3 c_p T_3 = m_{10} c_p T_{10} + m_{12} c_p T_{12}. \quad (20)$$

With the flow rate of the output stream m_{19} selected arbitrarily as a measure of the size of the system and with the given inlet temperatures $T_1 = T_{10} = T_1$, $T_6 = T_{16} = T_6$, there are altogether 36 equations for the 38 unknown temperatures and flow rates at points 1 through 19. Two of the latter can hence be selected as free parameters for optimization.

A computer program has been written to carry out the above calculation. In selecting m_{19} as unity, the temperature boost and the heat gain of the output stream are given by the same expression. It makes physical sense to select as the free optimization parameters the evaporation temperatures in the first and the second stage, T_8 and T_{18} . These are also the saturation temperatures corresponding to the pressures prevailing in the two stages. Figure 5 shows an operating map of the absorber/evaporator system in terms of these parameters. The figure describes the temperature boost ($T_9 - T_3$) as a function of the mass ratio between the rejected and useful streams (m_7/m_{19}). The working material is LiBr-water. The heat source temperature is fixed at $T_3 = 60^\circ\text{C}$ (140°F), and the solution from the desorber is at 54% and 57°C (134°F). Two sets of curves are shown for the low concentration at 50 and 52%.

*The indices used to mark the different points in the second stage correspond to those of the first, with 10 added to each. Thus, the entrance point of water to the evaporator is marked 1 in the first stage and 11 in the second stage; the evaporation temperature is 8 in the first stage and 18 in the second, etc.

It is clear from the operating map that the higher the evaporation temperature at both stages, the higher the temperature boost. For a given T_8 , raising T_{18} results in a significant rise in temperature boost without greatly increasing the mass of the rejected stream. The opposite is not as desirable: increasing T_8 at fixed T_{18} only slightly raises the temperature boost and considerably increases the mass rejected.

While T_8 and T_{18} can be selected independently, there is an upper limit on both. The upper limit for T_8 is the heat source temperature T_1 , and bringing the former close to the latter results in a rapid increase of the rejected stream m_2 . For each T_8 , the upper limit on T_{18} is set by the condition that the water temperature in both absorbers is always lower than that of the solution. When approaching this limit on T_{18} , the amount of water circulated between the first and second stage (e.g., m_{12}) becomes very large. All other operating parameters are reasonably practical. A good choice, therefore, would be to select T_8 somewhat below T_1 , so as to have a reasonable m_2 , and then select T_{18} somewhat below its limit corresponding to T_8 , so that m_{12} will not be too large.

The effect of the low concentration is also evident from Fig. 5. Lowering the concentration from 52 to 50% reduces the temperature boost by about 3.5°C for the same values of T_8 and T_{18} . It also reduces the maximum T_{18} attainable for each T_8 . The rejected stream m_2 is reduced somewhat, indicating a better heat gain of the upgraded stream under the lower C_L .

An analysis similar to the above has been performed for the system with LiCl-water as the working fluid. The high concentration of this solution has been selected at 41%, which has the same vapor pressure at 57°C as a 54% LiBr solution when leaving the desorber. The LiBr solution gives a larger temperature boost than the LiCl when desorbed under the same conditions.

DESORBER/CONDENSER PERFORMANCE

The performance of the desorber depends on the heat supply temperature and on the heat rejection environment as manifested by the vapor pressure and the corresponding saturation temperature. Ideally, the former temperature should be as high as possible and the latter as low as possible to achieve maximum concentration. In the following analysis (refer to Fig. 3 for notation), heat exchanger effectivenesses ϵ_d and ϵ_c are assumed for the desorber and condenser. The following equations may be written to describe the process in the desorber/condenser system:

