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John W. Andrews, Edward A. Kush, Philip D. Metz

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ABSTRACT

The use of heat pumps for the utilization of solar energy is studied. Two requirements for a cost-effective system are identified: (1) a special heat pump whose coefficient of performance continues to rise with source temperature over the entire range appropriate for solar assist, and (2) a low-cost collection and storage subsystem able to supply solar energy to the heat pump efficiently at low temperatures. Programs leading to the development of these components are discussed. A solar assisted heat pump system using these components is simulated via a computer, and the results of the simulation are used as the basis for a cost comparison of the proposed system with other solar and conventional systems.

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I. INTRODUCTION

Radiant solar energy is an abundant and environmentally attractive energy source, but it has never been effectively harnessed to the work of mankind because we have been unable to find an energy collecting method that will compete economically with other energy sources.

Fossil energy, solar energy stored by nature, has been also abundant and very inexpensive. The industrial development leading to today's technology has fed on this cheap energy source. But now the end of this convenient energy source is in sight and the upward energy cost spiral has started.

This upward cost spiral makes it possible for other energy sources to enter the energy market and compete. The energy market of tomorrow is not cornered and in many respects a competitive race is now underway between the remaining supplies of oil and gas and their competitors, coal, nuclear and solar.

Today we cannot identify the winner in this cost competition. The cost of both coal and nuclear energy have been driven upward. However, the cost of collecting solar energy has also been dissapointingly high leaving oil and gas still as the primary driving forces in today's energy market.

It is the task of the solar technical community to accelerate our effort to reduce the cost of solar energy collecting systems so that this abundant renewable energy source can enter the market soon, and at a low point in the cost spiral. The oil and gas supplies thus displaced will be saved for their more valuable chemical use and the pressure to develop our coal and nuclear options will be reduced.

Solar assisted heat pump systems have been analyzed before and have been identified as one of the potentially cost-effective solar energy collecting systems. This work will build on these previous analyses and extend consideration into two additional areas.

These new areas are:

A. Special Heat Pumps

The performance characteristics of an electrically driven vapor compression heat pump will not be limited to the characteristics of today's commercially available units. Instead, the potential of the vapor compression cycle will be analyzed to determine the performance characteristics of equipment built to meet the temperature and other special requirements of the solar assisted heat pump energy collecting system. Heat pumps designed to fulfill the requirements of a stand-alone application are not readily adaptable to the solar assist function and the thermodynamic potential of this concept is destroyed by forcing this mismatch. The development and cost-effective production of properly engineered heat pumps are within the state of knowledge of the vapor compression science, and this development has been started by contracts awarded in response to a current RFP requesting such work.

B. Special Collectors

The collector, the system component that intercepts the incoming radiant energy, is the major cost element in a solar energy system. Unless the cost of this component can be reduced there is little hope of attaining cost-effectiveness in the near future. However, through the introduction of a heat pump in the solar collecting system the possibility of major cost reduction in the collector is created. The heat pump becomes the energy distribution vehicle, relieving the collector from temperature requirements imposed by distribution. The collector can now operate over a wide temperature range including the lower temperatures easily developed by simple structures. This new set of low-cost collector options is the major hope for cost-effectiveness. Simple inexpensive structures including passive-like designs need to be examined. The use of shallow ponds for flat-roofed buildings is another cost-effective possibility. A large set of other possibilities exists which extends beyond this work. It is our hope that others will see this potential, ignite the American inventive spirit, and develop a good, long-life low-cost low-temperature solar collector.

II. PROBLEMS WITH PRESENT SOLAR ENERGY SYSTEMS

A. Heating

The major problem with present solar thermal energy systems is that they cost too much. A related problem is that they are not capable of space cooling. We now examine the cost of solar thermal energy systems for residential space heating. Three areas must be investigated. They are: present cost, ultimate cost and the amortization of the capital investment. The discussion is intended to be illustrative, not difinitive. Space cooling and domestic hot water are excluded.

The retail cost of liquid flat plate solar collectors suitable for space heating is presently about \$10 - \$20 per square foot (not installed).[1,2a,3a] Air collectors are cheaper,[3a] but less efficient.[4a] Generally, the best collectors are the most expensive. Installed system costs are about \$30 - \$40/ft^2 of collector area.[2b,3b] Let us see what this leads to in terms of residential system total initial cost. Consider, for example, a 1500 ft^2 house on Long Island (4854 degree days/year[4b]) with a moderate heating demand of 45.8×10^6 Btu/year. Using the method of Balcomb and Hedstrom,[4c] for a solar system to produce 50% of this heat demand requires 278 ft^2 of collector.[4b] The "99% temperature" for Suffolk County AFB is 9^oF.[5a] Thus, the heating load is:

$$\frac{45.8 \times 10^6 \text{ Btu/year}}{4854 \text{ degree days/year}} \times \frac{1 \text{ day}}{24 \text{ hr}} \times (65^oF - 9^oF) = 22016 \frac{\text{Btu}}{\text{hr}} \approx 2 \text{ tons}$$

The Mitre Corporation report gives, with this load and area, an initial cost of \$9517,[2c] or about \$34/ft^2. The cost of providing the same amount of heat via fuel oil at \$.48/gal (60% furnace efficiency) is about \$131/year.[5b] Natural gas would cost about the same. The present <u>savings</u> of this part of the solar system is, by comparison:

$$\frac{\$131/\text{yr}}{278\ \text{ft}^2} = \$.47/\text{ft}^2\ \text{year}.$$

What about amortization? What if we pay for and use the solar system over a very long period of time? Rising fuel prices and monetary inflation will enhance the value of the solar system while its initial (already expended) cost remains fixed. Clearly it is advantageous to amortize the system's cost over as long a period of time as possible. However, this time period is limited by the system's lifetime. A lifetime often used in the literature is 20 years (e.g., ref. 6, 2d). In this connection, A. and M. Meinel say, "...one should be cautious about using a value lifetime of more than 10 years. One reason for this conservatism is that achieving more than a 10-year operating lifetime for solar collectors is complicated by the need to use inexpensive materials. Even with expensive materials there is not proof of lifetime this early in the course of the development of a new energy option."[3c] The lifetimes of present household devices such as plumbing, faucets, and domestic hot water tanks also indicate that a fairer amortization period (for liquid solar systems) is 10 years. This greatly increases the effective cost of the solar system. For example, since most people would have to borrow money to buy the solar system, let us suppose a loan of $9517 at 9% annual interest (which is probably only available via government intervention and/or on new construction). Table II-1 presents a simple method to determine if this solar energy system can compete economically with the cost of oil. For various amortization periods, the annual cost of the mortgage on the solar system is compared to the fuel cost for the first year of the period and the fuel cost for the last year of the period, given various fuel price escalation rates. Roughly speaking, the economic "break even" point occurs if the annual solar system cost is somewhere near the average of the first year and last year fuel costs (more strictly speaking, somewhat below this average). If the solar system annual cost were below even the first year fuel cost, the solar system would already be economical and would grow more so with time. If the solar system annual cost were above even the last year fuel cost, the solar system would never be economical during the amortization period. It can be seen from Table II-1 that the solar system cannot compete economically given a 5, 10, or 15-year amortization period, even given a 15% per year fuel escalation rate. The solar system is marginally competitive given a 20-year amortization period and at least a 15% per year fuel escalation rate. The 20-year lifetime for solar energy systems is based less on fact than on necessity.

What about the future cost? Won't mass production bring prices down? The Mitre report projected substantial "ultimate" cost reductions. This is a controversial subject as it is difficult to assess what will ultimately happen. It is clear, however, that substantial near-term price reductions in conventional solar systems using flat plate collectors are unlikely. A recent article in Solar Engineering Magazine investigated this subject. They reported:

> [Sheldon] Butt [President of the Solar Energy Industries Association] notes that the typical solar collector is relatively material intensive and that these materials have been commonly used in large-scale production for many years. Their prices cannot be expected to change dramatically, he explained. The materials costs may account for about 50% of the cost of the collector, he said.... Some solar manufacturers are already in a position to buy those materials in large quantities....With the delivered costs of collectors accounting for about half of the costs of solar systems, Butt notes that one must look at the remaining costs of other components and the installation costs. Many of the other components....are standard 'off the shelf' products which are fully developed. There is no reason to expect any significant reduction in their cost.[7]

Later in this paper we examine why these cost problems may not persist in a

Table II-1
Simplified Economic Comparison of Solar vs. Oil Heating

Amortization Period (Years)	Annual Escalation Factor (Compounded Annually, %)	Fuel Cost for in First Year	Fuel Cost for in Last Year	Solar System Total Mortgage Payments per Year ($9517 @ 9%/year)
5	8	131	178	2371
	10	131	192	2371
	12	131	206	2371
	15	131	229	2371
10	8	131	262	1447
	10	131	309	1447
	12	131	363	1447
	15	131	461	1447
15	8	131	385	1158
	10	131	497	1158
	12	131	640	1158
	15	131	927	1158
20	8	131	565	1028
	10	131	801	1028
	12	131	1128	1028
	15	131	1864	1028

particular solar assisted heat pump system, and discuss cheap, efficient, low temperature collectors suitable for this system.

B. Cooling

There are various space cooling devices which are under development (i.e., absorption, Rankine, and desiccant) which can be solar driven (and this topic is of considerable interest).[8,2e] Alternatively, an electrically driven heat pump can be used for space cooling. In this case, the solar collectors are not used at all for cooling. In any case, some sort of "heat engine" is necessary for cooling. This adds considerably to the initial cost of a solar system which is already heavily capitalized. One might hope that cooling would be unnecessary in climates with significant heating demand. However, this is not the case. As of 1974, over 50% of all homes in the United States had some form of space cooling.[9] In households with incomes above $25,000 (i.e., where cost is less of a factor), 69% had some sort of space cooling (most of these were centrally air conditioned),[9] and in 1974, over 50% of all new homes were centrally air conditioned.[10]

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III. PROBLEMS WITH PRESENT HEAT PUMPS

The problems associated with present air-to-air heat pumps are performance and reliability. Cost is not included as a problem because it is generally agreed that heating with a heat pump is about as economical (on a life-cycle basis) as with an oil furnace.[1a] This makes it more expensive than natural gas, but cheaper than electrical resistive heating. We consider the reliability and performance problems together, as they are often closely connected. However, we identify two distinct categories of problems of present air-to-air heat pumps. They are: A. Correctable, and B. Intrinsic.

A. Correctable Problems

Correctable problems are mainly of a historical nature and are due to the newness of the industry and to the past availability of cheap electricity. The emphasis has been on cooling and low initial cost.[2a,3a] For example, money can be saved by skimping on heat exchanger size, but only at the expense of decreased efficiency.

Aside from problems with the machines themselves, installation and maintenance inadequacies are still widely discussed.[4] Various chronicles of this history have been written,[5a,6] with statistics such as military base annual heat pump failure rates of up to 30% in 1958[5a] emerging. In a much later (1968-1973) study, annual compressor failure rates of from 3.6% to 23.3% were found.[1b]

Undoubtedly, the economic reality of expensive energy will cause a premium to be placed upon good design and energy efficiency. If a lucrative market develops, a competent service industry is likely to follow, with improved service and lower maintenance costs resulting. However, there are other problems with the air-to-air heat pump as presently used which are inextricably bound up with the device itself, and with the way in which it is presently used. These are discussed below.

B. Intrinsic Problems

Some of these problems are illustrated by Figure III-1, which plots heat pump "Coefficient of Performance" (COP) versus source temperature. Coefficient of Performance can be defined here to be:

$$\text{COP} = \frac{\text{Energy Delivered}}{\text{Electrical Energy Consumed}}$$

For example, electrical resistive heating converts electrical energy into heat energy, but does not move any ambient heat indoors, and therefore has a COP = 1. Clearly then, a high COP is desirable as this indicates that the electrical energy input is small relative to the amount of heat delivered. The upper curve in Figure III-1 is the "Carnot Theoretical COP" for a heat pump operating from the indicated source temperature. A "split", that is the temperature differences between the two fluids in a heat exchange, of 30°F has been assumed for each (air-to-refrigerant) heat exchanger. The heat is delivered (as hot air) at 110°F. These are all typical values for present air-to-air systems. No real heat pump can, even in principle, have a COP higher than this Carnot curve, under the given conditions. The lower band in Figure III-1 is an envelope drawn from actual data for present heat pumps.[3b]

The first thing to note in Figure III-1 is that the COP decreases as the source temperature drops, both in theory, and in the real machines. Thus, a heat pump which must extract energy from a low temperature ambient source must necessarily have a low COP. This limitation is due to the second law of thermodynamics; actual machines incur additional limitations. Besides being economically unpleasant per se, the low COP means that the heat pump capacity

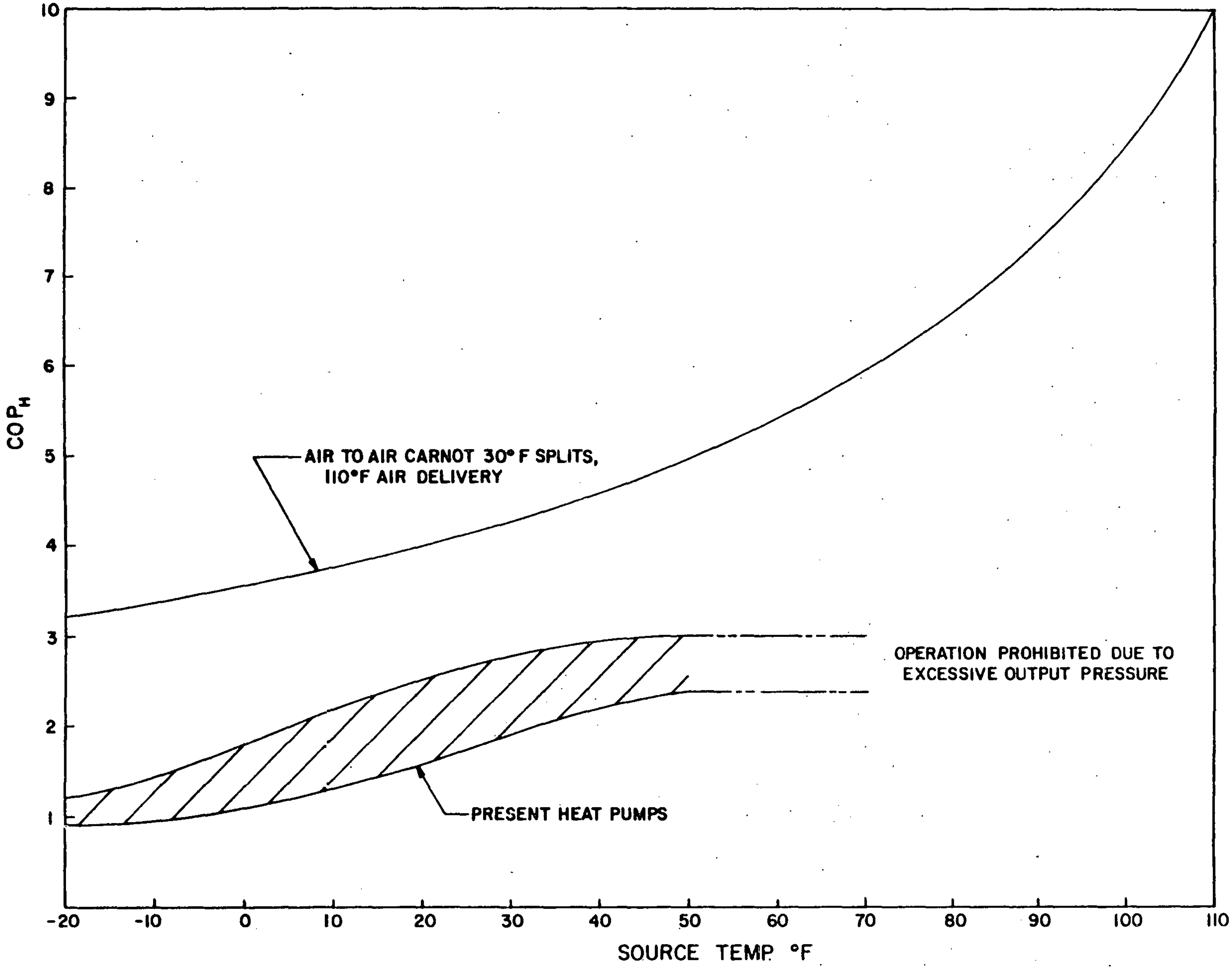

Fig. III-1. Carnot Theoretical vs Present Heat Pump Heating Coefficient of Performance.

drops with the ambient source temperature (for roughly constant electrical input). Of course heating demand <u>increases</u> linearly as the ambient temperature decreases. Therefore, there is a temperature, called the "balance point", below which the heat pump capacity is not adequate to meet the heating demand. This causes two problems. First, auxiliary heat must be supplied. This is usually electric resistive to origin, with a COP of 1, and is therefore very expensive. Secondly, to avoid using the resistive heating, the balance point is kept as low as possible. This is done by increasing the capacity, and hence, the cost of the heat pump. Thus, the heat pump, sized to give a low balance point, has excess capacity most of the time. This fundamental mismatch between capacity and demand has been widely discussed.[2b,2c,5b,6,7,8]

The next interesting thing to notice in Figure III-1 is that the COP of the present heat pumps levels off for temperatures above 40°F. Were it not for the presence of the theoretical Carnot curve, one might suppose that there is some fundamental thermodynamic reason for this behavior. However, it occurs because these heat pumps have been optimized by design to operate best when the ambient source is below 40°F. That is, to obtain a low balance point, and good efficiency (relative to Carnot) below 40°F, present heat pumps sacrifice efficiency above 40°F. This is quite reasonable as this lower temperature region is the critical regime for heating.

For temperatures below 40°F, the COP of present heat pumps can be seen to rise with temperature roughly in parallel with the Carnot COP curve. A heat pump with this property in the temperature region <u>above 40°F</u> would be able to attain an impressive COP as the Carnot COP increases monotonically with temperature at an increasing rate. A real machine can operate with high COP above 40°F , but <u>only if it is not burdened with the requirement of a low balance</u>

point. This is not a very desirable thing for a heat pump with an ambient source, because when the ambient temperature rises, less heat is needed, while the low balance point is essential to avoid resistive heating. So, we seem to have invented a very efficient but useless device - if it must extract heat from the ambient. However, consider the cheap, efficient, low temperature solar collectors alluded to at the end of section II and discussed in sections IV and VII, which can collect solar energy at temperatures of $40^{\circ}F$ to $120^{\circ}F$. These collectors can be used as a source for our very efficient heat pump. In short, present heat pumps, as stand-alone devices at temperatures below $40^{\circ}F$, cannot even in principle attain a high COP. Because of the design concessions made in order to obtain a low balance point, present heat pumps are not able to obtain the high COP's which are possible above $40^{\circ}F$, and therefore are not appropriate for use with a solar energy heat source.

In section VI we discuss the technical design modifications necessary to create heat pumps capable of operating with high COP's at source temperatures above $40^{\circ}F$. The technical reasons for the relatively low COP of present heat pumps, when operating from source temperatures above $40^{\circ}F$, have been discussed.[5c,6,9,10]

The final major problem with present air-to-air heat pumps is the defrost cycle. When the ambient temperature is in the range of $20^{\circ}F$ to $40^{\circ}F$, water in the air freezes to the outdoor coil since the outdoor heat exchanger coil (evaporator) temperature is about $30^{\circ}F$ lower. Frost builds up and eventually blocks the air flow across the heat exchanger, which prevents heat from being transferred. This is an especially severe problem under humid or wet conditions. Performance is hindered. Eventually, the buildup becomes so great that it must be removed. This is accomplished by temporarily reversing the refrigerant flow in order to use heat from indoors to heat the outdoor coil and melt the frost. Alternatively, this heat can be provided by electric resistance. Either method has a very negative effect on performance. During the "defrost cycle", the heat pump is operating (and drawing electrical energy), but is not providing heat to the building. Typically, a heat pump can spend 5% of the time in this mode when the ambient temperature is below $35^{o}F$.[4b] This decrease in performance is in addition to the fundamentally low "instantaneous" COP displayed in Figure III-1 for this temperature region.

The reverse cycle defrost technique causes severe reliability problems. Upon defrost, the superheated refrigerant (gas) from the indoor coil (now the evaporator) flows through the accumulator, which is a storage device for liquid refrigerant, whose need is discussed below. This boils the refrigerant in the accumulator and also boils the refrigerant saturated in the lubricating oil in the oil sump. This causes "foaming" which drives off the oil with the refrigerant. When this occurs, the compressor is inadequately lubricated. "This is a most dangerous running condition and can persist for some minutes at each defrost cycle."[3c] The EPRI epic also concluded, "...there are only two major externally induced causes for compressor failure: inadequate lubrication and inadequate cooling."[3d]

The defrost process proceeds slowly because the cold outdoor temperature produces a low heat pressure. This means that, "...only a trickle of liquid refrigerant flows back into the indoor coil to pick up defrosting heat...".[5d] So, during defrost, the low ambient temperature causes the liquid phase of the refrigerant liquid - gas phase equilibrium to be favored in the outdoor coil, and liquid refrigerant collects there. At the termination of defrost (beginning of the heating cycle), the refrigerant flow is reversed, and this liquid is sucked back toward the compressor.

