

Adiabatic Absorption and Desorption for Improvement of Temperature-Boosting Absorption Heat Pumps

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ABSTRACT

This paper is concerned with an improvement in the conventional absorption heat pump cycle that allows for achieving an increased temperature boost and a better coefficient of performance (COP). The improvement is obtained by adding adiabatic absorption and desorption steps to the absorber and desorber of the system, respectively. The adiabatic processes make it possible to obtain the highest possible temperature in the absorber before any heat is removed from it and the lowest temperature in the desorber before heat is added to it. This, in turn, allows for efficient utilization of the thermodynamic availability of the heat supply stream. Compared to the conventional cycle, the improved system can operate with a larger difference between the high and low concentrations, less circulation losses, and a more efficient heat exchange.

The concept of adiabatic absorption and desorption is described and compared for the conventional and improved cycles. Mathematical expressions are derived for the conditions at the adiabatic points, and design considerations are described for incorporating the improvements in a conventional system. Computer-generated performance curves of COP versus temperature boost are given for the heat pump to illustrate the performance enhancement by the adiabatic process.

INTRODUCTION

Very large quantities of waste heat at low temperatures are rejected daily from industrial plants throughout the world. Other sources of low-temperature heat are available (e.g., geothermal and solar) with a total energy value equivalent to many millions of tons of fossil fuel. In order to make effective use of this heat, it is often necessary to boost its temperature (typically from 60°C [140°F] to 120°C [250°F]). Different heat pump cycles have been considered for doing so, the most common of which require electricity as a source of power.¹⁻³ The absorption cycle is one of the more promising and has the advantage of using part of the waste heat to power itself, while boosting the temperature of the rest.

Low-grade-heat-actuated absorption heat pumps for temperature boosting have received growing attention in recent years. These systems are often referred to as "heat transformers." A one-stage ammonia-water unit was developed to operate on a 65°C hot water supply and 15°C cooling water and delivered 105°C output.⁴ An absorption-desorption heat pump using the same working fluids was described by Cohen, Salvat, and Rojey.⁵ It provided temperature boosts around 30°C with waste heat input above 70°C. A sulfuric acid-water heat pump with storage capability was described by Clark and Morgan⁶ which could deliver a temperature boost of 11°C to a process stream in a range of temperatures from 60°C to 190°C, using a heat sink at 10°C. Other units were studied for space heating, which normally calls for lower delivery temperatures than the industrial applications.⁷⁻¹⁰

A recent paper described a conceptual design study of an absorption heat pump for temperature boosting, performed as the first step toward the construction of a laboratory working system.¹¹ This was part of a more

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comprehensive work where different design alternatives were discussed, including closed versus open cycles, single- and double-stage systems, and various working materials.¹² In the course of the conceptual design, it became apparent that a considerable improvement in the system may be obtained by adding adiabatic absorption and desorption steps to the absorber and desorber of the system, respectively. These adiabatic processes make it possible to obtain an increased temperature boost at a better COP than in the conventional cycle.

This paper will describe the idea of the adiabatic absorption and desorption and compare the conventional and improved cycles. Performance curves of COP versus temperature boost will provide a quantitative measure of the enhancement brought about by the adiabatic processes.

GENERAL DESCRIPTION

Fig. 1 describes a one-stage absorption heat pump in its conventional form. The state points of the system are indicated on the thermodynamic equilibrium diagram of the working fluid described in Fig. 2. For the sake of the discussion, reference will be made to a common fluid pair — lithium bromide-water (LiBr-water). 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 of Fig. 1.

Water vapor at state 8 is produced in the evaporator (which may be of an open type, as shown, or of a closed type, as often used in absorption cooling) at a temperature somewhat below that of the supplied waste heat. The vapor is absorbed in the LiBr solution entering the absorber at the concentrated state 7 and leaving diluted at state 4. The heat of absorption is transferred to the stream of water entering the absorber at 3 and leaving at 9 and boosts its temperature. The dilute solution is transferred to the desorber, where it is concentrated again and returned to the absorber. Waste heat is applied in the desorber to supply the energy required for desorption from state 5 to 6, and a heat sink in the form of a condenser (or something similar) is provided to receive the desorbed vapor. A recuperative heat exchanger is also part of this system, transferring heat from the dilute to the concentrated solution, as shown.

