ABSORPTION TECHNOLOGY FOR SOLAR AND WASTE HEAT UTILIZATION

G. GROSSMAN

Faculty of Mechanical Engineering, Technion — Israel Institute of Technology, Haifa

ABSTRACT. Absorption heat pumps, first developed in the 19th century, have received renewed and growing attention in the past two decades. With the increasing cost of oil and electricity, the particular features of this heat-powered cycle have made it attractive for both residential and industrial applications. Solar-powered air conditioning, gas-fired domestic cooling and waste- heat-powered temperature boosters are some of the applications on which intensive research and development has been conducted. This paper describes the operation of absorption systems and discusses several practical applications. It surveys recent advances in absorption technology, including the selection of working fluids, cycle improvements and multi-staging, and fundamentals of the combined heat and mass transfer in absorption processes.

INTRODUCTION

Absorption heat pumps, first developed in the 19th century, have received renewed and growing attention in the past two decades. With the increasing cost of oil and electricity, the particular features of this heat-powered cycle have made it attractive for both residential and industrial applications. Solar-powered air conditioning, gas-fired domestic cooling and waste-heat-powered temperature boosters are some of the applications on which intensive research and development has been conducted.

One attractive application for absorption systems is solar cooling[1]. A single-stage absorption unit may be employed to produce chilled water for air conditioning, using for its source of power solar-heated water at below-boiling temperatures, obtainable from low cost flat-plate solar collectors. Chilled water is cooled in the evaporator where distilled water serves as a refrigerant, evaporates and is absorbed by a strong LiBr-water solution in the absorber. The solar-heated water is applied to reconcentrate the solution in the generator by desorbing the excess refrigerant, condensing it, and returning it to the evaporator. The system may be staged for better utilization of higher temperature heat sources. Also, the same system can use for its operation, instead of the solar heat, low grade heat from any other source, including industrial waste heat[2].

A similar system, operating in reverse, may be employed for temperature boosting, making it possible to convert industrial waste heat back to useable process heat[3]. This system, known as a heat transformer, uses part of the waste heat itself as a source of power. An application of particular interest is the conversion of low grade heat collected in solar ponds into process steam, at a coefficient of performance far exceeding that obtained in the more conventional electric power mode[4].

This paper will survey recent advances in absorption technology, including the selection of working fluids, cycle improvements, multi-staging, and fundamentals of the combined heat and mass transfer in absorption processes.

PRINCIPLES OF THE ABSORPTION CYCLE

An absorption system is described schematically in Figure 1 showing its major sub-units. The absorbent flow is indicated by a heavy line, to be distinguished
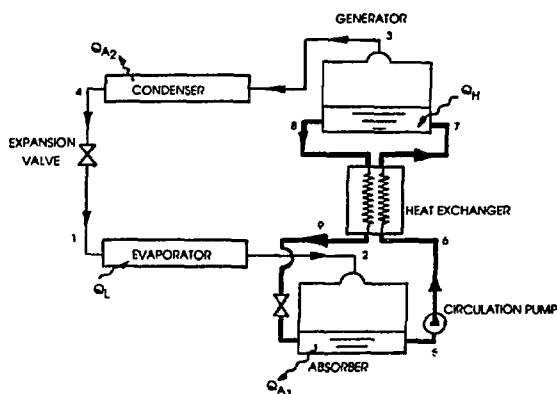

Fig. 1. *Schematic diagram of absorption cooling system*

from the flow of refrigerant. Liquid refrigerant entering the evaporator at state 1 evaporates at the low temperature T_L while removing heat Q_L from the cooled medium. The vapor at state 2 goes into the absorber where it is absorbed by a concentrated (strong) absorbent solution. The latter must be cooled during the process, to remove the heat of absorption. This is necessary for maintaining a low vapor pressure which allows for efficient absorption. The effect of this process on the refrigerant vapor is equivalent to the compressor suction in the vapor compression cycle.

The saturated (weak) absorbent leaves the absorber at state 5 and is pumped via the heat exchanger into the generator. Entering at state 7, it is heated to the high temperature supplied by the solar energy source and its vapor pressure increases to the point where it can no longer contain the amount of refrigerant it had. The latter is released at a high pressure (state 3) and flows into the condenser. The result produced by this process is similar to that of the compressor discharge in vapor compression. The refrigerant entering the condenser in the superheated state 3 condenses (state 4) and expands into the evaporator. The concentrated (strong) absorbent, having released the refrigerant, leaves the generator at state 8 and returns to the absorber at 9.

Since the absorbent solution must be heated in the generator and cooled in the absorber, a recuperative heat exchanger is provided between the two streams to recover heat from the strong and preheat the weak absorbent. A circulation pump is also needed to transfer the solution from the low pressure absorber to the high pressure generator. In some designs, the pump is replaced by a vapor lift system.

