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MARTIN MARIETTA

Development of a High-Efficiency, Gas-Fired, Absorption Heat Pump for Residental and Small-Commercial Applications

Phase I Final Report
Analysis of Advanced Cycles and Selection of the Preferred Cycle

B. A. Phillips

MANAGED BY
MARTIN MARIETTA ENERGY SYSTEMS, INC.
FOR THE UNITED STATES
DEPARTMENT OF ENERGY

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ORNL/Sub--86-24610/1

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DEVELOPMENT OF A HIGH-EFFICIENCY, GAS-FIRED, ABSORPTION HEAT PUMP FOR RESIDENTIAL AND SMALL-COMMERCIAL APPLICATIONS

PHASE I FINAL REPORT
ANALYSIS OF ADVANCED CYCLES AND SELECTION OF THE PREFERRED CYCLE

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Date Published: September 1990

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UNDER SUBCONTRACT 86X-24610C

Prepared for

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Oak Ridge, Tennessee 37831-6285
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MARTIN MARIETTA ENERGY SYSTEMS, INC.
for the
U.S. DEPARTMENT OF ENERGY
Under Contract No. DE-AC05-84OR21400

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ABSTRACT

The purpose of the total project is to develop a gas-fired absorption heat pump for residential and small-commercial applications that will produce at least 1.6 Btu of heating and 0.7 Btu of cooling per Btu of heat content in the gas being burned. These coefficient of performance (COP) goals are based on air-to-air heat pump operation at Air-Conditioning and Refrigeration Institute (ARI) rating conditions of 47°F outdoor ambient temperature in the heating mode and 95°F in the cooling mode.

The primary technology advances that can be used to attain the new goals are higher efficiency cycles, increased flue efficiency, and better fluids. Flue efficiency technology is well developed, and fan-assisted combustion systems with condensing heat exchangers can limit flue and insulation losses to the 10% range. If this 10% loss assumption is made, the resulting target cycle COPs are 1.78 in heating mode and 0.78 in cooling mode at the ARI rating conditions. The objective of Phase I was to analyze working fluids and absorption-cycle concepts that are capable of performing at the target COPs and are potentially competitive with existing space-conditioning products in cost, operating life, and reliability. The intent of the analysis was to select a preferred cycle and fluids for development in later phases of the total project.

Six advanced cycles were evaluated with ammonia/water as the fluid pair. Then additional analysis was performed with other fluid pairs to determine whether cycle ranking would change depending on which fluid was used. It was concluded that the preferred cycle/fluid was the generator-absorber heat exchange (GAX) cycle using ammonia/water as the fluid pair. A cost estimate made by an independent manufacturing engineering firm for a residential heat pump based on the cycle/fluid combination determined that the GAX heat pump could be cost competitive with existing products.

1. INTRODUCTION

1.1 BACKGROUND

The purpose of the total project was to develop a gas-fired absorption heat pump for residential and small-commercial applications that will produce at least 1.6 Btu of heating and 0.7 Btu of cooling per Btu in the gas being burned, as specified by the Oak Ridge National Laboratory Request for Proposal. These coefficient-of-performance (COP) goals are based on air-to-air heat pump operation at the Air-Conditioning and Refrigeration Institute (ARI) rating conditions of 47°F outdoor ambient temperature in the heating mode and 95°F ambient temperature in the cooling mode.

For year-round residential and small-commercial use, other requirements can be added. It is desirable that the heat pump be capable of providing, without supplemental heat, all the heating and cooling required by the building within the temperature ranges experienced by the majority of the population in the United States, i.e., at least from −10°F to 110°F. (In earlier developments, that had been found to be a capability of heat-actuated heat pumps.) The heat pump should also be potentially competitive with existing space-conditioning products in cost, operating life, and reliability.

The heat pump should be capable of meeting all national codes for residential installation. Because of the limitations of existing absorption-fluid pairs, this requirement for meeting national codes has made it necessary, in past absorption heat pump developments, to locate air-cooled absorption systems outdoors and to transfer the heating and cooling to the conditioned space by means of a suitable secondary fluid. The extra temperature drops involved in heat transfer to and from the secondary fluid reduce the effective COP. The cycle evaluation conditions were therefore adjusted to allow for such factors.

Absorption heat pumps developed in the past by Phillips Engineering personnel were designed for, and operated to, COPs of 1.25 in the heating mode and 0.5 in the cooling mode. One development used ammonia/water and the other used organic fluids. Both were taken to field-test stages. The potentials for elimination of supplementary heat requirements to temperatures of −10°F were proved on those heat pumps. Those past COPs were obtained with flue and insulation losses of about 25%. The primary advances that can be used to attain the new goals are higher efficiency cycles, increased flue efficiency, and better fluids. Improving flue efficiency to the ninety-plus percent condensing range could reduce flue and insulation losses to the 10% range. Anticipation of the 10% losses affects the Phase I investigations to the extent that it sets the minimum net cycle COPs at 1.78 for heating mode and 0.78 for cooling mode. The Phase I investigations were therefore directed toward cycles capable of net cycle COPs above those levels, and toward working fluids capable of achieving those and other objectives.

1.2 OBJECTIVES

The objective of the total project was to develop and demonstrate a practical prototype of a residential and small-commercial gas-fired absorption heat pump having gas COPs equaling or exceeding 1.6 in the heating mode and 0.7 in the cooling mode. The system is to have potential installed costs capable of competitive marketing with then-existing space conditioning equipment.

The objectives of Phase I were to analyze working fluids and absorption-cycle concepts with potentials for heat pump operation at cycle COPs of 1.78 or above in the heating mode and 0.78 or above in the cooling mode (both under standard testing conditions), and to select the preferred cycle and fluids. That cycle and fluid pair should be usable in a possible Phase II to develop and demonstrate the performance goals on a laboratory test unit of residential capacity, and in a Phase III on a subsequent prototype heat pump.

1.3 SCOPE

A variety of absorption working fluids and cycles were to be investigated and screened, separately or in combination, for capability to meet the performance and operational objectives of the project. That became possible within the constraints of this project because many of the working fluids and absorption cycles could be investigated and screened by classes.

In this investigation it quickly became apparent that independent evaluations of the fluids and cycles would be only partly effective if done separately. First, a fluid pair was needed for comparing cycles under a common set of conditions. Later, it might be necessary to determine whether any or all of the cycles would improve if operated with other fluids. Working-fluid combinations were therefore to be screened first for ability to meet the project objectives and for suitability for use in comparative analysis of the cycles selected for detailed evaluation.

The absorption cycles were to be compiled by thorough literature searches and then screened either individually or by classes for ability to meet the project requirements. If many cycles were thus found, further screenings were to be used to reduce the number of cycles to a number that would be workable within this phase. The final group of cycles were to be evaluated relatively in the detail required by the applicable criteria. Following those evaluations a reanalysis was to be made considering use of fluids that might be more suitable for each individual cycle, and any other changes that could improve any cycle. At that point, the cycle and working fluid that should best meet the project objectives would be selected for possible use in future phases. For the final steps in Phase I, a residential heat pump was designed in enough detail to be suitable for a cost estimate by an independent firm, and that cost estimate was made.

1.4 APPROACH

The approach was to thoroughly search for and screen fluids based on specifications that would meet the project objectives and the residential application. Simultaneously, absorption cycle searches were to be made and the results screened on the same bases of project objectives and residential use. The usability of a working-fluid combination in analyses of the resulting set of cycles was to be established. When the best fit of fluid to cycles was obtained, the cycles were to be analyzed under common conditions with the same fluid to allow selection of the best cycle.

Then the possibility of changing cycle rankings by changing fluids would be analyzed and the final selection of fluid and cycle would be made.

1.4.1 Working-Fluid Selection

Multitudes of working-fluid pairs have been proposed.[1] The physical and thermodynamic properties that are important for use in absorption refrigeration and heat pumping have been measured to varying degrees and to varying accuracies for many of these pairs.[1] Relatively few pairs have been tested in absorption systems and shown to perform well. Even fewer have met the competitive requirements of commercial products.

The fluid requirements established for this project were:

- optimum absorption properties, both thermodynamic and transport,
- ability to meet code flammability and toxicity standards,
- ability to operate to −15°F evaporator temperature,
- ability to operate at high generator peak temperatures for 20 years, and
- preferably, established background in refrigeration or absorption.

Optimum absorption properties were expected to be necessary to reach the COP objectives, which were well beyond previous air-cooled residential state-of-the-art. The flammability and toxicity requirements of the national code had to be met, but with the less hazardous fluids, they could be met by using a secondary fluid to transfer the heating and cooling to the house. The low evaporator temperature was considered necessary, because the ability of the gas heat pump to supply all the heat requirements of the home to below zero was considered an important competitive advantage. Stability at high temperatures was required because higher peak generator temperatures can be used to improve the COPs of most cycles. Past usage in refrigeration would reduce or eliminate the amount of testing and proofing of the fluids that might otherwise be necessary during the project. Working with those requirements, the selection of the trial working fluids was found to be quite straightforward.

Absorption working fluids can be categorized by whether they are inorganic or organic materials, or combinations of the two, and by their chemical type. This categorization can often indicate relative stability, flammability, toxicity, etc.

The large majority of the fluid combinations listed in the literature were found to consist of organic fluids, which as a group do not meet the requirements well. Flammability and toxicity are often problems, and their high-temperature stability is much lower than that of the inorganics. Their thermodynamic and transport properties are generally less suitable to absorption and are rarely fully determined. The other requirements in the list were less important for the trial working fluids, but, as a group, the organics were not outstanding for those requirements, either.

There are relatively few inorganic absorption working fluids, and only four refrigerants. Of the refrigerants, carbon dioxide has a critical temperature that is too low for air-to-air heat pumps. Sulfur dioxide has many disadvantages, including corrosion problems if ever humidified, and toxicity. The remaining two inorganic refrigerants, water and ammonia, have had broad use in absorption. The freezing point of water prevented its consideration for this low-temperature heat pumping application. Ammonia, however, has outstanding thermodynamic and transport properties for refrigeration in general and for absorption in particular. It readily met the requirements for low temperature operation and for use at high peak temperatures in the generator. Concerns are often expressed about decomposition of ammonia at high temperatures, but there have been no indications of such a problem, even in household absorption refrigerators which operated for over 20 years at generator temperatures up to 400°F.

The absorbents for ammonia were found to be of several types: water, salts, salt solutions in water, a few organic fluids, and some mixtures. From the thermodynamic and heat and mass transfer standpoints, water and ammonia are, individually, probably the best-characterized of fluids. Data on their solutions also were by far the most complete, although in some respects not as complete or accurate as wished. They had also had extensive long-term use in commercial products to the high and low temperatures listed in the requirements.

A quick review of the data on other ammonia/absorbent combinations showed that ammonia/water had no close competitors for use as the trial working fluid. There were other combinations that could have advantages in ultimate product use, however.

1.4.2 Cycle Selection

As with fluids, many absorption cycles of various types and for varied purposes had been proposed over the years.[2,3,4,5] The basic single-stage cycle and the absorber heat-exchange (AHE) cycle had been used in gas-fired, air-cooled, residential air-conditioner production units with COPs ranging from 0.3 to 0.5. The AHE cycle had also been used in experimental heat pumps of residential size using both organic fluids and ammonia/water.[6,7] Larger systems for commercial and industrial applications had used more complex cycles to achieve refrigerating COPs of more than 1.0 for water-cooled conditions.

In compiling the cycles to be evaluated, the known cycles were listed, plus those from searches of the absorption literature and the solar literature. Searches of the Dialog data bases and general libraries were also made, but no new cycles were found in those attempts.

To select cycles for evaluation, both the potential performance of the cycle and its suitability for the competitive residential market were considered. Broadly, preliminary screenings were made to eliminate cycles known to have or likely to have COPs below the project requirements, and also cycles likely to result in products having costs or complexities that would be excessive for the residential market. A plot of energy usage to meet a given load as a function of COP, Figure 1, showed that in the heating mode, COP increases over the project objective would have limited value, but that in the cooling mode they could be important. As shown by Alefeld, complex multistage, multieffect cycles have potentials for very high COPs.[2] They can be expected, however, to be costly and much more prone to service problems than the simple cycles. Hence, preliminary screenings using the two limits were made to reduce the number of cycles to be investigated to below ten. Preliminary calculations then reduced the number to six. Early calculations indicated that those six cycles were likely to exceed the project performance objectives comfortably. They also represented some of the fundamental methods of improving performance over the basic single-stage cycle and the AHE cycle.

The cycles included in the evaluation were:

- the double-effect cycle,
- the resorber-augmented cycle,
- the generator-absorber heat-exchange (GAX) cycle,
- the double-effect regenerative (2R) cycle,
- the variable-effect cycle, and
- the two-stage GAX cycle.

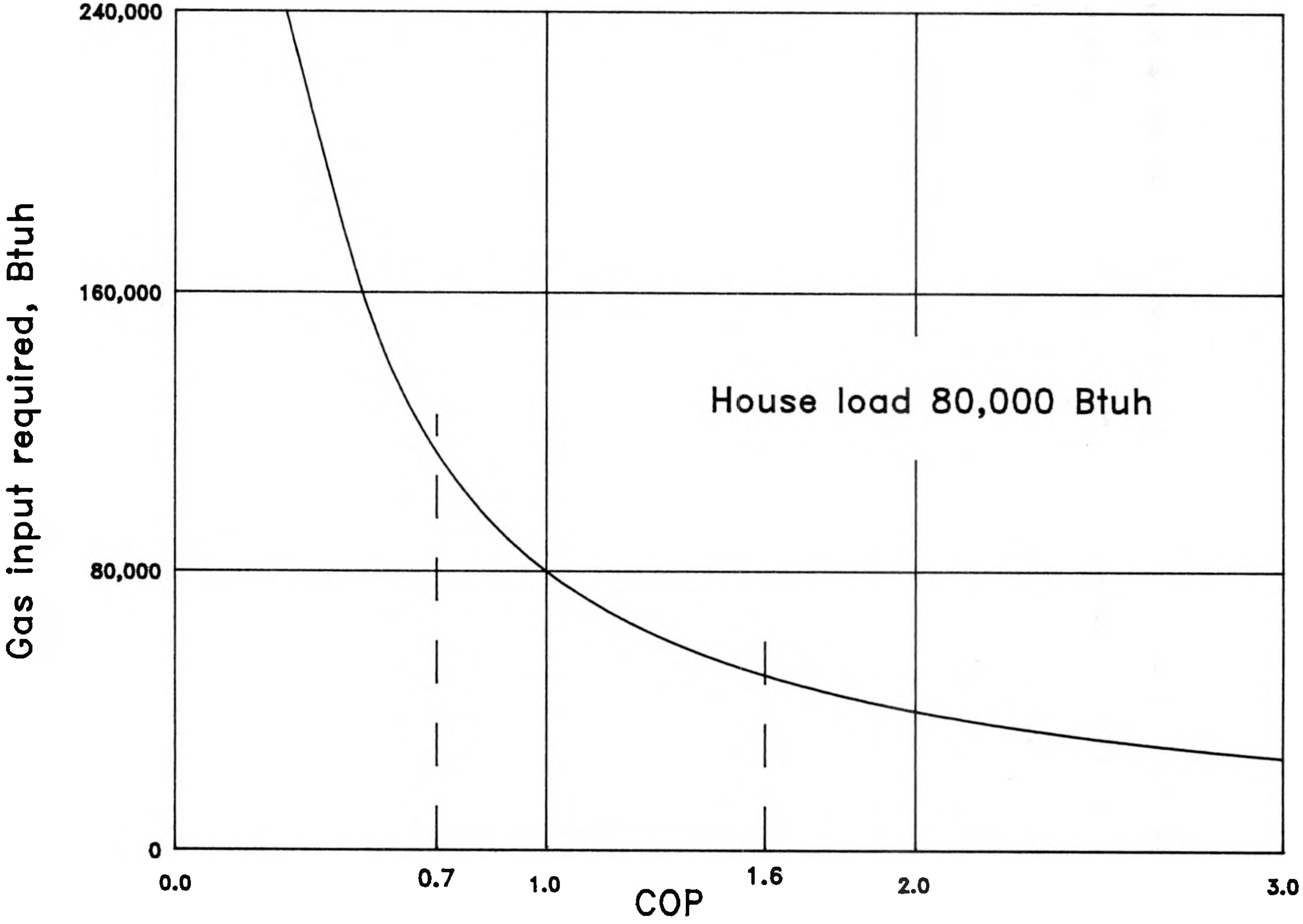

Fig. 1. Gas input required versus coefficient of performance, gas heating value basis.

The double-effect cycle, used in water-cooled lithium-bromide air conditioners, recovers heat of condensation at higher pressures to produce additional refrigerant. It represented the class of cycles that achieve performance gains by extending the cycle into the higher-pressure portion of the ammonia/water P-T-X chart. The resorber-augmented cycle, which had been used industrially,[8] is a method of evaporating and absorbing the refrigerant twice. It extends operation into the pressure range below that of the single-stage cycle. The GAX cycle uses the same pressure range as the single-stage cycle but operates to the higher temperatures available within the ammonia/water solubility limits. It and the three additional cycles recuperate heat from the absorber to the generator to reduce the operating heat input per unit of refrigerant generated. The variable-effect cycle uses an added intermediate pressure between the low and high pressures of the GAX cycle to increase the potential heat exchanged from absorber to generator.[9] The two-stage GAX cycle divides the absorbent loop into two circuits to allow easier optimization of the cycle.[10] The 2R cycle also uses two circuits in the absorbent loop.[11] Its high-temperature circuit uses a series of pressure levels between the high and low pressures as desorber/resorber stages, which serve to enrich the rich solution and to allow use of lower temperature heat as operating heat input. It had been developed for operation with solar heat.

As indicated earlier, equipment cost is a major factor in the residential market, so the cycles chosen for evaluation were not much more complex than the double-effect and the resorber-augmented cycles, even though more complex cycles have higher COP potentials. The limit on complexity was based partly on the limited value of incremental increases in COP, as indicated in Figure 1, and partly on the belief that first-generation gas heat pumps should be easy to make highly reliable.

The criteria used in the cycle evaluations included:

- COP,
- pumping power,

- total heat to be transferred,
- operational range and convertibility, and
- complexity and probable cost.

The evaluation criteria were defined as follows:

Coefficient of Performance. While the 0.78 cycle cooling COP and the 1.78 cycle heating COP were the lower limits, the objectives were to achieve the highest COPs possible within the requirements of the other criteria.

Pumping Power. Electric power costs could become a significant fraction of the operating cost. The small solution pumps and motors required for residential applications tend to be inefficient. While solution pumps can, in principle, be powered by the returning fluids, no commercially usable pump/hydraulic-motor combinations had been developed.

Total Heat To Be Transferred. The total amount of heat to be transferred in a cycle, externally and internally, is a first-order indication of the heat-transfer surface requirements and of the relative costs of the absorption systems.

Operational Range and Convertibility. All refrigeration cycles and fluid combinations have a limited range in which they can operate effectively. If the heat pump is to provide the total heating requirement of a building without recourse to a supplemental heat source, then the cycle range, or low-temperature cutoff point, is an important characteristic. However, if a cycle is convertible to one suited to lower-temperature operation, a high cutoff point need not eliminate an otherwise desirable cycle.

Complexity, Probable Cost, etc. This category was to include, if necessary, other factors affecting the marketability of the ultimate heat pump, including manufacturability, size, appearance, installation problems, probable life, maintenance and service requirements, etc.

2. CYCLE DESCRIPTIONS

For easier understanding of the cycle improvements being investigated, short cycle descriptions of the basic single-effect cycle and the AHE cycle are included, followed by those of the advanced cycles evaluated.