The above process takes place between two pressures: the high pressure prevailing in the evaporator and absorber is determined by the evaporation temperature in the former, as related to the waste heat temperature; the lower pressure prevailing in the desorber and condenser is determined by the condensing temperature in the latter, as related to the available cooling medium temperature. The solution concentration varies between a high value related to the desorber's ability to desorb water and a low value determined by the absorber.

As evident from Fig. 2, the highest temperature in the cycle is at the absorber outlet (state 4), where the dilute solution is in vapor pressure equilibrium with the evaporator. The solution at the inlet (state 7) is at a somewhat lower temperature reached in the passage through the recuperative heat exchanger. The maximum temperature of the boosted stream (state 9) is therefore limited so as to be lower than that of the solution at 4. From this point of view, it is desirable to raise the dilute solution concentration as much as possible so as to raise the temperature at 4, with the limit being the high concentration supplied by the desorber. However, from the performance point of view, it is undesirable to have the high and low concentrations close to each other, as this results in excessive circulation losses.

The improvement discussed herein is based on the fact that the solution at the absorber inlet (state 7) is always subcooled in the heat pump cycle. (This is not necessarily so in the reverse, more-common absorption cycle for cooling.) The improvement consists of a modification in the absorber where an adiabatic part is introduced prior to the heat-transferring part. This is illustrated in Fig. 3 and on the equilibrium diagram in Fig. 4. The solution entering at state 7 is allowed to absorb vapor adiabatically and reach an equilibrium, state 7e, before transferring any heat to the temperature-boosted stream. Due to the high enthalpy of the absorbed vapor at state 8, only a small amount of vapor will be absorbed in the adiabatic process before the solution reaches equilibrium, and, hence, the change in concentration with respect to that of 7 will be small too.

The following advantages are gained by introducing this adiabatic absorption step:

1. The highest temperature in the cycle is now at 7e, which is higher than at 4. This makes it possible to obtain a higher temperature boost.

2. The high temperature is now independent of the dilute solution concentration. This makes it possible to lower considerably the low concentration with respect to the high and, thus, reduce the circulation rate of solution. This, in turn, cuts the pumping power and the losses associated with the imperfect recuperator and finite heat of dilution. The overall performance is improved significantly.
3. The flows of solution and temperature-boosted water in the absorber can now be arranged in counterflow, as opposed to coflow before. This allows for better heat transfer at a smaller approach temperature.

A similar improvement, although less significant, may be obtained in the desorber by dividing it into an adiabatic and a heat-transferring part. The solution entering the desorber at state 5 is always superheated. By allowing it to desorb some of the vapor adiabatically, it will reach an equilibrium state, 5e, after which it can continue to desorb along the equilibrium line $5e \rightarrow 6$ while receiving heat. It is then possible to arrange the flow of solution and water in counterflow, which allows for better utilization of the waste heat for desorption. With given temperatures of the waste heat and cooling medium, the temperature at 6 can now be higher, allowing for a higher concentration out of the desorber.

It should be noted that adiabatic absorption and desorption take place to some extent in most conventional absorption chillers and heat pumps. As the absorbent solution emerges from the distribution device (e.g., spray nozzles) and before it comes into contact with the heat-transfer surface (e.g., tube bundle), it has a short adiabatic contact with the vapor. If during this contact the absorbent is subcooled or superheated, absorption or desorption, respectively, takes place. In absorption chillers, these adiabatic processes are of no consequence from the thermodynamic point of view.* In the heat pump, however, it is desirable to enhance these processes and provide enough contact area and time between the solution and the vapor for equilibrium to be reached under the adiabatic absorption and desorption steps.

ADIABATIC PROCESS CALCULATIONS

The conditions at the end of the adiabatic absorption and desorption steps (state points 7e and 5e, respectively) can be determined based on the inlet conditions and the vapor pressure in the absorber and desorber. Simple mathematical expressions for the temperature and concentration at these two points may be derived as follows.