The presence of an incompressible liquid in the compressor cylinder can cause tremendous mechanical shock and also break valves. To prevent this, the "accumulator" is installed to catch this "flood back" of liquid before it can reach the compressor. Although the accumulator ameliorates this problem, its presence in the line during the defrost cycle leads to the additional problem of foaming, as discussed above. The reverse cycle defrost technique has been widely identified as the leading scourge of present air-to-air heat pump reliability.[3e,5e] The EPRI study concluded, "The most positive means of improving compressor reliability is the adoption of an alternative method of defrost..."[3f] We note that the defrost cycle can be eliminated entirely by using a liquid source,[2d] and by not allowing the source temperature to drop below 40°F, i.e., by solar assist.

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IV. IDENTIFICATION OF PROPOSED SYSTEM

A. System Arrangement

There are two major design options for the solar assisted heat pump system. They are the "parallel" system and the "series" system.

The parallel system is shown schematically in Figure IV-1a. It contains the usual solar energy heating system (section II), and an ambient source heat pump for space heating and cooling (section III). The solar collectors provide heat energy which is placed in the storage device. If the storage is at a high enough temperature, it is used to heat the load. If not, the ambient source heat pump is used. When the heat pump is also inadequate, electrical resistive heating is used. This system can be viewed as a solar energy system which happens to use a heat pump for auxiliary heating instead of gas or oil. It is subject to the unpleasant economic realities of the solar system, and also to the performance and reliability difficulties inherent in ambient source heat pumps. As both components function best during warm, sunny weather (when heating demand is lightest), and worst during cold or cloudy weather (when demand is highest), they do not complement each other - indeed they are often redundant or inadequate. Additionally, expensive electrical resistive heating must often be used.

The series system is shown in Figure IV-1b. Here the solar energy is provided to the storage device which heats the load when possible, as in the parallel system. When this is not adequate, the heat pump removes heat from storage and delivers it to the load. The series system can have better performance - cost characteristics than the parallel system or the ordinary solar heating system of section II. There are two reasons for this.

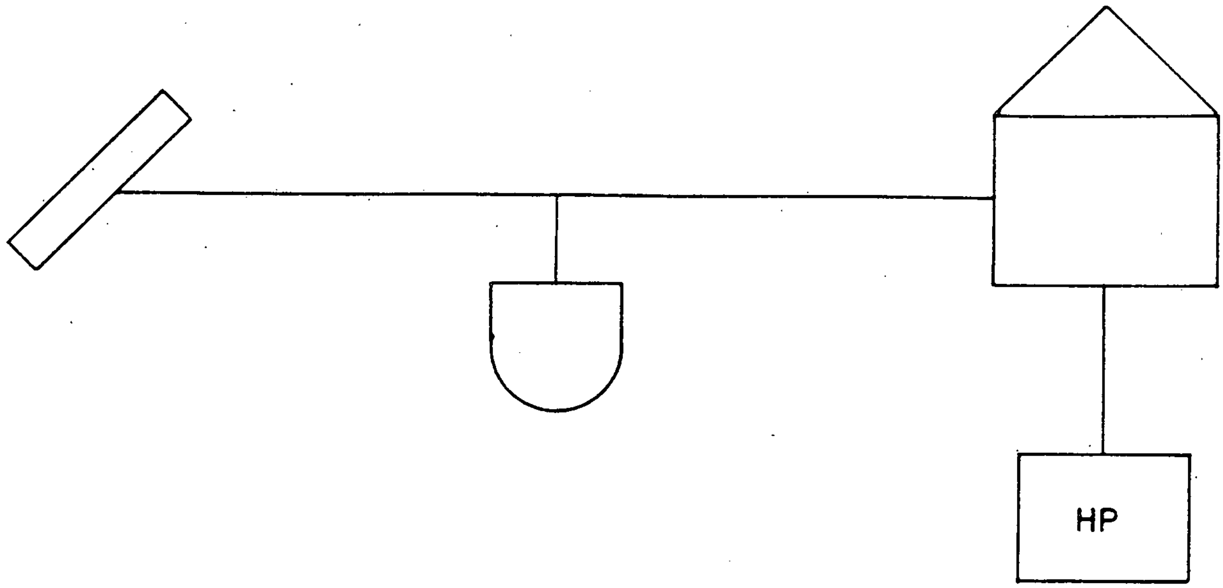

Fig. IV-1a. The Parallel System.

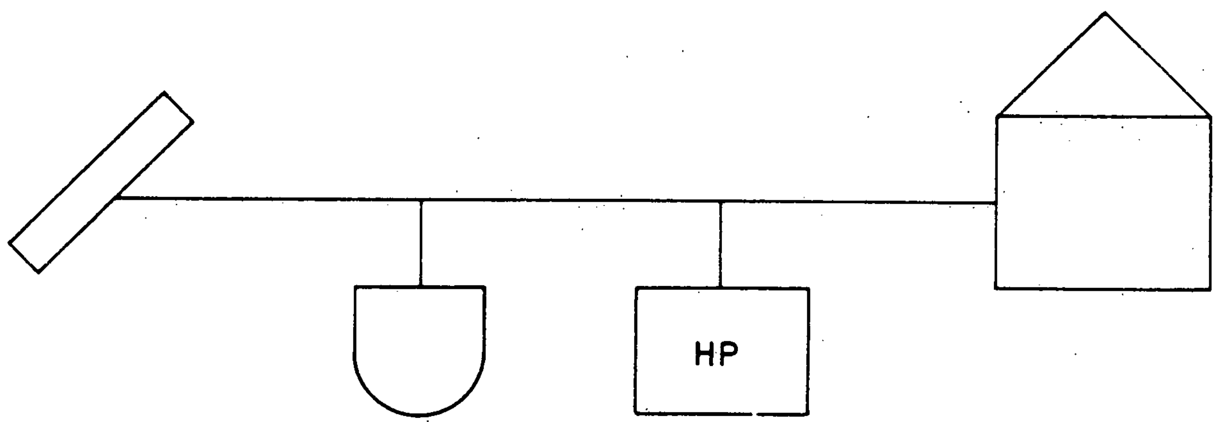

Fig. IV-1b. The Series System.

1. Cheaper collectors can be used. The dominant economic factor in all solar energy systems is the cost of buying and installing the solar collectors. In the series arrangement, the collectors do not have to deliver heat at a high enough temperature to carry the load to be useful. Relieved of this distribution requirement, cheaper collectors, which would be inadequate in the parallel system can be used.

2. The system efficiency can be greatly increased. If the storage temperature can be maintained above the ambient temperature, the heat pump can operate with a higher temperature source and at a correspondingly higher efficiency. A secondary benefit is that the solar collectors also operate at a higher efficiency. This is because the heat pump removes heat from storage which lowers the storage temperature. The fluid (air or liquid) which is then circulated through the solar collectors is thus also at a lower temperature. All collectors have the property that their efficiency <u>increases</u> as the average circulating fluid temperature <u>decreases</u> for given ambient temperature and solar insolation (see e.g., ref. 1). Storage losses are also reduced because of the lowered storage temperature.

The series-system cannot achieve lower cost and higher performance than other solar energy systems unconditionally. Two conditions are essential for the success of this system.

1. The collector-storage size must be adequate. In the parallel arrangement, the solar system size can be arbitrarily chosen to provide any desired fraction of the heating load, due to its independence from the heat pump. In the series arrangement, however, the heat pump relies upon the storage as its energy source. If the storage is depleted, electrical resistive heating must be used at a great expense. Thus, in the series arrangement, for a given load,

the solar collector has a minimum "critical size". It is obvious that the storage can be kept from depletion if the solar collector area is very large. The crucial question is: Can this condition be met at a reasonable solar system size and cost? A detailed discussion of the economics of this system compared to other solar and conventional heating systems appears in section XI and leads to the conclusion that the size condition can be met economically.

2. The heat pump must be specially designed for this system. In order to exploit the higher temperature source of the series system, a heat pump is needed whose efficiency increases significantly with source temperature above $40^{\circ}F$ as permitted by the Second Law of Thermodynamics. Existing heat pumps, as discussed in section III, do not have this property. Appropriate heat pumps are being developed, however, and are discussed below.

B. Optimized Solar Assisted Heat Pump System

The rest of this section concerns the design of components suitable for the series system, the operational modes of this system, and some options which appear to be attractive.

1. Collector

The energy collection system is pictured in block 1 of Figure IV-2. Since the burden of high temperature operation is removed, heretofore inadequate, but cheap, collectors may now be used. (These are surveyed in section VII.) Insulation is reduced. Single glazing is adequate. Nevertheless, more energy can be collected because the heat pump enables us to economically use this low temperature energy that is useless in the stand-alone solar system. Air collectors are used as they can be made much cheaper than liquid collectors,[2] mainly because they can be much less material intensive. They are also simpler,[3] which opens the door to on-site assembly. This eliminates the large cost markup associated with the marketing of prefabricated collectors. Handling costs

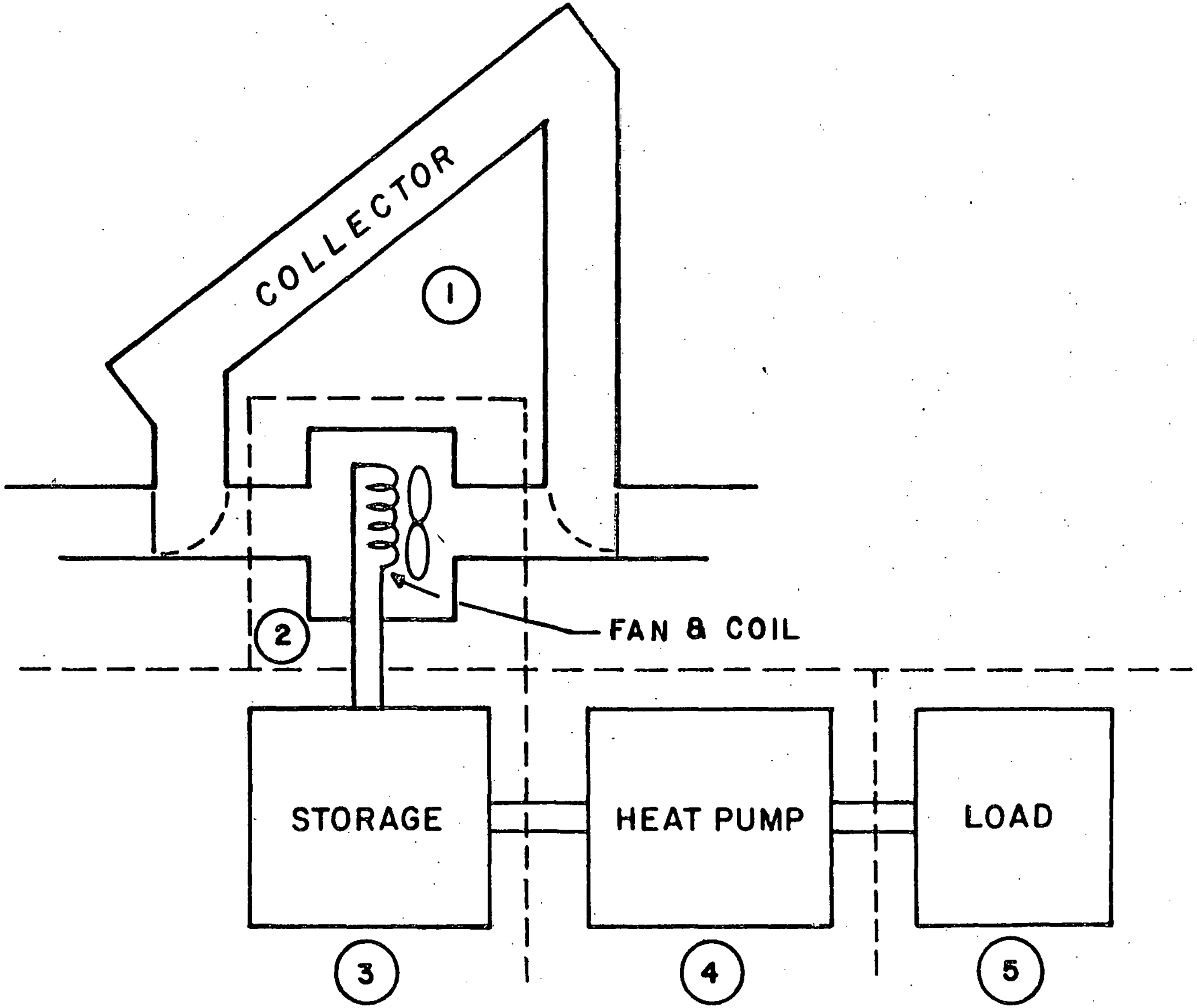

Fig. IV-2. Block Diagram of the Proposed System.

of these heavy, bulky, fragile items are reduced. Simplicity also helps to keep down the on-site labor costs. Earlier work has addressed the issue of site-assembled collectors in more detail.[4,5] Air collectors do not have to be protected against freezing, do not contain antifreeze (some of which is toxic, and some of which if leaked dissolves roof covering), and can be made much lighter than liquid collectors. This reduces the necessary weight-bearing capacity (and cost) of the structure and collector support assembly.

On the other hand, high temperature operation prevents stand alone air collector systems from accruing most of the cost reduction benefits discussed above [and also forces higher collector areas compared to liquid collectors (see e.g., ref. 6)]. It is difficult to simulataneously achieve adequate efficiency and high temperature with air collectors.[7]

2. Air-to-Water Exchanger

Block 2 of Figure IV-2 shows that an air-to-water exchanger has been used to heat water with the collected energy. Corrosion protection may be needed, but antifreeze is unnecessary since freezing is not possible with the entire water loop enclosed by the shell of the heated building. It is much cheaper to use and to insulate water pipes than air ducts (mainly because of the size difference). Since the heat pump now has a water source, its performance characteristics are improved as lower heat exchanger temperature "splits" can be obtained with a water-refrigerant heat exchanger than with an air-refrigerant exchanger. This raises the heat pump evaporator temperature which raises its efficiency.

3. Storage

Energy is stored as heated water. The expected temperature range is $40^{o}F$ - $120^{o}F$. The upper limit is about the highest temperature heat that the inexpensive collectors are expected to deliver efficiently during the heating season. Higher temperatures may be reached under warm ambient conditions. The lower limit arises from efficiency and reliability considerations. If the storage can be maintained above 40°F, important benefits arise:

a. The heat pump efficiency is increased over an ambient source heat pump due to the elevated source temperature.

b. The defrost cycle is completely eliminated. Recall that this is the greatest reliability problem of ambient-source heat pumps (section III).

c. The heat pump capacity is increased, or its cost can be reduced.

d. The need for electrical resistance heating is removed. This results in lower electrical costs to the owner. It also means that the electric utility is faced with a greatly reduced peak load. That is, in other solar systems (and in "conventional" heat pump systems) the heating method of last resort is usually electrical resistors. The utility must provide generating capacity for the occasions when these solar systems are inadequate, although this capacity is ordinarily unused. It can be seen that the necessary standby capacity is much lower for this system where the heat pump can operate from a 40°F source than for other systems where resistive heating must be used. The utility customer ultimately pays for any such excess capacity.

Storage at these moderate temperatures (40°F - 120°F) permits new cheaper storage options. Underground storage becomes feasible. This means that the storage does not occupy space in the dwelling. It also opens up the possibility of thermal coupling to the ground. In an ordinary solar system, this is not a good idea as the ground temperature is about $55^{\circ}F$ while the storage must be maintained at $100^{\circ}F$ - $180^{\circ}F$, so that any ground thermal coupling must be considered a loss. In the series heat pump system, however, the storage is expected to be driven down to about $40^{\circ}F$. When its temperature drops below the ground temperature, heat flows from the ground to the storage! Thus, the ground acts like a "buffer" and delivers heat to the storage when it is needed most. It is also possible to place heat in storage all summer, since the heat flows into the ground for retrieval when needed. This can reduce the collector area significantly.

The storage device can alternatively be used to increase the cooling-season heat pump efficiency. This is done by lowering the storage temperature via heat rejection to the nocturnal ambient. Then during the day, the heat pump uses the relatively cool storage as a heat sink. This enhances the value, and hence eases the amortization of the total solar system.

4. Heat Pump

The heat pump in this system must be specially designed to operate optimally with a source temperature above $40^{\circ}F$. As discussed in section III, present heat pumps are not satisfactory for this task because they have been designed to attain a low balance point, and consequently do not achieve high efficiency in the desired temperature range. However, contracts have already been let by the U. S. Department of Energy to develop suitable heat pumps for use in the series configuration. Hardware is expected within two years, at costs slightly above present units. The key idea is that these heat pumps will take previously unusuable energy from heretofore unsuitable, but cheap, collectors and deliver it at high efficiency as a result of the high efficiencies available with these source temperatures. This saves energy and money, and removes most present major heat pump problems. For design details, see section VI.

5. Distribution System

The heat delivery system (block 5 of Figure IV-2) uses a water loop with fan-coil units. This type of system is presently common in commercial, but not in residential, buildings. It has several advantages here. Water on both sides of the heat pump increases efficiency because water-refrigerant heat exchangers have lower temperature "splits" than air-refrigerant exchangers. Also, one heat exchanger can be optimized as an evaporator, and one as a condenser. Then, instead of reversing the refrigerant flow for cooling, the water loops are switched.

The water loop facilitates "zoning" - not heating unused rooms. Due to the expense of insulating and closing air ducts, and the need for a constant air flow across the heat pump condenser (if the delivery system is air), zoning is difficult in air delivery systems. In this way, the heating and cooling load can often be significantly reduced (e.g., by only heating sleeping areas at night) with no discomfort.

6. Major Operational Modes

a. Heating

When possible, heating is accomplished by circulating the stored hot water through the fan coil units. Failing this, the heat pump operates. It removes heat from storage and delivers it to the fan coil units. This is the dominant mode of operation. In principle, it is possible to use the ambient as a heat source. It is found, though, that the storage of this properly designed system is almost always at a higher temperature than the ambient and can be kept above $40^{o}F$. Very little resistive heating is necessary.

b. Cooling

The heat pump rejects heat to storage. In turn, by rejecting heat to the ambient at night, the storage temperature can be kept below the diurnal ambient temperature peaks. This increases the heat pump efficiency.

Note: The above operational modes and component size and performance are studied in a computer simulation in section X.

7. Other Options

a. Forced Air Distribution System

In some applications, the water loop distribution system may not be desirable. In this event, a forced air distribution system will be used.

b. Domestic Hot Water Preheating

This is a desirable addition. It allows for year-round use of the collectors. During the warm months, the collector can be used to heat hot water directly. In winter, the heat pump can be used to preheat hot water.

c. Storage Options

i. Ground thermal coupling (see discussion above and section IX)

ii. Two-component storage (see section IX)

d. Direct Connection to Collectors

It may sometimes be advantageous to connect the load or heat pump directly to the collectors. This could occur when the storage is at a low temperature, but solar energy is available to be collected. Due to its large "thermal inertia", much energy must be placed in storage to raise its temperature significantly. Instead of doing this, the collectors can be used directly to heat the load or as a source for the heat pump. Study is needed to determine if this is a desirable option, i.e., if this situation occurs enough of the time to be included as a control strategy.

e. Dessicant Dehumidification

Space cooling generally requires dehumidification, and this means that the evaporator temperature must be kept at about 45°F. If dehumidification can be accomplished by a dessicant (perhaps recharged by the collectors), the evaporator temperature can be raised, and the heat pump efficiency increased. Dehumidification also raises the comfortable "dry bulb" temperature, which lessens the space cooling demand.

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V. LITERATURE REVIEW

The series system occupies a curious position in the solar assisted heat pump literature. Some of its advantages have been widely appreciated, but it usually has not been identified as superior in either performance or cost when compared to the parallel system. Here, this situation is examined. Attention is focused on those papers that have judged the series system inferior, with special emphasis on the assumptions and analyses which led to this conclusion. It is found that there are two pervasive reasons for the above situation. First, analyses have used hardware components in the series system which are not appropriate for it. Secondly, they have failed to optimize this system in regard to component size. These factors have driven the cost-performance characteristics of the series system below those of the parallel system.

A. Components

1. Collectors

As explained in section IV, in the series system the burden of high temperature operation is removed from the solar collectors. This permits the use of much cheaper collectors (discussed in section VII). This possibility has not been widely appreciated. For example, Asbury and Mueller said, "...the break-even point for the solar collector component of the solar-assisted heat pump system will be lower than that of the solar/electric resistance heating system by the amount of the added capital cost of the heat pump."[1] The MITRE report said, "Costs for the solar assisted heat pump system will be the same as those reported for the solar heating and hot water system except that the auxiliary heating system costs will be replaced by the cost of the heat pump."[2a] Evidently, no allowance has been made for cheaper collectors in the heat pump systems relative to the ordinary solar systems in these papers.

This is a major oversight.

The situation is slightly improved in the series vs parallel analysis papers. Dubin[3a] suggested unspecified cost savings for the collectors in the series system. Although the Arthur D. Little-EPRI study indicated that the series system has "more potential for low performance, low cost collectors,"[4a] the only cost allowance actually specifically made was the use of a one-pane (rather than a two-pane) collector.[4b] Unfortunately, this is the usual situation in the literature (see e.g., 3b, 4c, 5a, 6a, 7a, 8a). The resulting collector cost reduction is very minor - $.50 per square foot (collectors cost about $10 - $20 per square foot uninstalled[2b,9,10]).