1. in the desorber:

$$\text{salt mass conservation: } m_{25}C_L = m_{26}C_H, \quad (21)$$

$$\text{mass balance: } m_{27} = m_{25} - m_{26}, \quad (22)$$

$$\text{energy balance: } m_{20}c_p(T_{20} - T_{21}) = m_{26}h_{26} + m_{27}h_{27} - m_{25}h_{25}, \quad (23)$$

$$m_{20} = m_{21} \quad (24)$$

$$\text{equilibrium: } T_{26} = f(T_{22}, C_H), \quad (25)$$

$$\text{approximate desorbed vapor temperature: } T_{27} = (T_{25} + T_{26})/2, \quad (26)$$

$$\text{desorber effectiveness: } \epsilon_d = (T_{20} - T_{21})/(T_{20} - T_{26}), \quad (27)$$

2. in the condenser:

$$\text{mass balance: } m_{27} = m_{22}, \quad (28)$$

$$\text{energy balance: } m_{23}c_p(T_{24} - T_{23}) = m_{27}(h_{27} - h_{22}), \quad (29)$$

$$m_{23} = m_{24}, \quad (30)$$

$$\text{condenser effectiveness: } \epsilon_c = (T_{24} - T_{23})/(T_{22} - T_{23}). \quad (31)$$

The enthalpies of the solution at the different states may be expressed in terms of the temperature and concentration. By specifying the inlet conditions of the solution (m_{25} , T_{25} , and C_L), the high concentration required (C_H), and the inlet hot and cooling water temperatures (T_{20} and T_{23}), the above set of equations may be solved for the flow rates and temperatures of all eight states, with one free parameter for optimization. It makes physical sense to select as that parameter the condensing temperature T_{22} , as was done in the analysis of the absorber/evaporator system.

Figure 6 shows an operating map of the desorber with LiBr-water, obtained from a computer solution of Eqs 21-31. The relative flow rates of hot water and cooling water are plotted for different cooling temperatures [the waste heat temperature is fixed at 60°C (140°F)]. The inlet low concentration and required high concentration are 52 and 54%, and the effectiveness of both the desorber and condenser is 0.80. It is clear from the figure that, as the cooling water temperature rises, larger and larger flow rates of both cooling and hot water are required to produce the desired change in concentration. The upper limit on the condensing temperature is set by the required high concentration and the waste heat temperature available through the equilibrium condition Eq 25. Figure 6 also shows the hot water flow rate to be dependent directly on the condensing temperature T_{22} and to increase very rapidly as the limit on T_{22} is approached. The higher T_{22} is, the smaller the amount of cooling water required.

An additional run of the computer program was made to obtain a similar operating map of the desorber for LiBr high and low concentrations of 54 and 50%. The results are very similar to those of Fig. 6, with the flows of hot water (m_{20}) and cooling water (m_{23}) per unit mass of entering solution (m_{25}) approximately doubled. This is to be expected, since the amount of vapor desorbed is now about twice as large as in the previous case. The main conclusion is that when the high concentration is fixed, the amounts of hot water and cooling water needed are a function of the amount of water desorbed, except for additional second-order effects.

An operating map similar to that of Fig. 6 was calculated for a LiCl-water solution. Here again, as in the absorber/evaporator analysis, the high concentration of the solution has been selected at 41%, which yields the same vapor pressure at 57°C as LiBr at 54%. It was clear from this diagram that, for the same condensing temperature, larger quantities of both cooling and hot water are required to produce a 2% change in concentration in LiCl than are required for the LiBr. This is to be expected, since the amount of water desorbed is higher in the case of LiCl, being 0.0488 kg per kilogram of entering solution as opposed to 0.0370 kg for LiBr. The ratio m_{20}/m_{23} remains about the same for both working materials.