Consider first the adiabatic absorption point 7e. A mass balance for the adiabatic absorber yields the following equations:

$$\text{conservation of salt: } m_7 C_7 = m_{7e} C_{7e} \quad (1)$$

$$\text{conservation of water: } m_{7e} - m_7 = m \quad (2)$$

where m is the mass of water vapor absorbed adiabatically. An energy balance yields

$$m_7 h_7 + m h_8 = m_{7e} h_{7e} \quad (3)$$

Eliminating m_{7e} and m between Eqs (1), (2), and (3) results in:

$$\frac{C_{7e}}{C_7} = \frac{h_8 - h_{7e}}{h_8 - h_7} \quad (4)$$

which provides one relation between the concentration and the enthalpy at the point 7e, as a function of the given inlet conditions at 7 and 8. Another relation between these two quantities must be provided from the equilibrium characteristics of the given fluid pair. From the two relations, one can determine C_{7e} and h_{7e} . Then, based on the two properties, the temperature at point 7e may be found.

*It is sometimes desirable to have a small inlet superheat in both the absorber and desorber to enhance an even distribution of the solution over the heat-transfer surface. The flashing associated with the superheat provides that enhanced distribution.

Eq (4) indicates the closeness of the concentration at the adiabatic point to that at the inlet. The enthalpy of vapor, h_B , is considerably larger than h_7 and h_{7e} ; hence, the expression on the right-hand side of Eq (4) is close to unity.

In most working fluid pairs, with moderate variations in concentration, the relation between the enthalpy and concentration may be expressed linearly as follows:

$$\frac{C_{7e} - C_L}{C_H - C_L} = \frac{h_{7e} - h_{BL}}{h_{BH} - h_{BL}} \quad (5)$$

where C_H and C_L are the high and low concentrations in the system, respectively, and h_{BH} and h_{BL} are the enthalpies of the solution at the high and low concentrations, respectively, when in equilibrium with the vapor at state 8. Note that $h_4 = h_{BL}$. Note also that the linear relation (5) is only an approximation. Fundamentally speaking, the enthalpy-concentration relation is considerably more complex. However, the linear approximation is a very good representation of reality for many materials, provided the difference between the high and low concentration remains within a few percent.

By solving Eqs (4) and (5) for the two unknown characteristics of the adiabatic point, noting that $C_7 = C_H$, we find:

$$C_{7e} = C_H \frac{(C_H - C_L)h_B + C_L h_{BH} - C_H h_{BL}}{C_H(h_{BH} - h_{BL}) + (C_H - C_L)(h_B - h_7)} \quad (6)$$

$$h_{7e} = \frac{C_H(h_B h_{BH} - h_7 h_{BL}) - C_L h_{BH}(h_B - h_7)}{C_H(h_{BH} - h_{BL}) + (C_H - C_L)(h_B - h_7)} \quad (7)$$

Similar expressions may be obtained for the concentration and enthalpy and, hence, temperature, at the adiabatic desorption point, 5e.

DESIGN CONSIDERATIONS

The actual design of a system involving the adiabatic absorption and desorption may be done in a number of different ways. In this paper, no attempt has been made to specify the best of them or all of them. One may design the adiabatic and heat-transferring parts of both the absorber and the desorber as two separate chambers in series or as one chamber. The main principle to follow in the design is that the absorbent solution at the inlet be given enough contact time and area with the vapor to achieve equilibrium adiabatically, before it reaches the heat-transfer surface.

Several devices may be used to "delay" the absorbent solution at the inlet long enough to achieve the adiabatic process. The schematic criss-crossed box in Fig. 3 can contain some form of ceramic or plastic packing, e.g., Raschig rings, as often used for this purpose in various gas-liquid contactors. Alternatively, one can use an arrangement of slanted fins or perforated trays. A very convenient arrangement for the conventional horizontal-tube type absorber would be an extension of its own tube bundle. Specifically, one can use a series of "dummy" tubes, which can be staggered and supported above the tube bundle on the same tube sheets used to support the latter. Then, no change has to be made in the distribution device; the absorbent solution is spread over the dummy tubes and drips from them on the heat-transferring tubes, thus eliminating the need for redistribution between the adiabatic and conventional absorbers. Generally speaking, a unit surface area of adiabatic absorber or desorber would be considerably cheaper than that of a heat-transferring unit.