Most of the works referenced in the above paragraph concluded that the parallel system was superior to the series system economically.[4d,5b,8b,6b] Cheap collectors unsuitable for other solar systems were not used. Due to the domination of collector related costs upon solar system economics, the potential for system cost reduction via significantly cheaper collectors can hardly be overstated. In section XI, the economic behavior of the series system is reevaluated using the appropriate cheap "low efficiency" collectors. The series system is found to be less expensive than the parallel system.

2. Heat Pump

The increased heat pump efficiency in the series configuration has been noted by some authors. However, this gain has been limited by most authors[3b,3c,4e,5c,7b,8c,11] to the very moderate efficiency increases resulting when high source temperatures are used with existing heat pumps. This is understandable as heat pumps especially designed for the solar assist function do not yet exist (contracts to develop them have been let and they are expected within two years). The use of "ordinary" heat pumps (sized and otpimized for a low balance point), while appropriate in the parallel system, severely penalizes the series system. Since the COP of these units rises little as the source temperature rises above 40°F (see section III for discussion), little advantage is taken of the elevated source temperatures of the series system. As a result, series systems with such heat pumps have efficiencies only slightly better than other systems with ambient source heat pumps. This stands in contrast to the greatly elevated COP expected from series systems with heat pumps designed for these systems. Although "high efficiency" heat pumps (more efficient than standard units, but with similar COP vs temperature characteristics) are discussed in the literature, no mention is made of heat pumps especially designed for solar assist - i.e., with COP vs temperature characteristics suitable for this function. These units are essential for the optimized series system.

It has been recognized that heat pump reliability is favorably affected in the series system.[4b] For example, the Westinghouse-EPRI study found that the series system "...has the greatest potential of the four systems [thermal storage, series solar, parallel solar, hybrid (series/parallel) solar] for improving heat pump operating conditions by the elimination of extreme low temperature operation and also the need for defrost. These characteristics favorably impact seven of the top eight heat pump component-related reliability problems."[5d]

3. Storage

Storage temperatures (40°F - 120°F) are reduced[6c] compared to the parallel system (100°F - 180°F). These moderate storage temperatures cause several benefits which have been appreciated to various extents. Collector efficiency is increased[3d,11] (see sections IV, VII, X), and consequently more energy is collected.[6d,11] This enhances the value of the collectors. A favorable interaction between heat pump and collector, where both gain in efficiency, has been noted.[3a] Finally, due to the lower temperature range, cheaper storage options are available (see e.g., ref. 12). As a simple example, it is clear that heavy insulation is less cost effective for lower storage temperatures. Storage options are discussed in sections VIII and IX.

In contrast, the series versus parallel analyses in the literature have assessed both systems with identical storage costs (see e.g., refs. 4f, 5e, 6a, 7c, 8d). This has unnecessarily penalized the series system economically.

It can be seen from the above that the norm in the solar assisted heat pump literature has been to use components which are appropriate for the parallel system in the series system. The major result of this has been a severe negative impact on the economic feasibility of the series system. Upon using correct components, the economic picture brightens markedly.

B. System Optimization

It was seen in section IV that it is very important to optimize the series system with regard to component size and control strategy. If this is not done, the cost-performance of the system is severely undermined. Here, the extent to which optimization has been carried out in the literature is examined. Light is shed on why the series system has been judged to have inferior thermal and economic performance and on some of the characteristics of the optimum series system.

The Arthur D. Little-EPRI study found "...a slight economic performance disadvantage for the solar-assisted [i.e., series] configuration [versus the parallel configuration]."[4d] What were the assumptions and analyses which led to this conclusion? First, "...comprehensive system concept matrices were prepared...to fully explore the arrangements that should be considered for experiments..."[48] However, "system optimizations were not performed in this phase but baseline systems were defined...."[4h] The series system evidently was not optimized. The parallel system was chosen as the baseline system because "for the purpose of the large number of calculations to be performed..., we have taken the baseline system to be the simpler parallel configuration".[4i] The upshot was that, "...these conclusions...are still not definitive enough to resolve the issue in a clear cut fashion."[4i] Indeed, in view of the lack of optimization of the series system, the economic advantage projected for the parallel system must be viewed as precarious.

A similar situation exists in the Westinghouse-EPRI Investigation. Their conclusion was: "Of three generic solar assisted heat pumps studied [series, parallel, and hybrid (2 evaporator heat pumps)], the 'parallel' system...appears clearly best even though program limits did not permit comparison of optimized species."[5f]

What factors contributed to the above conclusion? First, recall from part A above that the same components (appropriate for the parallel system) were used in both the series and parallel systems.[5g] Next, note that in lieu of a determination of optimized component size, the components must be sized in some other way. The method used was: "Solar-assisted systems with a seasonal solar contribution of approximately 40% were targeted as desirable."[5h] In the parallel system it is definitely reasonable to arbitrarily "target" the solar contribution to be almost any chosen fraction of the load because the collectors are independent of the heat pump. The solar fraction of the load is usually chosen to be small (<50%) due to the great cost of the solar system (see section II) and also to its diminishing returns as collector area increases.[13] Thus, the parallel system investigated was fairly well optimized with regard to component size.

It does not follow that the series system is optimized at this same collector area. The evidence is strongly to the contrary. The Westinghouse-EPRI computer simulation found that: "It is reasonable to conclude that the available heat from the collector is insufficient and the heat pump is essentially starved for heat....The picture would be changed if the heat supplied by the solar collectors was increased."[5i] Furthermore, they found: "The parallel system was shown to have essentially the same performance as the hybrid configuration (where the heat pump can choose between the ambient and storage for a source)...[5b] This stems from the fact that the solar heat available to the system is very limited and is being fully utilized by the parallel configuration."[5i] Clearly, the collector area in these simulations was inadequate except for the parallel system where the heat pump does not depend on the solar energy and, therefore, where any collector area is allowable. Due to this collector inadequacy, the series system storage temperature was driven down until "heat pump starvation" occurred whereupon it was necessary to use expensive resistive heating. Thus, it was found that "...the series system was least effective in reducing electrical energy consumption."[5d] Even more curious pathologies due to "starvation" have been found by other authors. For example; Cassel, Lorsch and Lior said, "A high - COP solar heat pump requires a larger solar collector or more resistance heating than a low - COP unit, and it may actually consume more purchased energy than a low - COP unit."[3e] This is the height of irony - a high efficiency heat pump resulting in greater electrical usage. Similar results were found in a computer simulation of a commercial building by the Syracuse Group.[14] Using quite small collector areas (less than 10% of building area), they found that increasing the heat pump COP by 1 resulted in lowered heat pump electricity consumption, but also caused almost compensating increased resistive electrical consumption. Starvation was clearly the cause of all these anomalies, as cooling electricity consumption dropped drastically when the heat pump COP was increased. It seems quite likely that the poor performance of the series system observed in the computer simulation of the Westinghouse-EPRI Investigation was not due to any intrinsic defect in the series system, but rather to a poor choice of collector area. It only remains to be shown that given adequate collector area, the electrical consumption of the series system can be reduced to an acceptable level.

There are two reasons why high electrical consumption has been projected for the series system: (1) high electrical resistive use due to depletion of stored energy and (2) overall high electrical consumption because the lower storage temperature permits less direct heating from storage. The computer simulation in section X indicates that resistive auxiliary heating is reduced practically to zero once the collector area is large enough. The economic analysis of section XI indicates that the required collector area is economically feasible. These two results dramatically change the solar heat pump picture. The experimental results of the Phoenix House of Colorado Springs also indicate that resistive auxiliary is negligible in a properly sized series system. This house has a collector area which is 35.6% of the house floor area.[15] In January, 1976, the solar heat pump system delivered a total of 16,369,000 Btu[16a] (4796 KWh) to the load. The electrical resistive consumption during this time was only 215.5 KWh,[16a] or about 4.5% of the total load. Note that January is typically the coldest month of the year, averaging 1128 degree-days vs a yearly total of 6423 degree-days in this location.[17] Additionally, this system used <u>no</u> resistive heating for the rest of the heating season from February through June 1976.[16b]

Thus, over the entire heating season, the fraction of energy provided by expensive electrical resistive heating was very small. It is difficult to compare different solar heat pump papers as each uses a different climate and load. However, the Phoenix House experiment clearly indicates that it is possible to design a series system with a reasonable collector area which does not suffer from heat pump starvation or require significant electrical resistance heating.

The resolution of the question of overall high electrical consumption due to less direct heating in the series system is twofold. First, once the collector area is large enough to prevent "heat pump starvation", one would expect a series system using a conventional low COP heat pump to have electricity consumption roughly equivalent to that of a parallel system with the same collector area. This is because, although the series system can use direct heating less, the heat pump in the series system operates at a higher COP due to its higher source temperature. Note that the parallel system heat pump operates at an even lower COP than a stand-alone heat pump because it operates only when the solar system is inadequate - i.e., when the weather is coldest. This expectation is verified by the work of the Madison, Wisconsin Group. In one paper, Karman, Freeman, and Mitchell evaluated the performance of a series and of a dual source, solar heat pump for a residential load via a computer simulation for various climates. They did not arbitrarily pick a collector area, but rather plotted F, which is "the percentage of the load carried by 'conventional' fuels"[7d] (in the absence of any furnace or hot water heater F is just the heat pump electrical input divided by the total heating load), versus collector area. This is a very informative number as it indicates how much energy must be <u>purchased</u> given a particular collector area. Generally, F decreases as collector area increases. In all cases studied, F for the series system declined to a value very similar to that of the dual source system as collector area increased. In the Madison, Wisconsin (7863 degree-days)[13] this occurred at a collector area of about 25% of the building area.[7e] In the Albuquerque (4348 degree-days)[13] simulation, this occurred at a collector area of about 10-15% of building area. In Charleston (2033 degree-days)[13] this happened at less than 10% of the building area. In all cases, this series system had a water storage system of about 8.7 gal/ft^2 of the collector area.[7f] The performance of this systen can be correlated to a parallel system by noting that a later paper by Mitchell, Freeman, and Beckman compared the dual source system performance to a parallel system. Both systems had almost identical performance at all collector areas presented.[6e] This agrees with the conclusion cited earlier from the Westinghouse-EPRI investigation.[5b] It follows, then, that according to those simulations, the series system, using a conventional low COP heat pump, requires about the same electrical input as the parallel system once the collector areas are as large as those indicated above.

The second important step toward decreased electrical consumption in the series system is the use of a heat pump designed to take advantage of the high source temperatures available. Given a collector area sufficient to prevent "starvation", this heat pump will operate at a very high COP. The thermal and economic performance of a series system using a heat pump with the characteristics expected for these units is evaluated in sections X and XI, respectively.

To summarize this section, the advantages of the series system has not been widely appreciated in the literature because components and component sizing appropriate for this system have not been used, leading to undermined thermal and economic performance. Given a heat pump specially designed to capitalize on the higher source temperatures, thus having a very high COP, the series system COP will also be very high, if the collector area is adequate to prevent heat pump starvation. This critical size is economically feasible because significantly cheaper collectors can be used.

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VI. THERMODYNAMICS AND PERFORMANCE

A. The Vapor Cycle

The vapor cycle in a refrigerator/heat pump is a reversed heat engine cycle in that it transfers heat from a lower temperature reservoir to one of higher temperature, and it, therefore, requires work input by the Second Law of Thermodynamics. The work is input by a compression process, and if this work can be minimized for a given heating or cooling load, the device's efficiency and energy conservation are maximized. Solar energy can be used to effect this work reduction in a heat pump.

Thermodynamically, the distinction between a refrigerator and heat pump vapor cycle is arbitrary, and the function depends on whether the conditioned space is being cooled or heated. A heat pump can provide either function by re-routing the flow of cycle fluid. A reversed heat engine cycle is shown in Figure VI-1. Heat energy, Q_2, is transferred from the lower temperature reservoir,

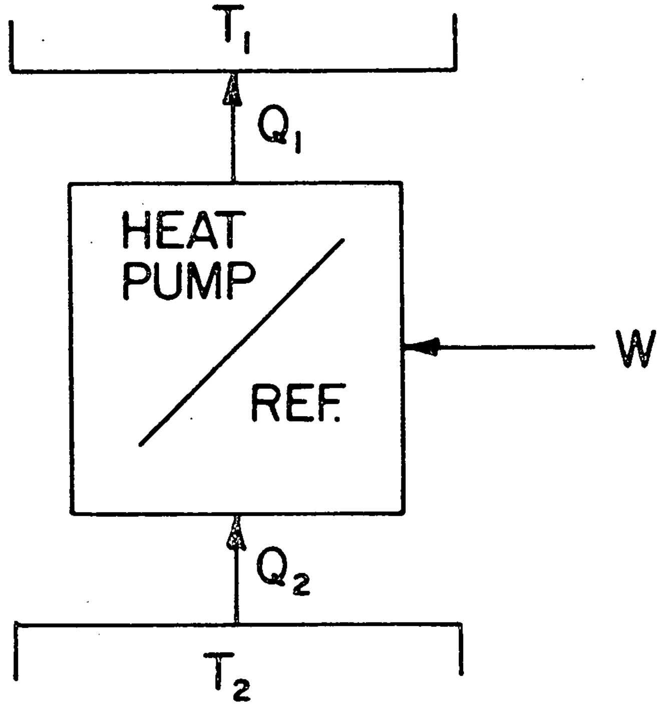

Fig. VI-1. Reversed Heat Engine

to the vapor cycle device and thence to the high temperature reservoir together with the heat equivalent of the work input W. By the First Law

$$Q_1 = Q_2 + W \tag{1}$$

If the application of the system is heating, then Q_1 is the useful heat quantity and the Coefficient of Performance (COP) for heating, the measure of efficiency, is

$$(COP)_{heating} = \frac{\text{Useful heat}}{\text{Work input}} = \frac{Q_1}{W} = \frac{Q_1}{Q_1 - Q_2} \tag{2}$$

If the application is cooling, Q_2 is the desired effect and

$$(COP)_{cooling} = \frac{Q_2}{W} = \frac{Q_2}{Q_1 - Q_2} \tag{3}$$

It is clear that high COP's are desirable. It can be shown readily from classical thermodynamics that the maximum COP's attainable for given T_1 and T_2 are the Carnot cycle efficiencies given by:

$$\text{Heating} \qquad (COP)_{max} = \frac{T_1}{T_1 - T_2} \tag{4}$$

$$\text{Cooling} \qquad (COP)_{max} = \frac{T_2}{T_1 - T_2} \tag{5}$$

These efficiencies are not attainable but represent upper limits which can never be exceeded no matter how well a heat engine device is designed. A measure of how well a practical heat engine (or reversed heat engine) is operating is by expressing its performance in "percent of Carnot". For a typical conventional air-to-air heat pump, T_1, the condensing temperature of the refrigerant, is on the order of 130^{o}F, and T_2, the evaporating temperature, is on the order of 45^{o}F (or less). Converting these to absolute temperatures and applying equation (4) shows an ideal heating COP of 6.94. If, however, the evaporator temperature were raised to, say, 75^{o}F by solar energy, the ideal COP is raised to 19.6.

Neither of these values is practically attainable, of course, but if the same "percent of Carnot" could be achieved for each, the latter case offers significant energy savings.

The basic components of a refrigerator/heat pump vapor cycle are shown schematically in Figure VI-2. The corresponding theoretical pressure-enthalpy diagram is seen in Figure VI-3. The discussion pertains to the heating mode. The basic cycle processes are:

1-2 Starting at point 1, saturated vapor is compressed to a pressure corresponding to saturation for the required condensing temperature. The ideal compression process is isentropic, given by $1\text{-}2_s$, but irreversibilities due to fluid friction and turbulence cause an entropy increase and require greater work input to the compressor, reflected by the compressor isentropic efficiency

$$\eta_{isen} = \frac{h_{2_s} - h_1}{h_2 - h_1} \qquad (6)$$

which typically is on the order of 0.8 to 0.9.

2-3 The superheated vapor at point 2 is cooled to saturation, then condensed at constant pressure, rejecting heat to the space to be warmed.

3-4 The saturated vapor is expanded via a throttling (constant enthalpy) process to the saturation pressure corresponding to the evaporator temperature required for absorbing heat from the scurce.

4-1 The bulk of the refrigerant evaporates, thereby absorbing heat that will be rejected into the conditioned space.

Pt. 2 The compressor suction condition, saturated vapor at the evaporator temperature.

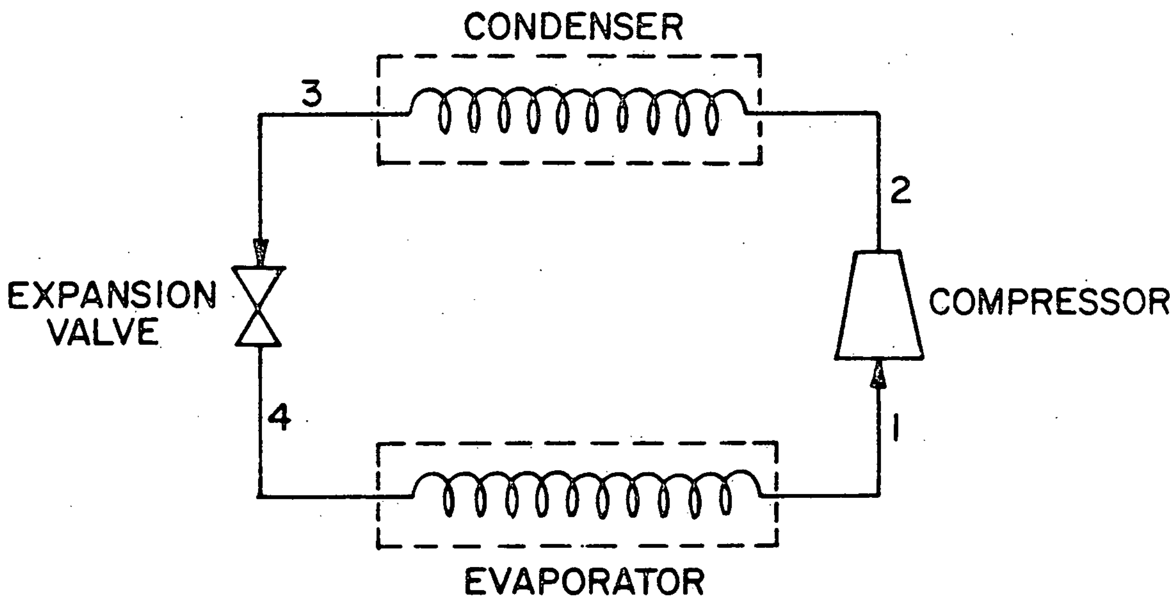

Fig. VI-2. Schematic of Simple Vapor Cycle Components.

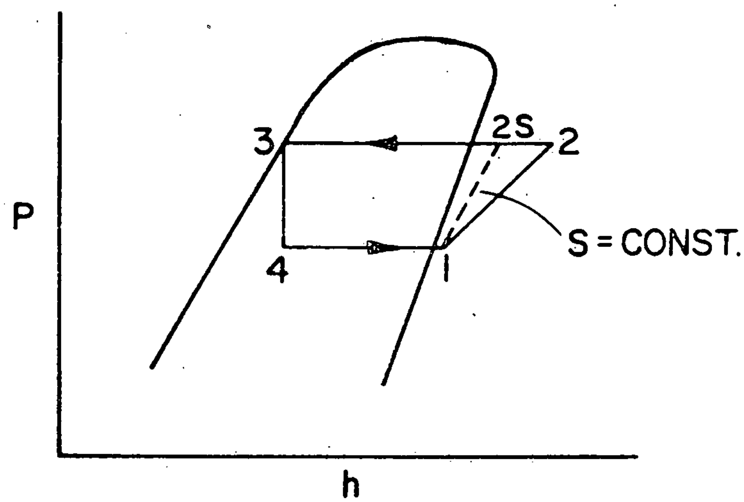

Fig. VI-3. Pressure-Enthalpy Diagram of Simple Vapor Compression Cycle.

For this cycle, $\Delta h_c = h_2 - h_1$ = enthalpy added by the compressor, (7)

$$Q_1 = Q_{cond} = h_2 - h_3, \quad Q_2 = Q_{evap} = h_1 - h_4 \quad \text{where } h_4 = h_3 \tag{8}$$

and $$COP_{heating} = \frac{h_2 - h_3}{\Delta h_c} = \frac{h_1 + \Delta h_c - h_3}{\Delta h_c} = 1 + \frac{h_1 - h_3}{\Delta h_c} \tag{9}$$

In an actual vapor-cycle/heat pump device there are a number of factors which modify the basic (simplified) cycle described above, all of which produce efficiency losses and lower COP. These include pressure drops due to flow resistance in the evaporator and condenser heat exchangers and piping, heat losses to the ambient surroundings, compressor volumetric efficiency (or inefficiency), compressor mechanical losses which are not accounted for by η_{isen} and prime mover losses. Additionally, there is superheat at compressor entrance and subcooling at throttling valve entrance. Before introducing the complication of detailing these factors, it is instructive to view the overall system defining the heat pump requirements.