OVERALL CYCLE PERFORMANCE

A computer program similar to those described above was used to evaluate the overall performance of the system. Figure 7 describes the temperature boost ($T_f - T_1$) and the thermal COP (Eq 1) for the two-stage system (Fig. 3) as a function of the low concentration. The waste heat temperature is 60°C and the condensing water temperature is maintained constant at 20°C. The working material is LiBr-water, and a series of curves is shown for different values of the heat exchanger effectiveness, assumed to be the same for all the heat exchangers in the system. Two sets of runs were made for each value of ϵ : in one, the high concentration is fixed at 54%; in the other, C_H is higher than C_L by a constant 2%. As expected, the temperature boost is independent of C_H and determined primarily by the low concentration. The broken lines indicate the temperature boost to increase with C_L and, of course, with the heat exchanger effectiveness. The COP turns out to be sensitive to the concentration difference. With $C_H - C_L = 2\%$, the change in COP with the low concentration is very small. However, for the case where C_H is kept constant, one can achieve a significant increase in COP by lowering C_L at the expense of a reduced temperature boost. The change in COP is more significant for the system with low heat exchanger effectiveness. There the losses due to solution circulation are highest, and, therefore, a larger gain is achieved by increasing the concentration difference. Once this difference becomes large, on the order of 8%, the COP is about the same for all ϵ 's.

Figure 8 shows the effect of the waste heat temperature and cooling water temperature on the performance of the heat pump. The COP is plotted as a function of the temperature boost for different values of the above temperatures, and the trade-off between these two performance criteria is evident. The working fluid here is LiBr-water, and the effectiveness of all heat exchangers is 0.85. The condensing temperature is assumed to be 5°C higher than the cooling water temperature; similarly, the high solution temperature in the desorber is 5°C lower than the waste heat temperature. It is clear from the figure how raising the waste heat temperature increases performance and raising the cooling water temperature reduces it. Trading off a lower temperature boost against a higher COP is achieved by reducing the low concentration; however, there is a limit to that when the COP reaches an asymptotic value of about 0.4. The theoretical COP obtainable with a Carnot heat pump, operating with 60°C waste heat and 15°C cooling water, is shown for comparison with the corresponding absorption heat pump curve. Figure 8 may be regarded as a summary of all previous calculations, describing the overall system performance for different operating conditions.

CONCLUSIONS

A thermodynamic analysis of an absorption heat pump cycle has been performed. The system operates on low-grade heat, with only a small input of electrical energy for auxiliary equipment. This feature alone makes it extremely attractive from the standpoint of energy conservation. The actual design of the system should attempt to minimize the parasitic power required for operation. Two working materials, LiBr-water and LiCl-water, have been considered.

The combination LiBr-water offers the important advantage over LiCl-water of achieving a higher temperature boost. This consideration may make LiBr solutions the preferred working fluid.

The analysis of the absorber/evaporator system indicates that in order to maximize the temperature boost, it is necessary to operate at high values of the low concentration (C_L), with a relatively small concentration change ($C_H - C_L$). A small concentration change, however, is undesirable, as it tends to increase circulation losses. In an actual design, it becomes necessary to trade off a high temperature boost for a lower COP. The absorber/evaporator performance maps presented in this paper are a useful guide in selecting practical operational conditions that can be met without increasing the parasitic power requirements beyond reasonable limits.

The analysis of the desorber/condenser shows that the higher the cooling water temperature, the larger the flow rates of both cooling and waste hot water required to produce a given change in concentration. The most favorable operational conditions from the point of view of reduced flow rates occur when relatively cold cooling water is available. A maximum heat sink temperature exists for each given waste heat temperature, above which the system cannot perform.

The overall system analysis confirms and expands the conclusions extracted from the component analysis: A large temperature boost calls for large values of the low concentration and small concentration changes. For increasing concentration changes, the temperature boost is decreased, but the thermal COP increases. Also, the effectiveness of the heat exchangers has a greater influence on the thermal COP than on the temperature boost. This is a favorable circumstance, since the size of the heat exchangers may then be reduced. Although efficiency will decrease, a large temperature boost will be maintained.

The overall conclusion of this study is that absorption cycles offer a good opportunity to recover waste heat for temperature boosting applications. Previous work in the field of waste heat utilization, carried out at Oak Ridge National Laboratory, indicated that absorption cycles have good potential applications for enhancing energy conservation via heat recovery.³ As such, they were identified as an appropriate subject for research and development. Within that framework, the findings of this study indicate that the development of absorption heat pumps for waste heat recovery is both feasible and desirable.