2.1 SINGLE-EFFECT CYCLE

The basic single-effect, two-pressure absorption cycle consists of four major heat and mass transfer components: condenser, evaporator, absorber, and generator, as shown in the schematic diagram of Figure 2. It also has a solution pump, a refrigerant flow-control device, and a weak-liquid flow-control device. Cycle diagrams may be drawn on various types of fluid-property charts, the most common being a pressure-temperature-composition (P-T-X) chart as in Figure 3. The processes occurring are marked on the diagram. Evaporation and condensation are shown as isobaric and isothermal, but they may not be completely so. The absorber and the generator operate within a small temperature range, the width of the range depending on the working fluids and the design of the cycle.

In addition to the four components listed in the previous paragraph, three other heat-exchange components are helpful in improving performance. They are commonly assumed to be part of the cycle, even when not shown in schematics like Figure 2. The three are the precooler, the liquid heat exchanger, and the solution-cooled rectifier.

The precooler is a counterflow exchanger used to cool the condensed refrigerant flowing toward the cold evaporator. The vapor and unevaporated refrigerant flowing from the evaporator to the absorber cool the condensate, but are also themselves warmed and evaporated before entering the warm absorber. The precooler may not be practical in vacuum units like the lithium-bromide systems. In Figure 4, the precooler heat transfer is shown by the arrows labeled P.

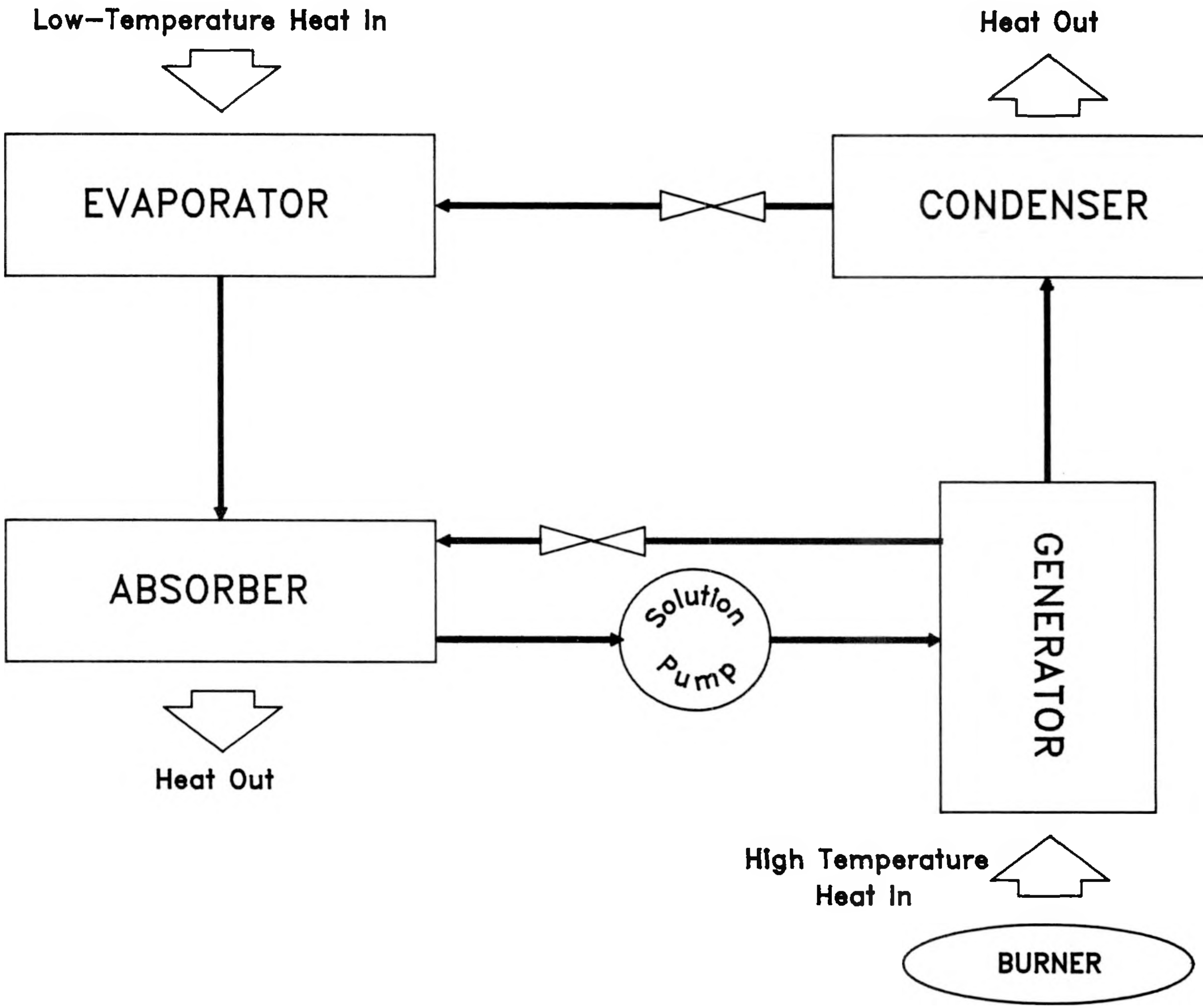

Fig. 2. Schematic of basic single-stage absorption unit.

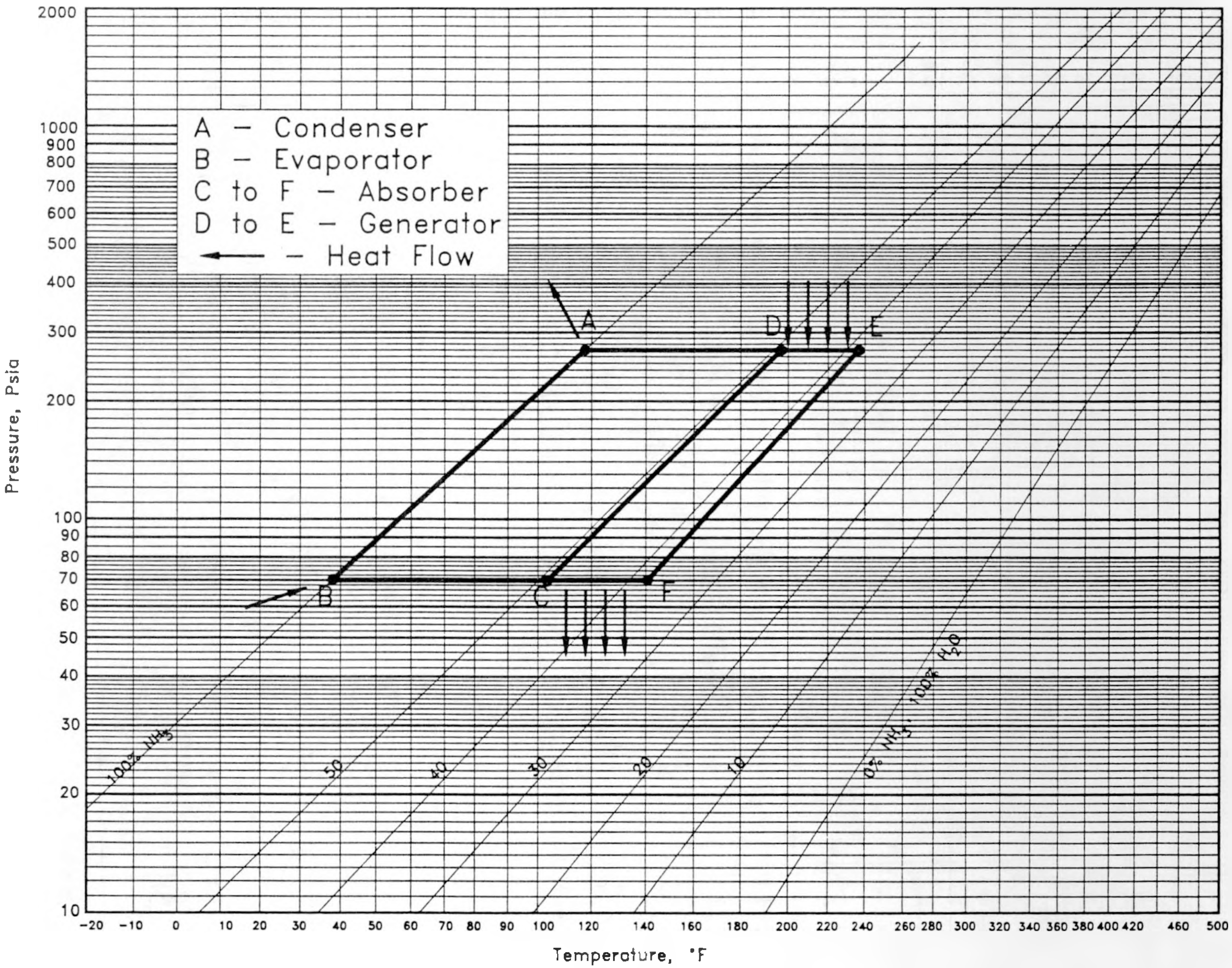

Fig. 3. Pressure-temperature-composition diagram of basic single-stage cycle.

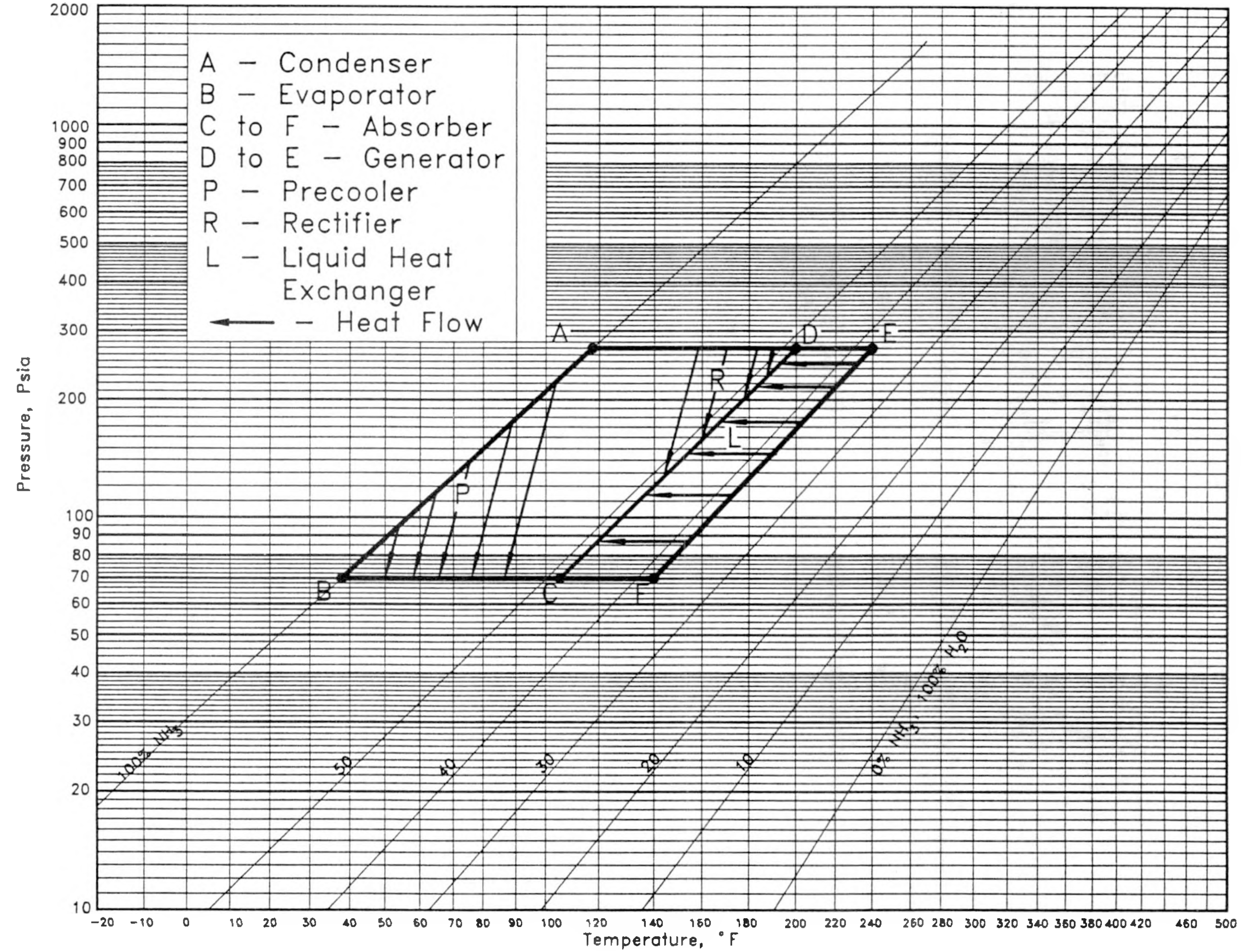

Fig. 4. Pressure-temperature-composition diagram of single-stage absorption cycle secondary heat exchangers.

The liquid heat exchanger produces the heat transfer shown at L in Figure 4. It, too, is counterflow, using the heat in the weak solution flowing from the generator to the absorber—from point E to F— to heat the rich solution being pumped from the absorber to the generator, at C to D. In the process, the weak solution is cooled so that it does not flash on entering the absorber. Because there is less weak liquid flowing than rich liquid, normally less heat is available from the weak liquid than is needed to raise the rich liquid to its boiling point at generator pressure. This lack of heat brings the third type of exchanger into use.

If the absorbent of the chosen pair has an appreciable vapor pressure, as in ammonia/water, the generator of the absorption system must be a distillation column. Above the feed plate, generally point D, a small portion of the vapors must be condensed to rectify the vapor to the required purity. That heat of partial condensation can be partially recuperated into the cycle by transferring it to the rich solution. That transfer is shown by the arrows labeled R in Figure 4. It may be done by a separate exchanger, but it can be more efficient if combined with the liquid heat exchanger into a so-called Triple-X.

2.2 ABSORBER HEAT-EXCHANGE CYCLE

If the peak temperature of the generator (point E in Figure 3) of a basic cycle is increased to try to take advantage of Second Law potentials, the COP of the basic cycle is found to decrease rather than increase.[5,12] One reason for the decrease is that the temperature difference between the fluid streams in the liquid heat exchanger is increased, reducing the exchanger efficiency and increasing Second Law losses. The AHE cycle is a method of resolving that difficulty.[3,5,13]

The AHE cycle, Figure 5, uses absorber heat to warm the rich liquid, and utilizes the sensible heat of the weak liquid as heat input to the generator. At many operating conditions, liquid heat-exchanger heat and rectification heat are also needed to warm the rich liquid, as Figure 5 indicates.

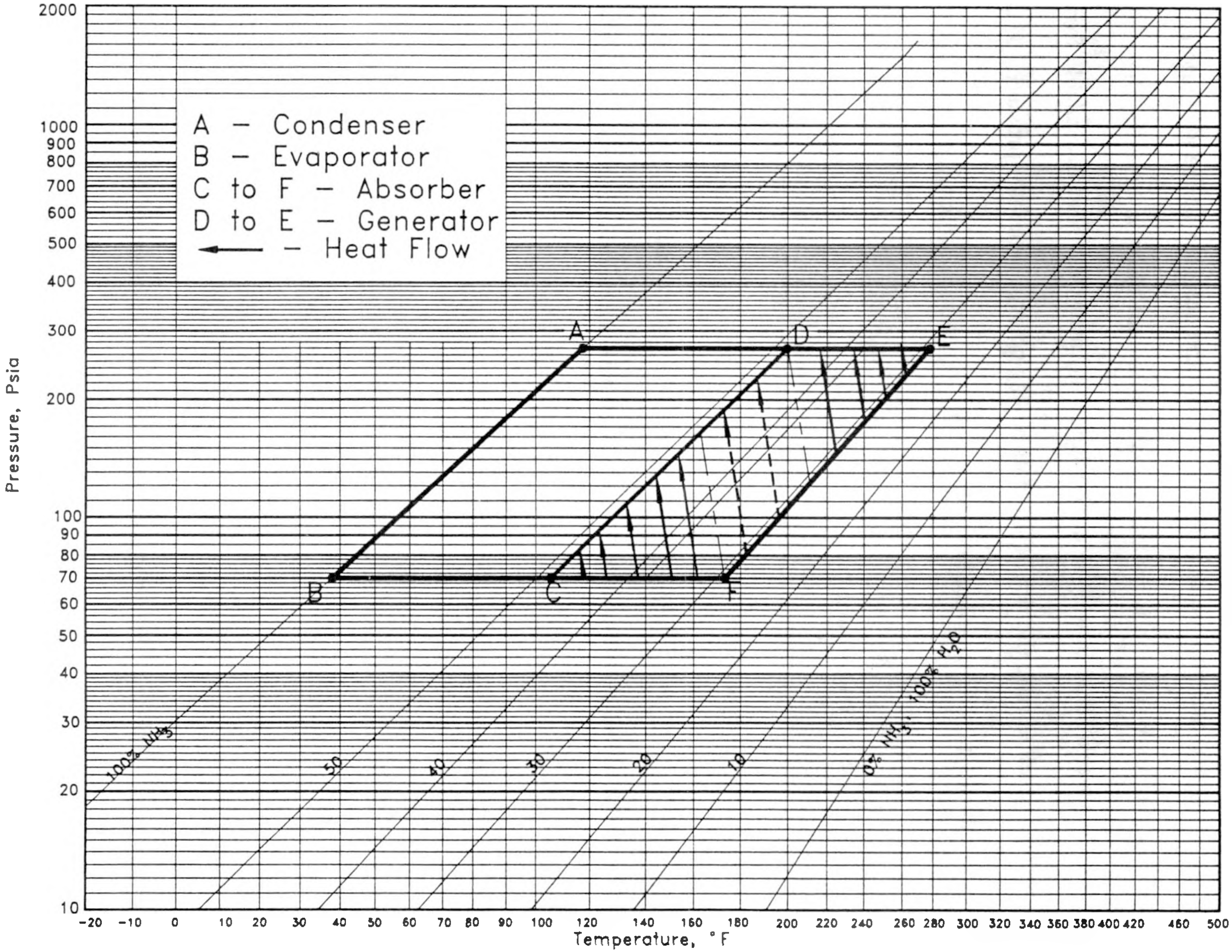

Fig. 5. Pressure-temperature-composition diagram of absorber heat-exchange cycle.

The AHE cycle is at its optimum operating condition when the absorber temperature range is just sufficient to equal the temperature rise of the rich liquid from the absorber outlet to its saturation temperature at condenser pressure, point D (plus the temperature differential required for heat transfer), as shown in Figure 6. That absorber temperature range also results in a generator temperature range sufficient to allow all the weak-liquid sensible heat to be transferred to the generator. Under those conditions, a liquid heat exchanger is not required.

Air conditioning systems that use AHE may operate close enough to the optimum conditions to be built without a liquid heat exchanger, but a liquid heat exchanger is needed to increase the range to heat pump requirements.

The AHE cycle has been used in ammonia/water air conditioner production units for 20 years as well as in experimental ammonia/water and organic-fluid heat pumps. The heat pumps operated at cycle COPs of 0.65 to 0.7 in the cooling mode and 1.65 to 1.7 in the heating mode, which translated to gas-fired COPs of up to 0.5 cooling and 1.25 heating.[6,7]

2.3 DOUBLE-EFFECT CYCLE

The double-effect cycle utilizes the concept of multieffect distillation, used in the chemical industry and in water desalination, in which the heat of condensation at a high pressure in a first effect provides the heat input to the boiler of a lower-pressure next effect. The high affinity that absorption fluids have for each other causes the boiler temperature to be much higher than the condensation temperature. The number of effects possible in absorption with known fluids is therefore much smaller than those possible in water purification.