Figure VI-4 presents a scale of the temperatures intrinsic to heat pump operation in the heating mode. The basic "given" quantities are (2), the design temperature required of the heating system terminal devices, which is typically on the order of 120 to 125° F and (5), the effective source temperature. For efficient performance, the quantity (e) should be minimized consistent with the practical constraints, since it defines the "lift" required of the compressor, which operates at a pressure ratio approximately equal to the saturation pressure corresponding to T_{cond} divided by the saturation pressure corresponding to T_{evap}. This can be accomplished in several ways:

1. supplying a high source temperature
2. reducing quantities (a) and (d) by using sufficiently large and efficient heat exchangers, and

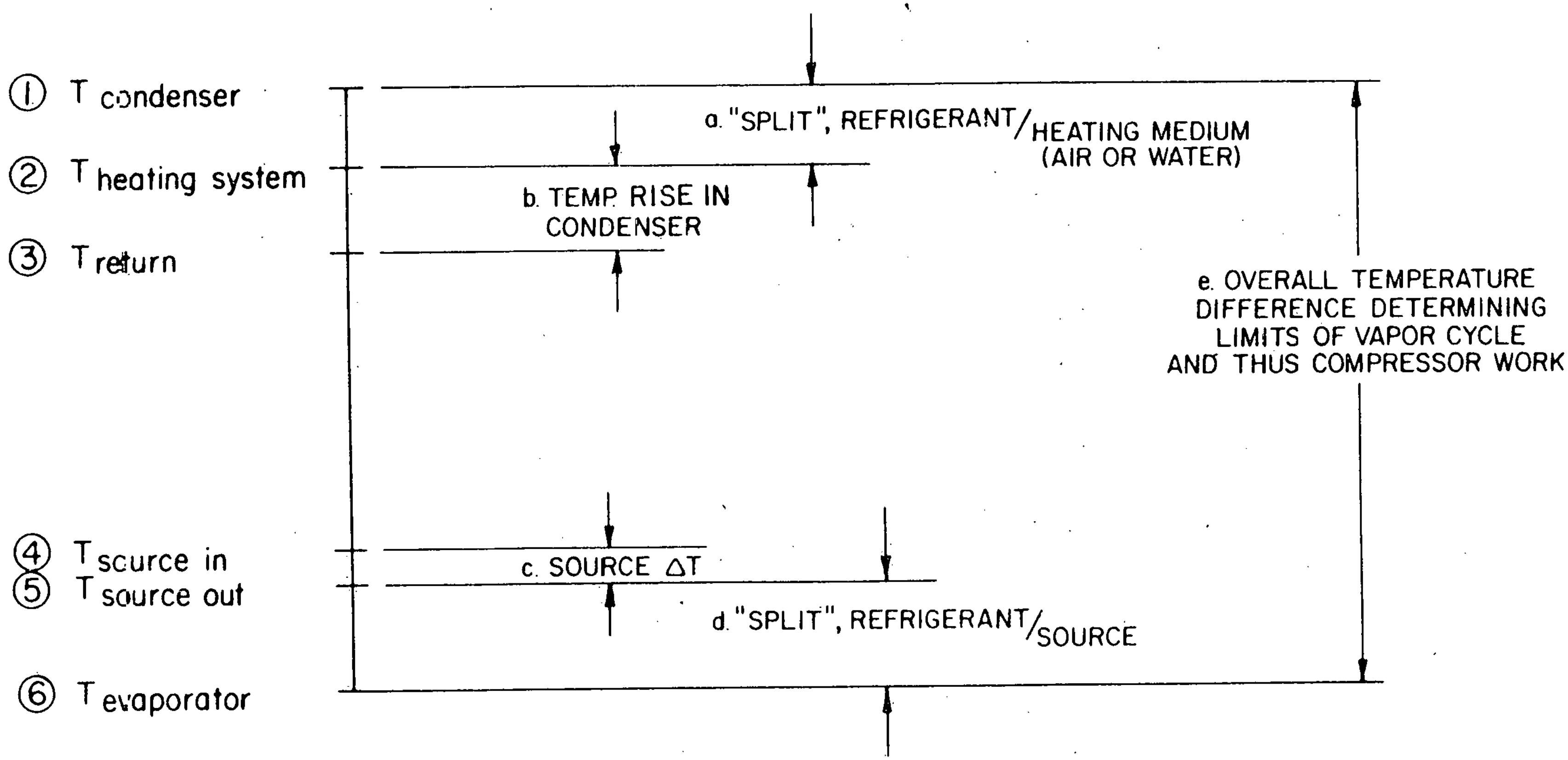

Fig. VI-4. Temperatures Affecting Vapor Cycle Operating Limits.

3. designing the heating system (and building) to operate at as low a temperature as practical.

Items 2 and 3 can be done independently of solar energy, and should be for good conservation practice, but they are particularly important in solar systems for providing more efficient usage of "free" solar Btu's (e.g., allowing the collectors to operate at a lower temperature). Item 1 lies at the heart of the present discussion in that solar energy can readily provide a high source temperature.

Given the temperature range of the cycle, as shown in Figure VI-4, several different COP's can be related to it.

I. The Carnot cycle COP as given by equation (5).

II. The theoretical vapor cycle COP for the cycle of Figure VI-5, which with $\eta_{isen} = 1.0$ will be called the Ideal Vapor Cycle.

III. The actual COP of a practical working machine which is considerably lower than (II) because of the inefficiencies previously listed and additional efficiency reducing measures incorporated as practical means of reducing machine cost.

Now (I) can be simply calculated from equation (5) and is independent of working fluid. The calculation of (II) requires use of property tables (or charts) of the given refrigerant and follows the description accompanying Figure VI-5. The enthalpy input from the compressor is evaluated by tracing a constant entropy line, such as 1-2s in Figure VI-2, from P_{evap} to P_{cond}, giving the isentropic Δh (or using the analytic expression for Δh_s in an isentropic process). This is the work input for the Ideal Vapor Cycle. For the actual COP, (III), performance data of existing heat pumps can be used for evaporator temperatures up to 55^oF, but above that point extrapolation or a lengthy calculation accounting for the system details is necessary.

Figures VI-5 and VI-6 show the results of a series of calculations for condensing temperatures of 105 and 135°F, respectively, and evaporator temperatures ranging from 40 to 100°F. For case (II), refrigerants R-12 and R-22 were treated as the working fluid, and the plotted curve represents an average for the two, which fell quite close together. Also plotted are curves of 80% and 60% of Ideal Vapor Cycle COP. The 80% curve could represent a heat pump with η_{isen} = .8 and no other losses, and the 60% curve could represent the same machine with the additional listed losses accounted for. For case (III), compressor data from several manufacturers for R-12 were used for evaporating temperatures up to 50°F, and then extrapolated to the higher temperatures by backing out compressor isentropic, volumetric, and mechanical efficiencies from the data and extrapolating them. Additionally, it was assumed that the heat exchangers were large enough to accommodate the large heat loads of the higher temperatures. An electric motor efficiency of 0.85 was included, but the power run pumps or blowers were not charged against the COP herein.

It was found that these values fell very close to or above the 60% Ideal Vapor Cycle efficiency curve as given in Figures VI-5 and VI-6. It is realistic then, to take this 60% ideal curve as a lower limit for the COP performance of practical SAHP's, since this performance can be raised relative to that of today's typical machines by incorporation of mechanical and thermal efficiency improvements which are cost-effective on a life-cycle basis. An upper bound, or goal, may be taken as the 80% ideal curve, and case (III) performance is shown in Figures VI-5 and VI-6 as a band between the 60 and 80% curves. (Note that the ideal curve itself can be raised somewhat by modifications to the basic cycle which are not discussed here in order to preserve brevity and clearness, the points being demonstrated not being affected.)

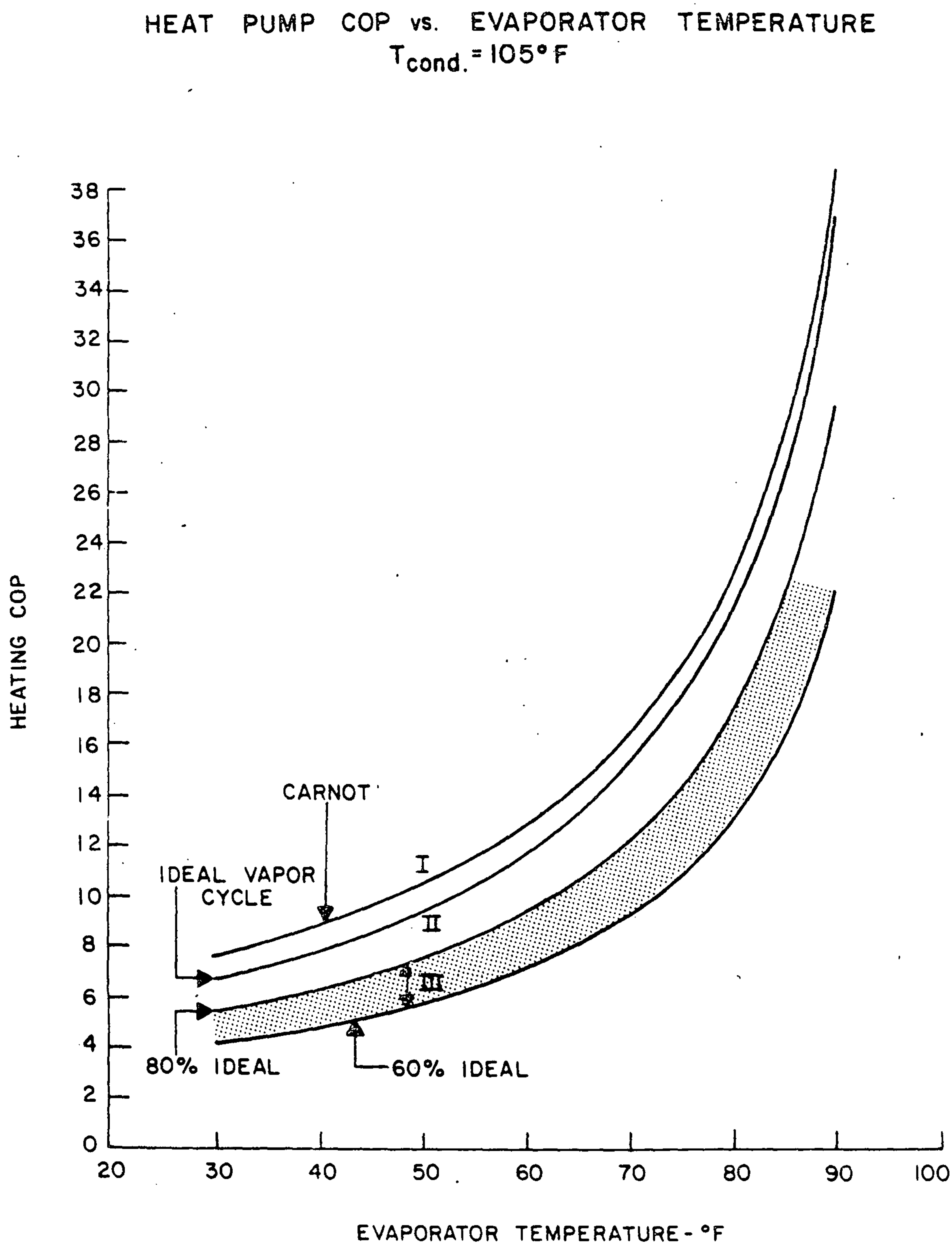

Fig. VI-5. Heat Pump COP vs Evaporator Temperature, $T_{cond.} = 105^{\circ}F$.

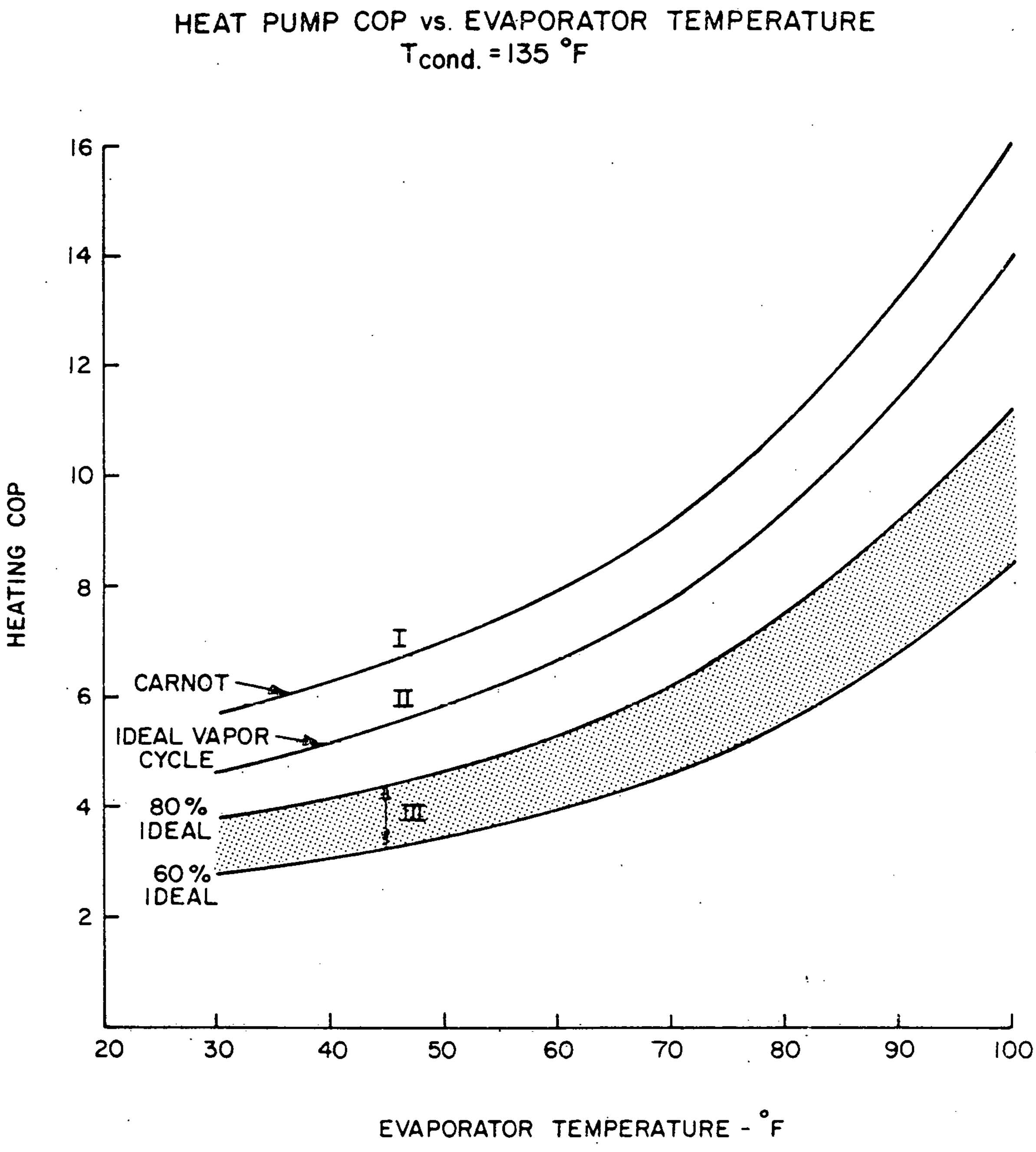

Fig. VI-6. Heat Pump COP vs Evaporator Temperature, $T_{cond.} = 135^{\circ}F$.

B. Practical SAHP's

Once accepting the viability of high COP performance of the SAHP, the challenging problem of making practical hardware to realize these advantages must be undertaken. At its heart is the necessity to make the system operate efficiently over a wide range of these temperatures (approximately 40 to 100°F). As previously discussed, current heat pumps can not perform this task because the high vapor densities produced by the high pressures attendant to these temperatures raise the mass flows to levels which can not be properly accommodated by their compressor, heat exchangers, and expansion valves. These machines can be forced to operate at the high solar-supplied evaporator temperatures, but only through the use of energy-inefficient techniques, e.g., hot gas bypass of the condenser, and the potential high COP's are not realized. That is, the COP's at suction temperatures above 50°F are the same, or even less, than that at 50°F and do not monotonically increase like the curves of Figures VI-5 and VI-6. Thus, as heretofore stated, simulation studies comparing parallel and series SAHP's using the COP's produced by current heat pumps do not account for the true potential of the series system.

A heat pump which will properly utilize solar heat must, therefore, incorporate some significant changes which accommodate the higher suction vapor densities by use of energy conservative techniques. These changes must imperatively produce a higher first cost of the machine but are justified by the energy savings which will provide a lower life-cycle cost. To attain them is well within the technology and capability of current manufacturers, given the incentive of a suitable market. As an obvious first step, larger (or multiple) heat exchangers and expansion valves may be employed. This is a necessary, but not sufficient, condition because the operation of the compressor and its ability to modulate the system is the key factor in a heat pump. Use of some form of capacity control appears tantamount to success. A salient choice is that of variable speed control of positive displacement machines, not a new technique at all. But previously it has been utilized to allow operation over a wide band of suction temperatures toward the low side. In a SAHP it can be used to extend the range of efficient suction temperatures toward the high side, that is 60 to 90°F or higher, with the lower speed used to reduce system mass flow at these higher temperatures. High speed operation can be used in the cooling mode (evaporator temperatures on the order of 45°F) and in the heating mode when available solar-supplied storage temperatures are at the lower end of the useful range, from 40°F to approximately 60°F. A continuously variable speed machine would be desirable from a theoretical viewpoint, but in practice a two-speed 2/4 pole or 4/6 pole motor producing a discrete step in capacity would probably be satisfactory. The suction temperature at which the step in capacity occurs must be defined as a function of the climate, collector and storage size, and building and heating system design and can be optimized theoretically by computer simulations and verified by field demonstrations.

Alternatively, compressor-supplied capacity modulation could be produced by cylinder unloading. In this method, gas flow to one or more of the cylinders is blocked off when high evaporator temperature is creating excess mass flow. Additionally, there is the possibility of using dual compressors in parallel. These methods, too, have been employed previously to some degree in vapor cycle machinery, and like the two-speed motor, produce a step in capacity at some designated evaporator temperature. Each of the three methods has its relative merits and disadvantages which will not be discussed here. As a further step, a compressor having variable compression ratio corresponding to that most efficient for the temporarily available source and required outlet temperatures can be developed. This machine could employ relatively sophisticated valve configuration and control, heretofore considered cost ineffective. It would give a continuous capacity output versus source temperature, and could be of either the reciprocating or rotary type, the latter corresponding to larger sizes.

In the Solar Assisted Heat Pump the throttling valve can assume an increasingly important role. The thermostatic expansion valve is the most likely candidate, but alternative, and perhaps innovative, designs which provide good mass flow control over a wide range of pressure ratios can be used. Two parallel valves or a single valve with an auxiliary bypass may be employed. The selection of refrigerant charge in the bulbs of externally balanced thermostatic valves can provide additional flexibility for regulation characteristics.

In addition to the design of the basic heat pump machine, which is vital to the success of an energy and cost-effective system, attendant design steps which incorporate multiple source and compound or cascaded vapor cycles can be developed to further increase SAHP system effectiveness.

To implement the development of effective Solar Assisted Heat Pumps, the Solar R&D branch of the Department of Energy's Solar Division issued RFP EG-77-R-03-1467 in 1977. Three companies have been chosen as successful respondents to carry out two-year development programs which will result in prototype hardware. Brookhaven National Laboratory (BNL) is responsible for technical monitoring of these programs. At BNL, a Solar Assisted Heat Pump simulator is being designed and constructed to assist in evaluating the resulting designs. In conjunction with the simulator, a laboratory model Solar-Assisted Heat Pump is being assembled which will be used to verify the operation of the simulator and to conduct laboratory experiments of SAHP performance. These experiments will investigate attainable COP's as a function of evaporator and condensing temperature, refrigerant, compressor capacity control technique, heat exchanger size, and expansion valve configuration for both steady state and transient operation.

VII. COLLECTORS FOR SOLAR ASSISTED HEAT PUMP SYSTEMS

A major advantage of the series solar assisted heat pump concept is its ability to effectively utilize energy from simple low cost solar energy collectors.

Parallel systems and solar systems which do not include heat pumps require collectors which can supply the load via direct heating. Collector outlet temperatures in the 130-180°F range are needed. Collectors capable of supplying such temperatures at reasonable efficiencies are on the market and typically cost about \$15 per square foot of collector delivered to the site but not installed.

The series system, since it can make use of lower temperatures, can utilize low temperature, low cost collectors. Low temperature operation of the collectors means that many of the pains normally taken to maintain high efficiency at high collection temperatures can be dispensed with. Because of these relaxed design requirements installed costs of \$5/ft^2 or less are foreseeable for this type of collector.

The use of low temperature, low cost collectors in a series system has two ancillary advantages which should be pointed out. First, they have high efficiency when high efficiency is most needed, in the winter months. It is then that the temperature difference between storage and ambient is lowest. Secondly, these collectors are suitable for summertime direct heating of domestic hot water. Stagnation temperature experienced by these collectors are acceptable without additional safety precautions.

As a preliminary step in our investigation of the potential for this type of collector, we conducted a survey of collector manufacturers and researchers to determine whether collectors having acceptable characteristics of performance, cost, and reliability could be obtained. The characteristics sought were:

1. collector priced to the customer at $\$5/ft^2$ or less,

2. collector efficiency of 75% or more at $\Delta T/I = 0$, and dropping to zero efficiency at $\Delta T/I = 0.5$ ft^2-hr-F/Btu or more, where ΔT is the difference between the average fluid temperature and the ambient temperature, in ^oF, and I is the insolation striking the collector, in Btu/ft^2-hr.