NOMENCLATURE

C_H	=	high solution concentration in the system (%)
C_L	=	low solution concentration in the system (%)
COP	=	coefficient of performance (dimensionless)
$(COP)_{max}$	=	maximum possible coefficient of performance, obtainable with a Carnot cycle (dimensionless)
$(COP)_{th}$	=	thermal coefficient of performance, excluding parasitics (dimensionless)
c_p	=	specific heat of water (kJ/kg·°C)
ϵ_a	=	heat exchanger effectiveness in absorber (dimensionless)
ϵ_c	=	heat exchanger effectiveness in condenser (dimensionless)
ϵ_d	=	heat exchanger effectiveness in desorber (dimensionless)
ϵ_r	=	heat exchanger effectiveness in recuperator (dimensionless)
h_j	=	enthalpy at point j in the cycle (kJ/kg)
m_j	=	flow rate at point j in the cycle (kg/s)
Q	=	useful heat gain of the temperature-boosted stream (kJ)

T_f	=	final temperature of the upgraded heat ($^{\circ}\text{C}$)
T_g	=	temperature of solution leaving the desorber ($^{\circ}\text{C}$)
T_1	=	temperature of the input waste heat ($^{\circ}\text{C}$)
T_j	=	temperature at point j in the cycle ($^{\circ}\text{C}$)
T_s	=	heat sink temperature ($^{\circ}\text{C}$)

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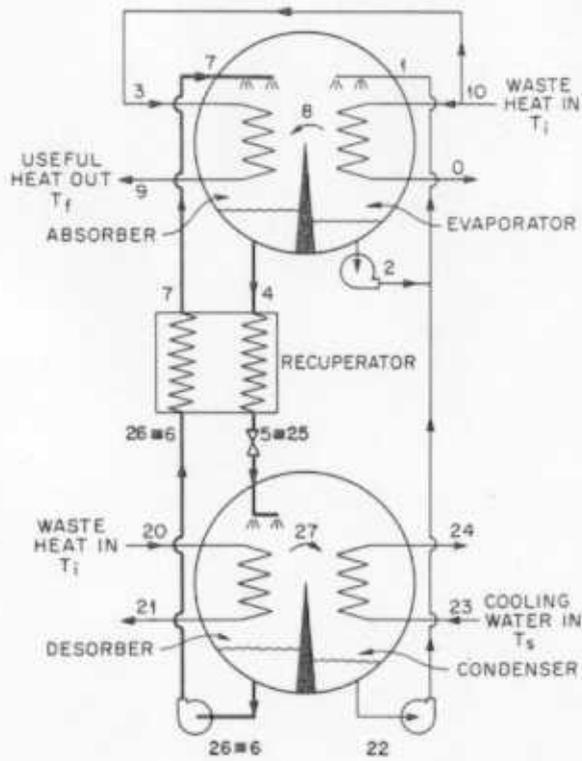


Figure 1. Schematic of a single-stage absorption heat pump with main components enclosed in two separate shells