In the cycle diagram of Figure 7, heat is input to the system only at line D'E', the first-effect generator. The first-effect pressure must be such that condensation at A' will occur at a temperature higher than that of the second-effect generator, DE. Second-effect condensation occurs at the

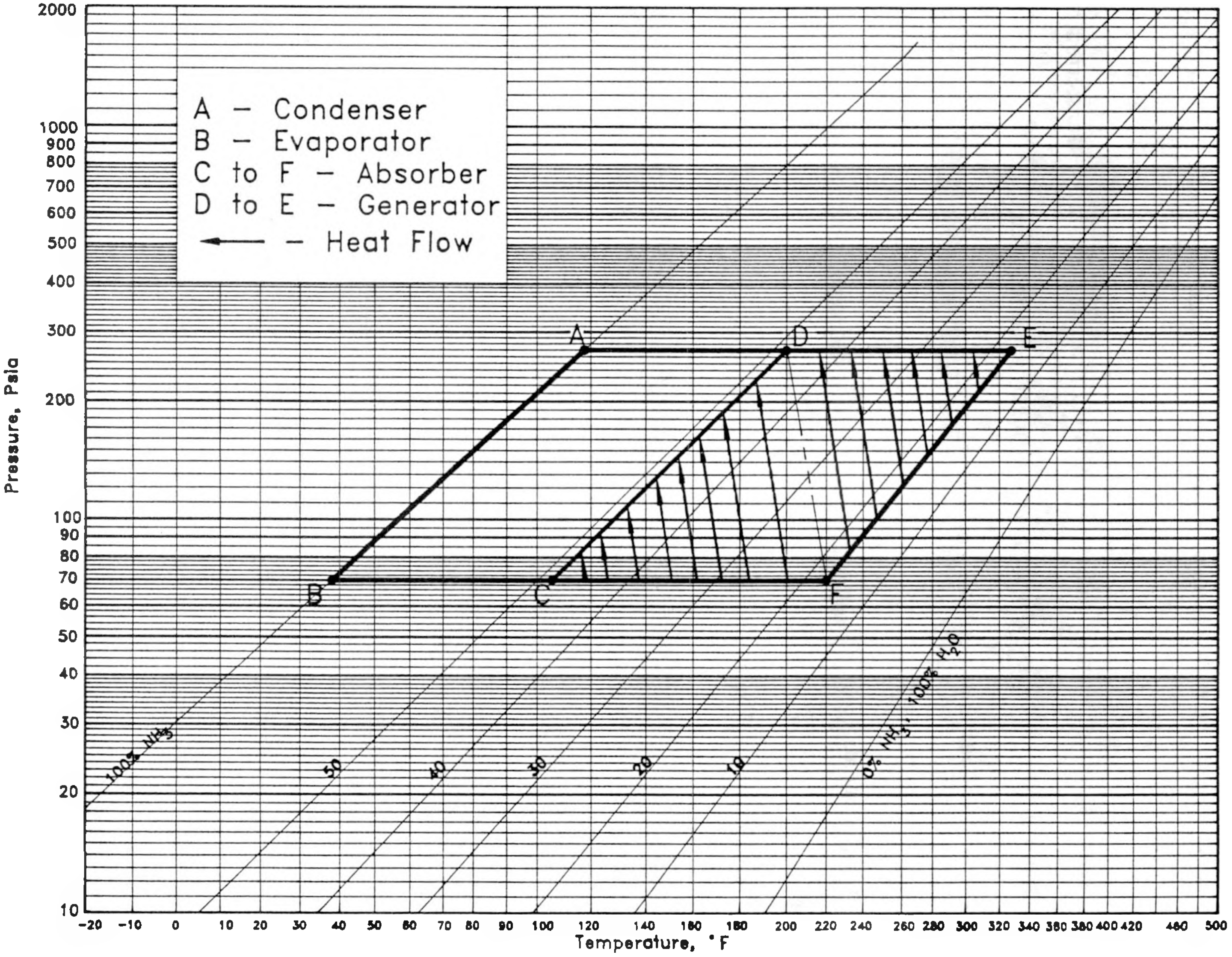

Fig. 6. Pressure-temperature-composition diagram of the optimum absorber heat-exchange cycle.

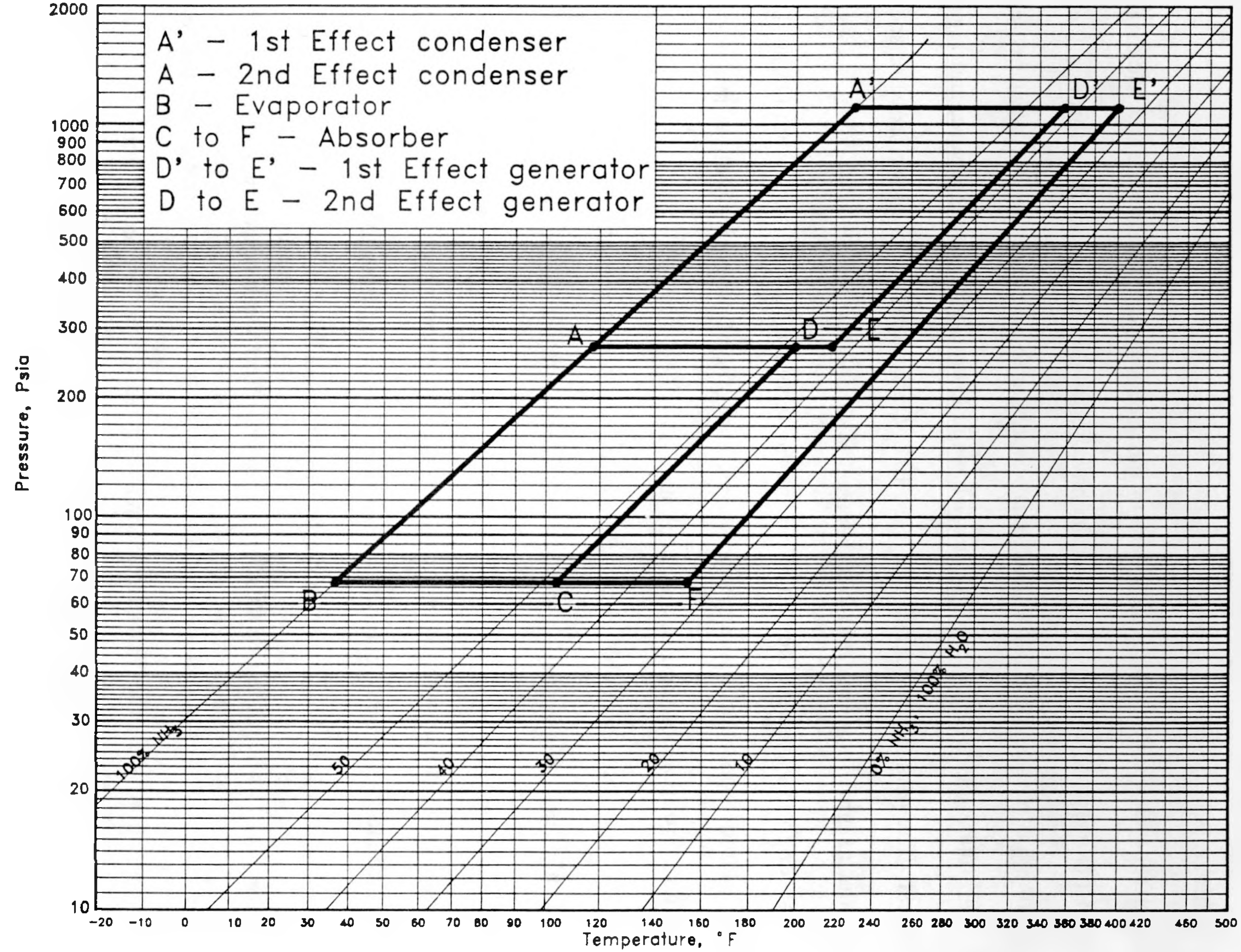

Fig. 7. Pressure-temperature-composition diagram of the type 1 series-flow double-effect absorption cycle.

normal condenser temperature at A. The evaporation of both refrigerant streams occurs at B, and all absorption occurs at CF. The heat output of the cycle is thus primarily at A and CF. A schematic diagram of this cycle is shown in Figure 8.

Double-effect cycles can be connected in any of three basic types of circuits. The cycle diagrams are shown in Figures 7, 9, and 10. Two circuits have series flow of the solution through the two effects, and one has parallel flow. With ammonia/water, the three cycles all operate at first-effect pressures over 1000 psia. The circuit of Figure 7 has the lowest first-effect pressure, and Figure 9 has the highest. The parallel flow circuit of Figure 10 operates at an intermediate pressure and was found to produce slightly better COPs, so it was used in the evaluations.

As can be seen from Figure 10, the first-effect and the second-effect circuits are both basic single-effect cycles. Fluid properties, operating pressures, and other factors rarely allow much spread between the concentrations of the rich and weak liquids.

The basic concept of the double-effect cycle is to recuperate the heat of the high-pressure condensation only, but in this evaluation all other possible methods of heat recuperation were utilized to optimize the cycle COP. The precooler was used within its temperature limits, but, in addition, the sensible heat of the condensate from the high-temperature condenser flowing from A' to A in Figure 10 was used to heat the rich solution for both circuits flowing from the absorber outlet to the low-pressure generator, C to D. The second-effect rectifier heat and the weak liquid from the two effects were also used to heat the rich solution. The first-effect weak liquid used part of the first-effect rectifier heat to heat the rich liquid to the temperature D'. The majority of the first-effect rectifier heat, together with the first-effect condenser heat, was applied to the second-effect generator. Using counterflow allowed the first-effect condenser to be at a somewhat lower temperature and pressure than would have been possible otherwise.

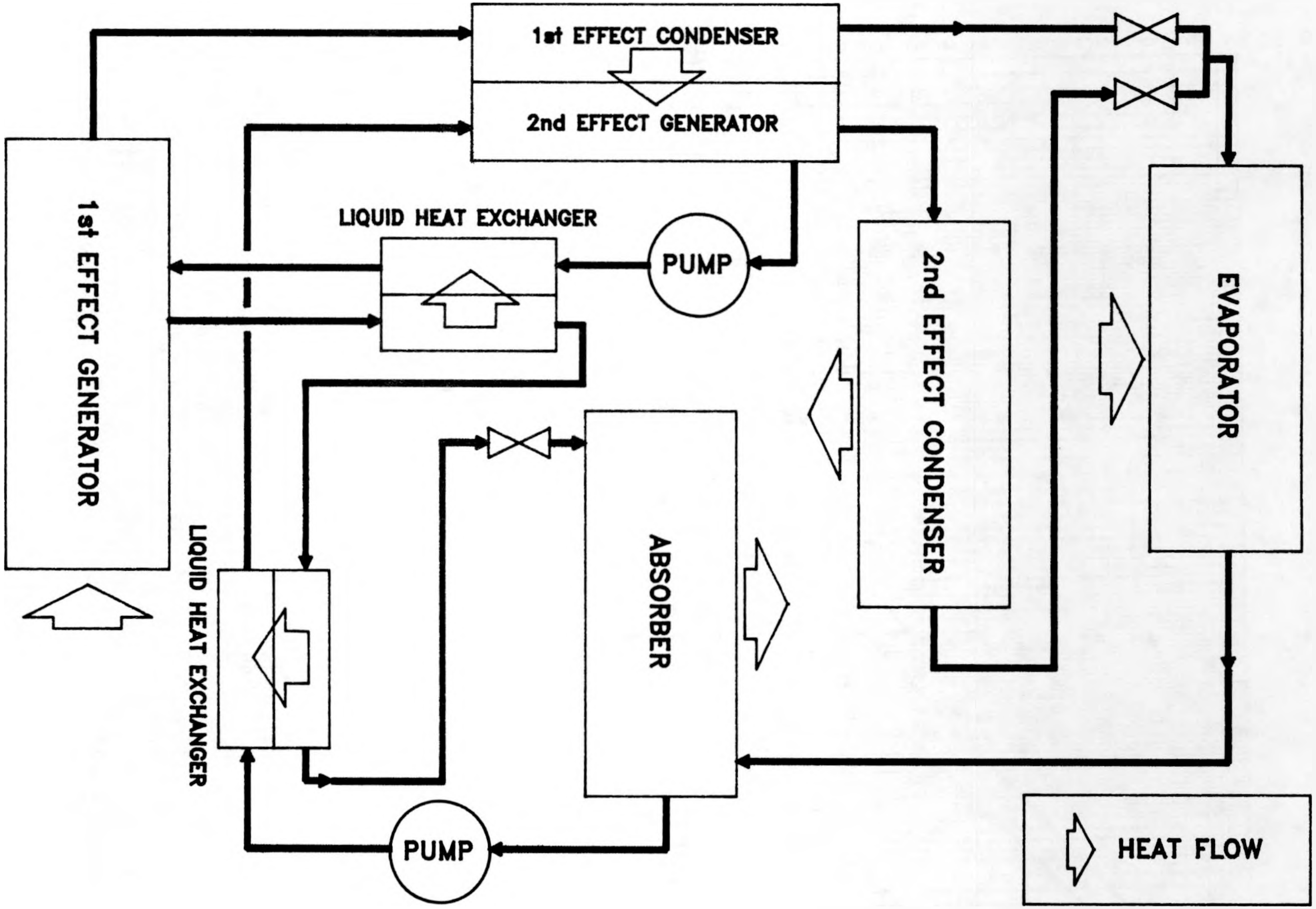

Fig. 8. Schematic of type 1 double-effect absorption unit.

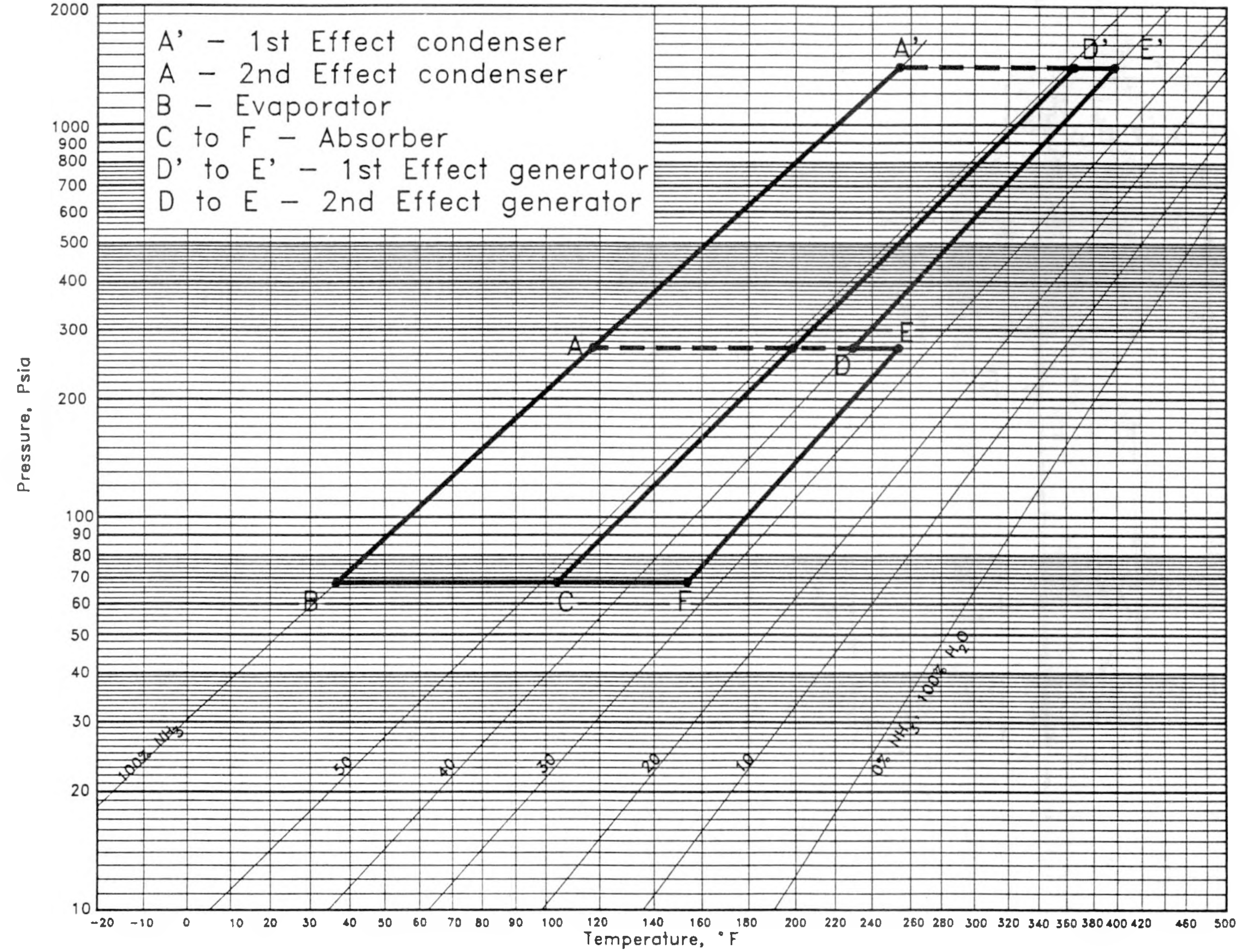

Fig. 9. Pressure-temperature-composition diagram of the type 2 series-flow double-effect absorption cycle.

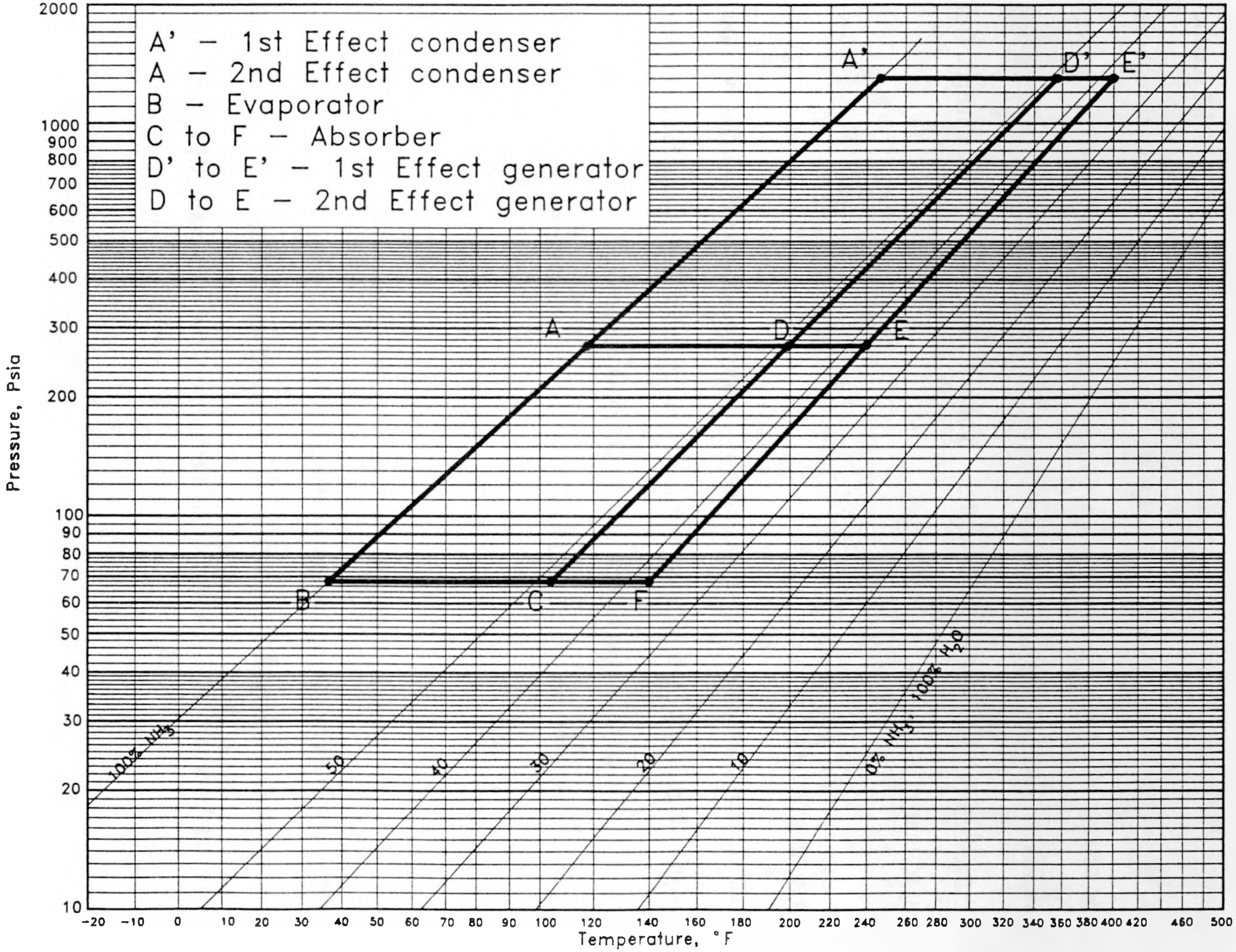

Fig. 10. Pressure-temperature-composition diagram of the parallel double-effect ammonia/water cycle.