Figure 2 is an equilibrium diagram for the absorbent-refrigerant solution, where vapor pressure is plotted as a function of temperature for different solution concentrations. To exhibit the thermodynamic states at the different points in the system, a log P vs. 1/T plot based on the Vant Hoff equation is employed where the abscissa is the reciprocal of the absolute temperature and the ordinate is the logarithm of the vapor pressure. In these coordinates, lines of constant concentration are practically straight. Numbers in the diagram correspond to those in Figure 1 designating the states at the different points in the system. A number in parentheses indicates a non-saturation or non-equilibrium state.

Points 1 and 2 characterize the inlet and outlet conditions in the evaporator, both at saturation; points 3 and 4 describe the same in the condenser. The refrigerant at state 3 is superheated, hence the parentheses around the number 3. The pressure P_C in the condenser and the generator is represented by the
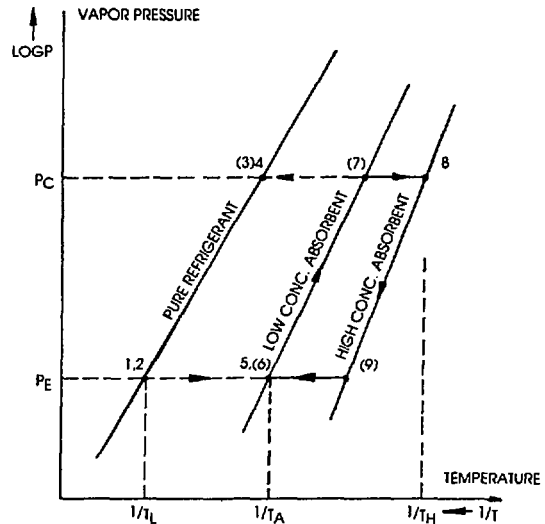

Fig. 2. *Thermodynamic equilibrium diagram of the absorbent-refrigerant solution in an absorption system*

ordinate through point 4, corresponding to refrigerant saturation at the heat rejection temperature T_A. Similarly, the pressure in the evaporator and absorber, P_E, depends on the cooled medium temperature T_L. The absorbent cycle is composed of the two isobars 7-8 and 9-6 in the generator and absorber respectively and of two fixed concentration lines in the passage through the recuperative heat exchanger. The solution entering the absorber at the non-equilibrium state (9) is diluted from the high to the low concentration. The absorption process terminates at point 5 where pressure equilibrium is reached at the heat rejection temperature T_A. Similarly, the termination point 8 of the desorption process in the generator is determined by equilibrium between the condenser pressure P_C and the heat supply temperature T_H. Thus, the low and high concentrations of the cycle are determined by equilibrium between P_E and T_A and between P_C and T_A, respectively, while P_E and P_C are themselves determined by T_L and T_A. In this discussion we have ignored, for simplicity, the temperature differentials between the transfer and working fluids, and assumed ideal recuperative heat exchange between the strong and weak solutions. In practice there is normally some superheat or subcool in the solution entering the absorber and generator, respectively. Furthermore, points 4 and 5 are shown to be at the same temperature T_A. This is correct only if the condenser and absorber are supplied with cooling water from the same source in parallel. Where the cooling tower is limited in size, their cooling water passages would be arranged in series

and points 4 and 5 will not have the same temperature.

If the cooling water was warmer, keeping T_H and T_L the same, T_A and the pressure P_C would be higher. This would cause point 5 to move to the right and point 8 to the left, resulting in a decrease in the high concentration and an increase in the low concentration. With the difference in concentrations between the weak and strong solutions reduced, one would need to circulate a higher flow of absorbent to carry the same cooling load. At some point, with a further increase in the cooling water temperature, the lines of low and high concentration would coincide bringing the operation to a halt. It hence becomes clear that in a given system a minimum generator temperature is required for given values of cooling water and cooled medium temperatures. The value of this minimum T_H is high at the higher T_A and the lower T_L.

The coefficient of performance of the conventional absorption device, defined as the ratio of cooling capacity to heat input, has a theoretical limit of unity. The heat supplied to the generator is used to evaporate the refrigerant formerly absorbed in the solution, and hence must provide about the same heat of evaporation removed from the cooled medium. In addition, the generator must overcome the heat of dilution and the heat losses in heating and cooling the absorbent at each cycle, due to the non-perfect recuperative heat exchanger. For this reason the C.O.P. of a single stage system is less than unity.