3. twenty-year life.

We wrote letters to the 185 collector manufacturers listed in the April, 1977 edition of "Solar Collector Manufacturing Activity"[1] published by the Federal Energy Administration and to the twenty announced winners of the research proposal competition conducted by the U.S. Energy Research and Development Administration under PRDA EG-77-D-29-0001.[2]

Approximately fifty replies were received, a few without price information. It was found that modularized, factory-built collectors having the specified performance and life are not generally obtainable at an installed cost of $\$5/ft^2$ or less. Unglazed swimming pool type heaters can be obtained for less than this figure, but their performance cannot be expected to meet even these minimal criteria, and the plastics from which they are generally constructed may be expected to have a short lifetime.

Two responses from researchers doing work under contracts resulting from the PRDA give hope that the above specifications can be met. The first of these is a design developed at West Virginia University.[3] It consists of a Foamglass™ absorber/insulating plate into which vee-grooves are cut, giving a ratio of absorber area to glazing area of 3 to 1, thereby improving the heat transfer rate from absorber plate to the collector air stream. This collector is available in manufactured form for $4.50 per square foot. Installation and any contractor's markup are not included. However, improved manufacturing methods such as molding the glass foam material to the proper shape rather than cutting it, should result in reduced cost.

The second concept is a black liquid collector being developed at Battelle-Columbus Laboratories. This concept is still in the early stages of development, but the principal investigator has estimated[4] that the installed cost will be less than $5 per square foot.

Measured and estimated performance data for these concepts are shown in Figure VII-1, along with our minimum criterion and the performance curve used in the computer simulation discussed in section IX. If this performance is confirmed, and if the collectors prove sound over twenty-year operating lifetimes, then one or two collector candidates have been identified.

Discussion of these two concepts should not be taken to exclude other approaches to low cost collection of solar energy. Indeed, it is likely that a variety of concepts are worthy of investigation. When the development effort on the special heat pump becomes more generally known, development activity on low cost, low temperature collectors may be expected to increase.

The identification of these two collector candidates indicates that low cost collectors with acceptable life and performance characteristics can be developed. Such collectors are not generally available on the market today because the collector manufacturers have tended to take the same path, a path which had led to expensive collectors designed to meet other objectives. The need for a collector for the series system has not been clearly identified because of the absence of the proper heat pump component. That is, a feedback relationship appears to exist between heat pump and collector development. Heat pump manufacturers have not had the incentive to develop the

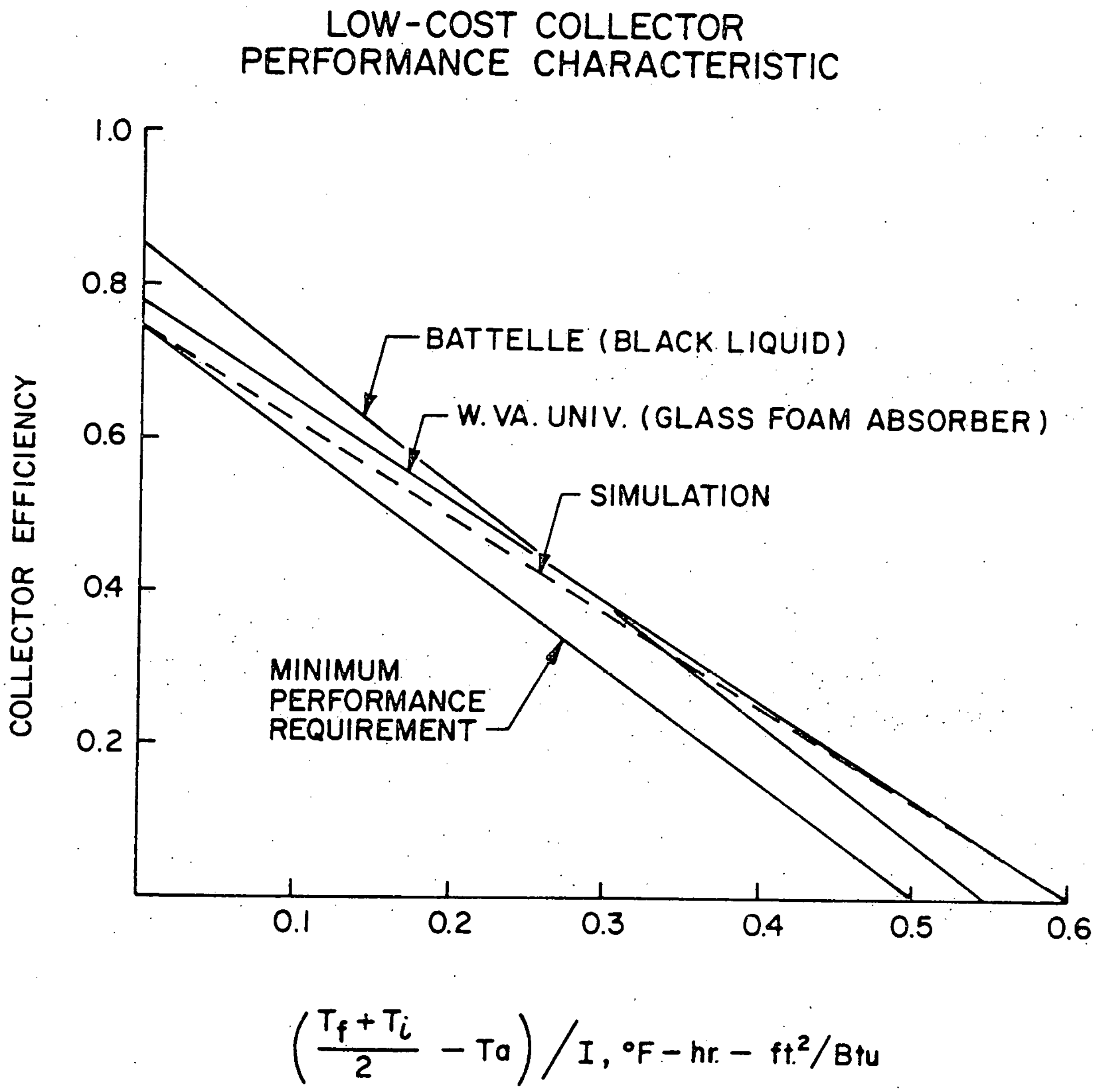

Fig. VII-1. Collector Performance Curves Obtained from Research Projects Surveyed.

advanced heat pump needed for the series solar assisted heat pump system because the low cost collectors were not available. Collector manufacturers did not have the incentive to develop the low cost, low temperature collectors because the advanced heat pump needed to incorporate them into an efficient heating system was not available. An important role of the U.S. Department of Energy is to break this loop by supporting development of both of these needed components.

Analysis of collector cost factors has revealed some further interesting observations. Collector manufacturers generally attempt to produce as much of the collector package in the factory as possible, reducing the on-site labor to a minimum. This strategy has been influenced by the higher cost and lower efficiency of on-site labor relative to factory labor. While this principle seems sound, its application in this case has not resulted in low collector costs. There are two reasons for this. The first of these has been the need for structural rigidity during transport and installation of the collectors, which has resulted in the production of material-intensive solar panels weighing as much as eight pounds per square foot. There is a direct relationship between weight and cost in the fabrication of structures, and heavy panels are more costly.

Another factor which stems from the same root has been the method of distribution of prefabricated collectors through a network of middlemen, each of whom has his own markup. The result has been that the consumer pays a price which can be as much as five times[5] the manufacturing cost. This markup evidently more than compensates for any labor cost savings and must be carefully considered in future production plans.

The construction of collectors on-site from pieces which are distributed through the building materials industry needs to be reexamined. The pieces from which the collector is constructed can now be relatively light in weight, and these pieces, distributed in the same manner as building materials, will have much lower price markups. By careful design, site labor requirements can still be kept low. It is likely that these ideas will find their most immediate application in low temperature air-heating collectors since dimensional quality control requirements can be relaxed. Precise fits, close tolerances, and caulking needed to achieve high temperatures are not required.

We have examined four basic designs[6] embodying these principles. Dubin-Bloome Associates have been retained to determine the installed costs of these collectors; their report[7] indicates that the lowest cost of the four designs can be built at a $4.88/ft^2 installed cost to the consumer. They have further indicated design changes which will reduce this cost significantly. The import of the Dubin-Bloome study is that major breakthroughs can be made in costs for the type of collector needed by the series solar assisted heat pump system.

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VIII. THERMAL STORAGE FOR THE SERIES SOLAR HEAT PUMP SYSTEM

In the series heat pump system, energy is collected in the form of low temperature ($\sim 40^{\circ} - 120^{\circ}F$) heat. As there is no cheap, efficient way to convert this heat into another storable form of energy, energy is stored thermally. There are two general types of thermal storage. "Sensible" heat storage means depositing energy by raising the temperature of a storage medium. In "latent" heat storage a reversible phase change or chemical reaction is effected to store heat at constant temperature.[1]

Whatever type of thermal storage is used, there are certain conditions that the storage must satisfy in order to be suitable for the series system:

1. It must be able to store heat in the general temperature range of $\sim 40^{\circ}F - 120^{\circ}F$.

2. Generally, it is desirable to keep the storage temperature as high as possible in order to provide the heat pump with a high temperature source. The computer simulation in section X constrains the storage temperature to be above $35^{\circ}F$. Ideally, a similar condition will be maintained in real systems. However, in certain instances, a lower minimum storage temperature may be the only alternative to extensive resistive heating. For example, storage space or collector area might be unavoidably inadequate. In these instances, an antifreeze solution must be used, or given a specially designed storage vessel, freezing can be permitted. The latter course is preferable (if an economical device can be built) as the water-ice phase change is exploited to help keep the temperature from dropping further.

3. The storage system must be economical. Economic viability must ultimately be judged in terms of the entire system.

4. The space requirement for the storage system must be reasonable. The

definition of "reasonable" depends on the value and size of the space occupied.

A. Latent Heat Storage

Energy storage as latent heat has two major advantages. It is possible to achieve a higher energy density than is possible with sensible heat storage because there are many processes that involve latent heats which are large compared to the volume heat capacities of common sensible heat storage materials. For this reason, latent heat storage has potential for reducing storage space requirements, container size, and cost. Secondly, the storage element can be operated over a smaller temperature range which is desirable in certain applications. For example, space cooling requires a temperature of no more than about $45^{\circ}F$ in order to accomplish dehumidification. A latent heat "cool storage" device at this temperature would therefore have application.

While there are many interesting research projects being conducted in this field, no latent heat storage system is presently commercially available which meets the conditions set forth above for the series heat pump system. Due to the great potential of this type of storage, a brief discussion is presented to indicate the areas which have been explored to date, and the present "state of the art". References are provided for the interested reader.

Certain chemical reactions produce very large amounts of heat per unit volume. Therefore, storage via the latent heat of chemical reaction has ultimate potential as an extremely high energy density storage medium. This storage method is not yet well developed. One major problem which must be addressed is that the most energetic (and thus most desirable) reactions involve gases. This presents containment and pressure problems. On the other hand, in reactions involving a solid phase, mass transport and hence heat transport are inhibited.[2] Reference 2 discusses various chemical reactions (and other types of thermal storage) which may have heat storage applications. The incipient nature of the field of chemical thermal storage is evident. Other references on chemical storage have been presented,[3a] work on this subject has been discussed,[4a-c] and the need for more research has been indicated.[4d] The present work in this field primarily involves finding and studying suitable reactions.

Work in latent heat storage via phase change materials is at a more advanced state than the research on chemical reactions discussed above, but here again more research and development is needed. The potential value of a compact, high capacity, economical storage device is immense, so that there is great interest in this field. On the average, liquid-gas phase changes involve the most energy. As an extreme, but very common example, the water-steam phase change involves about 980 Btu/lb (depending upon the temperature). By comparison, the sensible heat capacity of water is 1 Btu/lb ^{o}F. (This is very high as sensible heat capacities go.)

Because of the containment problems created by the production of gases, most efforts toward latent heat storage via phase change materials have used the solid-liquid phase change. References have been listed in the literature[2b] and various phase change materials relevant to space heating and cooling have been studied and catalogued[2c, 4e-1] (especially with regard to their melting point and heat fusion). The materials of special interest include salt hydrates, clathrates, various paraffins and other organic chemicals, and water. A characteristic set of problems presently common in liquid-solid phase change devices has been identified elsewhere:[3b]

1. Supercooling

   The liquid continues to cool below the nominal freezing temperature instead of freezing. Nucleating agents have been added to rectify this.

2. Incongruence

   The two phases separate, often due to gravity. This can reduce the heat transfer rate. A similar problem is separation of the nucleating agents (used to cure 1). Attempts have been made to resolve this problem by using thickeners.[4m]

3. Complex Melting

   Repeated cycling results in less materials undergoing the phase change after repeated cycling.

4. Reduced Energy Density

   Various schemes to alleviate the above problems result in enlarged devices which are no longer smaller than equivalent sensible storage devices.

A few analyses and experiments have been conducted on phase change material thermal storage as applied to solar energy and energy conserving heating schemes. It was judged that "a salt hydrate phase change material has been used successfully for three years with air-to-air heat pumps in 'Solar One' at the University of Delaware...".[5a] However, elsewhere it was claimed that this system suffered from problems 2, 3, and 4 above.[3c] Another author reported abandonment of a salt hydrate "cool storage" system due to "instability of the salt".[5b] The use of paraffin has been discussed,[5c] but has also been discounted elsewhere due to flammability.[3d] (Similar fears apply to many organic materials.) A recent computer simulation compared storage devices for solar heating systems using paraffin, sodium sulfate decahydrate ($Na_2SO_4 \cdot 10\ H_2O$), rockbeds, and water tanks. A conclusion was: "A system utilizing paraffin wax may require a slightly larger storage volume than a system of comparable performance with a water tank. Systems utilizing $Na_2SO_4 \cdot 10\ H_2O$ require roughly one-half the storage volume of a water tank system."[6] They also concluded that "the choice between sensible and latent heat storage will be decided solely by economic considerations."[6] Studies of more elaborate storage devices are also being undertaken.

General Electric reports work on "rolling cylinder latent heat storage" which consists of "...a horizontal cylinder filled with latent heat storage materials revolving slowly on a set of rollers. A thin tubular nucleator connects to one end at the axis rotation."[3e] No "hard" data has been presented. The state of the art of phase change storage systems has been evaluated[4n] (1975): "A wide range of candidate materials and configurations has been identified. Of these, a goodly number, perhaps one-tenth to one-fifth, have received some experimental attention and a very few have actually been employed in experimental buildings. Success is application has not been uniform and economic competitiveness has not been demonstrated. Other candidates require additional research, development, and demonstration."

Water has also received attention as a liquid-solid phase change thermal storage material. Two schemes using this storage method have been discussed recently. One, the "Dual Phase Annual Cycle" (DPAC)[7,8] has two very large tanks. A conventional solar system heats the water in one of these year round for heating season use. The other tank is frozen during the winter for summer cooling use. The reader unfamiliar with annual storage should appreciate that the storage volumes used are typically the size of a large room or even of an entire basement for a residential load. The disadvantages of DPAC, as indicated by the author are: "Large storage tanks are expensive and may be difficult to situate....The cold system may need a refrigeration unit for use in mild winter climates."[7a] Another approach is to use a heat pump to extract heat from water to make ice. This is the basis of the "Annual Cycle Energy System" (ACES).[4o,9-13] Here the ice is stored and used for summer space cooling. As in the DPAC above, this requires a very large, well insulated, storage vessel. To ameliorate this problem a "radiant/convector panel"[11a] may be used. This is a sort of unglazed, uninsulated solar collector which provides energy to melt the ice in order to reduce the needed storage volume (and also the summer cooling potential) required. Even so, the 2000 ft^2 demonstration house contains a large "water tank" which is roughly 19' x 17' x 10'[11b] for a volume of about 3200 ft^3. The initial cost is likely to be high for this system. Also, the summer cooling is "free" (neglecting parasitics) only to the extent that the cooling load matches the ice stored (after storage losses). Finally, as noted in the General Electric Study, "...it will always have the disadvantage of the 32°F freezing temperatures being too low for optimum performance."[3f]

B. Sensible Heat Storage

The method analyzed here for energy storage in the series solar heat pump system outline in section IV is heated water. Rockbed storage has been used in some other solar energy systems. Water is the first choice here for the following reasons:

1. Water is a more desirable heat transfer medium than air. Water pipes are cheaper to use and to insulate than air ducts and occupy less space.
2. Water-refrigerant heat exchangers can obtain smaller "splits" than are possible with air-refrigerant exchangers. Thus, the heat pump source temperature is effectively increased which raises the heat pump COP.
3. Water storage occupies less space than is required for rockbed storage.
4. A water storage system can be designed which satisfies all of the conditions set forth at the beginning of this section. The design details, costs, and possible options associated with this system are discussed below.

C. Proposed Water Storage System Description

The proposed water storage system consists of an inexpensive, uninsulated vessel which is buried underground.

1. Storage Volume

The storage volume projected for the solar source heat pump system is on the order of 1 to 10 gallons of water/ft^2 of collector area. A more definitive judgement has not been made yet because the computer simulation in section X uses "average weather functions" instead of actual weather data. While these functions offer computational advantages and lead to accurate estimates of many system parameters, they do not contain the solar insolation fluctuations of real weather. Thus, computer simulations using these weather functions are relatively insensitive to storage capacity variations if the storage size is at least large enough to carry the system through one night. Computer simulations by the Wisconsin Group indicate a noticeable improvement in system performance when a storage volume of 8.7 gal/ft^2 is used compared to 2.0 gal/ft^2, both at moderate collector area. The advantage disappears at larger collector area.[14] The use of thermal coupling between the water storage and the ground (discussed in section IX) further complicates the issue of storage size. This is an especially difficult interaction to model for widespread application due to the site dependence of ground thermal conductivity, ground water presence, and ground water flow. The Phoenix House (Colorado Springs) uses a water storage volume of about 8 gal/ft^2 in a ground coupled scheme with excellent results.[15]

Intuitively, there is a tendency toward larger storage in the series solar heat pump system than is common in conventional solar systems ($\sim$ 1-2 gal/ft^2). This is because the need to keep the storage temperature above $\sim 40^{o}F$ under many of the worst weather conditions and the lower dependence on direct solar heating both indicate that a large storage "thermal inertia" is desirable. On the other hand, if enough energy can be stored in the ground and a high enough heat transfer rate can be obtained when desired through a ground coupling scheme (see section IX), smaller storage volumes may be possible in the heat pump system.

2. Storage Construction and Cost

Two inexpensive storage vessels have been identified which are suitable for the series system. Both can be used in ground coupled installations or as conventionally insulated storage devices.

Steel tanks, similar to those used to store home heating oil, are one cost effective storage possibility. The price range typically quoted for solar system storage costs is ~ \$1 - \$2/gal (e.g., reference 16). A price quotation for steel tanks from a Long Island dealer in late 1977 was ~ \$.25/gal in the 550 to 3000 gallon range, almost independent of size.[17] This price does not include installation or insulation. Additionally, coatings have been recommended for the inside of steel tanks used to hold water and for the outside of those buried.[4p] It is possible, however, that interior coatings may not be necessary. One way to avoid coating is to use a fluid other than water. This may be expensive. More interestingly, the Phoenix House experience has been that corrosion is not a problem. After three years and four months of operation, the buried steel tank was drained of water and inspected. The uncoated interior was free of corrosion except for a thin "skin" of iron oxide.[18] The exterior, which had been coated with a thin layer of pitch, showed no sign of deterioration whatsoever.[18] In cases where underground installation is not feasible, steel tanks can be placed in a basement (probably not in retrofit), garage, or outdoors. Insulation is then necessary.

Various types of concrete tanks may be even more cost effective. The cheapest tank of this type is made from precast concrete cesspool rings. These rings are cylinders commonly 8 or 10 feet in diameter and 4 feet high. One 8-foot ring holds about 1600 gallons, and a 10-foot ring, 2500 gallons. Other sizes are available. A published cost estimate for buried, precast septic tank storage, including installation and waterproofing (but not insulation), is ~ \$.30 to \$.34/gallon (1975) in the 1000 to 1500 gallon capacity range.[4q] This author projected a cost reduction due to volume production and development of 25%.[19] Tanks made from precast cesspool rings are presently used in some areas to store water. Thus, no new technology is needed to use this type of storage. A recent installation on Long Island used two 10-foot diameter rings for a volume of roughly 5000 gallons. Typical underground installation is as follows:

1. The excavation is made. One ring is placed in the hole. Alternatively, the ring can be placed on the ground in which case soil is removed from the inside of the ring, which settles by gravity into the hole. This method is feasible only for large rings (8 ft or 10 ft).
2. A bed of mortar is placed on top of the first ring.
3. The next ring is placed, and the above process is repeated.
4. The bottom of the tank is formed from poured concrete.
5. A precast concrete slab with a manhole can be used as a lid. (These are also widely available in the appropriate sizes.)
6. The interior is waterproofed, if necessary.