The improvement associated with adiabatic absorption and desorption may be applied to multistage absorption heat pumps. Single- and double-stage systems have been discussed in Refs 11 and 12. The two-stage cycle is an extension of the simpler one-stage described in Sect. 2, aimed at achieving a higher temperature boost. A preferred configuration of a two-stage heat pump in the conventional form is illustrated in Fig. 5. The system employs two absorber-evaporator stages, both served by one desorber-condenser. The improved version of this system is described in Fig. 6.

PERFORMANCE CALCULATIONS

The improvement associated with the adiabatic absorption and desorption steps and its various advantages have been described in a qualitative manner. A computer program has been developed that performs a full simulation of the heat pump, making it possible to compare the improved cycle with the conventional one on a quantitative basis and to evaluate the magnitude of the improvement in temperature boost and COP. Some typical results are described below.

Figs. 5 and 6 illustrate the two-stage absorption heat pump in the conventional and improved configurations, respectively. Both configurations have been compared here on the basis of equal operating conditions, as follows:

1. Low-grade heat is supplied to the heat pump at a constant temperature of 60°C (140°F).
2. The heat rejection (condensing) temperature is maintained constant by the cooling water at 20°C (68°F).
3. The evaporation temperatures in the first and second stages (T_B and T_{1B} , respectively) have been selected so as to flash-evaporate 1% of the mass of the entering stream.
4. All the heat exchangers in the system have been assumed to be designed with the same closest approach temperature (CAT), i.e., the same smallest temperature difference between the cold and the hot streams. Two sets of curves are presented for each heat pump configuration, for a CAT of 2.5°C (4.5°F) and 4.0°C (7.2°F).
5. Thermodynamic equilibrium is obtained at the end of the adiabatic processes (states 7e, 17e, and 25e) and also at the absorber and desorber outlets, points 4, 14, and 26.
6. Pressure drops in the system are negligible.

The above conditions fix the operating state of the heat pump completely, leaving only one degree of freedom — in selecting the low absorbent concentration. Different selections of that concentration, within a limited range, provide for different temperature boosts at different COPs.

Fig. 7 describes the results of the computer simulation performed for the conventional and the improved cycles operating under the above conditions. It shows the overall COP as a function of the temperature boost for both systems, for two values of the CAT. The solid lines indicate the improved cycle and the broken lines, the conventional cycle.

The curves demonstrate clearly the advantage of the improved cycle in providing a higher temperature boost at a higher COP than the conventional one. In both configurations, a better COP may be obtained at the cost of a lower temperature boost by reducing the low solution concentration. The range for this is smaller in the improved cycle than in the conventional cycle, indicating that the temperature boost and COP are fairly insensitive to the low concentration in the former.

A typical operating point with the improved cycle, selected from Fig. 7 at CAT = 2.5°C, would yield a temperature boost of 73.0°C at a COP of 0.38. Such a temperature boost is not possible with the conventional cycle. At the same COP of 0.38, the conventional cycle would give a temperature boost of only 45.0°C.

In an actual working system with a finite transfer area, full thermodynamic equilibrium cannot always be achieved at the end of an absorption process. This is known from experience with absorption chillers, where the absorbent solution at the absorber outlet is often somewhat subcooled. A deviation from equilibrium may well occur in an adiabatic absorber or desorber. It is therefore important to investigate the effect of partial adiabatic absorption and desorption on performance.

Fig. 8 describes four performance curves generated by the computer simulation program for the improved two-stage heat pump of Fig. 6. The curve for an unimproved system is plotted along for comparison. The operating conditions assumed are as stated earlier in this section for Fig. 7, with CAT = 4.0°C (7.2°F), except that a deviation from thermodynamic equilibrium has been introduced at the end of the adiabatic processes. This deviation is expressed in terms of a temperature shift from the equilibrium temperature corresponding to the concentration in question under the given vapor pressure. The four curves shown correspond to deviations of 0, 1, 2, and 3°C. As expected, the greater the deviation from equilibrium, the greater the decrease on the obtainable temperature boost.