As mentioned earlier, the absorption system described above may be operated in a reverse mode for temperature-boosting of low-grade heat. Figure 3 describes schematically the single-stage heat transformer. The system consists of four primary components — evaporator, absorber, desorber and condenser, in which the heat input or output causes a change of state of the working fluid. In addition, there is a recuperative heat exchanger, pumps and piping. For simplicity, the system is described in a form suitable for operation with a nonvolatile absorbent such as lithium bromide-water. Absorbate vapor at state 8 is produced in the evaporator, heated by the low grade heat source. The vapor is absorbed in the absorbent solution that enters the absorber at a strong state 7 and leaves weak 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 weak solution is transferred to the desorber, where absorbate is removed from it, then returned in a strong state to the absorber. Low-grade heat is supplied to the desorber to provide the energy for desorption from state 5 to 6, and cooling water is applied to the condenser, which
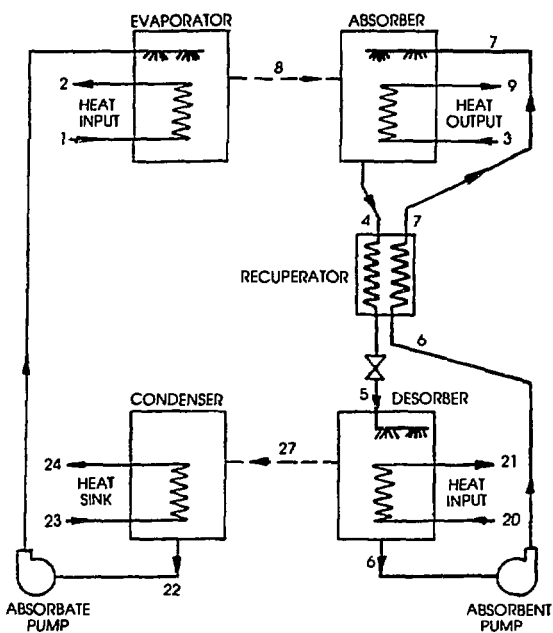

Fig. 3. *Schematic description of a single-stage heat transformer*

provides the heat sink for the desorbed vapor. The condensate is returned to the evaporator by the absorbate pump. A recuperative heat exchanger transfers heat from the weak to the strong solution, as shown. The coefficient of performance of this system (useful heat output per waste heat input) is around 0.5.

WORKING FLUID COMBINATIONS

The selection of the refrigerant/absorbent combination is always the starting point in the design of an absorption system. Several properties are desired in the working fluids, some of which are absolutely necessary for efficient operation.

The most important characteristics may be summarized as follows:

1. 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 generator conditions. If this requirement is not met, large amounts of absorbent solution must be circulated between the absorber and generator in order to transfer the quantity of refrigerant needed to produce the cooling effect. This in turn causes excessive internal heat losses leading to poor performance;

2. The heat required to release the refrigerant from the solution in the generator should be close to the heat of vaporization of the

refrigerant in the evaporator. The ratio of the former to the latter is normally greater than unity due to the finite heat of dilution which must be rejected in the absorber. The heat which must be spent in the generator in order to release the refrigerant from solution over and above that which is required to vaporize it, goes to waste;

3. The properties of the absorbent-refrigerant solution which affect heat and mass transfer, such as viscosity, diffusion coefficient, thermal conductivity etc. should be favorable;

4. The absorbent should be non-volatile, or considerably less volatile than the refrigerant, to prevent the transfer of absorbent into the condenser and evaporator. When this condition is not fulfilled, an absorbent separator, also known as a rectifier, must be installed between the generator and the condenser;

5. The solidification temperature of the absorbent/refrigerant solution, either by crystallization or by freezing, must be lower than and out of the range of the operating conditions of the device;

6. The working fluids should be chemically stable, nonflammable, non-toxic and non-corrosive to common materials of construction;

7. The cost of the materials should be minimal.

Of the many fluid pairs considered and experimented with for absorption systems, only two have found their way to large scale use in commercial units, having satisfied most of the above criteria. These are the absorbent-refrigerant combinations a water-ammonia and lithium bromide-water, respectively. The latter is preferred in air conditioning, since ammonia is toxic and therefore dangerous for this purpose. The lithium bromide-water combination seems more promising for solar applications in particular, as it yields higher coefficients of performance with temperatures obtainable from flat plate solar collectors. Its disadvantages are in that (1) lithium bromide salt crystallizes out of solution when the temperature is lower and the concentration higher than at saturation, thereby clogging pipes and passages. (Even though the machine is designed for operation away from crystallization, this may still occur during transient conditions.) (2) Lithium bromide salt is highly corrosive to metals, particularly in the presence of air, and (3) the salt is relatively expensive. The aqua-ammonia combination is used more in refrigeration, where the low temperatures required rule out the use of water as a refrigerant due to freezing. These systems operate at pressures considerably higher than atmospheric, as opposed to

the LiBr-water operating under vacuum. A rectifier is required downstream of the generator as the water evaporates along with the ammonia.

A number of other working fluid combinations has been tested in experimental setups, trying to obtain better performance under the solar operating conditions. Systems were built employing the binary mixtures sodium thiocyanate-ammonia[5,6], lithium chloride-water[7], lithium nitrate-ammonia[8], calcium chloride-ammonia[9] and others. It appears at present that none of these fluid pairs has shown a definite advantage justifying its utilization in solar cooling installations in place of the lithium bromide-water or aqua-ammonia solutions.