The 5000 gallon installation mentioned above was waterproofed with a sealing compound which was troweled onto the inside of the tank. The total cost was estimated by the contractor to be ~ \$1500 or about \$.30/gallon. This installation has certain features unnecessary for the envisioned storage tank so that the cost should represent a conservative upper limit. The contractor noted that good quality precast concrete must be used and specified a local dealer. He also suggested that the cost could be reduced by using waterproof, precast concrete so that coating would be unnecessary. Another contractor provided an estimate for an unlined tank made from three 10-foot rings for a volume of

~ 7500 gallons. The price quoted (not including mortar work or top insulation, which is probably desirable) was $733, or less than $.10/gallon. This is probably a lower limit and is partially a result of the large size. In any event, with the use of waterproof, precast concrete rings in desirable sizes, underground uninsulated storage costs should be about $.20/gallon or less in the 1000 to 10,000 gallon range.

Concrete tanks are very cost effective when used in basements, etc. Concrete block or poured concrete can be used[4r] in retrofit applications where steel tanks might not fit through doorways. Precast rings could also be appropriately sized to fit through doors. They would then be stacked to form a tank, and insulation applied. However, the cost of basement storage is higher than for buried storage as the tank must now be structurally strong enough to support the weight of the water and as insulation is now required. One must also equitably account for the value of space occupied by the storage tank. This is not always done in solar system economic analyses. The reality is that basement space is not free. In fact, in an effort to reduce the "base cost" of new homes, it has become common to sell basements and garages as "options" at extra cost. In the case of underground storage, the only expense, besides the tank itself, is for excavation. Living space, purchased at considerable expense, is not occupied.

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19. This reduction applies to insulated tanks. Insulation cost reductions must be extracted.

IX. GROUND COUPLED STORAGE

There is evidence to indicate that the introduction of thermal coupling between the storage tank and the ground can improve the performance and reduce the initial cost of the series solar heat pump system, The ground acts in two roles in order to achieve these improvements. When the storage temperature is below the ground temperature, the ground provides heat and thus behaves as a "buffer" to help raise the storage temperature. This "smooths out" the storage tank temperature fluctuations and raises the annual minimum storage temperature. As a result of the latter, resistive heating is reduced or eliminated. The ground is also used as a "quasi-annual" storage device which permits greater usable energy collection and storage. This enchances the value of the collectors, or put another way, a smaller collector area can supply a given load. The amount of money saved in this way can be significant. The annual average storage temperature is also elevated which makes more direct heating possible and raises the heat pump COP.

A. Qualitative Analysis

The suitability of the ground for the roles envisioned above is now examined. Some of the results of the computer simulation in section X are illuminating in this connection. Figures 5 and 11 of that section plot average storage temperature (among other quantities) versus time for the optimally sized systems for New York (500 ft^2 of collector) and for Washington, D.C. (333 ft^2 of collector), respectively. It can be seen from these figures that for each city the storage temperature curve is roughly sinusoidal (about ½ cycle during the heating season) as is the curve which describes the annual ambient temperature variation. These "sine curves" are in phase, i.e., they reach their minimum temperature on almost the same date (sometime in January). The curves are in phase because it is not economically feasible to provide enough storage to carry the load for more than a few days. Hence, the storage temperature is pulled down as the ambient drops, and the heating demand rises, despite infusions of collected energy. The result is that during about two months of the "hardcore" heating season (roughly December 10 to February 10), the storage temperature is below 55°F.

A few feet below the surface, the ground temperature averages about 55°F throughout much of the United States. Evidently, the heat pump system would benefit from thermal contact between the storage and the ground during this period. The total amount of heat transferred via this route is limited by the low temperature and low thermal conductivity of the ground. Experiments using the ground as a heat source/sink for heat pumps tried to overcome this intrinsically low heat transfer rate by using buried serpentine coils through which flowed the heat pump refrigerant or a heat transfer fluid (which delivered heat to a "normal" heat pump heat exchanger). Many papers have been written on this subject.[1] It has been claimed that even in these cases most of the heat obtained was derived by freezing water in the soil.[2] In at least one test, however, water was used as the heat transfer fluid so that in this case at least "no advantage was realized from freezing the earth."[3] A more serious problem is that if a significant energy is withdrawn and not replaced, the ground temperature gradually drops and prolonged operation is not possible.[4a] It is conceivable that a "grid" of serpentine coils could be spread over a very large area in order to ameliorate this last problem.[4b] However, calculations indicate that this is probably not economical. In many cases, the required land area might not even be available.

The above situation is improved markedly if heat is added to the ground as well as withdrawn from it. In a sense, the conductivity of the ground, which limits its use as a heat source per se, makes the ground an ideal heat storage device. This can be seen by noting that the ground temperature varies sinusoidally with an annual period as does the ambient temperature (see, e.g., Figure X-5). The reason for this is that the atmosphere (and ultimately the sun) is the heat source which "drives" the seasonal ground temperature variations. Interestingly, though, the underground temperature is not "in phase" with the atmospheric temperature. One recent paper indicates that the temperature four feet below the surface in Oklahoma peaks in October and reaches its minimum in March.[5a] During the "hardcore" heating season, the ground temperature is very close to its annual average. Since the ambient temperature peaks in July and reaches its minimum in January, it can be seen that for the case of Oklahoma, the variation of the temperature four feet underground "lags" three months behind the ambient temperature fluctuation which produces it. This result, or "flywheel effect" as it has been called,[5a] transcends the "natural" flow of heat from the atmosphere into the ground, and occurs when any source (such as a solar collector) transfers heat to the ground. It is important that this phase shift appears to be on the order of a few months. A "time constant" of this size is desirable as it indicates that energy delivered to the ground during the summer and fall will not diffuse "too far" (on the average) in a few months and will therefore be partially retrievable during the winter when it is most needed. If the phase shift were much shorter than a few months (e.g., a few hours or days), the energy transferred to the ground would quickly be lost. A shift longer than a few months would make it difficult to transfer a significant amount of heat to the ground in a few months, i.e., an insulated storage vessel would be approximated.

In a system with conventional storage, during the summer and early fall the solar collectors (purchased at great expense) are largely unused, even though external conditions are optimal for energy collection. This is because the heating demand is light at this time so that the limited storage capacity is usually saturated. The addition of thermal coupling between the storage tank and the ground augments the limited "purchased" storage with the virtually limitless storage capacity of the ground. The value of the collectors is enhanced as they can now collect energy almost year round. In the winter, whenever the storage tank temperature drops below the now elevated ground temperature, some of this energy is retrieved. The storage of heat in the ground for up to several months is termed "quasi-annual storage". The retrieval rate is greatest when the tank temperature is the lowest, i.e., when extra heat is needed the most. In this way the ground acts as a buffer to provide heat for situations that cannot economically be prepared for in any other way.

B. Existing Evidence

Up to this point, the discussion of ground coupled storage has been almost entirely qualitative. The factors that will determine whether this storage method is practical are largely quantitative and also site dependent. Important questions include: Can a significant amount of energy be transferred to the ground? Can it be stored for a long enough time to be useful? Can energy be stored at all in some locations, e.g., in the event of groundwater flow? (Note that dependable groundwater flow at a reasonable temperature is itself an excellent heat pump heat source.) Can the stored heat be retrieved from the ground when needed at a fast enough rate? Finally, and supremely, if ground coupled storage is possible, is it economically attractive?

In the last few years, a number of analyses have been written on various aspects of underground heat storage.[6-13] The Phoenix House experiment has a large (approximately 7000-gallon) buried steel storage tank.[14a] The tank is surrounded by an "envelope" of sand about one-foot thick to increase thermal conductivity in the region of the tank. The ground cover is four feet so that the tank is below the frostline penetration level. Data from November, 1975 to August, 1976 indicate that this series solar heat pump system with ground coupled storage was able to successfully heat the load with only a very small amount of resistive heating.[14b] For this season, no heat was placed in storage until late September, 1975.[15] During the period from January, 1976 to August, 1976, "39% of the heat put into the tank was transferred to the ground [much of this was retrived as discussed below], 57% was extracted by the heat pump, and 4% was stored in the water."[14c] Evidently, it was possible to store a large amount of energy in the ground. The storage losses to the ground were low during the winter (January - March, approximately 10-20%).[14c] In fact, during the worst part of the heating season, heat flowed from the ground into the storage tank.[14d]

The concept of "quasi-annual storage" was tested the following heating season. Through the spring, summer, and fall of 1976, heat was placed in storage. The minimum storage temperature reached during the subsequent winter of 1976-1977 was 58°F.[72] This is to be compared with 40°F the previous winter (when storage charging started in late September).[15] Note that the heat pump simulated in section X has a heating COP of 4.6 when operating from a 58°F source (Table X-1).

C. Other Options and the Future of Ground Coupled Storage

The use of buried serpentine pipes to transfer collected heat to and from the ground is one major hardware option. Experiments have been conducted by Bose and verify that an adequate heat transfer rate is obtainable.[5b] Indeed, a high heat transfer rate due to the large surface area involved is the forte of this approach. The problems which must be addressed by future research are cost-effectiveness and low storage energy density. The first problem is due to the hundreds of feet of trenches and pipe that are required. The second problem involves the distribution of the stored energy over a large land area. That is, the temperature of retrieved energy would be expected to be lower on the average than the energy stored in a small region around a tank. Thus, given a heat pump that can capitalize upon high source temperatures, this diffuse storage may be a performance disadvantage. The resolution of this problem is not clear and further investigation is required to accurately assess the economic tradeoffs of a high heat transfer rate versus high temperature storage.

The natural next step is ground coupled storage devices involving a tank <u>and</u> serpentine pipes. The possibilities here are myriad. Again, cost-effectiveness is possible only if the storage component costs can be justified by performance improvements or by cost reductions in non-storage components.

The availability of the ground as a "free" large capacity storage device resurrects one storage strategy discussed earlier in this section - the dual phase system. For example, a system could contain one storage device sized for a heating system and another only large enough to contain the heat rejected by one day of summer cooling. The "hot storage" is charged all year with heat which is stored largely in the ground. The "cool storage" is charged with "cold" during the winter. (The use of ice is possible but additional storage difficulties are involved.) Then, in the summer, the load is cooled and the heat rejected to storage for later rejection to the nocturnal ambient. The ground around this tank acts largely as a buffer to help lower the storage temperature. Research is needed to evaluate attractive configurations. A fundamental performance question is prominent: Is the increased cooling efficiency due to the low "cool storage" temperature worth more than the heat that would otherwise be rejected to storage (in a "single phase" system) for the next winter?

Ground coupled storage for the series solar heat pump system appears very promising. What is needed now are models of the heat transfer processes involved validated against experimental results in a variety of geographical locations and soil types. Then, the performance characteristics and economics of the various configurations above can be evaluated and improved upon.

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X. SOLAR ASSISTED HEAT PUMP COMPUTER SIMULATION STUDY

A. Introduction

We have modeled, using the transient simulation computer program TRNSYS,[1] a solar assisted heat pump system embodying the principles which we have discussed in this paper. This simulation effort had two objectives:

1. to understand the dynamic interactions of collector area and storage volume upon seasonal performance factors of the system in the heating and cooling modes,
2. to provide a basis for an economic comparison of the proposed system with other solar and nonsolar systems for heating and cooling.

Simulations were carried out for New York City and Washington, D.C., with varied storage volume for New York.

B. General System Description

The system as modeled with TRNSYS is shown schematically in Figure X-1. A site-fabricated air-heating collector is applied to the roof, and its inlet and outlet openings are connected via a short length of ductwork in which is incorporated a fan/coil unit. The use of an air-heating collector and a water-source heat pump, both discussed previously, dictates the air-water heat exchange at this point. The short duct length should make placement of the system in a residence easier and should reduce the parasitic power requirement The placement of the heat exchanger inside the insulated portion of the building means that the hydronic portion of the system need contain no antifreeze. A set of airflow-activated louvers in the ductwork will inhibit reverse thermosymphoning when the sun is not shining.

Dampers near the collector inlet and outlet are used to switch from the heating to the cooling mode. In the cooling mode air is drawn in at night from

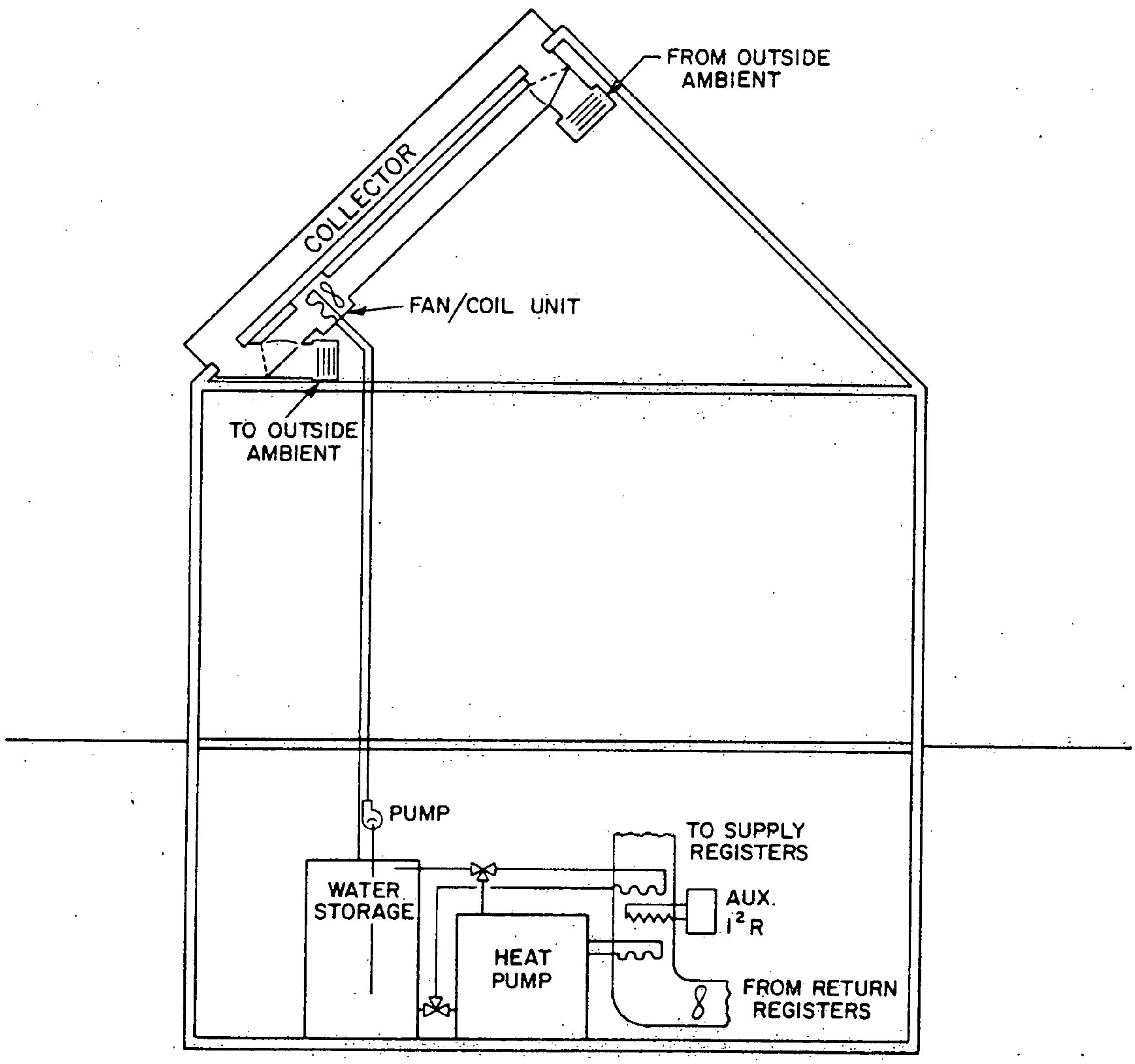

Fig. X-1. Series Solar Assisted Heat Pump System as Modeled with TRNSYS.

the outside, passed through the fan/coil unit, and returned to the outside at a point located well away from the inlet opening. Storage water, into which heat has been rejected by the heat pump during the day, is passed through the coil, giving up its heat to the night air.

In the heating mode, heat can be delivered to the load from storage in two ways, as shown. If the storage temperature exceeds the value needed to heat the house directly, storage water is passed through a heating coil located in the ductwork. If the storage water temperature is less than this value, storage water is passed through a water-to-refrigerant heat exchanger where heat is given up to heat the heat pump evaporator. The storage temperature at which direct heating becomes possible is taken to be 105°F in this simulation.

In the cooling mode, the coil in the ductwork serves as the evaporator, and heat is rejected via the water-refrigerant heat exchanger to storage. The dissipation of heat from storage to the outside ambient at night allows advantage to be taken of the lower nighttime temperature of the ultimate heat sink. With the exception of a pair of dampers and two short lengths of ductwork, all of the necessary apparatus for nocturnal dissipation is the same as that used in the heating mode, resulting in a considerable cost saving.

The heat pump is assumed to have the energy efficiency characteristics of an advanced unit such as is now being developed under programs funded by the U.S. Department of Energy and discussed in section VI of this paper.

The load is a reasonably well insulated one-story house with basement which incurred an annual heating load of 45.8 million Btu in New York City (4854 degree-days) and 34.7 million Btu in Washington, D.C. (4192 degree days). The cooling loads were 7.6 million Btu in New York and 12.4 million in Washington. These loads were generated by subroutines in TRNSYS which utilize the ASHRAE transfer-function method[2a] of calculating conduction heat gains and losses and the effective sol-air temperature method[2b] of accounting for heat gains due to direct insolation. The latent portion of the cooling load was assumed to equal 30% of the calculated sensible load, in accordance with ASHRAE recommendations.[2c]

Weather data were obtained from the Syracuse University generalized weather functions[3] which we have incorporated into our copy of the TRNSYS code. These functions give temperature and insolation values based on long-term averages of weather conditions for thirteen U.S. cities.

As discussed in section VII, the collectors can be used in the summer to heat domestic hot water, and this use can be extended to late spring and early autumn when the heating loads are small. A separate set of high-temperature panels could be included in the system to heat water in the winter, but as these would give needed energy only half the year (being superfluous the other half) they would probably not be cost effective. Hot water heating is not shown in Figure X-1, but a pair of valves, a run of coiled pipe through the hot water tanks, and an addition to the control system to switch the dampers from day to night operation would be the only additional equipment needed. Since the storage water does not pass through the collector and need contain no antifreeze or toxic inhibitors, an isolation heat exchanger ought not be necessary.

In summary, the system as described possesses the following advantages:

1. the use of an advanced type of water-source heat pump having high coefficients of performance and needing no defrost cycle;
2. the use of low temperature simple air-heating collectors which can be constructed at very low cost and easily site-assembled;
3. the achievement of energy-efficient cooling using storage as a heat sink for the advanced heat pump during the day and rejecting the heat

from storage at night when the ambient air is cooler;

4. the low temperature collectors will be suitable for hot water heating during the warm half of the year and for water pre-heating via the heat pump during the cold winter months.

We were not in a position to model the effect of coupling the thermal storage to the ground, although we anticipate that this step can result in a significant improvement in system performance with a corresponding reduction in collector area. Ground-coupling will be a major area for further work and is discussed in section IX.

C. Component Parameters

In this section we list the more significant component parameters and their values which served to define the characteristics of the system being simulated.

1. Collector.

The transmissivity, absorbtivity, collector efficiency factor, and collector heat loss coefficient were given the values $\tau = 0.90$, $\alpha = 0.95$, $F' = 0.877$, $U_L = 1.425$ Btu/ft^2-hr-^{o}F, respectively, resulting in a collector efficiency equal to 0.75 at $\Delta T/I = 0$ and dropping to zero at $\Delta T/I = 0.6$ ft^2-hr-^{o}F/Btu, where ΔT is the average collector air temperature minus the ambient temperature, and I is the insolation rate on the collector surface (c.f. Figure VII-1). The relatively high value of F' for an air-heating collector should be achievable at low cost, we believe, by providing a sufficient ratio of absorber surface to glazing surface, as discussed in Refs. 6 and 7 of section VII.

2. Heat Exchanger, Pump, and Fan.

The UA of the heat exchanger was set at 10 Btu/hr-^{o}F per square foot of collector. Water flow rate was 20 lb/hr (0.04 gal/min) per square foot of collector, while air flow rate was 24 lb/hr (5 scfm) per square foot of collector. With the collector operating at 50% efficiency at an insolation rate of 300 Btu/ft^2-hr, these values would result in temperature drop in the air of 25°F, a temperature rise in the water of 7.5°F, and a heat-exchanger effectiveness of 0.73. The airflow rate is somewhat higher than is generally used in air-heating collectors; however, the short duct run with no rockbed is expected to keep the pressue drop and fan power within acceptable limits.

3. Storage

It is our intent to incorporate uninsulated, in-ground storage into the system. However, at the present time we have not had a validated model of in-ground storage to use in our simulation. Instead, we used a simple unstratified storage tank with a surface heat loss coefficient of 0.10 Btu/ft^2-hr-$^{\circ}$F, placed in an environment at 55°F. In our simulation, energy leaving the tank is lost for good, whereas in the actual case much of this energy would be regained at a later time.

The minimum allowable tank temperature was set at 35°F. If the tank temperature reaches this low a temperature, electric resistance heat is used to supply any additional heating load; for this purpose an electric heating element is placed in the air supply duct.