When deviation from thermodynamic equilibrium exists only in the adiabatic absorbers, and not in the desorber, one can operate anywhere along the performance curve (although it is advantageous to operate somewhere in the "knee" of the curve). When the deviation exists in the desorber also, only the solid part of the curve is acceptable for operation. The larger the deviation, the larger the part of the curve shown in a broken line, where the temperature at the adiabatic desorption point (state 25e) is too high to transfer heat to the waste heat stream supplied for regeneration.

CONCLUSIONS

This paper has described an improvement in an absorption heat pump cycle obtained by introducing adiabatic absorption and desorption steps into the absorber and desorber, respectively. These adiabatic processes make it possible to obtain the highest possible temperature in the absorber before using its heat for temperature boosting and the lowest temperature in the desorber before applying low-grade heat to it. This, in turn, allows for efficient utilization of the thermodynamic availability of the heat supply stream.

The advantages of the improvement to the cycle are the following:

1. Higher temperature boosts than in the conventional cycle are obtained under the same waste heat and cooling medium conditions.
2. Better coefficient of performance is obtained, which is equivalent to more energy output in the temperature-boosted stream per unit energy input.
3. Parasitic power losses are reduced.

These advantages have been illustrated in the performance curves of COP versus temperature boost for the conventional and improved system in Fig. 7.

NOMENCLATURE

C_H	high system concentration (%)	h_{8L}	enthalpy of solution at the low concentration, C_L , in equilibrium with vapor at state 8 (J/kg)
C_i	concentration at state point i (%)	h_i	enthalpy at state point i (J/kg)
C_L	low system concentration (%)	m	of refrigerant vapor absorbed adiabatically (kg/sec)
h_{8H}	enthalpy of solution at the high concentration, C_H , in equilibrium with vapor at state 8 (J/kg)	m_i	mass flow rate at state point i (kg/sec)

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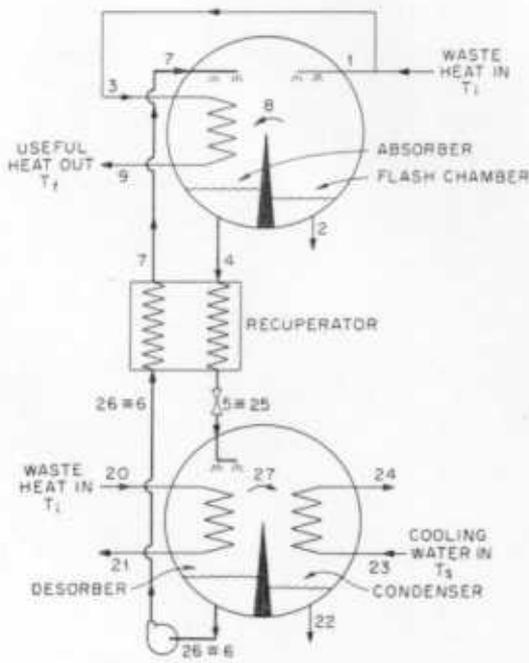


Figure 1. Schematic description of a single-stage temperature-boosting absorption heat pump in conventional form