Finally, the AHE types of recuperation were also used: absorber heat was used to heat the combined rich-liquid flows leaving the absorber, first-effect weak-liquid heat was applied to the first-effect generator, and sensible heat from the combined weak liquids was applied to the second-effect generator. The net result was that all heat quantities in the cycle at temperatures above those of the absorber and second-effect condenser were reused in the cycle at the highest possible temperatures. Not all of these heat exchange steps may be cost effective in a residential product, but the best cycle COP for the conditions was desired for this analysis. The schematic for the cycle changed from that of Figure 10 to that of Figure 20.

2.4 RESORBER-AUGMENTED CYCLE

In the resorber-augmented cycle,[5,8] a low-pressure resorption-desorption circuit is added to a single-stage cycle (Figure 11). The refrigerant vapor from the single-stage evaporator is absorbed in a resorber in which the absorbent is kept at a high ammonia concentration. The enriched solution from the resorber then flows through a liquid heat exchanger to a desorber at a lower pressure. The pressure of the enriched solution in the desorber is to be the pressure at which the refrigerant can desorb from the solution at the evaporator temperature and below. The refrigerant vapor flows to the main absorber, which has absorbent at a low enough concentration to reduce the pressure to the desorber needs. A schematic diagram is shown in Figure 12.

The desorber-resorber circuit thus requires a liquid heat exchanger and solution pump in addition to the resorber and desorber. Its advantage is that it evaporates the refrigerant twice—once pure and the second time from solution.

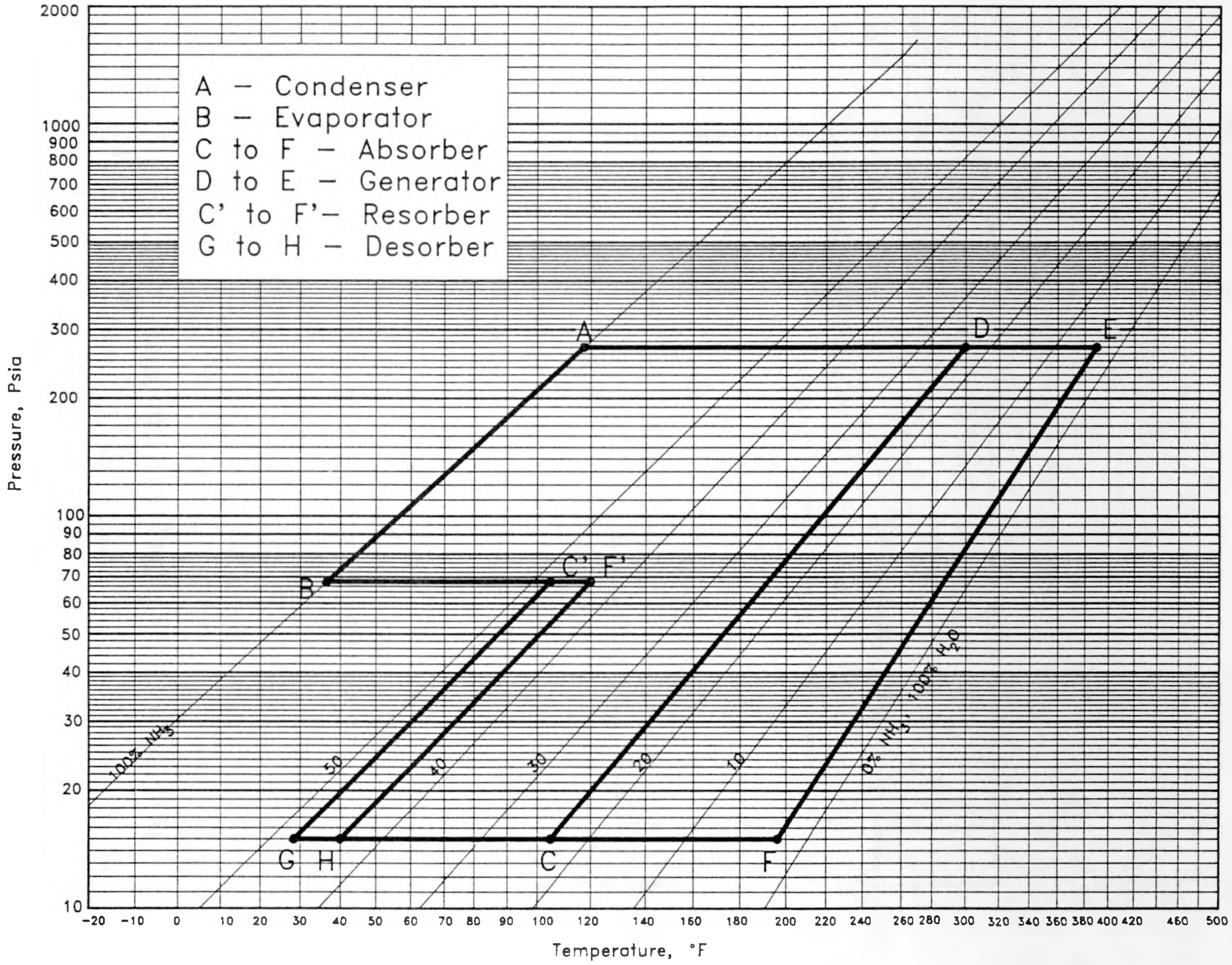

Fig. 11. Pressure-temperature-composition diagram of the resorber-augmented cycle.

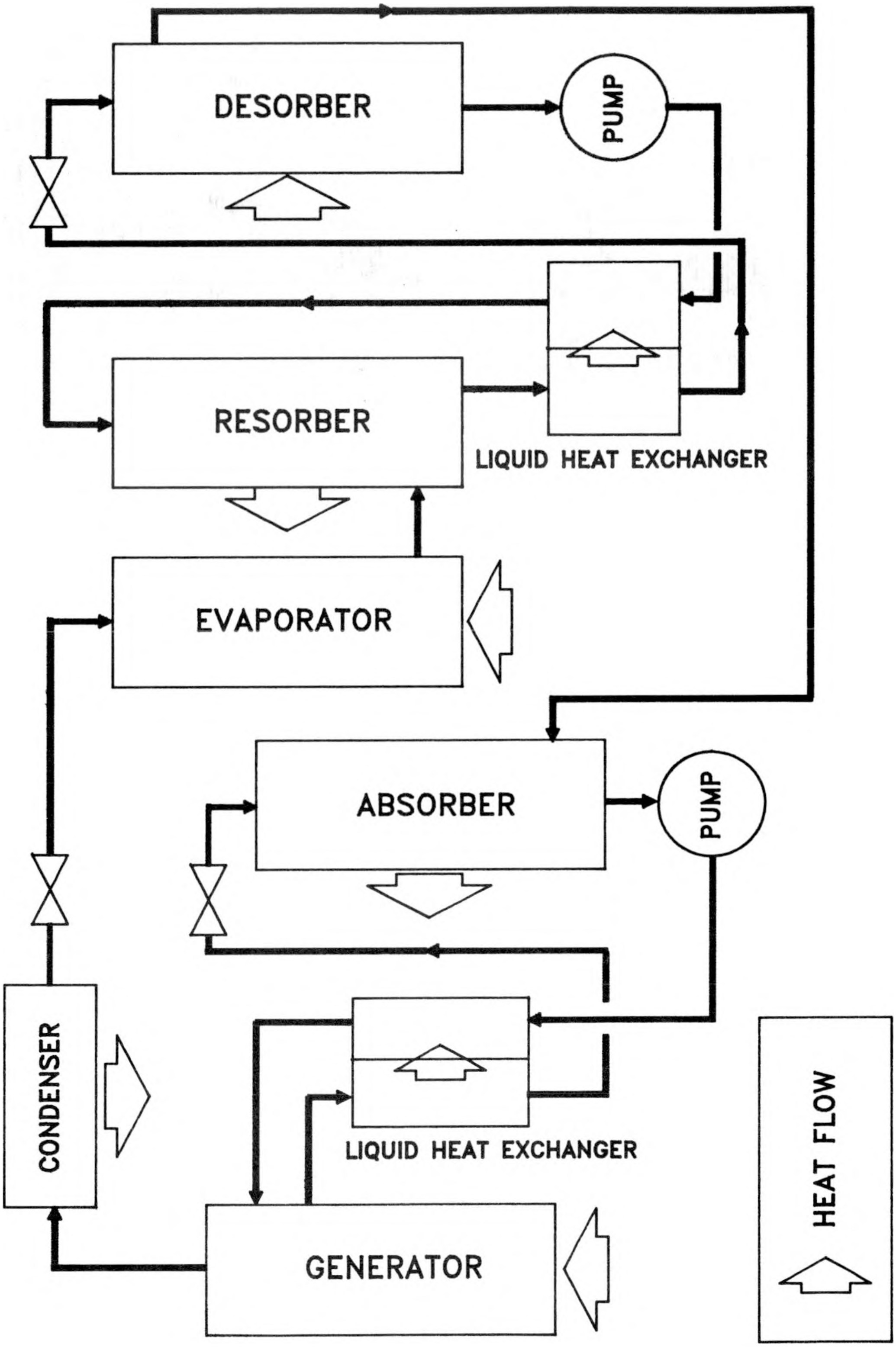

Fig. 12. Schematic of resorber-augmented absorption unit.

2.5 GENERATOR-ABSORBER HEAT-EXCHANGE CYCLE

The generator-absorber heat-exchange (GAX) cycle utilizes heat recuperation beyond that of the AHE cycle.[3,5] In the AHE cycle, the recuperation of absorber heat is limited by the sensible heat of the rich solution. Similarly, recuperation heating of the generator is limited to the sensible heat in the weak solution leaving the generator. In the GAX cycle, those sensible heat gains are still used, but by increasing the absorber and generator temperature ranges so that the two temperature ranges overlap, absorption heat at the overlap temperatures is also transferred to the generator. This direct transfer can occur whenever the weak solution concentration is decreased to the point that the temperature of the cool end of the generator is below that of the hot end of the absorber. All the heat of absorption in the overlap range can be utilized by the generator; the only reduction is the allowance that must be made for heat-transfer temperature differences.

The cycle is diagrammed on the chart of Figure 13. A schematic is shown in Figure 14. In Figure 13, the usable overlap between the absorber and generator temperature ranges is marked by the arrows indicating the heat transfer; the necessary temperature difference for heat transfer is indicated. Figure 14 shows the GAX heat-transfer coils and pump, which use a circulating fluid to transfer heat from the absorber to the generator. Because the vertical temperature gradients in the absorber and generator are reversed, it is necessary to cross-connect the lines between the two components, as shown. A number of other means of transferring the heat, other than by circulating a fluid, have been devised and are the subject of patent disclosures.

The value of being able to recuperate all the absorption heat in the overlap range is so great that it is tempting to search for ways of increasing the overlap. An obvious way is to use a fluid pair with a larger operating range. That will be discussed later. Another way to increase overlap would be to somehow divide the GAX cycle into a group of vertically stacked circuits as indicated in Figure 15.

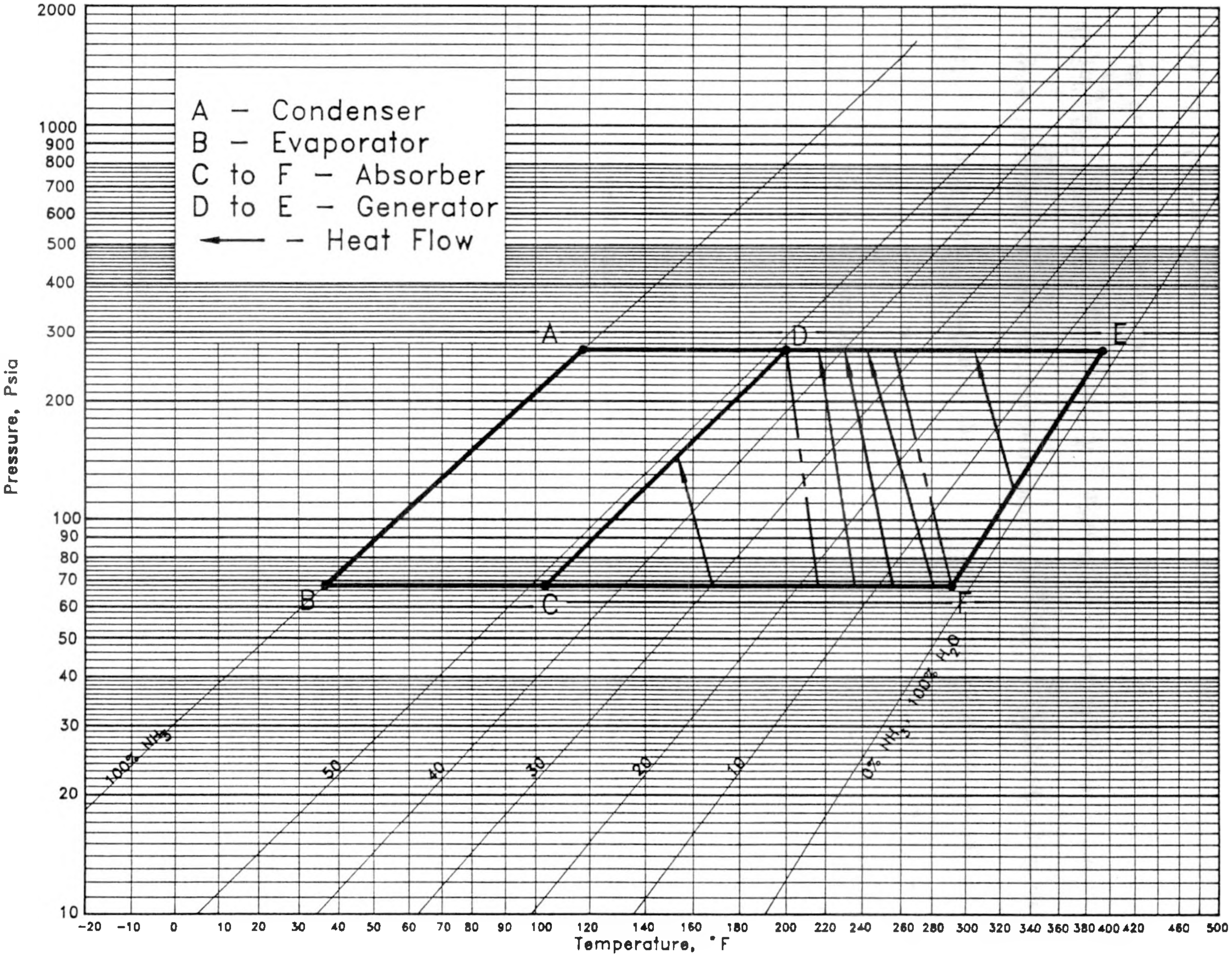

Fig. 13. Pressure-temperature-composition diagram of the generator-absorber heat-exchange cycle.

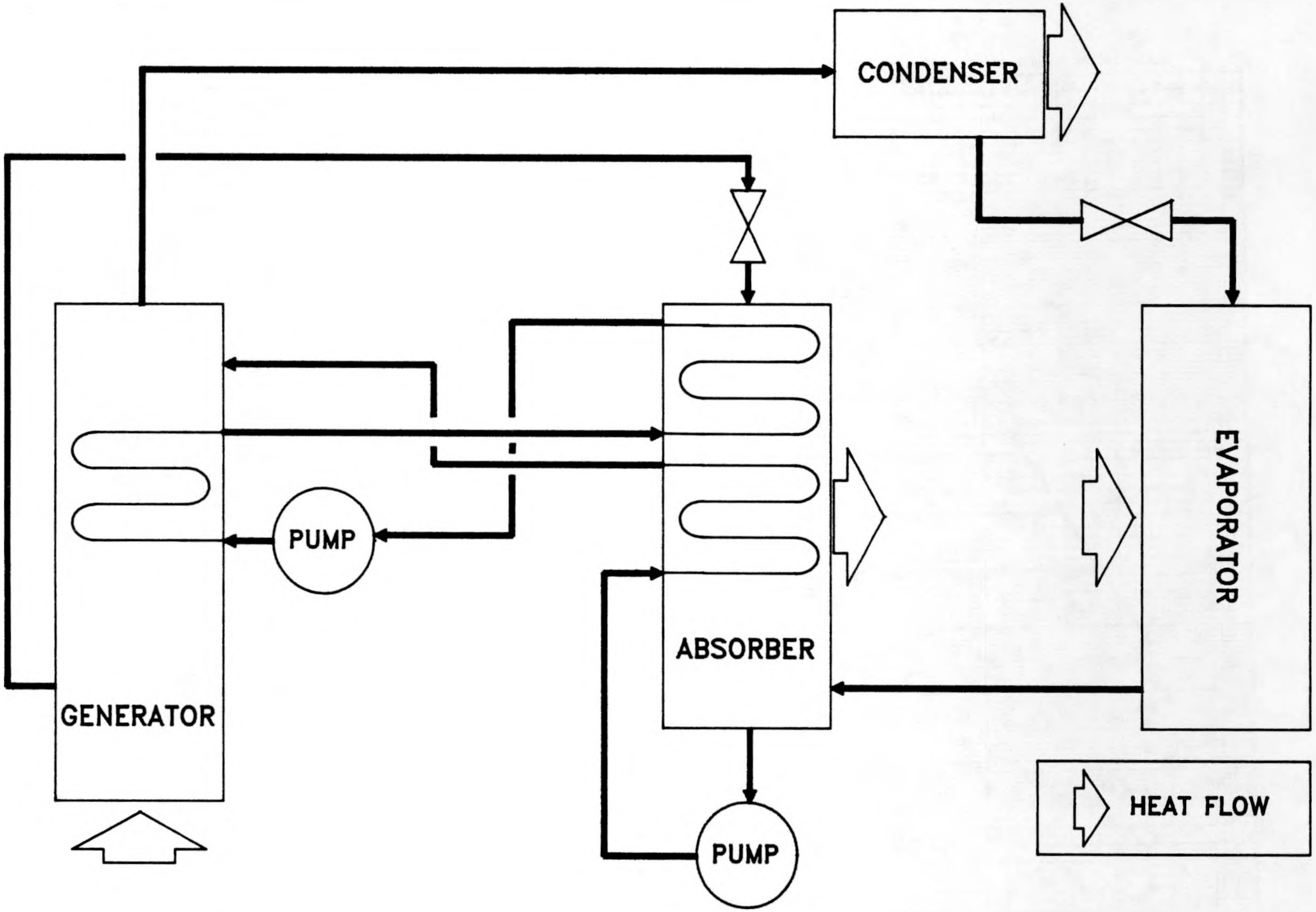

Fig. 14. Schematic of generator-absorber heat-exchange absorption unit.