4. Heat Pump

The heart of the system is an advanced heat pump which is now being developed under contracts let by the U.S. Department of Energy resulting from the research proposal competition under RFP EG-77-R-03-1467, "Solar Assisted Heat Pumps Projects for Solar Heating and Cooling Applications".[4] The heat pump was assumed to have an efficiency equal to 68% of the ideal vapor-compression cycle using refrigerant R-12, which is very similar to that of R-22. This is derived from a compressor efficiency of 80% and a motor efficeincy of 85%. The heat pump efficiency modeled here averaged approximately 50% of Carnot, whereas informed estimates have indicated that 60 to 65% of Carnot can be obtained with development effort. The rate of heat transfer through the condenser was limited to a maximum of 36,000 Btu/hr to avoid unrealistically high heat transfer rates through the evaporator and condenser. This artificially constrained the coefficient of performance of the heat pump to be no greater than 9.7. This maximum is reached at source temperatures above 85°F.

It was assumed that provision for dual capacity would be needed to achieve the high coefficients of performance which we have discussed. This assumption was incorporated into the data used in TRNSYS to model the heat pump.

The TRNSYS heat pump subroutine requires that the heat absorbed by the evaporator, the heat rejected by the condenser, and the work input to the compressor be specified for each of a set of evenly spaced evaporator temperatures, in the heating mode, and for each of a set of evenly spaced condenser temperatures, in the cooling mode. Condenser temperature is held fixed at 115°F in the heating mode; evaporator temperature is fixed at 45°F in the cooling mode.

It was assumed that 80% of the compressor work was converted into enthalpy, and that the temperature "splits" between water and refrigerant in the evaporator and condenser were 5°F.

Tables X-1 and X-2 show the heat pump data which were used in our TRNSYS simulation runs. The heat pump capacity and COP are plotted in Figures X-2 and X-3 for the heating and cooling modes. Figure X-2 should be compared with Figure III-1, which gives COP's obtainable from currently available air-to-air units. The losses in the electric motor are not reflected in columns 3, 4, and 5, which concern the vapor compression cycle only, but these losses are included in the electrical energy to the compressor in Tables X-6 and X-7. Therefore, the

Table X-1

Heat Pump Performance Data
Used in TRNSYS, Heating Mode

Source Temp. °F	Evaporator Temp. °F	Heat Absorbed Btuh	Heat Rejected Btuh	Compressor Work Btuh	COP Heat Pump* (Includes 85% motor efficiency)	COP VC Cycle (Does not include motor losses)	COP Ideal Vapor Compression Cycle	COP Carnot
35	30	19268	24338	6338	3.3	3.84	4.8	6.8
40	35	20789	25859	6338	3.5	4.18	5.1	7.2
45	40	22817	27887	6338	3.7	4.40	5.5	7.7
50	45	24845	29915	6338	4.0	4.72	5.9	8.2
55	50	27888	32958	6338	4.4	5.20	6.5	8.8
60	55	30930	36000	6338	4.8	5.68	7.1	9.6
65	60	17747	20282	3169	5.4	6.40	8.0	10.4
70	65	20242	22817	3169	6.1	7.20	9.0	11.5
75	70	23324	25859	3169	6.9	8.10	10.2	12.8
80	75	26873	29408	3169	7.9	9.28	11.6	14.4
85	80	32704	35239	3169	9.5	11.12	13.6	16.4
90	85	33465	36000	3169	9.7	11.36	16.5	19.2
95	90	33465	36000	3169	9.7	11.36	20.2	23.0
100	95	33465	36000	3169	9.7	11.36	25.9	28.8
105	100	33465	36000	3169	9.7	11.36	35.3	38.3

*Values in this column used in computer simulation.

Table X-2

Heat Pump Performance Data
Used in TRNSYS, Cooling Mode

Sink Temp. °F	Condenser Temp. °F	Heat Absorbed Btuh	Heat Rejected Btuh	Compressor Work Btuh	COP Heat Pump* (Includes 85% motor efficiency)	COP VC Cycle (Does not include motor losses)	COP Ideal Vapor Compression Cycle	COP Carnot
55	60	33465	36000	3169	9.0	10.6	27.6	33.7
60	65	33465	36000	3169	9.0	10.6	20.2	25.2
65	70	33465	36000	3169	9.0	10.6	16.2	20.2
70	75	33211	35746	3169	8.9	10.5	13.1	16.8
75	80	28648	31183	3169	7.7	9.0	11.3	14.4
80	85	24845	27380	3169	6.7	7.8	9.8	12.6
85	90	21296	23840	3169	5.7	6.7	8.4	11.2
90	95	13253	20788	3169	4.9	5.8	7.2	10.1
95	100	31943	37013	6338	4.3	5.0	6.3	9.4
100	105	38901	33971	6338	3.9	4.6	5.7	8.4
105	110	25859	30929	6338	3.5	4.1	5.1	7.8
110	115	23831	28901	6338	3.2	3.8	4.7	7.2

*Values in this column used in computer simulation.

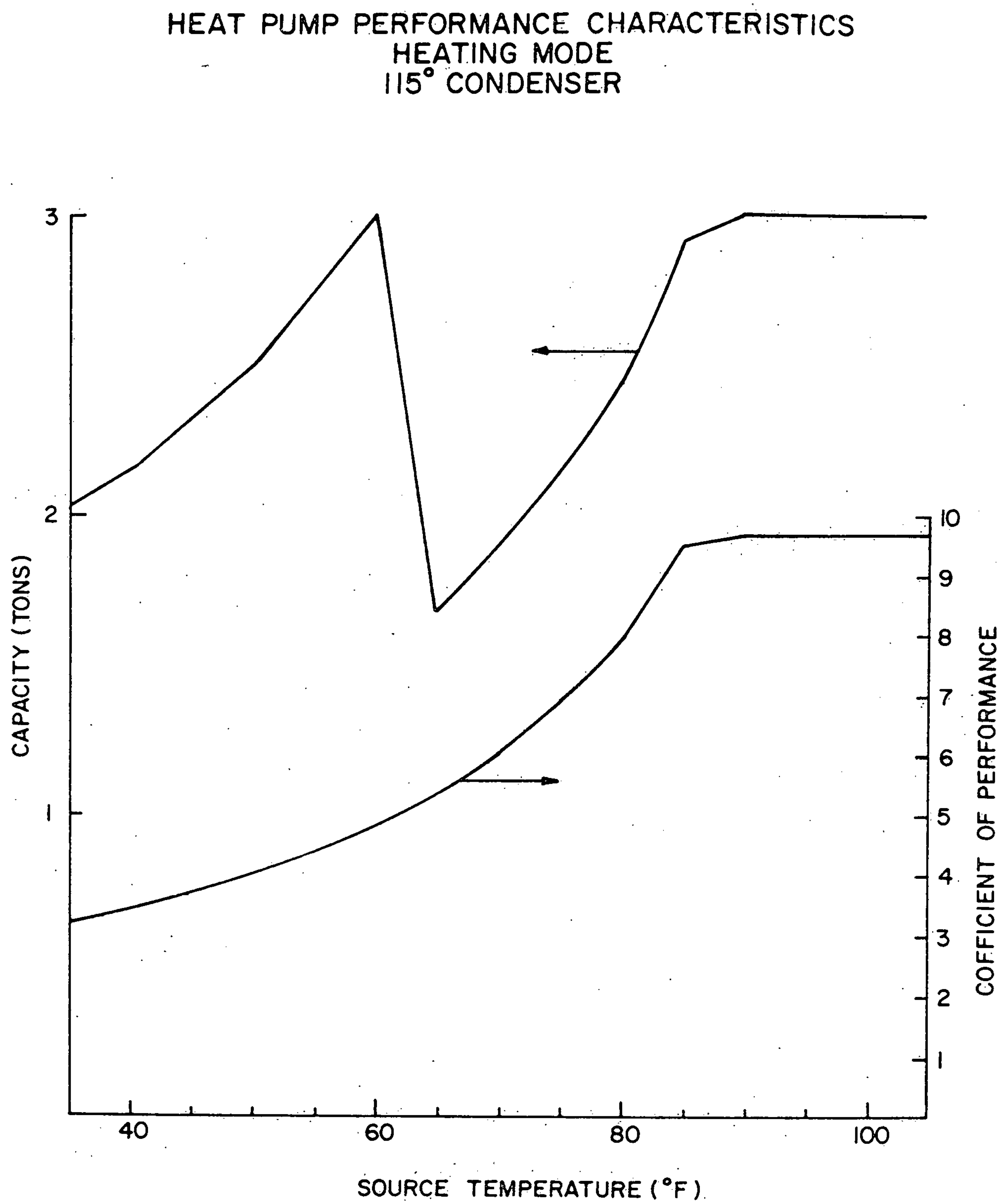

Fig. X-2. Capacity and Coefficient of Performance of the Heat Pump in the Heating Mode.

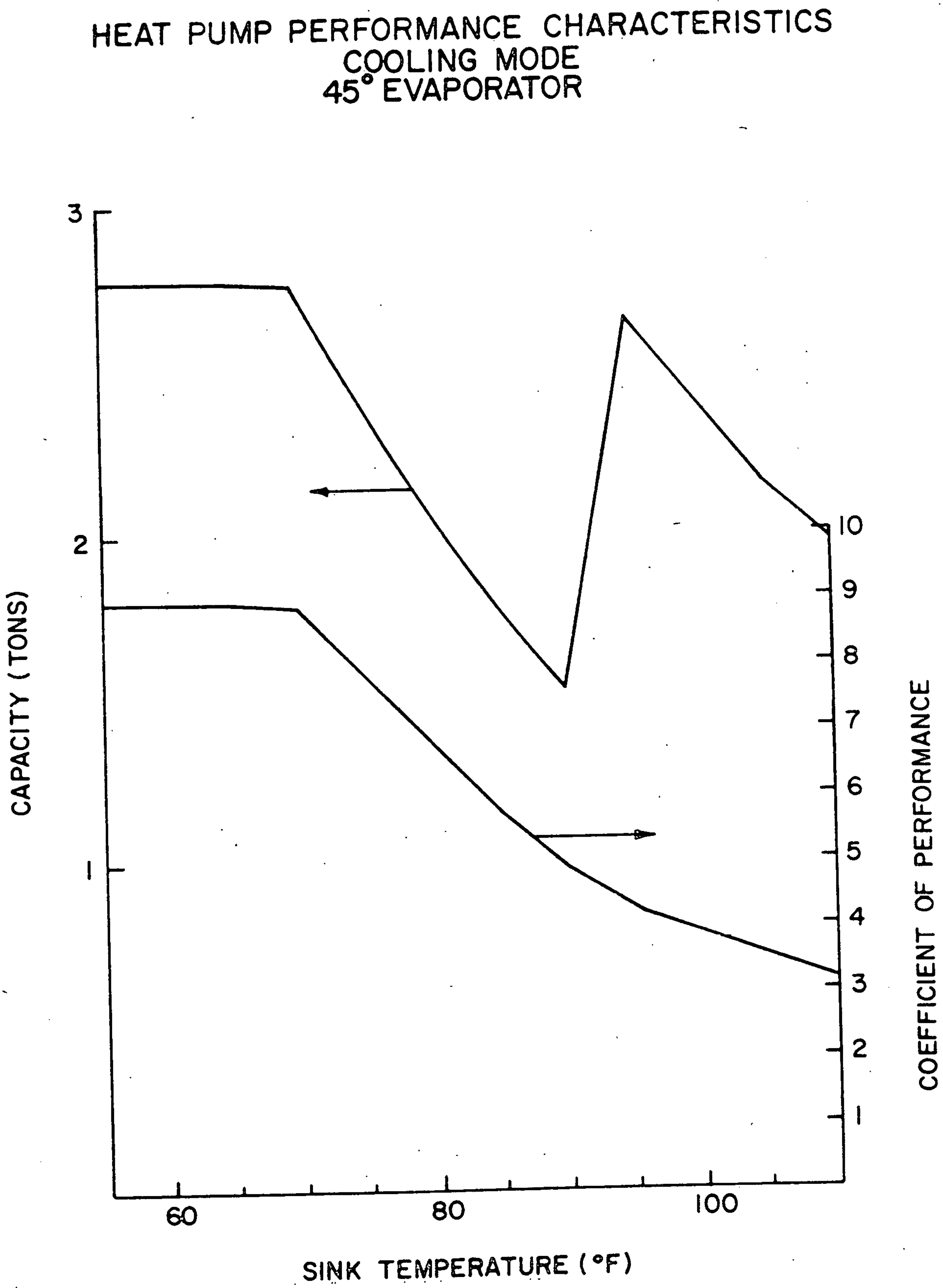

Fig. X-3. Capacity and Coefficient of Performance of the Heat Pump in the Cooling Mode.

relevant coefficient of performance for this analysis is that shown in column 6. Columns 7, 8, and 9 are shown for purposes of comparison only. The transition from the higher capacity to the lower capacity occurs in the 55 to 60°F evaporator temperature interval in the heating mode, and in the 95 to 100°F condenser temperature interval in the cooling mode. The capacity of the machine in the cooling mode varies from 1½ tons to 2-3/4 tons, and so it appears reasonable to characterize this heat pump as a 2-ton machine.

5. Load

The load was simulated using subroutines TYPE 17 (Walls), TYPE 18 (Pitched Roof and Attic), and TYPE 19 (Room and Basement) in TRNSYS. For the walls, we used the transfer function coefficients of ASHRAE wall number 36,[2d] with 3 in. of an insulating material having a thermal conductivity K of 0.025 Btu/ft-hr-°F. The thermal conductance or U-value of this wall is 0.081 Btu/ft^2-hr-°F. For comparison, ASHRAE Standard 90-75[5] specifies a maximum U-value of 0.23 Btu/ft^2-hr-°F for detached residential construction in a 5000 degree-day environment. The ceiling contained 6 in. of insulation, as specified through TRNSYS subroutine TYPE 18. The total wall area was 1420 ft^2, while the ceiling and floor areas were 1425 ft^2 each. Window area was 106 ft^2 on the south and 85 ft^2 on the other three sides. Infiltration of outside air occurred at the rate of one air change per hour. Internal heat was assumed to be generated at an average rate of 500 watts, in addition to the heat generated by an average occupancy of two people. This would correspond to a family of four, each of whom is in the house half the time, on average. The latent load for cooling was assumed to equal 30% of the calculated sensible load, as discussed previously in this section.

D. Simulation Results

The collector areas and storage volumes studied are shown in Table X-3. Each system was simulated for a period of one year, with a 15-minute timestep. As a consistency check, energy balances on the storage tank and on the load were computed as shown in Tables X-4 and X-5 for the heating mode. For the cooling mode, energy balances are shown in Table X-6. As can be seen, the imbalances on the storage tank averaged 1.1% for heating and 2.1% for cooling. Energy imbalances on the heating load averaged less than 0.1% while the cooling load balances were exact.

The results of the simulations are shown in Figures X-4 through X-12. For each simulation we show, at five-day intervals, the heating or cooling load, the average storage tank temperature, the average collector efficiency, and the average ambient temperature.

The tank temperature is started at 70°F at the beginning of the heating season, which is taken as September 21 in New York and September 26 in Washington. The tank temperature rises rapidly, and as it does so the collector efficiency declines due to the higher operating temperatures to which the collector is subjected. As the heating load becomes significant, the tank temperature begins to fall, and the collector efficiency rises, so that by midwinter the tank temperature has fallen to its minimum value and the collector efficiency has risen to a maximum. Thus, we see one of the strengths of the system, that the collectors are most efficient when high efficiency is most needed. Note that the efficiencies shown are average efficiencies; maximum collector efficiencies would be higher.

As winter moves on into early spring, the storage temperature rises once again. At a time when the heating load has dropped almost to zero (April 29

Table X-3

Collector Areas and Storage Volumes Simulated in New York City and Washington, D.C.

Storage Volume (ft^3) \ Collector Area (ft^2)	225	333	500	750
200	WASH	NY WASH	NY WASH	NY
450		NY	NY	NY

Table X-4

Energy Balances on the Storage Tank during the Heating Season (All Energies in Millions of Btu)

City, Collector Area, Storage Volume	Energy Supplied by Collector =	Energy Delivered to Direct Heat Coil +	Energy Delivered to Heat Pump +	Energy Lost to Environment +	Increase in Internal Energy of Tank	RHS Total
NY, 333 ft^2, 200 ft^3	36.43	1.86	31.39	2.04	0.67	35.96
NY, 333 ft^2, 450 ft^3	38.69	1.45	33.32	2.74	1.24	38.75
NY, 500 ft^2, 200 ft^3	43.69	3.44	35.28	3.31	0.79	42.82
NY, 500 ft^2, 450 ft^3	46.16	2.74	36.75	4.88	1.62	45.99
NY, 750 ft^2, 200 ft^3	47.61	6.03	35.09	4.79	0.91	46.82
NY, 750 ft^2, 450 ft^3	50.90	4.87	36.27	7.23	1.80	50.17
WASH, 225 ft^2, 200 ft^3	29.77	1.95	24.84	2.18	0.75	29.72
WASH, 333 ft^2, 200 ft^3	35.19	3.70	26.65	3.66	0.89	34.90
WASH, 500 ft^2, 200 ft^3	38.78	7.05	24.81	5.32	1.00	38.18

Table X-5

Energy Balances on the Heating Load
(All Energies in Millions of Btu)

City, Collector Area, Storage Volume	Heating Load =	Direct Heating from Storage +	Heat Pump Heating +	Auxiliary Heat	RHS Total
NY, 333 ft^2, 200 ft^3	45.74	1.86	37.55	6.32	45.73
NY, 333 ft^2, 450 ft^3	45.75	1.45	39.96	4.33	45.74
NY, 500 ft^2, 200 ft^3	45.72	3.44	41.32	0.94	45.70
NY, 500 ft^2, 450 ft^3	45.72	2.74	42.97	0.	45.71
NY, 750 ft^2, 200 ft^3	45.68	6.03	39.65	0.	45.68
NY, 750 ft^2, 450 ft^3	45.69	4.87	40.81	0.	45.69
WASH, 225 ft^2, 200 ft^3	34.61	1.92	29.66	3.00	34.61
WASH, 333 ft^2, 200 ft^3	34.58	3.70	30.88	0.	34.58
WASH, 500 ft^2, 200 ft^3	34.54	7.05	27.49	0.	34.54

Table X-6

Energy Balances on the Storage Tank
during the Cooling Mode
(All Energies in Millions of Btu)

City, Collector Area, Storage Volume	Energy Rejected by Heat Pump =	Energy Rejected to Outside Ambient +	Energy Lost to Environment +	Increase in Internal Energy of Tank	RHS Total
NY, 333 ft^2, 200 ft^3	8.32	7.20	1.08	0.13	8.41
NY, 333 ft^2, 450 ft^3	8.55	6.40	1.84	0.15	8.39
NY, 500 ft^2, 200 ft^3	8.28	7.12	1.02	0.13	8.27
NY, 500 ft^2, 450 ft^3	8.29	6.52	1.73	0.25	8.50
NY, 750 ft^2, 200 ft^3	8.28	6.85	1.01	0.11	7.97
NY, 750 ft^2, 450 ft^3	8.28	6.56	1.66	0.22	8.44
WASH, 225 ft^2, 200 ft^3	13.71	12.68	1.48	0.10	14.26
WASH, 333 ft^2, 200 ft^3	13.62	12.49	1.36	0.11	13.96
WASH, 500 ft^2, 200 ft^3	13.60	12.30	1.30	0.12	13.72

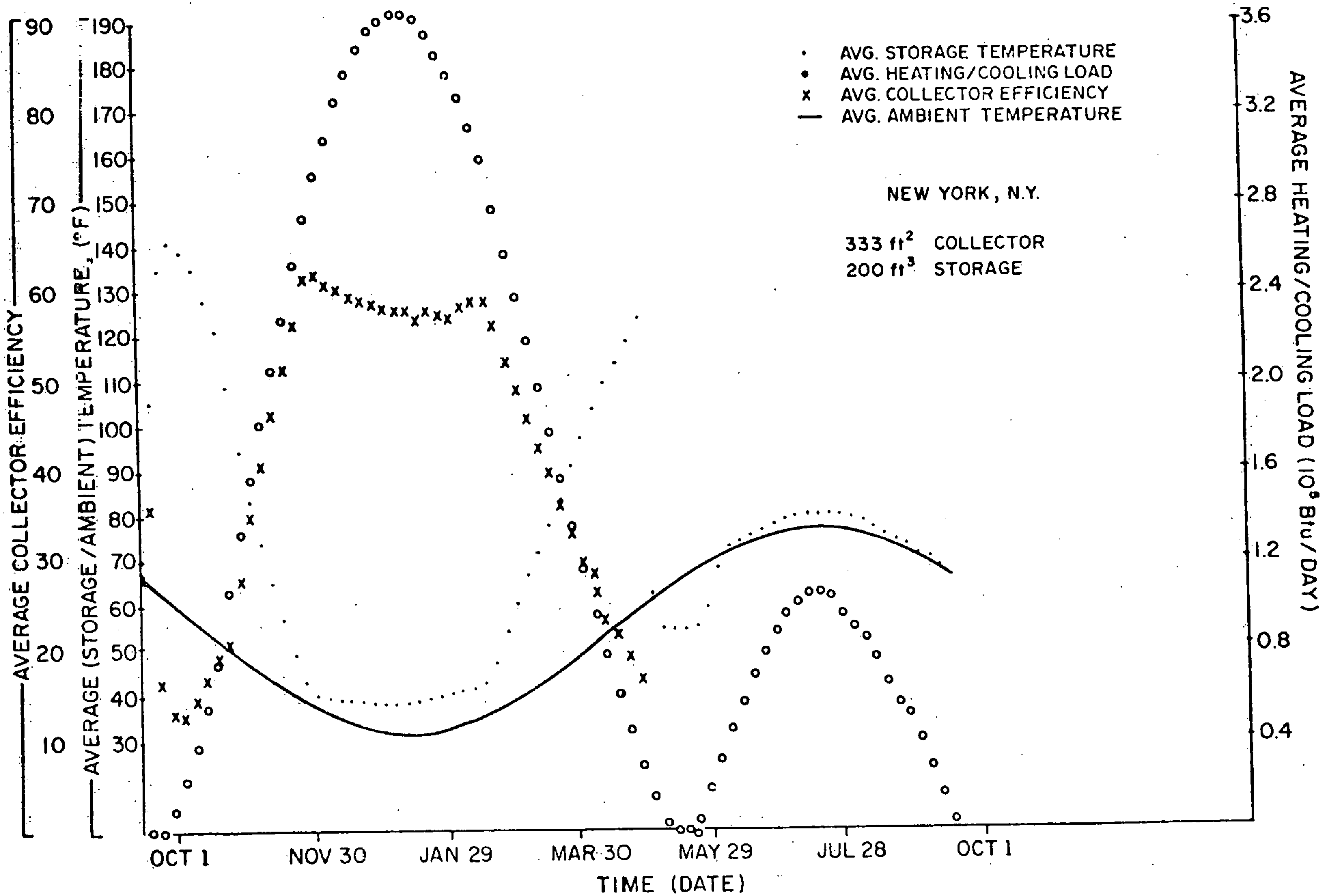

Fig. X-4. Simulation Results for New York City, 333 ft^2 Collector Area, 200 ft^3 Storage Volume.