An interesting suggestion to use a three materials combination in absorption was made by Radermacher and Alefeld[10], who experimented with a tertiary mixture of lithium bromide, water and ammonia. The presence of lithium bromide in the water reduces the equilibrium vapor pressure of the mixture at a given temperature and "holds" the water in solution in the generator, while the ammonia is free to evaporate. Thus the amount of water carryover in the refrigerant may be reduced to the point where a rectifier is no longer needed. The operating pressures of the entire system are also reduced considerably, simplifying construction. This can be achieved if the amount of LiBr in solution is small enough to keep away from crystallization. It appears that tertiary mixtures have an important promise as working fluids in absorption systems.

ADVANCED ABSORPTION CYCLES

In describing the principles of the absorption cycle it was shown that the coefficient of performance in a conventional one-stage system is theoretically limited to unity, and is practically always below that (around 0.7 with lithium bromide-water). Furthermore, a minimum generator temperature is needed to operate the system at given evaporator and cooling water temperatures. These two limitations do not exist in the vapor compression cycle.

It is possible to increase the C.O.P. of absorption devices considerably and overcome the minimum high temperature requirement by a multi-stage arrangement. A variety of these kinds of schemes has been proposed. They offer a great deal of flexibility in adapting absorption cooling for different applications and modes of operation. Richter[11] described several such systems employing aqua-ammonia for refrigeration. In the solar application the main purpose of multi-stage operation is to make better use of the available solar heat which may vary in quantity and temperature at different times.

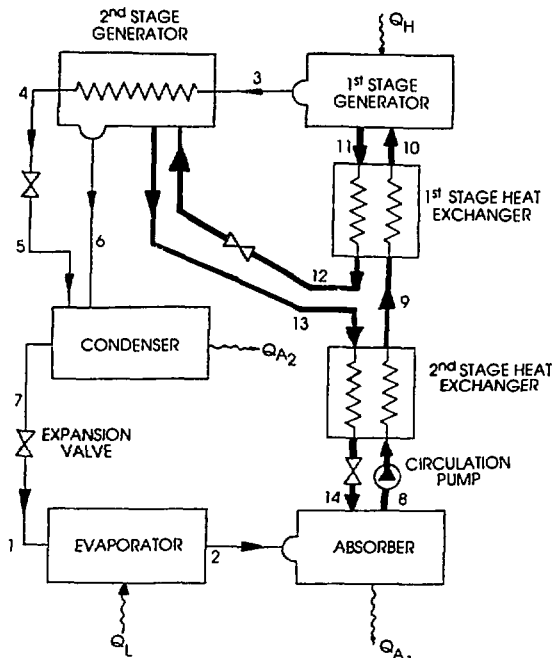Figure 4 describes schematically a double-stage

Fig. 4. *Schematic diagram of double-effect absorption cooling system*

Fig. 5. *Thermodynamic equilibrium diagram of the absorbent-refrigerant solution in double-stage absorption cooling*

system designed to increase the C.O.P. over that which may be obtained under the same operating conditions with a one-stage unit. The changes in state of the working fluids are indicated on the log P-1/T diagram of Figure 5. Reference should be made to Figures 1 and 2 describing the one-stage cycle for comparison. This two-stage system has a single evaporator, absorber and condenser, the former two operating as in the one-stage cycle. The first generator receives from the absorber the dilute (weak) solution and concentrates it partly, to a medium concentration. The refrigerant vapor desorbed from the first stage condenses in the second generator and releases heat which is used to concentrate the solution further. The vapor desorbed therefrom passes over to the condenser along with the condensate from the first stage. Liquid refrigerant from the condenser expands into the evaporator and the cycle continues as in the one-stage system. Two recuperative heat exchangers are provided for the first and second stages.

The net result of the above process is that each unit of heat supplied to the generator produces almost twice the amount of refrigerant vapor it would in a one stage system: once in the first stage, by virtue of its own energy and again in the second stage by the heat of condensation of the vapor produced in the first. This results in almost doubling the coefficient of performance which is theoretically
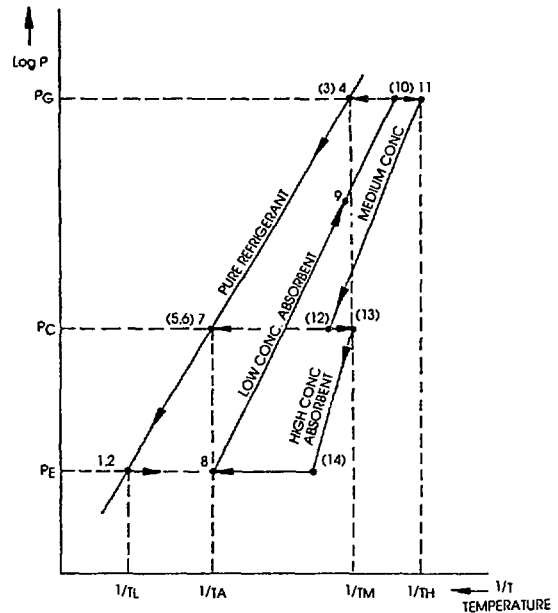
less than 2.0 and in practice around 1.2. However, the additional stage requires a higher temperature heat source, which rules out the use of flat plate collectors and calls for the concentrating or the evacuated-tube type for the solar application.