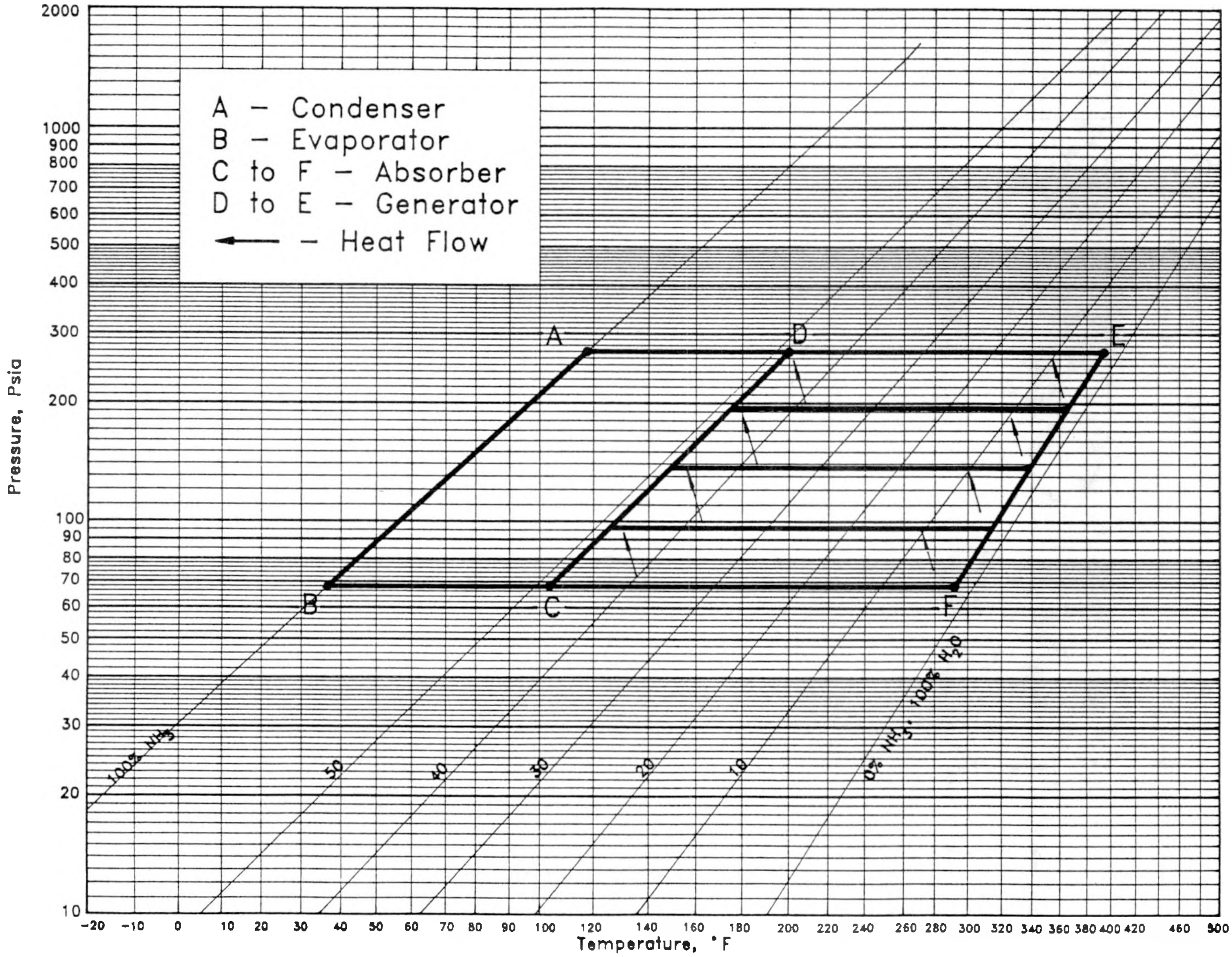

Fig. 15. Pressure-temperature-composition diagram showing potential of multiple cascaded generator-absorber circuits.

The majority of the temperature range of each of the absorbers would then be GAX overlap heat. The cycle diagrammed in Figure 15 is obviously not a true cycle; it is included only to point the reader in a tempting direction. The next three cycles are true cycles that utilize other methods of obtaining similar gains.

2.6 VARIABLE-EFFECT CYCLE

The variable-effect cycle also recuperates absorber heat to generate vapor, but does so with a two-step process.[9] The cycle uses an intermediate pressure range between the evaporator and condenser pressures. A cycle diagram is shown in Figure 16 and a schematic in Figure 17. The rich solution from the absorber is pumped to an intermediate pressure, defined by Kauffman as $\sqrt{Pe \times Pc}$ (ref. 13). At that pressure the liquid is heated in a desorber by overlap heat from the absorber and by heat from the rectifier, generating vapor that is transferred to a resorber. The partially stripped liquid is then pumped to the generator at normal condenser pressure.

The rich liquid, flowing from C to G, is heated to the saturation temperature at the intermediate pressure by AHE heat, but the intermediate liquid, H to D, is heated to generator temperature by a liquid heat exchanger and by rectifier heat. The generator is heated by the external heat source, by the heat of resorption (shown by the arrows), and by the heat given up by the weak solution as it cools to the resorber temperature, from E to K. The resorber operates at a pressure slightly below that of the desorber. Resorption of the desorber vapor into the weak liquid produces the heat transferred to the generator and enriches the weak solution. The enriched weak liquid then flows through the liquid heat exchanger, J to F, heating the intermediate liquid, H to D, and enters the absorber.

In this cycle, the fraction of absorber temperature range that falls in the overlap range is greater than in a GAX cycle at the same overall operating conditions.

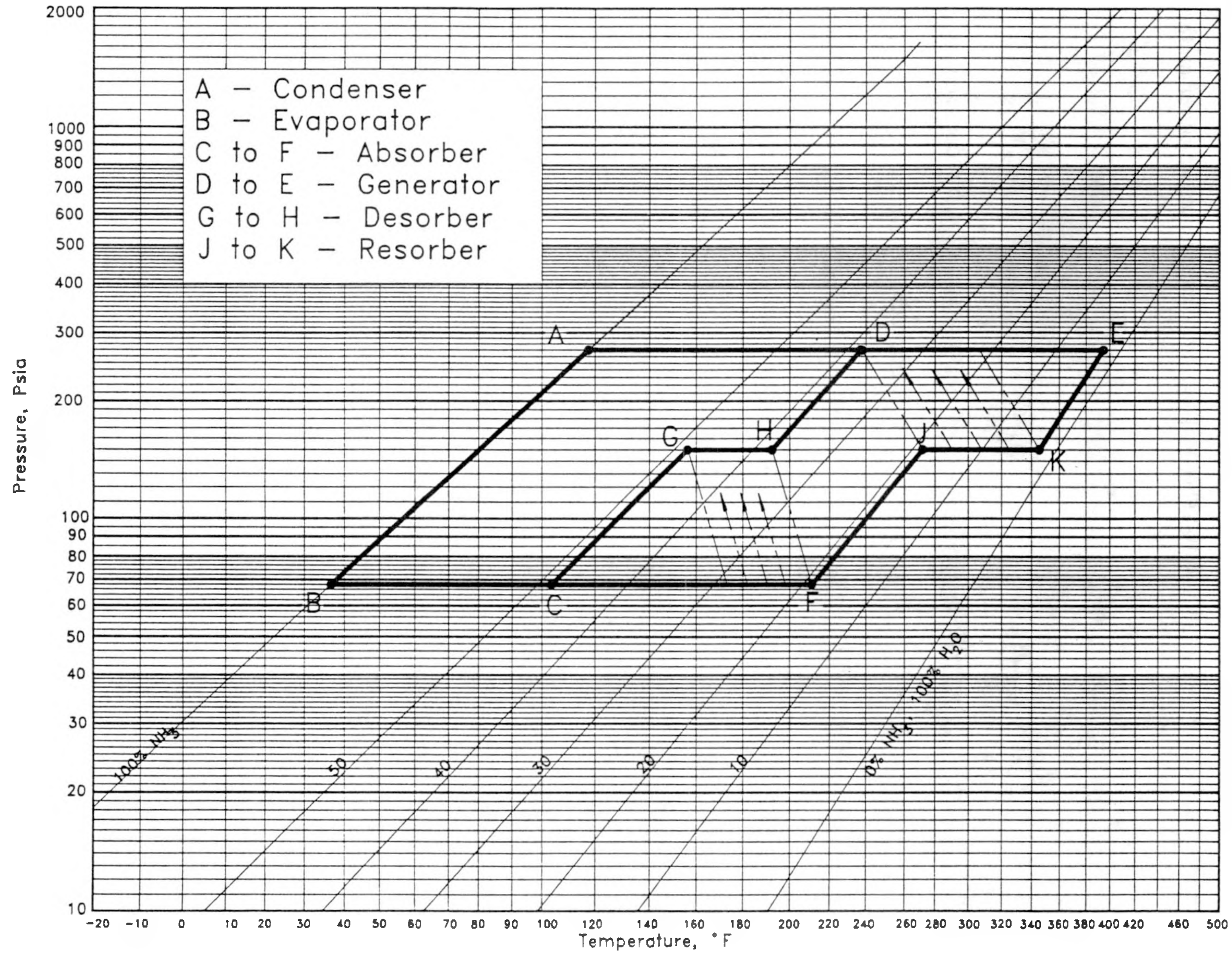

Fig. 16. Pressure-temperature-composition diagram of the variable-effect cycle.

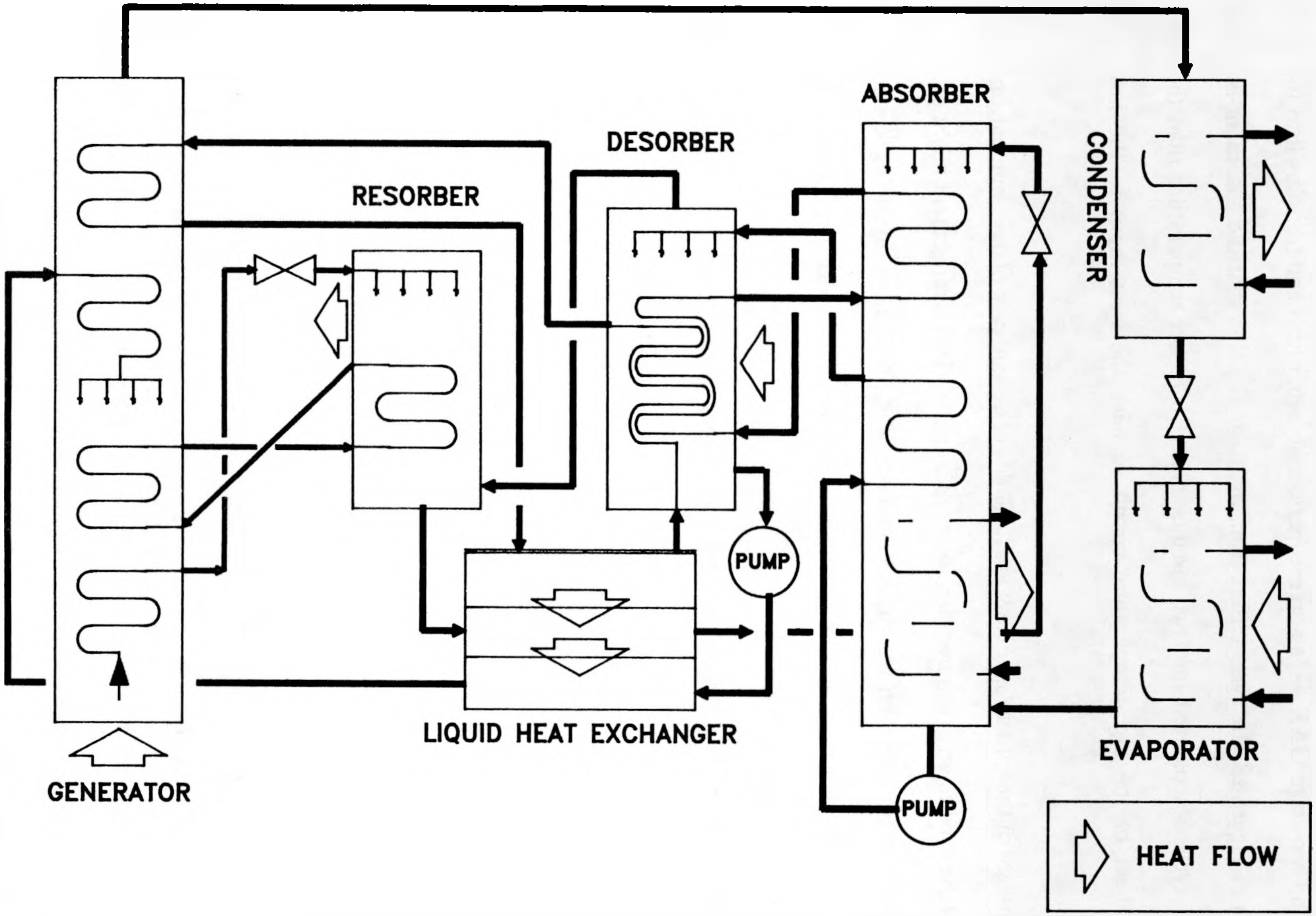

Fig. 17. Schematic of variable-effect absorption unit.

2.7 TWO-STAGE GENERATOR-ABSORBER HEAT-EXCHANGE CYCLE

The two-stage GAX cycle also recuperates absorber heat and transfers it to the generator.[10] As shown in Figure 18, the cycle differs from the basic GAX in that the absorbent-liquid circuit of the GAX cycle is separated into two circuits, one at a higher temperature than the other. The absorber heat of the high-temperature circuit supplies the heat input to the low-temperature generator.

Advantages of the split arrangement are flexibility of design, ease of development, and opportunity to control the fluid flows in the two circuits independently. This arrangement might provide a higher COP, allow a closer approach to Carnot conditions, and provide opportunities to simplify the heat transfer from absorber to generator. An obvious disadvantage is that two solution pumps and two liquid heat exchangers are required.

The rectifier losses from the high-temperature generator can be expected to be large. In the COP calculations they were used as input heat to the low-temperature generator and as heat for the rich solution of the high-temperature circuit.

2.8 DOUBLE-EFFECT REGENERATIVE (2R) CYCLE

The 2R cycle is also a two-stage recuperative cycle.[11] As in the two-stage GAX cycle, the condenser and evaporator are common to both stages, and there are two independent generator-absorber circuits.

The low-temperature circuit is a standard single-stage circuit, but the high-temperature circuit uses a new approach (Figure 19). It enriches the rich solution by resorption processes at a series of pressure levels between the absorber pressure and the generator pressure. The vapor to be resorbed is obtained from desorption stages in the weak-liquid circuit that correspond to the resorption stages (at slightly higher pressures to provide for the vapor flow). The heat input to the cycle occurs at the

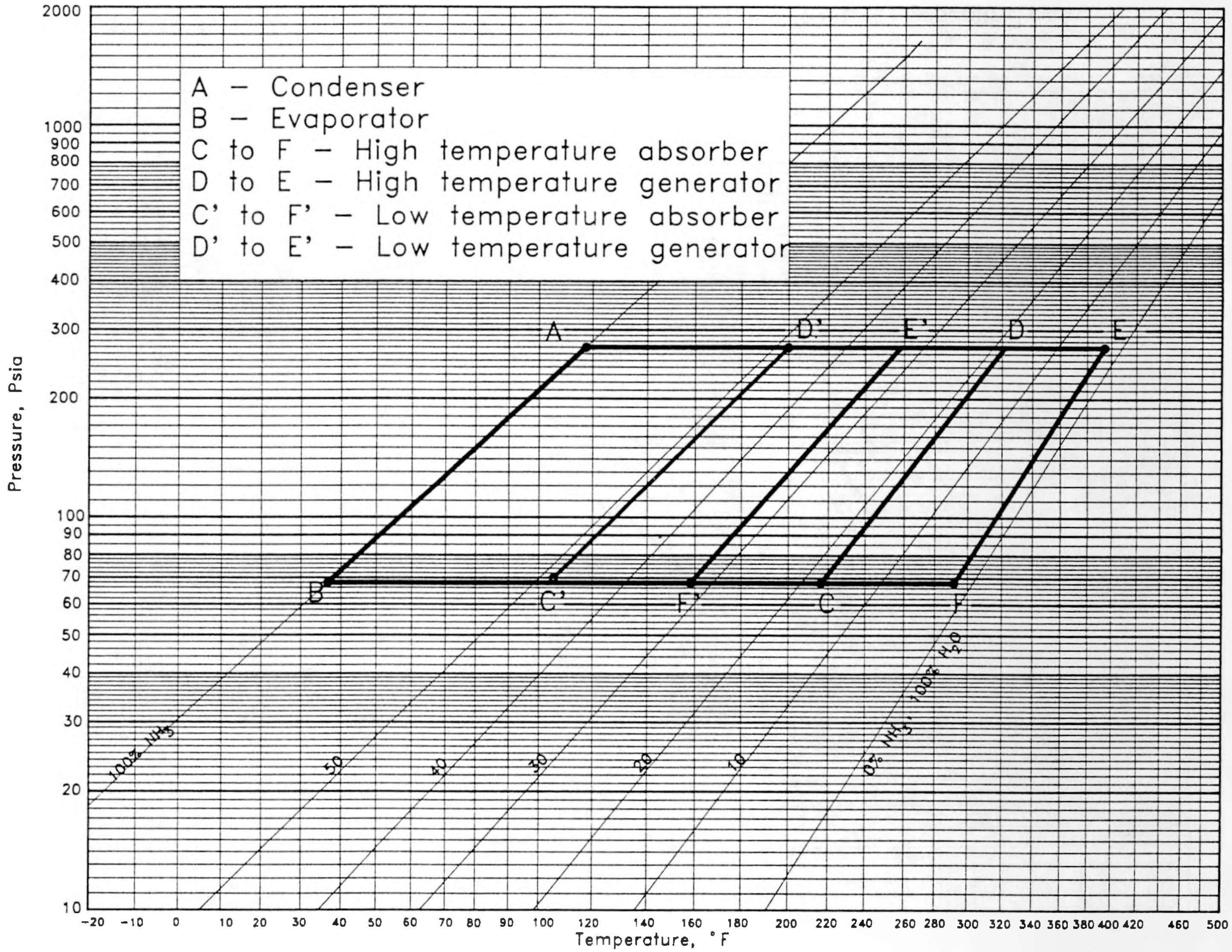

Fig. 18. Pressure-temperature-composition diagram of the two-stage generator-absorber heat-exchange cycle.