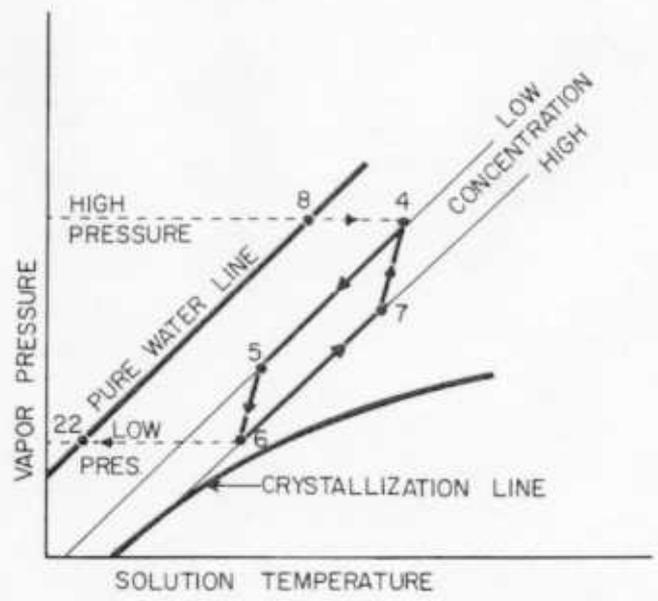


Figure 2. Cycle diagram for the heat pump described in Fig. 1

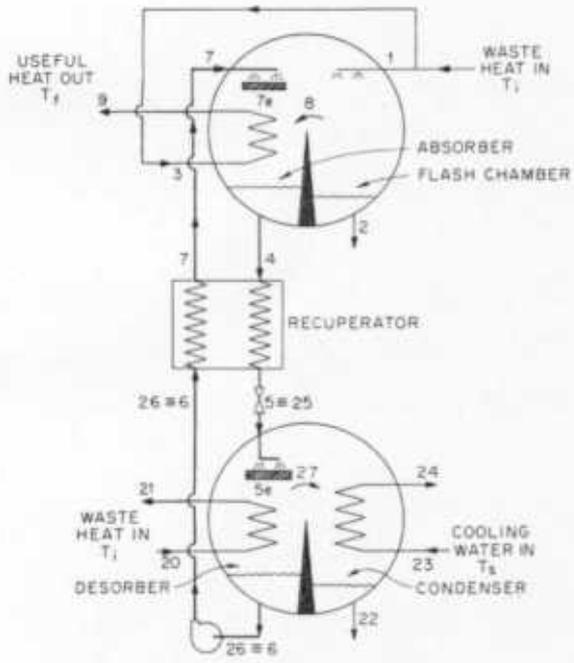


Figure 3. Schematic description of a single-stage temperature-boosting absorption heat pump with adiabatic absorption