It should be noted that the system just described is only one version of many possible multi-stage schemes[11,12]. Despite their potential for improved performance, the additional complexity of those systems has thus far limited their use in solar cooling. Multistaging is equally effective in heat transformers covering a wide variety of industrial applications[13].

OPERATING CHARACTERISTICS

Under the Section on Advanced Absorption Cycles, it was shown that the coefficient of performance of a single-stage absorption unit has its theoretical limit at unity due to the fact that the heat supplied to the generator must vaporize at least the amount of refrigerant previously evaporated in the evaporator. Figure 6 describes the C.O.P. of a typical solar lithium bromide water absorption chiller[14] as a function of the hot water temperature supplied to the generator, for different cooling water temperatures and for a fixed temperature of the chilled water. The curves were obtained by computer simulation taking into account the actual heat transfer areas and coefficients in the machine and were checked by
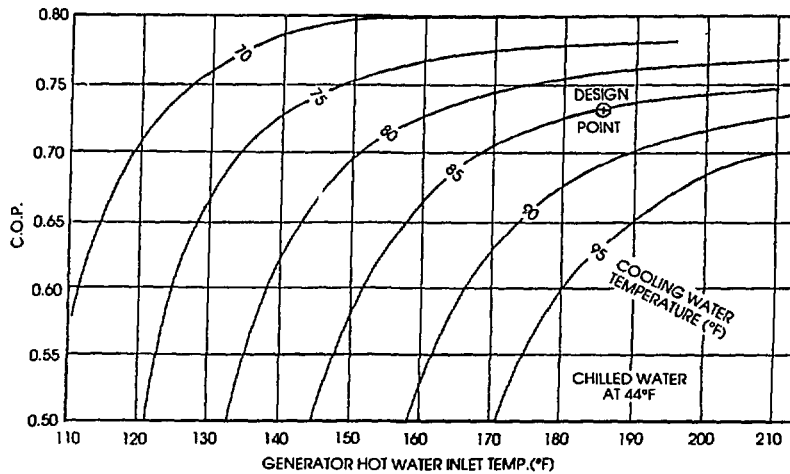
Fig. 6. *C.O.P. curves of a typical LiBr-water absorption chiller*

solution are relatively high. However, when operating at full design capacity, most of the heat supplied to the generator goes to evaporation of the refrigerant and supplying the heat of dilution associated with its desorption. Since this heat is fairly sensitive to generator or cooling water temperatures, and is proportional to the amount of refrigerant evaporated, the ratio of evaporator to generator load tends to be constant.

Figure 7 shows the cooling capacity of the above system[14] as a function of the same parameters as with the C.O.P. The capacity is seen to increase linearly with the temperature of hot water to the generator, and to be affected favorably by a decrease in cooling water temperature. This means that whenever cooling water at a lower than the design temperature is available, the hot water temperature can be reduced for the same capacity. A reduction in the available cooling water temperature usually coincides with a reduction in the required cooling capacity and consequently the hot water temperature can be reduced even further. A control system designed to automatically adjust the hot water temperature in relation to the cooling water temperature and the required cooling capacity would therefore result in improved C.O.P. of the absorption machine (Figure 6) and also in improved efficiency of heat collection by the solar system where more heat can be collected at lower temperatures. Further improvements in system performance may be obtained by the division of the hot water storage into a number of different temperature compartments and the provision of different types of solar collectors: the evacuated tube or concentrating type collectors for high water temperatures and flat plate collectors for reduced temperatures.
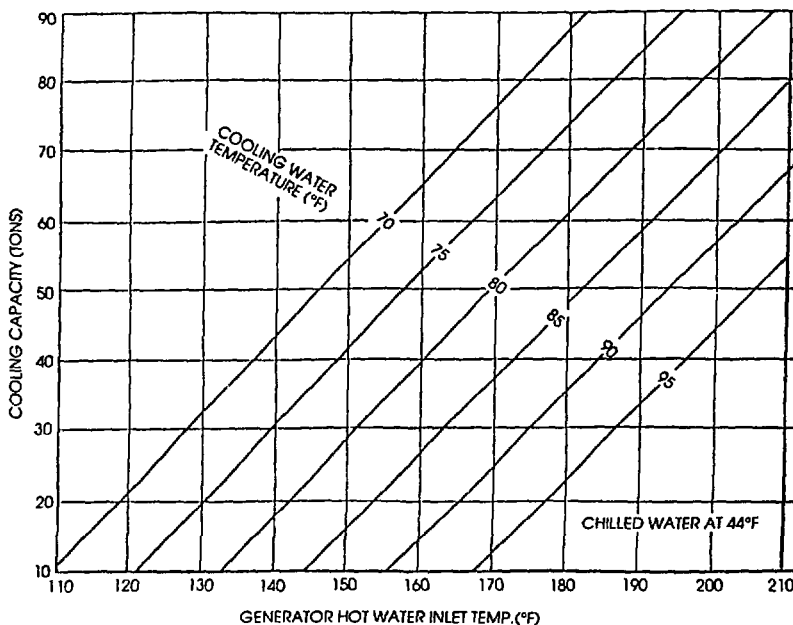
experimental data. They show the C.O.P. to increase rapidly with the hot water temperature up to a certain point, and then level off and reach an asymptotic value. Higher generator temperatures will not improve the C.O.P. Lower cooling water temperatures lead to better performance. However, one can see that the dependence on this temperature is more significant at low generator temperatures but less in the asymptotic range. This behavior is due to the fact that at low generator temperatures the system operates at low capacity and the internal heat losses associated with sensible heating and cooling of the