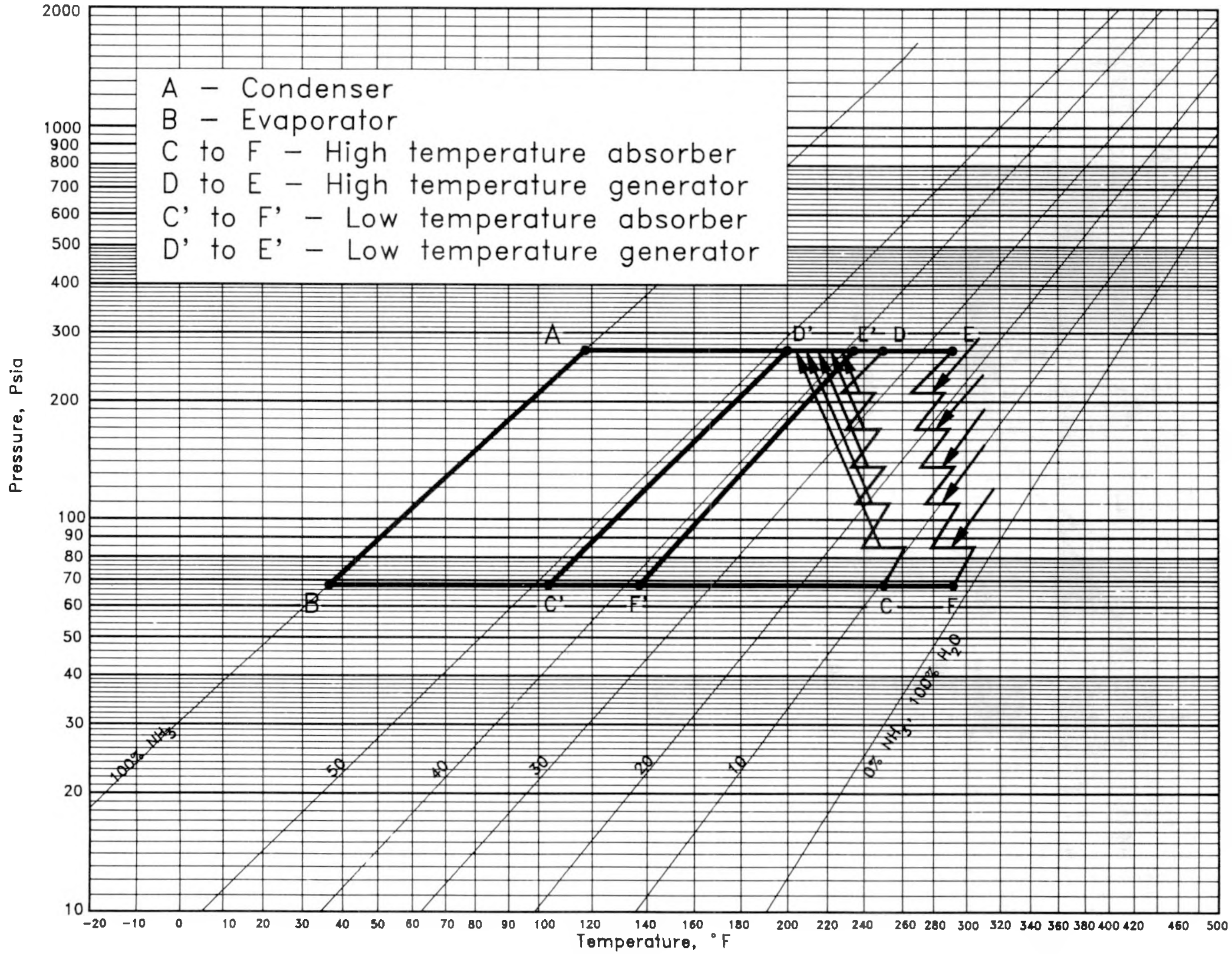

Fig. 19. Pressure-temperature-composition diagram of the double-effect regenerative cycle.

multiple desorber stages—six to eight depending on the design. The heat output from the resorber stages serves as the heat input to the low-temperature generator, D to E. In the Phase I analysis the resorber temperatures were set higher than the temperature of the low-temperature cycle generator by the temperature difference required to transfer their heat outputs.

The enrichment used in the COP calculations for Phase I was sufficient to lower the temperature of the generator to below the temperature of the absorber. The purpose was to use the heat of absorption to provide all the heat required by the generator.

This cycle was devised for use with solar heat.[11] As can be seen in Figure 19, the heat input temperature is lower than that of the other cycles. The cycle outlined in Figure 19 is a simplification to indicate the main concept. Additional details may be found in reference 10.

3. CYCLE EVALUATIONS

3.1 PROCEDURE

The cycle evaluations were made in a series of steps for each cycle, using ammonia/water as the working fluid. Cycle diagrams, primarily of the P-T-X type (and others when helpful), were first analyzed for means of optimizing the cycle performance. In addition to the process unique to each cycle, all other means of heat recovery that fit the cycle were applied if helpful.

Material and heat balances for each component were made around the cycle. Table 1 shows an example. Mass flow rates were calculated entering and leaving each component on the basis of one pound of ammonia refrigerant. The vapor and liquid compositions, based on temperatures that were adjusted to allow for heat-transfer temperature differentials and mass-transfer pressure differentials, were determined from existing ammonia/water P-T-X data.

The heat and mass balances were made at temperatures representative of the ARI rating temperature of 47°F outdoor ambient for heating. Calculations made for air conditioning operation at the 95°F ARI rating temperature were found to be so close to the cooling performance at the 47°F heating operation that it was decided to use the refrigeration and heating COPs at the 47°F conditions for comparing the cycle performances. The heating and cooling COPs therefore differ by 1.0 because the calculations were made on the cycle basis.

The refrigeration and heating COPs were calculated from the heat balance results. Using a heat output of 80,000 Btu/h as a basis, the theoretical pumping power of each cycle was calculated from the operating pressures and the rich-liquid flow rates. For the same 80,000 Btu/h output, the total heat quantities to be transferred in the operation of the cycle were calculated from the heat balances.

Table 1. Heat and Mass Balance Summary for an Absorber Heat-Exchange Cycle

	Composition Wt. Fraction	Pressure psia	Temp. °F	Enthalpy Btu/lb	Lb	Heat
Condenser		Condensing Temp 115°F				
Vapor in	0.995	265.2	150.8	584.6	1.0	584.6
Liquid out	0.995		105	82.9	1.0	82.9
					Condenser Heat Out	501.7
Evaporator-precooler		Evaporating Temp 37°F				
Liquid in	0.995		105	82.9	1.0	82.9
Vapor out	0.995	68.3	95	582.7	1.0	582.7
Liquid out	0.515	68.3	95	−39.8	0	0.0
					Evaporator Heat In	−499.2
					Maximum Precooler Heat	82.9
Absorber						
Weak liquid in	0.1829	68.3	211.3	127.3	1.724	219.5
Vapor in	0.995	68.3	95	582.6	1.0	582.6
Liquid in	0.515	68.3	95	−39.8	0.0	0.0
Rich liquid out	0.481	68.3	105	−29.9	2.724	−81.4
					Total Absorber Heat Out	883.5
					AHE Recovery (Rich Liquid, 105°F to 198.3°F)	285.9
					Net Absorber Heat Out	597.6
Generator-stripping section						
Rich liquid in	0.481	265.2	198.3	75.1	2.724	204.5
Vapor out	0.9684	265.2	205.3	635.3	1.058	672.2
Condensate in	0.510	265.2	188.3	64.9	0.058	3.8
Weak liquid out	0.1829	265.2	317.4	244.7	1.724	421.9
					Total Generator Heat	−885.9
					Weak-Liquid Heat to Generator (WL, 317.4° to 211.3°F)	202.5
					Net Generator Heat	−683.4
Rectifier						
Vapor in	0.9684	265.2	205.3	635.3	1.058	672.2
Condensate out	0.510	265.2	188.3	64.9	0.058	3.8
Vapor out	0.995	265.2	150.8	584.6	1.0	584.6
					Rectifier Heat	83.8
					Total Heat Out	1183.1

COP Cooling: 0.73
COP Heating: 1.73

The results are given in Table 2 together with listings of the operating pressure levels, number of pumps, number of major components, and low-temperature cutoff limits. The AHE cycle listed was included in the tabulation partly for comparison, but also because by increasing its heat transfer areas beyond normal to reduce Δt's and Δp's, the COPs of the AHE cycle can be raised to just above the project objectives. A major component was defined as being either a phase-change, mass-transfer component or a larger-than-usual heat exchanger. The cutoff limit is the evaporator temperature at which the cycle just ceases to operate in heating mode, assuming a constant indoor temperature. If the cutoff limit was above -15°F, the possibility and difficulty of converting the cycle to one capable of operating to the lower temperatures was investigated. The low-temperature cutoff was determined by heat balances in a few cases, but P-T-X diagrams were sufficient for most cycles. In a number of cases, the increases in pumping power at low temperatures could make the cycle impractical at temperatures above the cycle cutoff.

The calculated figures in the double-effect cycle column in Table 2 were placed in parentheses because of the uncertainty of the ammonia/water properties at the high pressure. Most of the ammonia/water experimental data were measured up to only 300 psia, with a few more to 500 psia and scattered points to 700 psia.[1] For these calculations, extrapolations were made up to 1300 psia. The extrapolations appeared reasonable relative to the data of pure ammonia and water, but there was no sure way of judging their accuracy.

When all the data had been compiled, the advantages and disadvantages of each cycle were compared and a preliminary selection was made. The possibilities of developing higher performances or of eliminating disadvantages by using other fluids, etc., were then investigated to determine whether those possibilities would affect the rankings. Based on all the information assembled, the cycle preferred for development in the possible Phase II and Phase III was then selected.

Table 2. Summary of Cycle Analyses

Results for Heat Pumps with 80,000 Btu/h Output

Cycle	Double Effect*	Resorber Augmented	Absorber Heat Exchange, Low Δt	Generator-Absorber Heat Exchange	Variable Effect	Two-Stage GAX	2R
COP Refrigeration	(1.11)	0.83	0.79	1.03	1.04	1.06	1.03
COP Heating	(2.11)	1.83	1.79	2.03	2.04	2.06	2.03
Theoretical Pumping Power (watts)	(388.4)	87.5	39	36	35	45	110
Total Heat Transfer (Btu/h)	(391,550)	288,904	197,935	220,033	263,512	293,608	275,339
Pressure Levels (psia)	(67/285/1250)	15/68/265	68/265	68/265	68/135/265	68/265	68/265 Plus Six Intermediate
No. of Pumps	2	2	1	1	2	2	2 1 MultiStage
No. of Major Components	8	7	4	5	6	7	8
Cutoff Limit (°F)	(20)	10	−50	5	−15	10	20
Convertibility	Complex	Complex	No Need	Automatic	No Need	Complex	Complex

---

*Data are in parentheses because of uncertainty about ammonia/water properties at the high pressures.

3.2 RESULTS

The results given in Table 2 show that five of the cycles are capable of operating at refrigerating cycle COPs slightly above 1.0 at the specified conditions. The AHE and the resorber-augmented cycles were found to be capable of meeting the minimum COP objective of 0.78 (the AHE cycle only when operating under the special low-Δt conditions), but both had COPs significantly lower than those of the other five cycles. Those two cycles were the first to be eliminated.

The COPs of the other cycles were relatively close together, so close that the other criteria had to become the primary bases for choosing between them. The double-effect cycle had the highest COP, but it also had high pressures, high pumping power, high heat-transfer area requirements, unusual complexity at the optimized COP design (shown in Figure 20), a high cutoff temperature, and difficulties in converting to a single-stage cycle.

The two-stage GAX cycle had a small COP advantage, but it required two pumps, two good-sized liquid heat exchangers, and two absorbers and generators. Its cutoff point was relatively low, but rapidly rising pumping power would probably set the practical cutoff at higher temperatures. Conversion to a simpler cycle would require three-way valving to connect the generator of the high-temperature circuit to the absorber of the low-temperature circuit, plus arranging solution concentrations and more rectification.

The variable-effect cycle was found to have an important advantage: a low cutoff limit that met the objective with no need for convertibility. On the negative side, two pumps were required, and the heat-transfer surface was high. It operated at three pressure levels, requiring (1) accurate synchronization of the two liquid flow-control valves and the pumps, and (2) means to accurately control the intermediate pressure at all operating conditions.

The 2R cycle had a COP equal to those of the four other top cycles, and because its heat input temperature was lower than those of the others, its COP was the closest to Carnot. The 2R cycle,

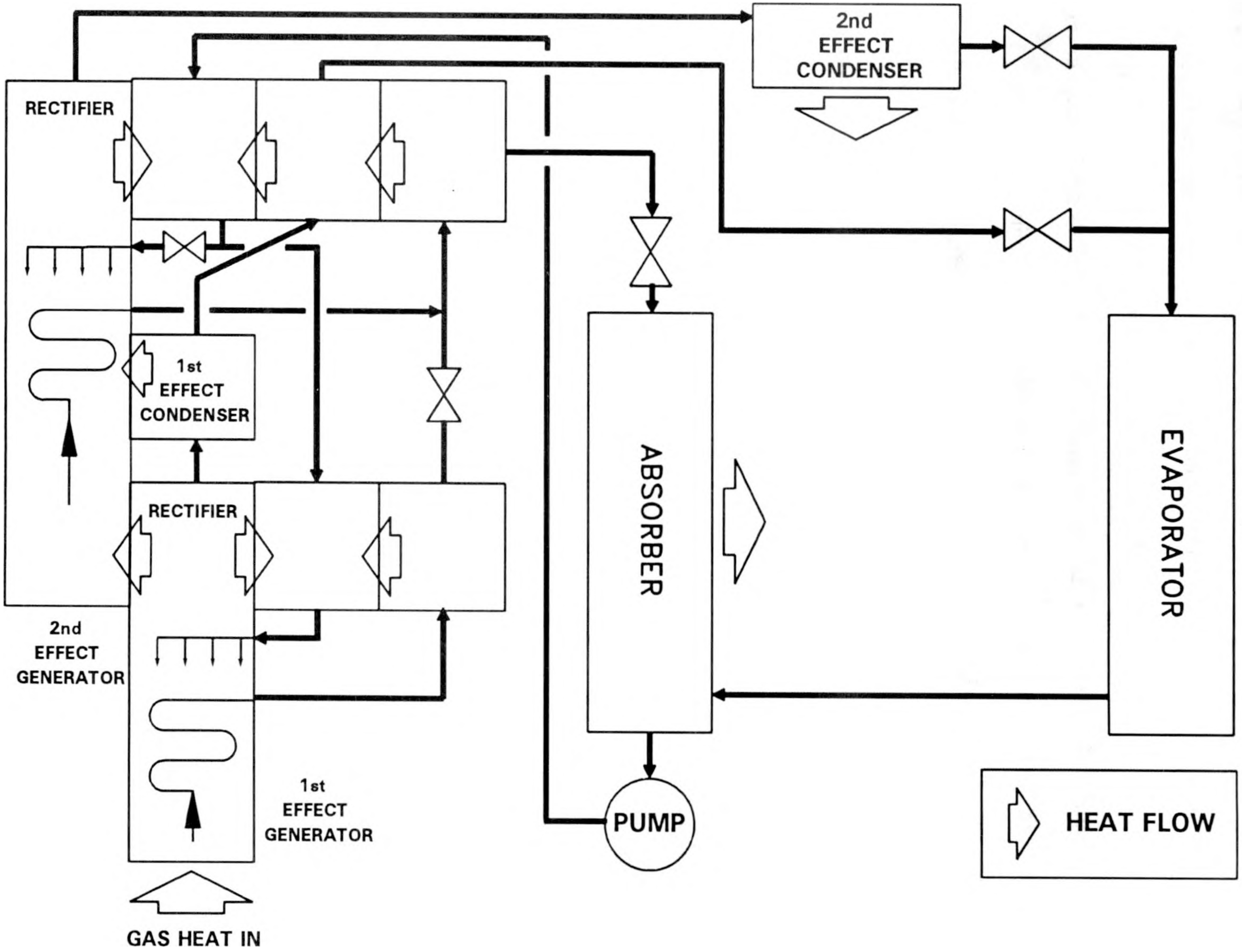

Fig. 20. Schematic of double-effect absorption unit with recuperation.

however, had a high cutoff temperature. It would be complex and expensive to add a gas-fired generator to operate the low-temperature cycle independently when outdoor ambients drop below the cutoff temperature. The cycle also had the other disadvantages of the two-stage solution circuit, plus the complexities of the multistage pump and resorber/desorber system.

Overall, the GAX cycle was found to be equal to or better than the other cycles in all but one category. The one problem area was that its cutoff temperature was 5°F instead of the desired −15°F. At the 5°F outdoor temperature, the pressure of the evaporator and absorber would drop to 34 psia. The cool end of the absorber, point C of Figure 13, would remain at 105°F, causing point D to move to about 245°F along the line AE, while point F, in dropping to 34 psia on an extension of the line EF, would also be at about 245°F. Under those conditions the overlap and the GAX effect would end—but the cycle would then be an AHE cycle. The AHE cycle has been operated to very low temperatures, well below −15°F. From the hardware standpoint, the conversion from the GAX cycle to the AHE cycle could, in principle, also be simple, requiring only that the GAX heat-transfer process from the absorber to the generator be turned off. The GAX cycle was thus found to meet all requirements and to have important advantages over each of the other cycles.

3.3 POTENTIALS FOR CYCLE IMPROVEMENTS

Table 2 indicated that the GAX cycle was the best choice for the residential application; however, the evaluation conditions had placed differing constraints on individual cycles. We therefore desired to determine whether there were means of further improving the COPs of individual cycles and whether this improvement might change the rankings.

Fluids tailored to the individual cycles should help increase the COPs beyond those calculated with ammonia/water. Many cycles, but especially the GAX, the variable-effect, the two-stage GAX, and the 2R, should be helped by using working fluids with a wider temperature spread between the refrigerant and absorbent vapor-pressure lines on the P-T-X diagram. The R-22/E181 fluid pair has that wider spread, but does not have high-temperature stability. The ammonia/lithium-bromide-water ternary mixture also has the wider temperature spread;[14] operation up to 500°F and slightly higher appeared possible if corrosion and/or materials problems could be solved. The absorber-generator overlap would be increased greatly. The other three recuperation cycles should accrue similar advantages. The published data on the ternary fluid,[14] especially enthalpy data, were not in a form that could be used in conjunction with the ammonia-water data to make accurate calculations around the whole cycle during this investigation. Rough estimates indicated that any resulting COP improvements should not differ greatly among the four recuperative cycles. The selection among the four cycles, then, would still be determined by the other criteria.

Possible gains for the double-effect cycle were developed in a different manner. In the evaluation, the double-effect cycle was calculated with a 400°F peak generator temperature because of ammonia/water corrosion limits with low-carbon steel. The other cycles were also kept to that limit, but at the pressures of the recuperating cycles, the 400°F limit with ammonia/water was caused both by corrosion and by the location of the vapor-pressure line of water. In the case of the double-effect cycle, the only limit on the peak generator temperature was caused by corrosion, hence a fourth double-effect cycle variation was designed as shown in Figure 21. For this cycle, a new fluid was not required, only that the ammonia/water corrosion problems be solved to 500°F. A heat-balance calculation was made with the cycle of Figure 21 at a peak boiler temperature of 500°F. It resulted in COPs of 2.4 in the heating mode and 1.4 for refrigeration. That was an important gain, but with ammonia/water, the high-pressure-related problems and corrosion difficulties remain. This analysis determined, however, that if a suitable low-pressure working fluid could be found, an air-cooled double-effect cycle would also have improved potentials.