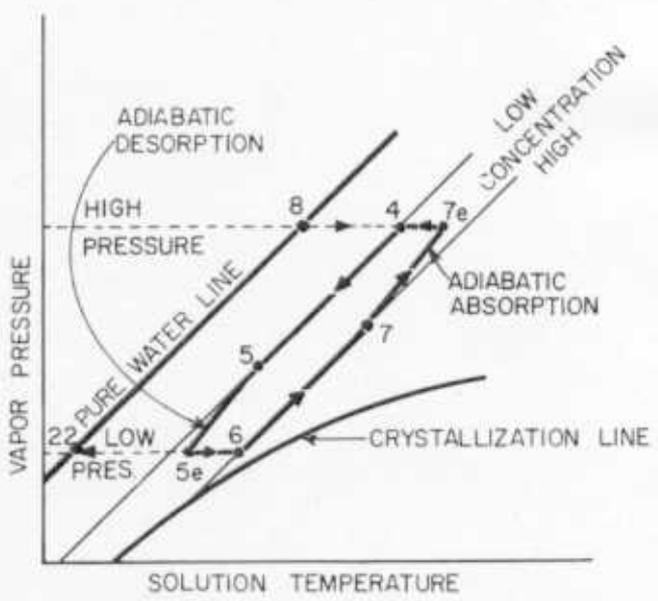


Figure 4. Cycle diagram for the heat pump described in Fig. 3

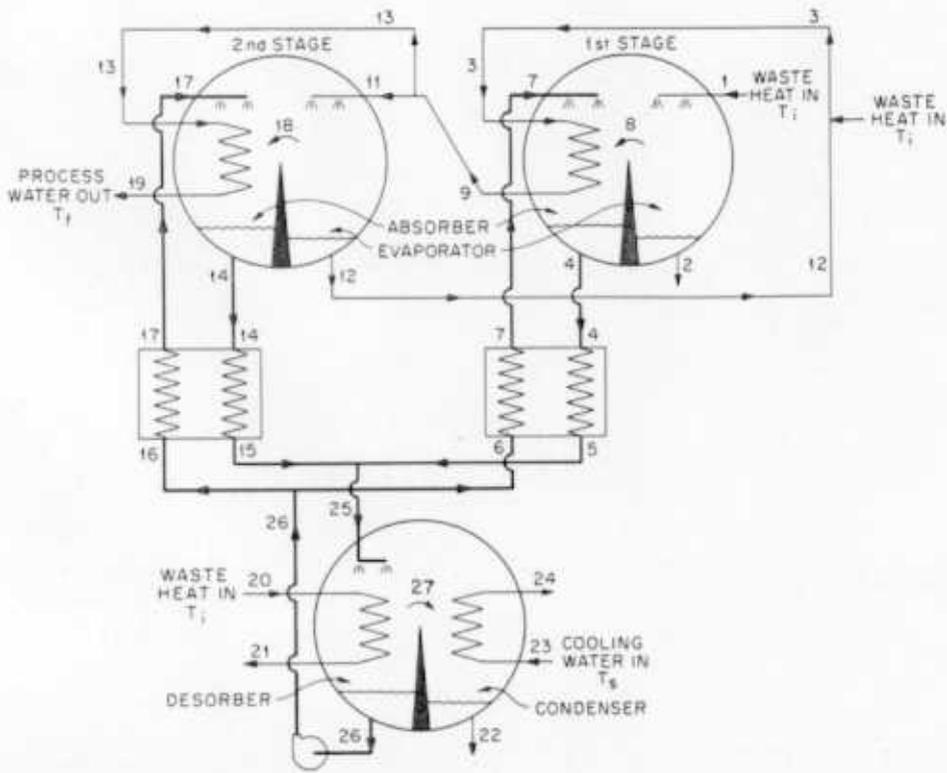


Figure 5. Schematic description of a double-stage temperature-boosting absorption heat pump in conventional form

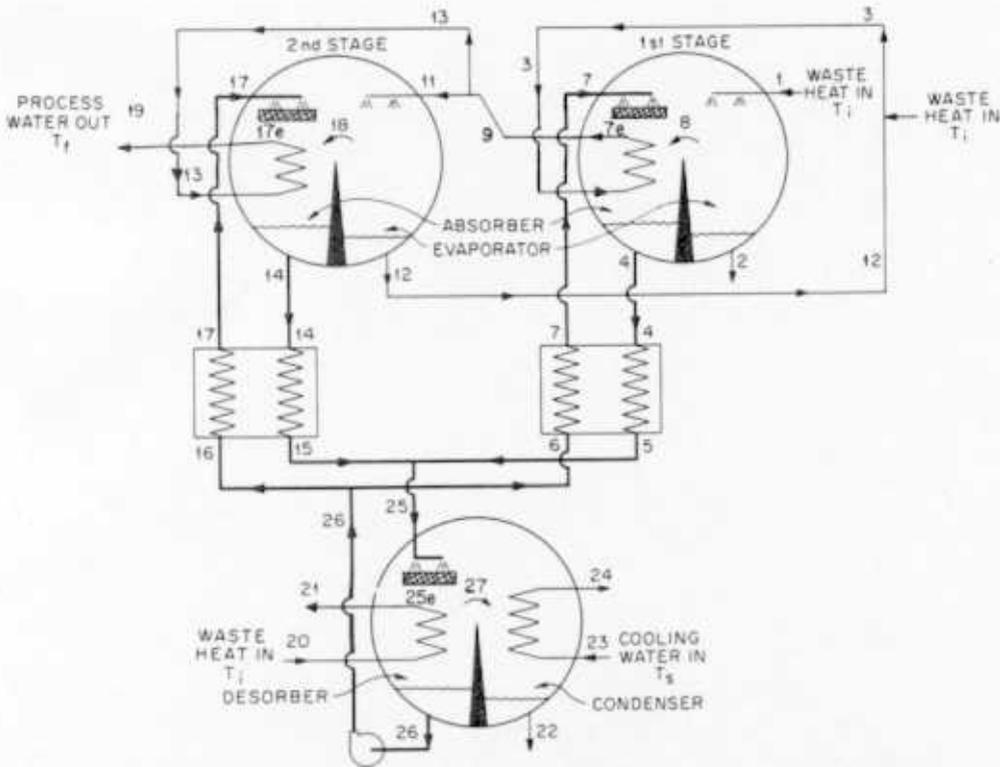


Figure 6. Schematic description of a double-stage temperature-boosting absorption heat pump with adiabatic absorption and desorption steps