The upper limit for the capacity which may be obtained from a given machine is set by practical difficulties occurring

Fig. 7. *Cooling capacity curves of a typical LiBr-water absorption chiller*

in off-design conditions. In lithium bromide-water systems, an attempt to raise the generator temperature or lower the cooling water temperature excessively, would lead to crystallization. In all types of practical systems the large flow of refrigerant vapor associated with larger-than-design capacity can cause carryover of liquid refrigerant from the evaporator to the absorber, thus disrupting operation.

ABSORPTION SYSTEM SIMULATION

The need for computer simulation models of absorption systems has become apparent with the renewed interest in these systems during the past two decades. The advantage of the absorption cycle for heat pumping applications, where heat may be utilized as a source of power in preference over mechanical energy, has been recognized. A variety of advanced cycles in multistage configurations has been proposed to fully utilize the thermodynamic potential of the heat source at hand. The development of working systems based on these cycles and the desire to investigate novel cycles and working fluid created the need for effective and reliable simulation.

Several computer simulation codes of absorption systems have been developed and described in the literature. The results of some of them have been compared with experimental data and found to be in good agreement. The above codes were all system-specific, that is, they were written to simulate one particular system with a particular design, flow arrangement and working material. Their structure did not allow easy modification to model other systems. In order to do that, major parts of the program and particularly the iterative sequence would have to be rewritten.

This section describes a flexible computer simulation code capable of simulating absorption systems in varying cycle configurations and with different working fluids. The need for such a code grew out of repeated attempts under absorption research programs to evaluate new ideas for advanced cycles and working substances and compare them with existing ones. The work is described in greater detail in Refs. [15,16]. The code may be used not only for evaluating new cycles and working fluids but also to investigate a system's behavior in off-design conditions, to analyze experimental data and to perform preliminary design optimization.

The objective of developing a flexible and user-oriented simulation code has led to two basic requirements with respect to the program structure. First, the program had to be modular in nature to enable the user to specify different cycle configurations, different working fluids and obviously different unit sizes and operating conditions. Second, the input requirements had to be kept simple and straightforward.

Recognizing that each absorption system consists of standard components (e.g. absorber, evaporator, desorber, condenser, etc.), the following logic was developed for the code: each basic component is simulated by a unit subroutine providing a mathematical expression of the physics of that component. Each unit subroutine contains all the physical equations required to fully describe of its behavior, such as energy balance, conservation of mass for each material species, heat and mass transfer and thermodynamic equilibrium. The main program calls the unit subroutines and links them together in a form corresponding to the user's specification. Each call to a unit subroutine is equivalent to collecting all the equations associated with it, without attempting to solve them as yet. When the calls to all the unit subroutines have been completed, and all the equations have been established, a mathematical solver routine is employed to solve the set of non-linear equations simultaneously.

Figure 8 is a schematic description of the program architecture. The user input conveys to the program an "image" of the cycle to be simulated: the number and types of units contained in it, their interconnections and size or transfer characteristics,
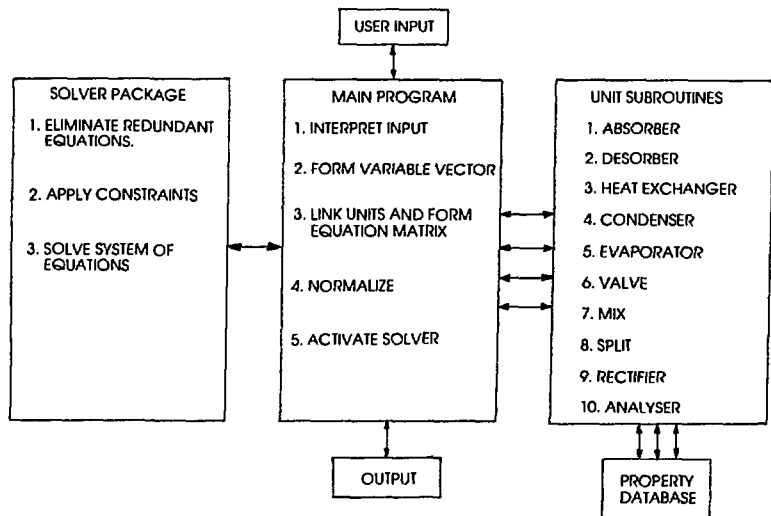