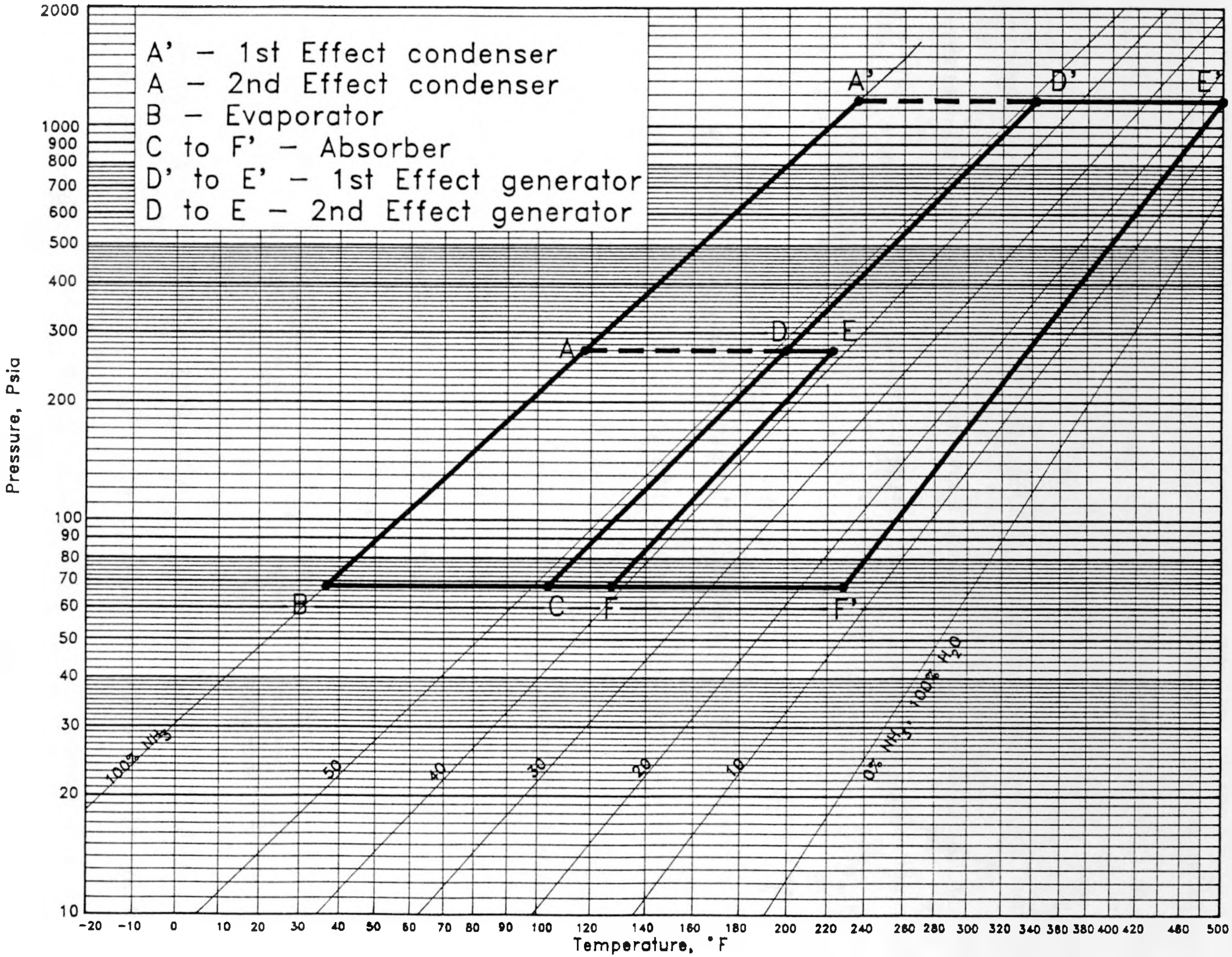

Fig. 21. Pressure-temperature-composition diagram of the double-effect cycle extended to 500°F.

It would also be desirable for the refrigerant to have a high critical pressure and temperature. With ammonia, the double-effect performance was limited by the proximity of the critical point at about 1640 psia. As shown in Figure 22, the latent heat drops rapidly as the critical point is approached. At 1250 psia (243°F) the heat of condensation/vaporization of ammonia drops to about half that at 115°F–120°F, greatly lowering the heat available for the second-effect generator. In the calculations for Table 2, that loss was partly compensated for by the use of rectification heat and the sensible heat of the condensate. Unfortunately, a low-pressure, high-temperature fluid pair with a high-critical-point refrigerant was not available for the double-effect cycle.

3.4 CYCLE AND FLUID SELECTION

Based on the results of the cycle evaluations and the improvement potentials discussed above, the GAX cycle was selected as the cycle to be proposed for use in the Phase II and Phase III heat pump developments.

The ammonia/water pair was selected as the working fluid. During the analyses and evaluations, a continuing search had revealed that no other fluid combination could suit the application as well. The ammonia/lithium-bromide–water ternary was considered a possible eventual replacement, but it was not suited for a near-term development of the GAX heat pump. The disadvantages of the ammonia/water pair are the toxicity and flammability of ammonia. Many of the other known pairs also have one or both of those disadvantages.

The national refrigeration codes allow ammonia to be used in space conditioning equipment under specific conditions.[15,16] A convenient way to satisfy those conditions for residential applications is to install the ammonia-containing part outdoors and to transfer the cooling and heating effects to the conditioned space by means of a secondary heat transfer fluid. These code requirements for the use of ammonia have been very effective. A recent survey of past toxicity incidents in refrigeration and

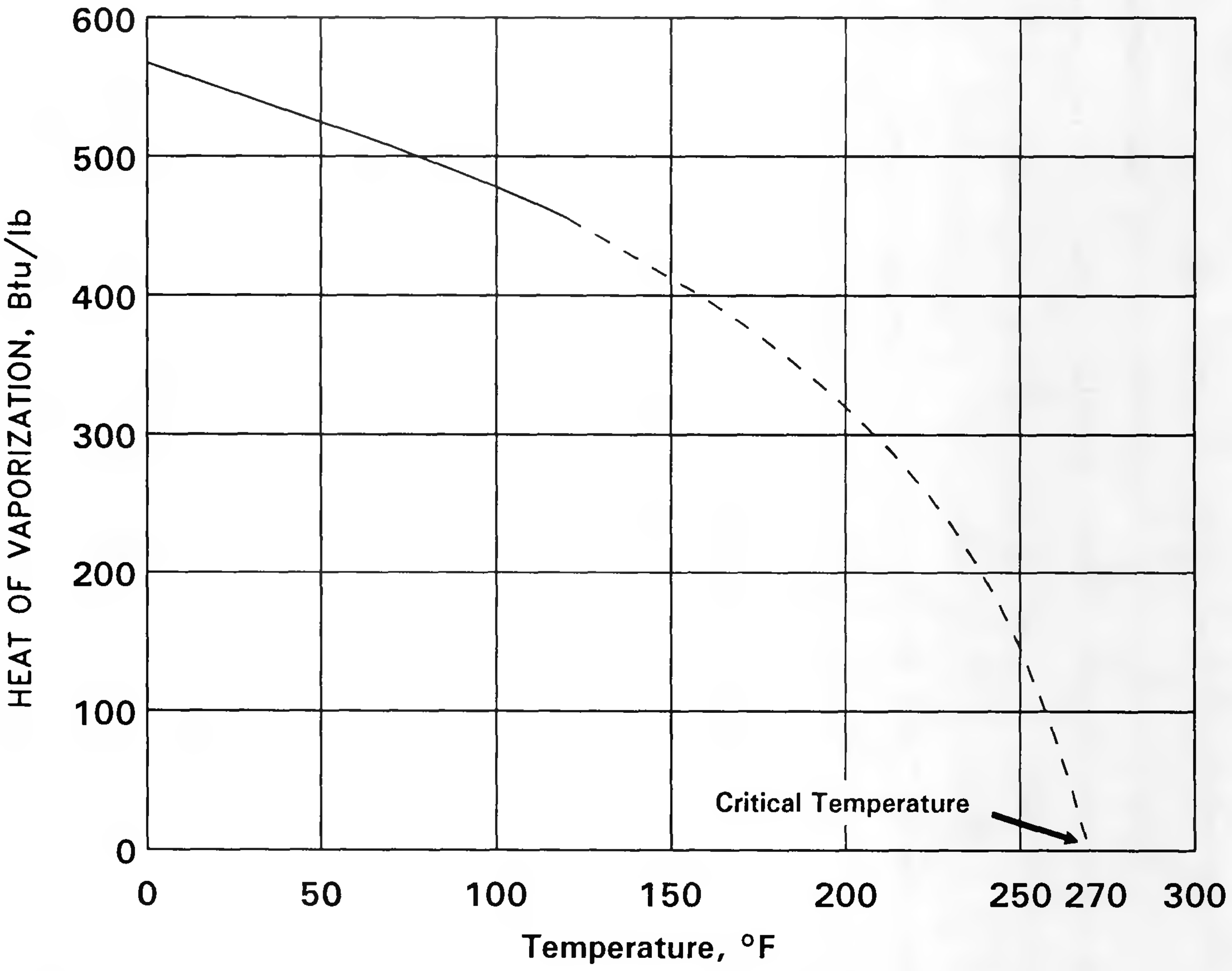

Fig. 22. Liquid ammonia heat of vaporization.

air conditioning found a minimum of problems: about the same for ammonia as for chlorofluorocarbons.[17] It is our understanding that the ammonia incidents occurred in industrial refrigeration rather than in residential refrigeration and air conditioning.

The Group 2 rating of ammonia is based on toxicity rather than flammability. Ammonia will burn, but it is listed as nonflammable by the Code of Federal Regulations and by the U. S. Department of Transportation for transportation purposes. The conditions under which ammonia will burn are limited and are unlikely to occur.[18,19] The presence of water in absorption systems can reduce or eliminate the flammability.[18] (In large industrial ammonia compression systems the lubricating oil can atomize upon leaking and thereby lower the flammable concentration to 4%.[20])

No carcinogenic or other long-term toxicity has been attributed to ammonia. Its acute toxicity at high exposure levels is serious, but that hazard is essentially eliminated by the outdoor installation and by ammonia's penetrating and alarming odor. It is detectable at 0.04 ppm; the eight-hour working limit is 25 ppm.[19]

It is thus very improbable that even a major break of a tube or the opening of the pressure relief valve of the outdoor unit would have any serious effect. Because ammonia can burn, however, and can affect the mucous membranes of the eyes, nose, mouth, lungs, etc., the manufacturer will want to ensure that ammonia from any source will be rapidly diluted and directed upward.

3.5 ADDITIONAL PERFORMANCE CALCULATIONS AND ANALYSES

After selection of the GAX cycle, a broader analysis of its potential operating performance was made. Calculations of cycle performance were made as a function of peak boiler temperature at the 47°F ambient temperature condition. That performance is plotted in Figure 23. The value of high heat input temperature is well indicated.

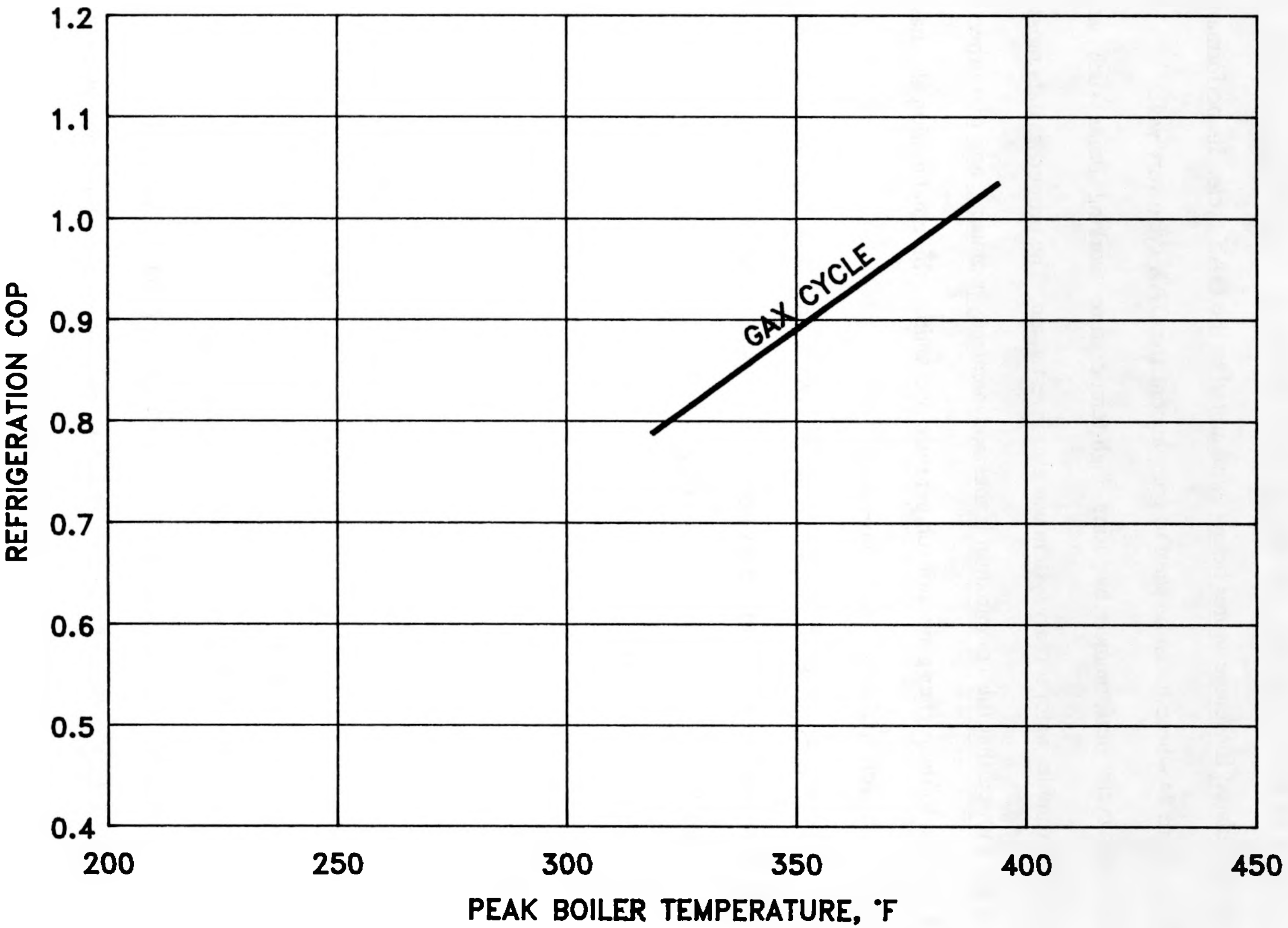

Fig. 23. Refrigeration COPs for heat pump operation at 47°F ambient.

The performance was then calculated for a range of outdoor ambient temperatures using a peak boiler temperature of 395°F. Those results are shown in Figure 24. The performance of an AHE cycle was also calculated for temperatures below those suited to the GAX cycle. Its performance is also shown in Figure 24 where it can be seen to complement the GAX cycle very well.

Improving the cycle performance by using high-temperature working fluids such as the ammonia/lithium-bromide–water ternary system was also investigated. The properties data measured by Radermacher showed that this combination is thermodynamically promising, and that others may be also.[9] In addition to the higher generator temperature possibilities, this combination also has less water in the vapor, thereby reducing rectification heat.

It was impossible to make accurate heat balances with the ternary fluid, because, as mentioned earlier, the Radermacher data could not be properly matched with the ammonia/water data during this Phase I program. Rough calculations (partly based on extrapolations and ammonia/water data), at a 500°F peak boiler temperature, however, allowed estimation of the performance plotted in Figure 25.

As can be seen, the heating COP is significantly higher at 47°F, and, equally important, this gain carries through to the low-temperature range, lowering the cutoff point to −10°F. Possible overlap ranges with the ternary fluid at 47°F and −10°F ambient temperatures are shown in the P-T-X diagrams of Figures 26 and 27. In those two figures, the vapor pressure line for the 60% lithium-bromide / 40% water absorbent is shown superimposed at the lower right corner of the ammonia/water P-T-X chart. Operation to a peak boiler temperature of 500°F can be fit in comfortably within that line. As a result, the usable overlap temperature range of the absorber increases greatly at the 47°F ambient conditions of Figure 26. An overlap still occurs at a −15°F evaporator, but it is too small to be of value, as seen in Figure 27.

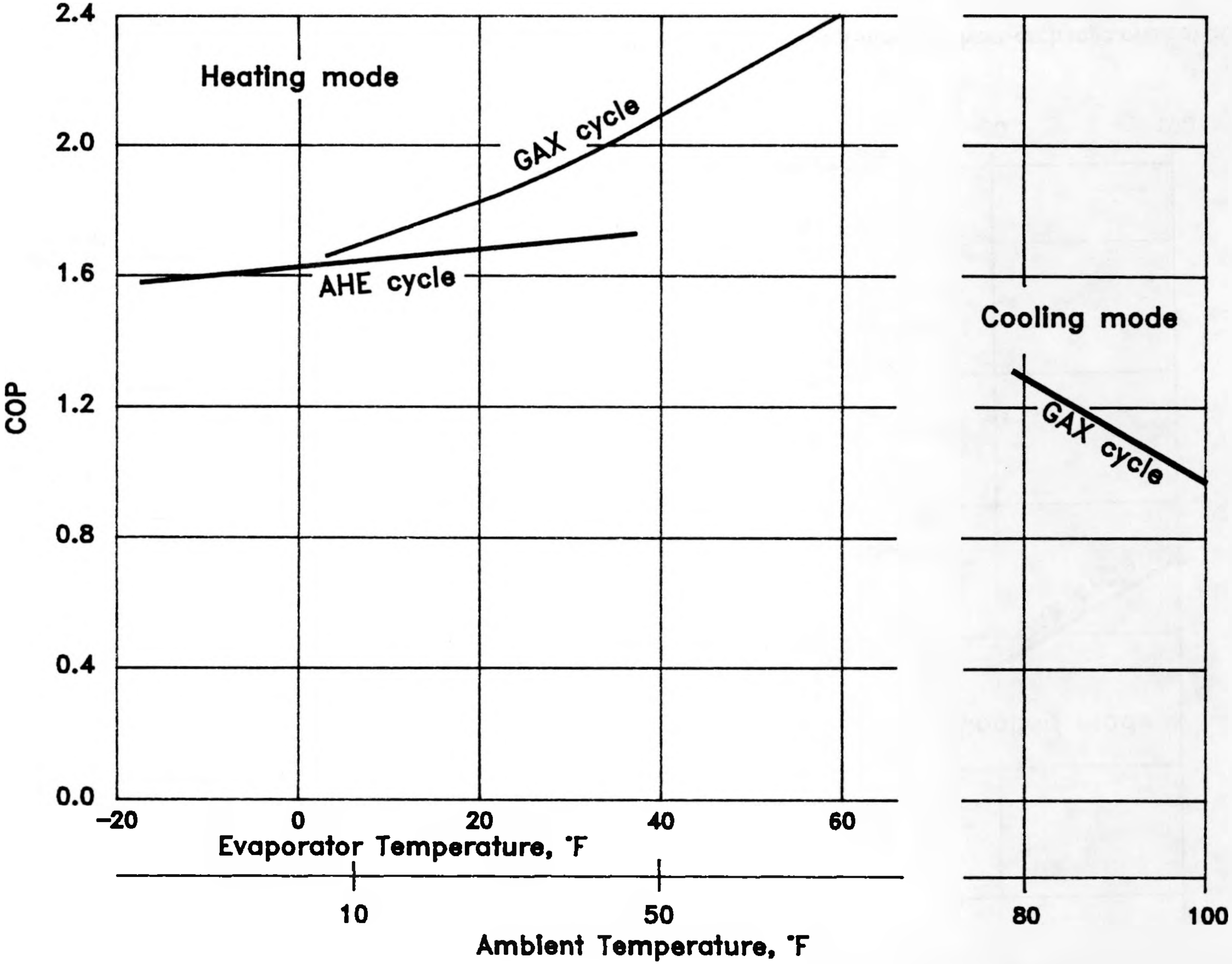

Fig. 24. Coefficient of performance versus ambient temperature for generator-absorber heat-exchange cycle at 395°F peak boiler temperature.