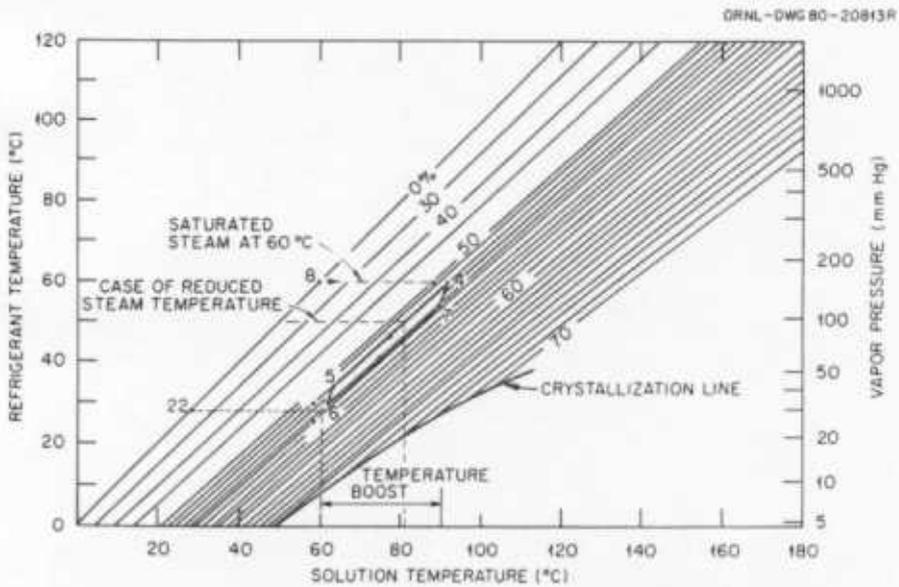


Figure 2. Property diagram for a LiBr-water solution; state points correspond to the heat pump cycle of Fig. 1