Fig. 8. *Structure of modular absorption simulation code*

where applicable. The input must also contain the values of the parameters set fixed by the user, e.g. temperatures, flowrates, etc. at specific state points. The main program interprets the user input and creates a variable vector containing all the independent unknown quantities. It then calls the unit subroutines, thereby establishing the system's governing equations. The unit subroutines refer to the property database as illustrated. All the equations and variables are normalized to be brought to the same order of magnitude. The solver package is then activated, and the values of the unknowns calculated to a user-specified accuracy. Recognizing that the set of non-linear equations may have more than one mathematical solution, a set of constraints is applied to direct the solver toward the physically valid solution. The output contains the temperature, enthalpy, flowrate, concentration, pressure and vapor fraction at each of the cycle's state points as well as the heat duty and transfer characteristics of each unit.

The ten unit subroutines contained in the code are listed in the right box in Figure 8, and are illustrated, with their respective state points[15]. The thermodynamic properties of working fluids have been incorporated in the code in the form of a property database external to the unit subroutines. Each unit subroutine, when invoked, calls the database several times and retrieves from it the properties required by its various equations. This separation between the unit subroutines and property subroutines provides the flexibility for each unit to operate with different working fluids, as specified by the user.

The program has been used to simulate a variety of cycles of interest with the working fluids presently in the database. Some results have been described[16].

The code was found to be quite robust with all the above systems. Run times are on the order of several seconds on a mainframe computer such as IBM 3081. Most systems with a working fluid involving a non-volatile absorbent usually converge without difficulty; some problems are still encountered concerning fluids with a volatile absorbent such as H_2O-NH_3 due to the complex behavior of these materials at large concentrations of the volatile component in the vapor phase. Efforts are underway to overcome this problem.

HEAT AND MASS TRANSFER IN FILM ABSORPTION

The absorber represents a major critical component in absorption heat pumps. However, while predictive techniques and data are good to adequate for other components of these systems (generators, condensers, evaporators and solution heat exchangers), the information available to quantitatively predict the absorption rates in the absorption process is strikingly inadequate for design purposes. This is largely due to the complexity and lack of fundamental understanding of the combined heat/mass transfer process taking place.

This section describes the approach to theoretical analysis of the combined heat and mass transfer process taking place in absorption systems. The two transfer phenomena are strongly coupled here. The purpose of the analysis is to relate, quantitatively, the heat and mass transfer coefficients to the physical properties of the working fluids and to the geometry of the system. The preferred configuration is that of a falling film of liquid on a metallic surface which serves to transfer heat from the absorbent in contact with the vapor of the absorbate[17].

The information summarized in this section originated from an extensive program on absorption heat pumps. In trying to optimize the design of absorbers and desorbers for these systems, it was recognized that fundamental, quantitative information on the combined heat and mass transfer process is lacking. Correlations of empirical nature (not always publicly available) exist for determining the transfer coefficients for specific fluid pairs in specific geometries. Theoretical models backed by experimental data exist on gas absorption in liquid films, dealing, however, with isothermal mass transfer only and neglecting the effect of heat transfer. The aim of the theoretical analyses has been to produce the kind of fundamental understanding available for pure heat transfer or pure mass transfer, with regard to the combined process. A theoretical analysis is the first step in this direction. Our model calculates the temperature and concentration distributions in the films and the resulting transfer coefficients for typical flow regimes, geometries, and boundary conditions. One of the important results of the analysis is the scaling laws of the system.

The system analyzed is described schematically in Figure 9. A film of liquid solution, composed of substances I (absorbent) and II (absorbate), flows down over an inclined plane. The film is in contact with stagnant vapor of substance II at constant pressure P_v. At $x=0$, the liquid solution is at a uniform temperature T_o and composition C_o (moles of II per unit volume) corresponding to an equilibrium vapor pressure P_{vo} lower than P_v. As a result of this difference, absorption takes place at the liquid-vapor interface. The substance absorbed diffuses into the film; the heat generated in the absorption results in a simultaneous heat transfer process. Two cases of practical interest are considered: in one, the wall is kept at a constant temperature T_o; in the other, the wall is adiabatic.
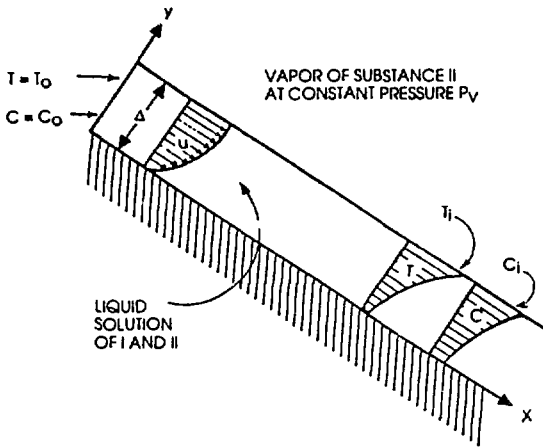
Fig. 9. *Schematic description of falling film absorbent in contact with absorbate vapor*