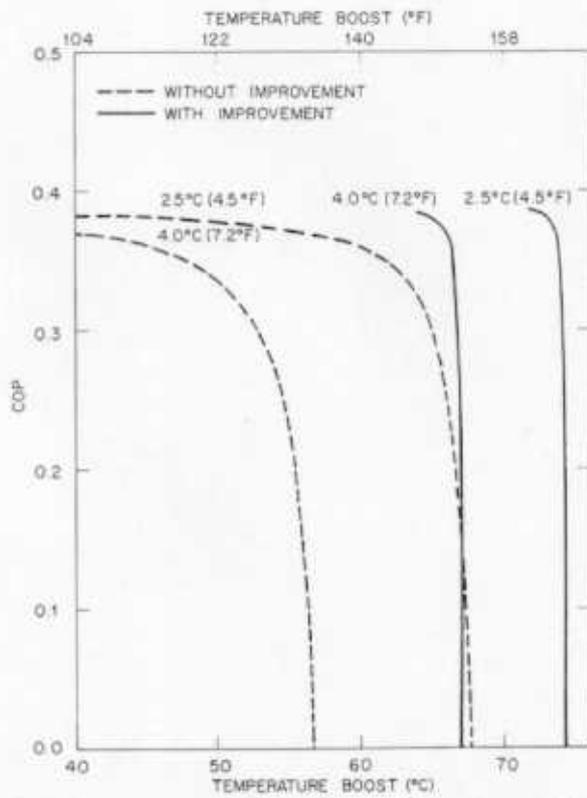


Figure 7. Performance curves for the double-stage absorption heat pump in the conventional and improved forms

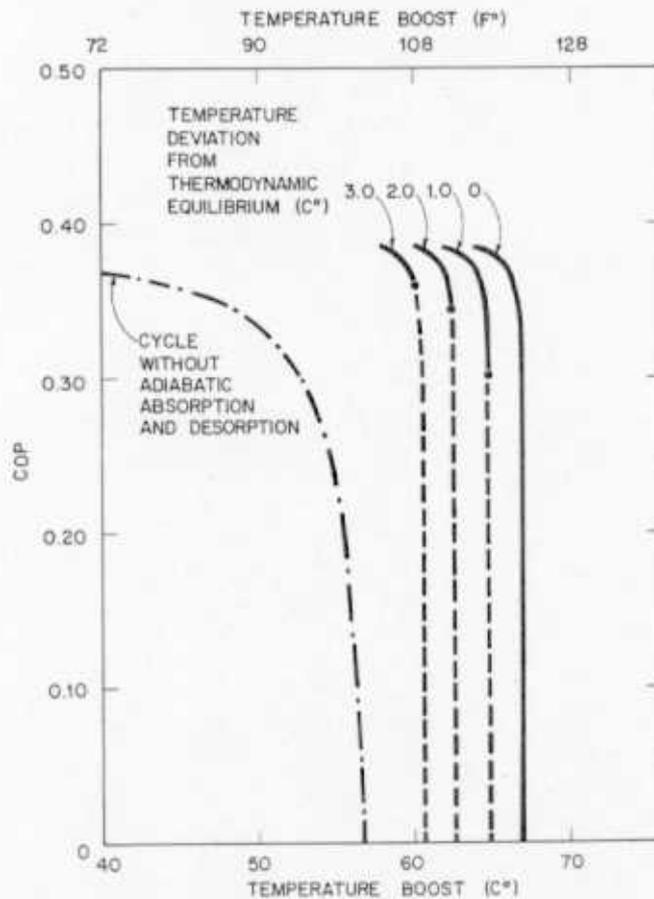


Figure 8. Effect of partial adiabatic absorption and desorption on performance for a double-stage absorption heat pump; deviation from thermodynamic equilibrium is expressed in terms of a temperature shift