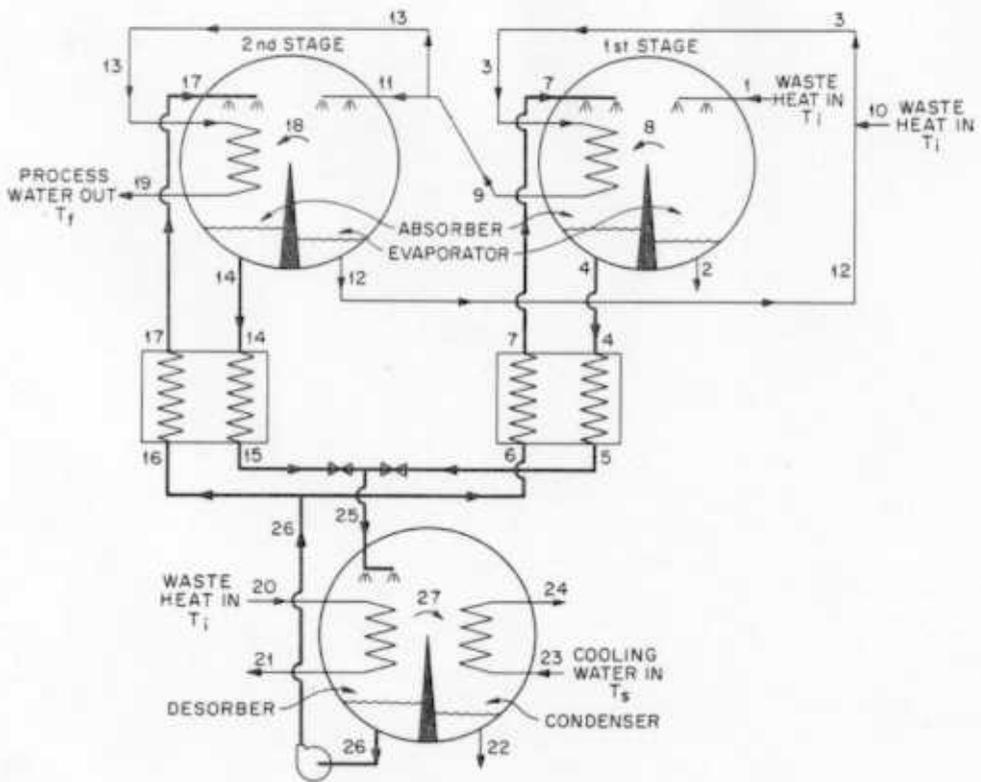


Figure 3. Schematic of a two-stage heat pump with flash-evaporators

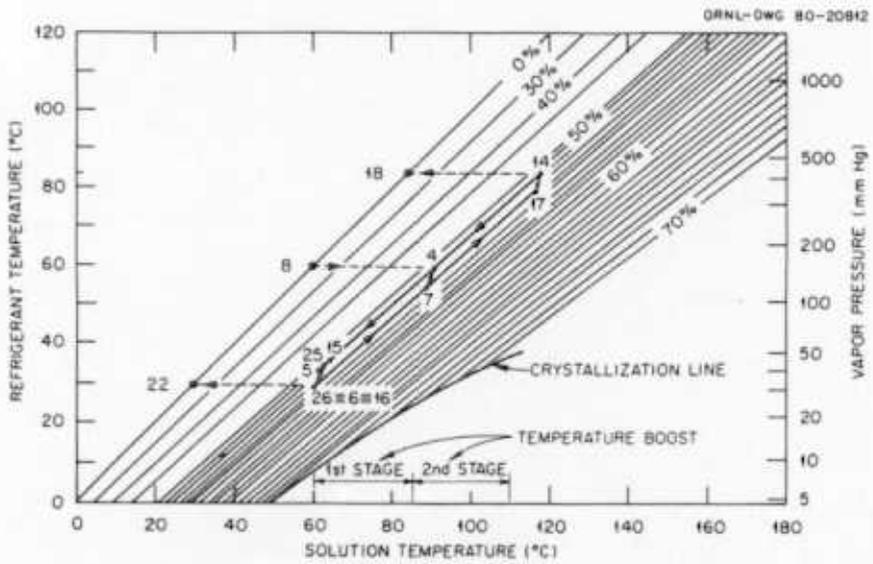


Figure 4. Property diagram for a LiBr-water solution; state points correspond to the two-stage cycle of Fig. 3

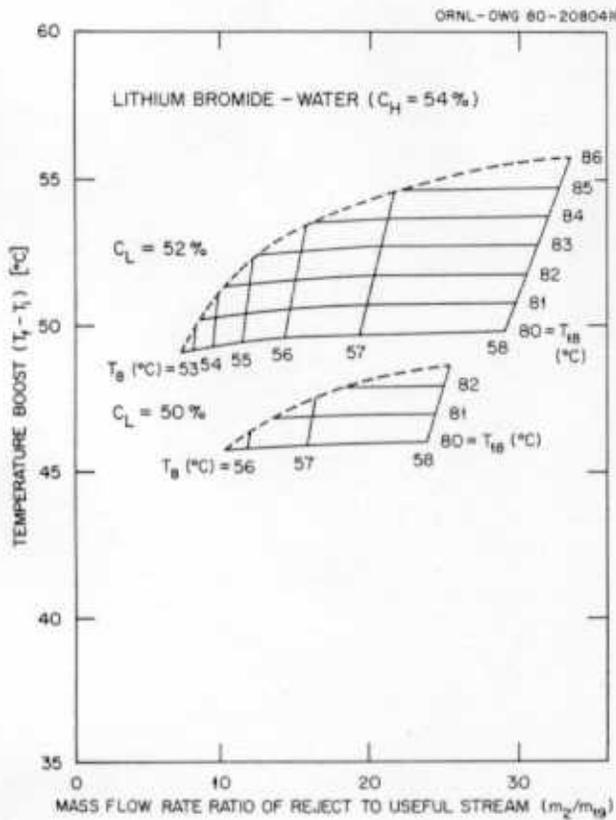


Figure 5. Performance map for the absorber/flash chamber combination with a LiBr-water solution