In formulating the model, the following assumptions have been made:

- the liquid solution's properties are constant and independent of temperature and concentration;
- the mass of vapor absorbed is small compared to the mass flow rate of the liquid. Therefore, the latter is constant, and so is the film thickness;
- there is no heat transfer in the vapor phase;
- there are no natural convection effects in the film due to temperature or concentration differences;
- diffusion thermal effects are negligible;
- vapor pressure equilibrium exists between the vapor and liquid at the interface.

Under the above assumptions, the simultaneous heat and mass transfer in the system at steady-state is described by Energy and Diffusion equations:

$$u(y) \frac{\partial T}{\partial x} = \frac{\partial}{\partial y}\left(\alpha \frac{\partial T}{\partial y}\right) \qquad (1)$$

$$u(y) \frac{\partial C}{\partial x} = \frac{\partial}{\partial y}\left(D \frac{\partial C}{\partial y}\right) \qquad (2)$$

where diffusion and heat conduction in the x-direction have been neglected with respect to those in the y-direction. The following boundary conditions apply:

- at the entrance plane, x=0: $T=T_o$ and $C=C_o$ (3a)

- at the wall y=0:

$\frac{\partial C}{\partial y} = 0$; $T=T_o$ for constant temperature wall

or $\partial T/\partial y = 0$ for adiabatic wall   (3b)

- at the vapor-liquid interface, $y = \Delta$:

$f(P_v, C, T) = 0$ (equilibrium);

$$k(\partial T/\partial y) = H_a D\left(\frac{\partial C}{\partial y}\right) \qquad (3c)$$

where u(y) is the velocity profile in the film, D,α and k are the diffusion coefficient, thermal diffusivity, and thermal conductivity, all properties of the solution, and H_a is the heat of absorption per mole of the vapor.

The typical shapes of the velocity, temperature, and concentration profiles for the case of laminar flow in the film are depicted in Figure 9.

The model developed is quite general and may be solved for a variety of flow regimes (laminar, turbulent, transition), and fluid properties. It can be extended to other wall and interface conditions. In our first model [18] the energy and diffusion equations were solved for laminar flow with the boundary conditions (3). A numerical solution based on finite differences in a two-dimensional grid was developed. Also, an integral method of solution was employed, which made it possible to obtain approximate analytical expressions for most parameters of interest. The solution was carried out for a linear absorbent — a mixture with a linear temperature-concentration equilibrium relation and a constant heat of absorption. Both techniques of solution are suitable, however, for non-linear absorbents with given characteristics.

The results of the solution describe the development of the thermal and concentration boundary layers and the variation of the temperatures, concentrations, and heat and mass fluxes. These quantities in their normalized, dimensionless form depend on two characteristic parameters of the system: the Lewis number Le and the dimensionless heat of absorption λ. The length in the direction of flow is normalized with respect to the Peclet number and the film thickness.

Heat and mass transfer coefficients for the system were calculated. The Sherwood number for mass transfer from the vapor-liquid interface to the bulk of the film reaches a constant value of 3.63 with fully developed boundary layers for both the adiabatic and constant temperature wall. The Nusselt number for heat transfer from the interface to the bulk reaches under the same conditions values of 3.63 and 2.67 for the adiabatic and constant temperature wall, respectively. The Nusselt number for heat transfer from the bulk to the wall reaches 1.60.

Additional results have been obtained for turbulent films and several other boundary conditions [17].

CONCLUSIONS

This paper has attempted to describe the technology and applications of absorption heat pump, and to survey recent developments in some important areas. Absorption systems have enjoyed renewed and growing interest in recent years due to their potential for CFC replacement in heat pumping applications such as refrigeration, heating and air conditioning. With the increasing cost of fossil fuels and the growing concern about the environment, the particular features of this heat-activated cycle have made it attractive for both residential and industrial applications. Solar-powered air conditioning and natural gas-fired heating and cooling are some of the applications in the first category; waste heat recovery and water desalination are promising applications in the second. Industrialized countries are conducting increasing, intensive research and development in this area. Absorption technology will play an important role in reducing CO_2 emission by recovering waste heat to generate process heat and/or refrigeration by co-generation of heat and power.

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