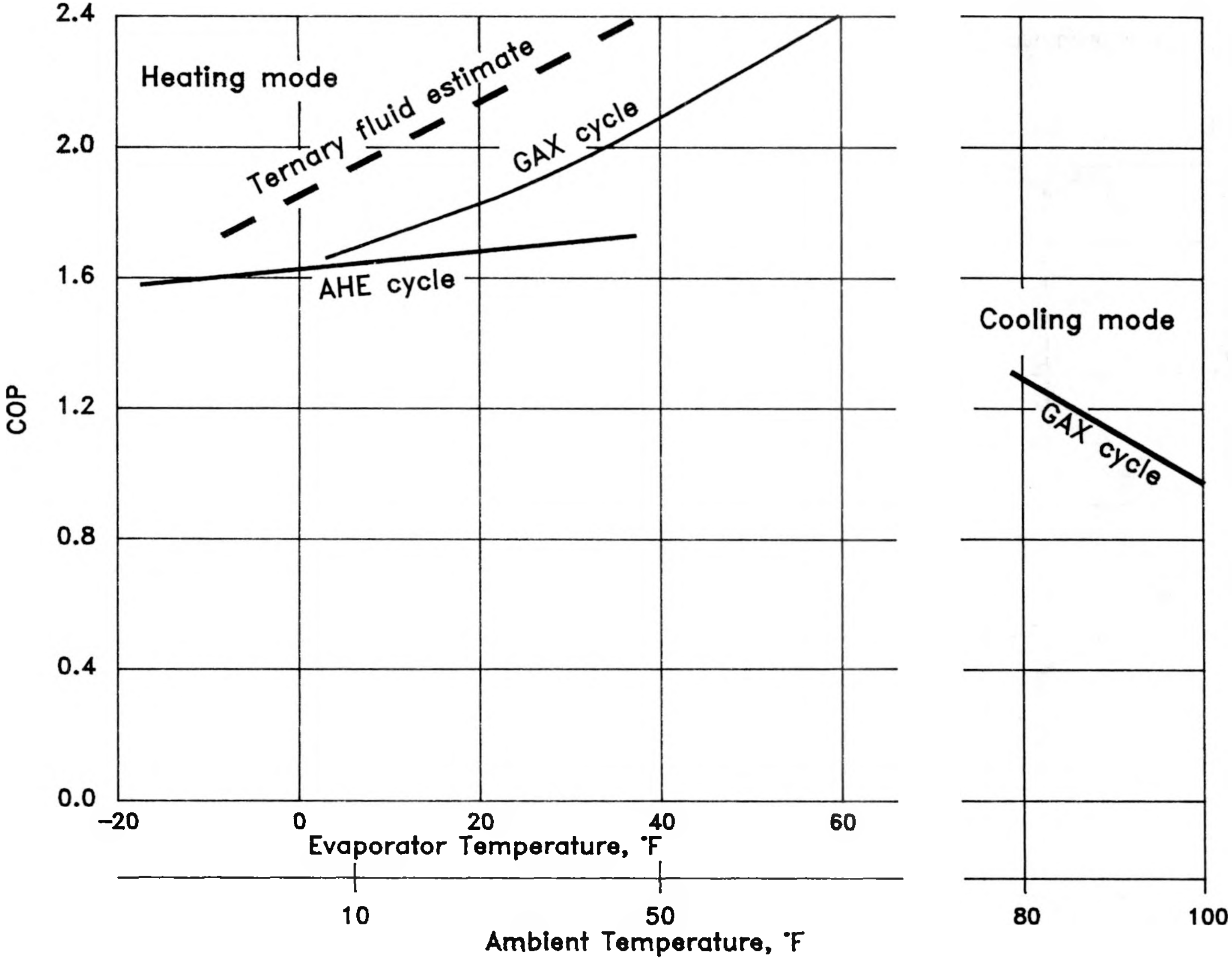

Fig. 25. Estimated coefficient of performance versus ambient temperature for generator-absorber heat-exchange cycle at 500°F peak boiler temperature with ternary fluid.

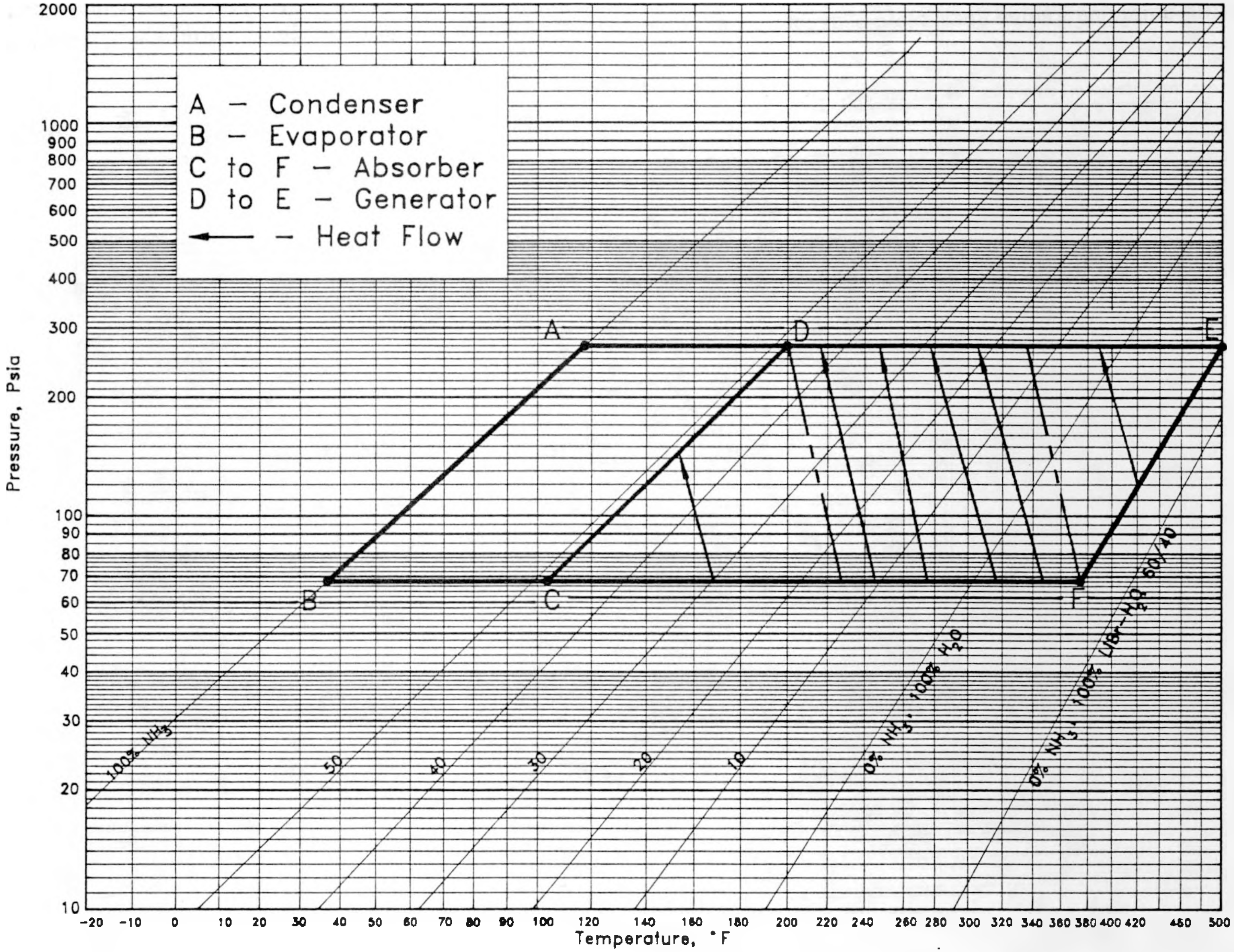

Fig. 26. Pressure-temperature-composition diagram of generator-absorber heat-exchange cycle using ammonia/lithium-bromide-water at 47°F ambient temperature and 500°F peak boiler temperature.

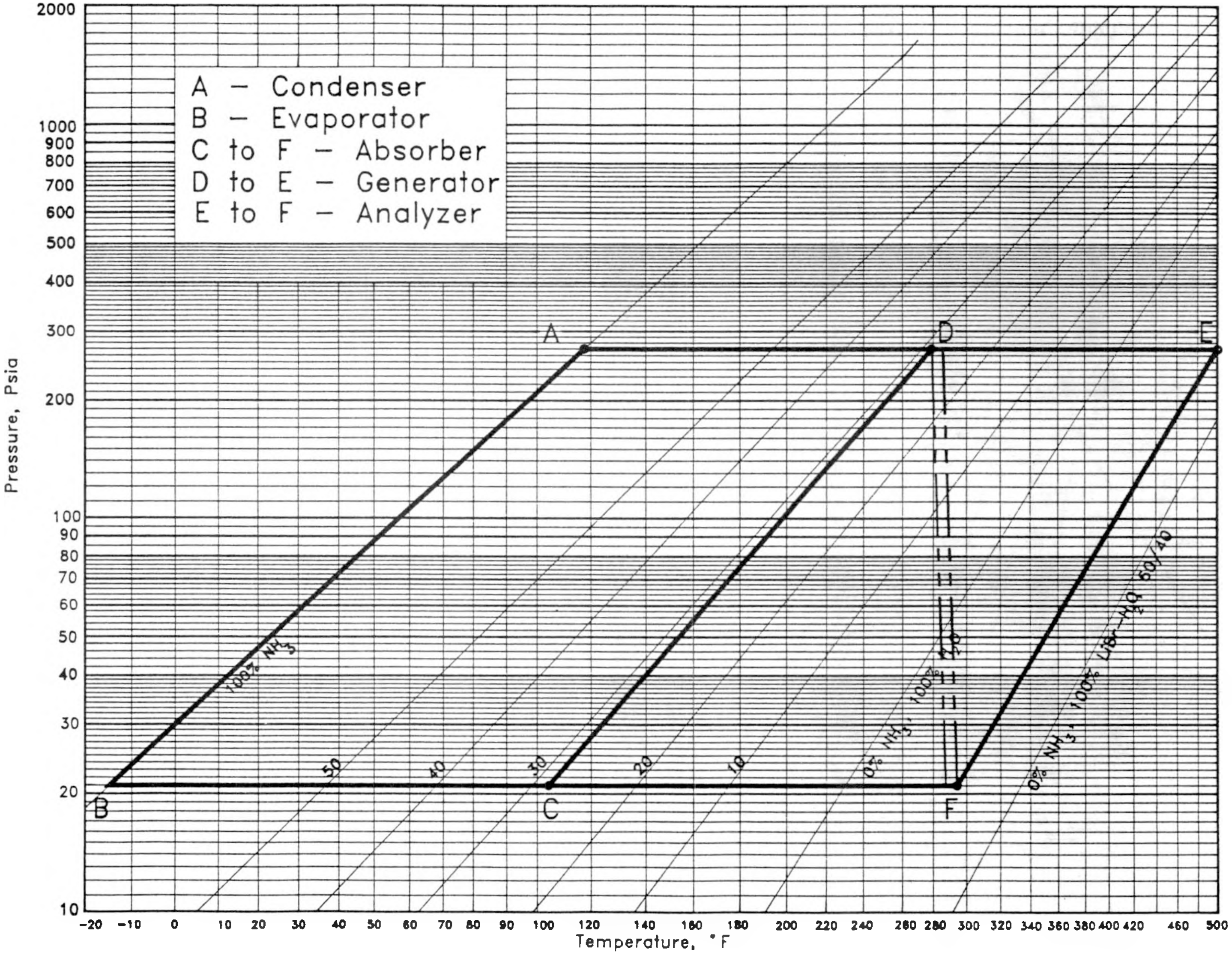

Fig. 27. Pressure-temperature-composition diagram of generator-absorber heat-exchange cycle using ammonia/lithium-bromide–water at –10°F ambient temperature and 500°F peak boiler temperature.

Another possible gain for the GAX cycle concept was also devised. It is based on increasing the GAX heat to be transferred. The heat discharged by the absorber across its overlap range had been found to be less than the heat that the generator could utilize over the same overlap range. The main reason for that inequality is that greater quantities of absorption solution flow at the rich ends of the generator and absorber temperature ranges than at the weak ends. The inequality was indicated in Figure 13, in which the overlap temperature range of the absorber was shown to transfer heat to a shorter section of the generator temperature range.

For the new concept, the cycle was modified to increase the absorber heat output in the overlap range so that it could provide all the heat that the generator could utilize. This modification was done by increasing the flow of absorbent for the overlap range alone—by adding a solution circuit, as shown in Figure 28. In Figure 28, the GAX range of the generator extends from D to I. To provide the amount of heat that DI represents, the flow of absorbent is increased in the absorber section G to F by adding the fluid circuit, FGHEF. The necessary increase in absorbent fluid flow over that circuit is to be achieved by adding a pump to take liquid from the absorber at G to the high-side pressure at H. Heating the liquid from its temperature at G to that at H can be done in various ways, but the addition of a liquid heat exchanger may always be necessary. A heat balance calculation for the enhanced cycle indicated a gain in cycle COP of 17%. That gain should be independent of gains obtained by using improved fluids such as the ternary. The GAX cycle is thus suited to modifications for higher COPs, but probably with added complexity and cost.

3.6 COST ESTIMATE

After the GAX cycle was selected, designs of a 3-RT heat pump were drawn using ammonia and water as the working fluids. The design of the absorption module unit was based on heat and mass transfer data that had been obtained from a 3-RT, AHE-cycle, ammonia/water heat-pump test unit

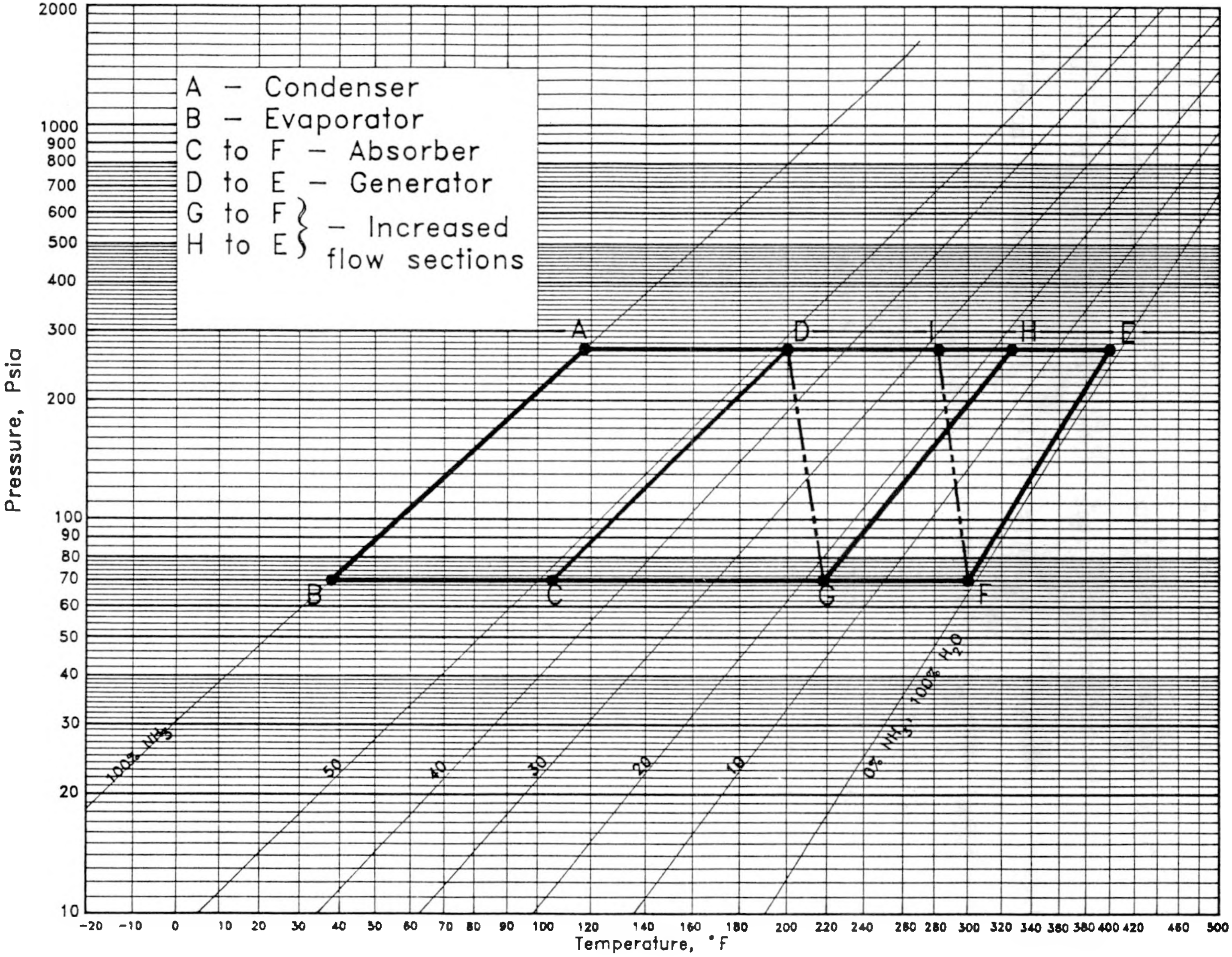

Fig. 28. Pressure-temperature-composition diagram of the modified generator-absorber heat-exchange cycle.

developed and tested independently by Phillips Engineering Company in preceding years. A bill of materials was assembled for the complete heat pump. Component drawings of the absorption unit were made in sufficient detail for costing. The remainder of the heat pump system utilized (1) design details developed and costed independently under Contract 86X-24610C, Organic Fluid Heat Pump Development, and (2) vendor parts such as pumps, motors, finned coil, etc.

Manufacturing cost estimates for the designs were developed by Pioneer Engineering and Manufacturing Company at a production level of 50,000 units per year. Pioneer is a Detroit company that is highly experienced in cost estimating for large manufacturers. They had costed other heat pump developments, including the absorption heat pump developed under the Allied Chemical Organic Fluid Heat Pump program. They estimated base factory costs for the outdoor unit of the GAX system to be $890.00 ± 10% (1985 dollars). That factory cost represented labor, material, factory burden, and scrap.

To those costs would be added costs for warehousing, warranty, shipping, and corporate overheads and profit, to reach the manufacturer's price. Home installation would also require an indoor blower coil, connecting tubing, indoor controls, wiring, etc., as needed for the application. The customer's cost would also include an installation charge plus the dealer's and the distributor's expenses and profits. The installed cost thus estimated at $3200–$3800 for a nominal 3-RT unit appeared competitive with the installed costs of existing heating and cooling systems.

4. CONCLUSIONS AND RECOMMENDATIONS

4.1 CONCLUSIONS

The study has determined that five of the cycles evaluated had cycle COPs in the 1.0 to 1.1 range for refrigeration and in the 2.0 to 2.1 range for heating, both at 47°F outdoor ambient conditions and 395°F peak boiler temperature. Preliminary studies of methods of improving the COPs beyond those obtained under the evaluation conditions were encouraging.

Developing improved working fluids could be an important means of increasing the performance of the cycles. The ammonia/lithium-bromide–water ternary combination can be expected to significantly improve the performance of most of the better cycles. However, corrosion and materials problems should be expected. Other fluids or ternary combinations can probably be developed if the corrosion problems of ammonia/lithium-bromide–water are found to be too severe for normal materials and inhibitors.

Of the five, the GAX cycle was the one best suited for the residential heating and cooling applications. Gas-fired air-to-air absorption heat pumps using the GAX cycle appear to be capable of appreciable energy savings and to be marketable at prices competitive with existing residential heating and cooling equipment. The overall results emphasize that there are a number of directions for developing more efficient residential heating and cooling systems, systems capable of important reductions in annual operating costs and national energy usage.

4.2 RECOMMENDATIONS

Most of the cycles evaluated have major efficiency advantages over past absorption refrigeration cycles at heating and cooling conditions. The GAX cycle has been selected as the most suitable for residential heating and cooling. It is recommended that this cycle be used to develop an advanced absorption heat-pump test unit for residential applications in Phase II, using ammonia/water as the working fluid.

It is also recommended that low-cost, service-free methods of transferring heat from the absorber to the generator be investigated and developed, as well as means of operating the heat pump with a condensing flue.

It is further recommended that the ammonia/lithium-bromide–water ternary combination be investigated. The investigation should correlate the existing ternary data with ammonia/water data, determine the corrosion characteristics with various materials of construction, and measure heat and mass transfer properties under absorber and generator conditions.

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