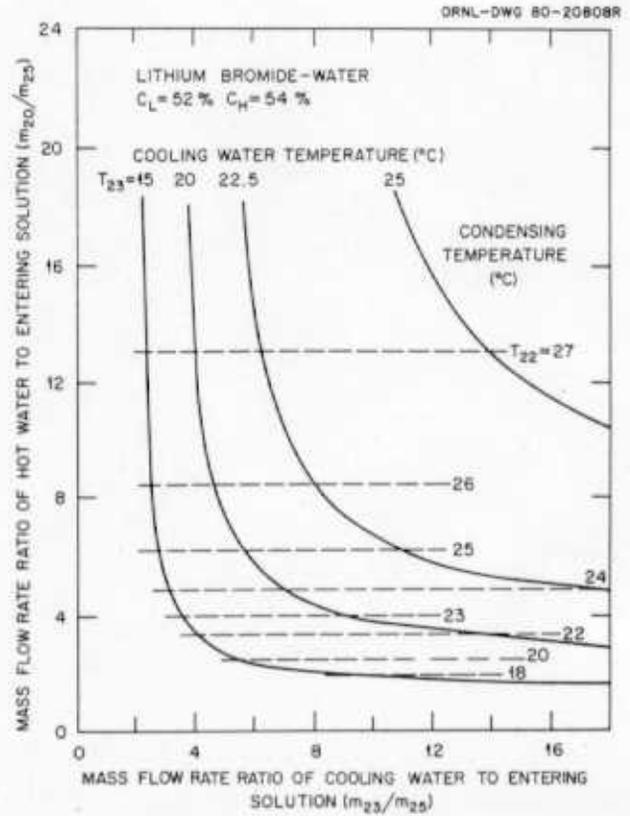


Figure 6. Performance map for the desorber/condenser combination with a LiBr-water solution

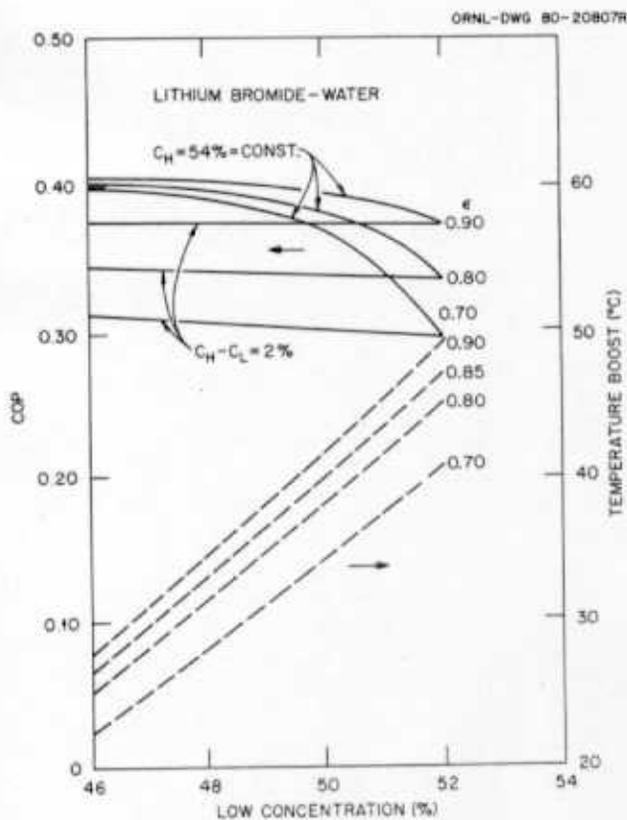


Figure 7. Performance map showing the effect of concentration on COP and temperature boost for a LiBr-water solution for the two-stage absorption heat pump

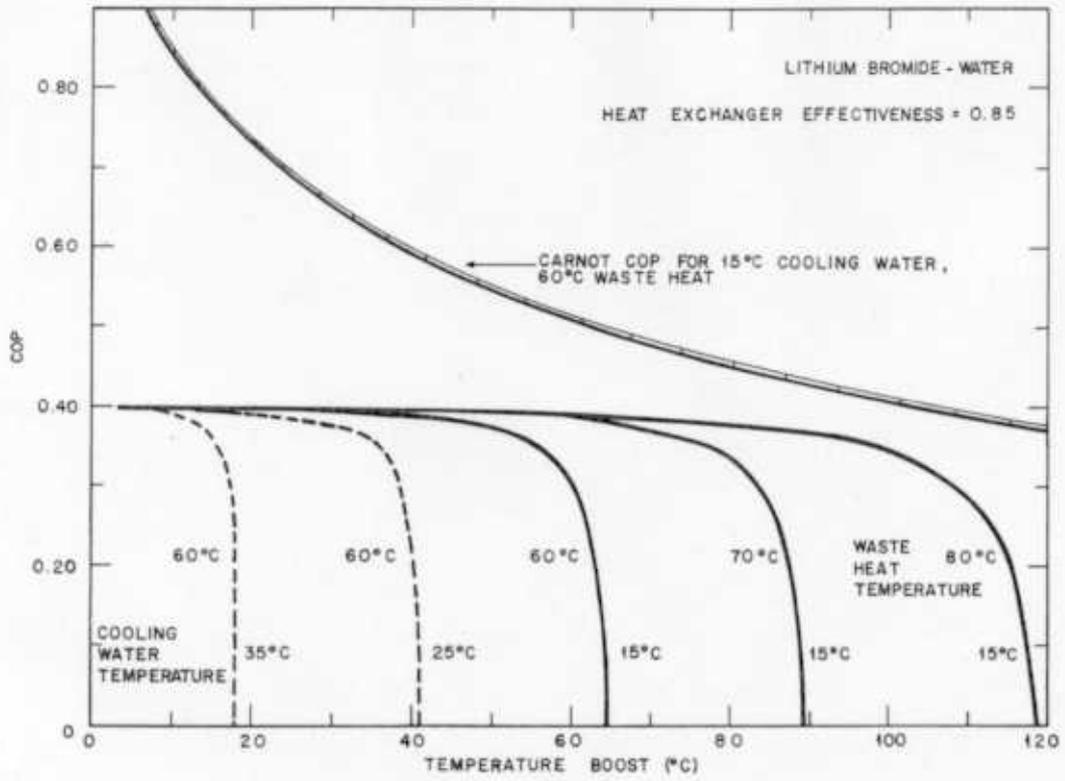


Figure 8. General performance map showing the effect of various heat source and heat sink temperatures for the two-stage heat pump

DISCUSSION

D. HARTMAN, Booz Allen, Bethesda, MD: Have any analyses been done to identify the industries and specific thermal processes that would be compatible with the temperature requirement of this conceptual design heat pump?

G. GROSSMAN: In a separate study, our group has looked at opportunities in different industrial sectors where waste heat is available with an accompanying need for process heat. This information may be found in the report, ORNL/TM-7655, Oak Ridge National Laboratory, 1981.

HARTMAN: Was there any comparison of the weight and size per ton for absorption heat pumps with centrifugal heat pumps?

Absorption machines are normally much larger and heavier on a per ton basis than vapor compression machines. This could have significant ramifications for initial costs.

GROSSMAN: The absorption heat pump has approximately twice the heat transfer area of a vapor compression heat pump, since, in addition to an evaporator and a condenser, the absorption heat pump requires an absorber and a generator of comparable size to the former. However, the compressor and driving motor or engine are eliminated. One must consider the fact that the vapor compression heat pump requires primary energy as a source of power; the absorption system uses the waste heat for this purpose.

U. BONNE, Honeywell CTC, Bloomington, MN: What would you consider to be typical ranges and/or limits of source and sink temperatures for absorption recovery systems based on $\text{LiBr-H}_2\text{O}$ and $\text{LiCl-H}_2\text{O}$ mixtures?

GROSSMAN: Source and sink temperatures affect each other in their influence on performance. The larger the difference between them, the better the performance, but too large a difference can cause crystallization in $\text{LiBr-H}_2\text{O}$ systems and even more so in $\text{LiCl-H}_2\text{O}$ systems. Fig. 8 in the paper describes a reasonable operating range and performance.

J.G. SMELE, P.E., Tech. Consult., Dupont Canada Inc., Kingston, Ont.: The working fluids for system were clearly presented and the questions well answered.

GROSSMAN: Thank you.