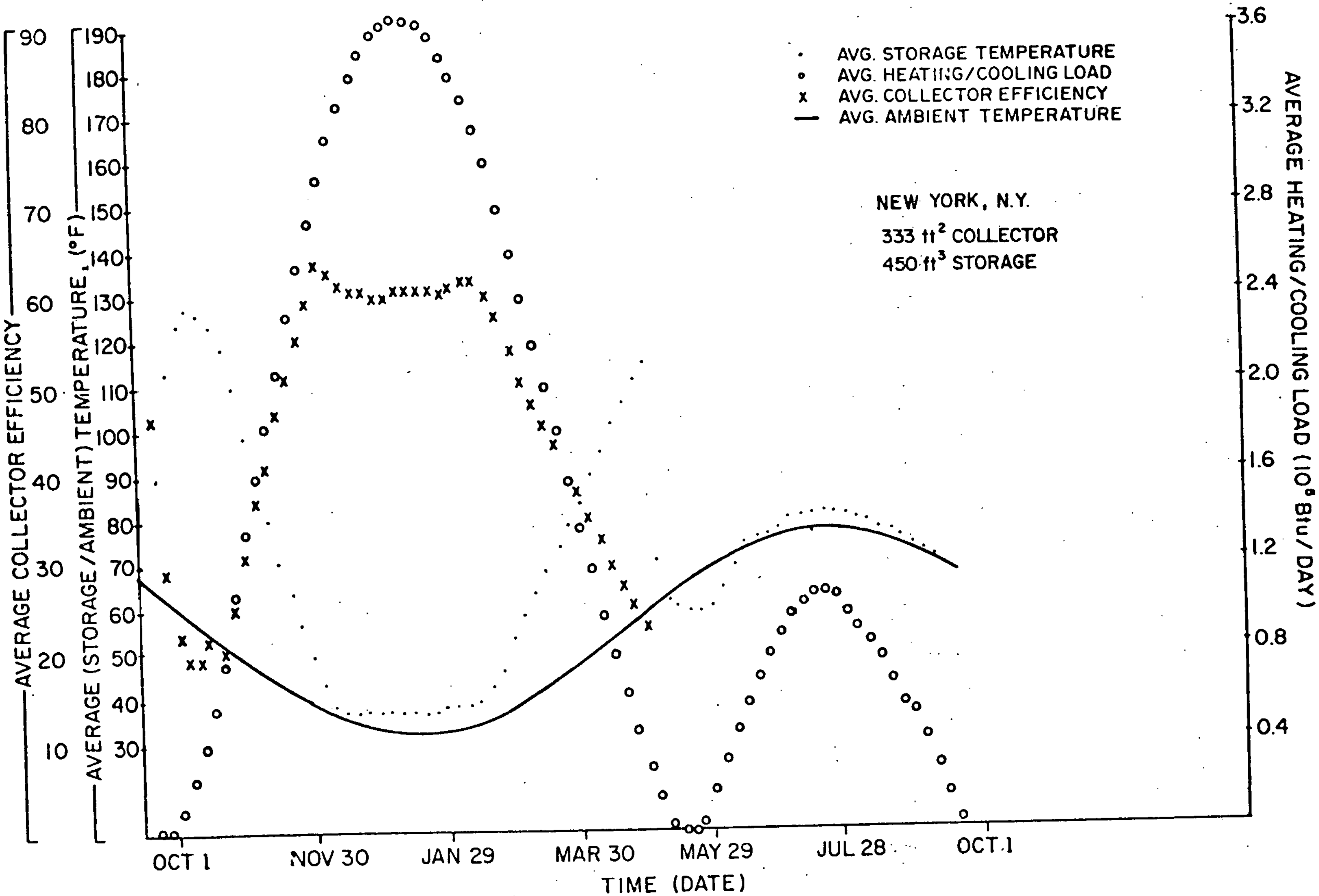

Fig. X-5. Simulation Results for New York City, 333 ft^2 Collector Area, 450 ft^3 Storage Volume.

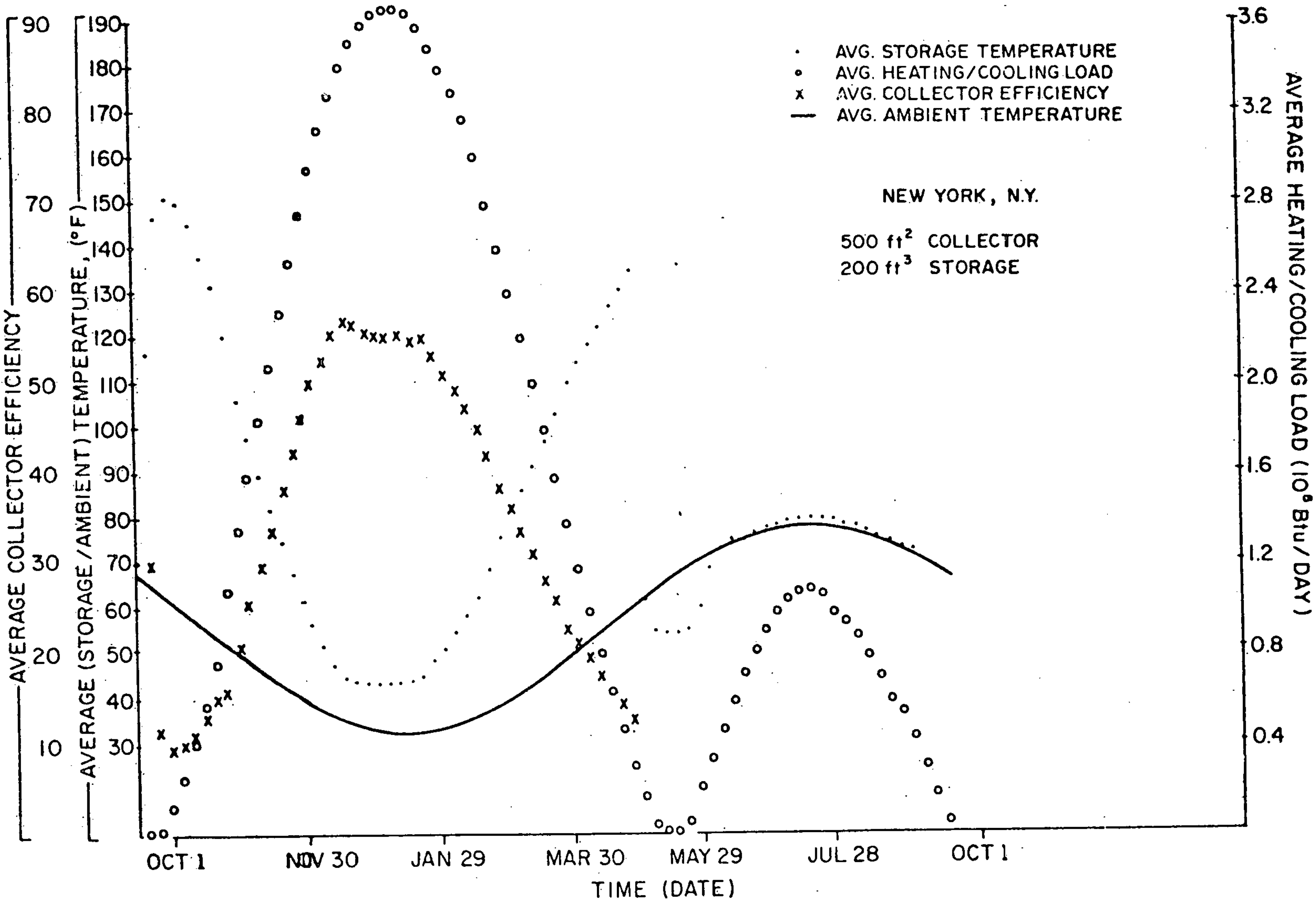

Fig. X-6. Simulation Results for New York City, 500 ft^2 Collector Area, 200 ft^3 Storage Volume.

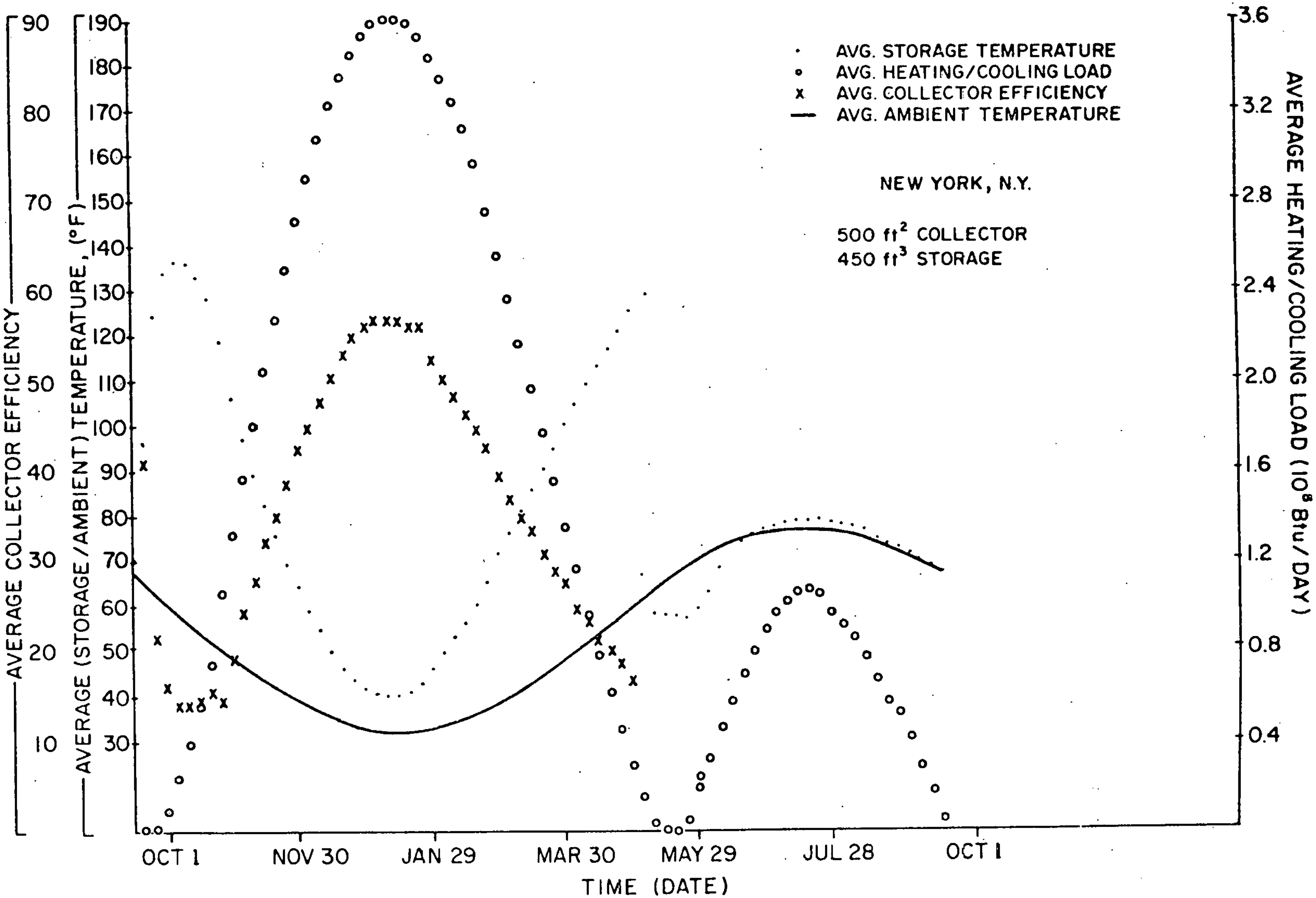

Fig. X-7. Simulation Results for New York City, 500 ft^2 Collector Area, 450 ft^3 Storage Volume.

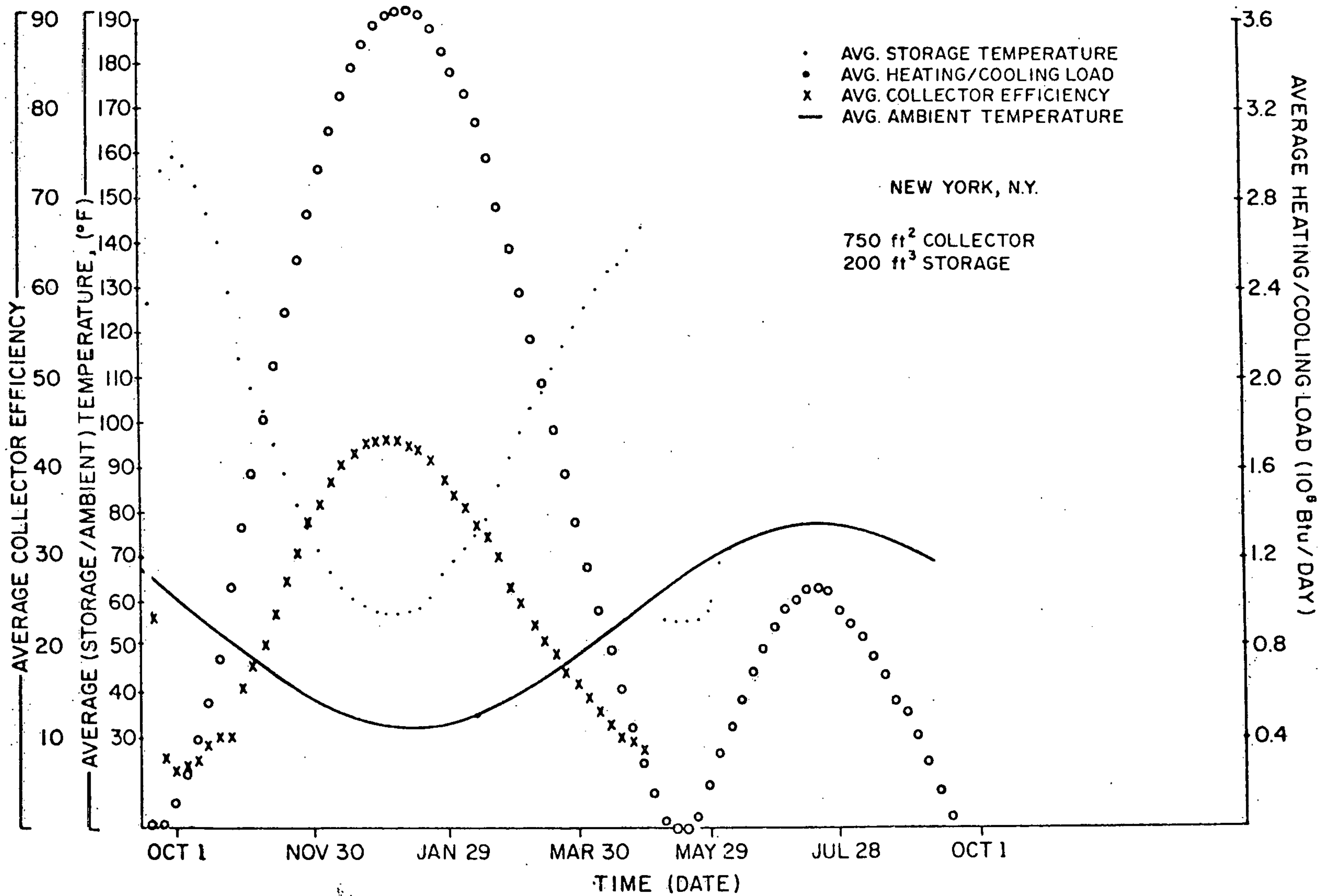

Fig. X-8. Simulation Results for New York City, 750 ft^2 Collector Area, 200 ft^3 Storage Volume.

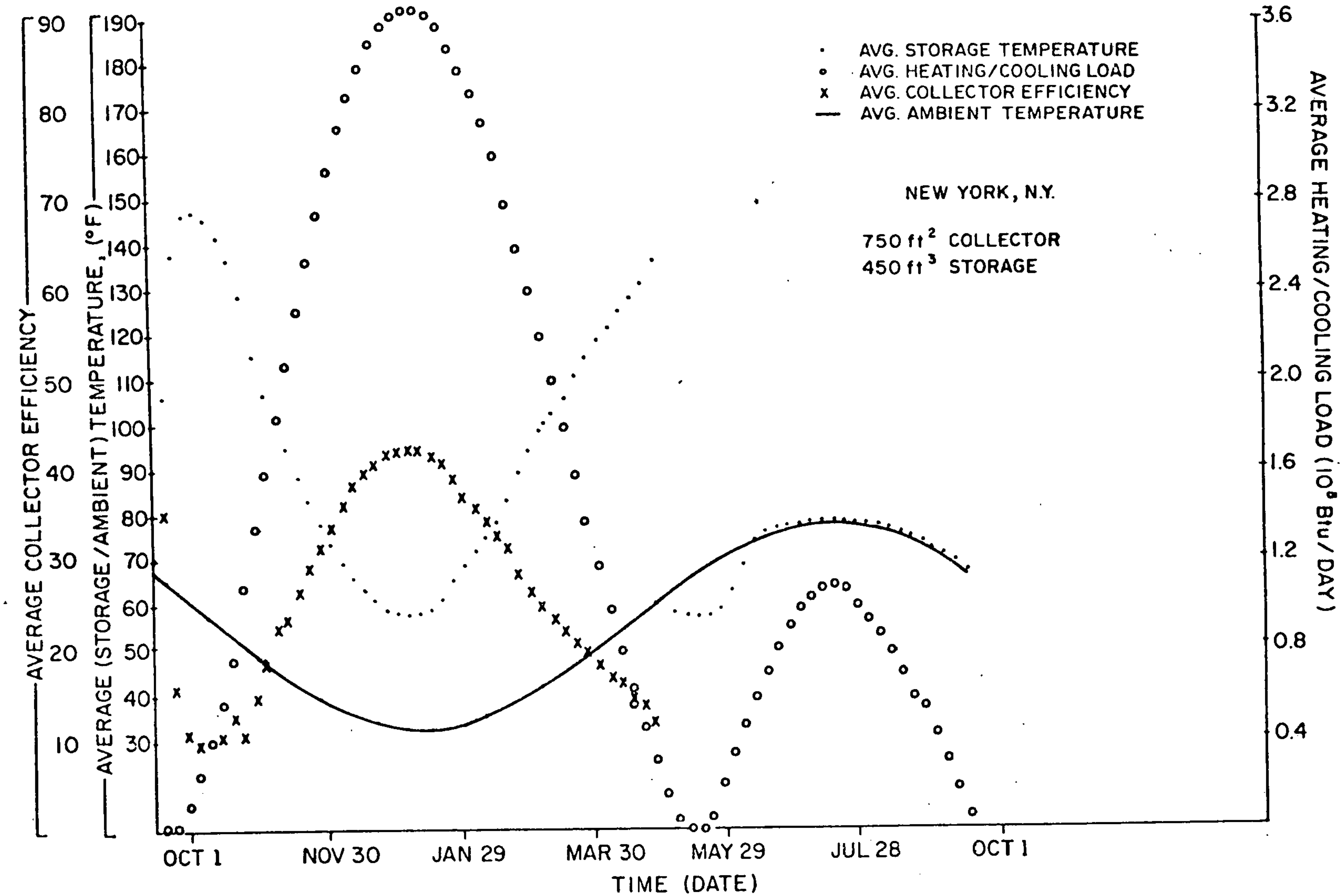

Fig. X-9. Simulation Results for New York City, 750 ft^2 Collector Area, 450 ft^3 Storage Volume.

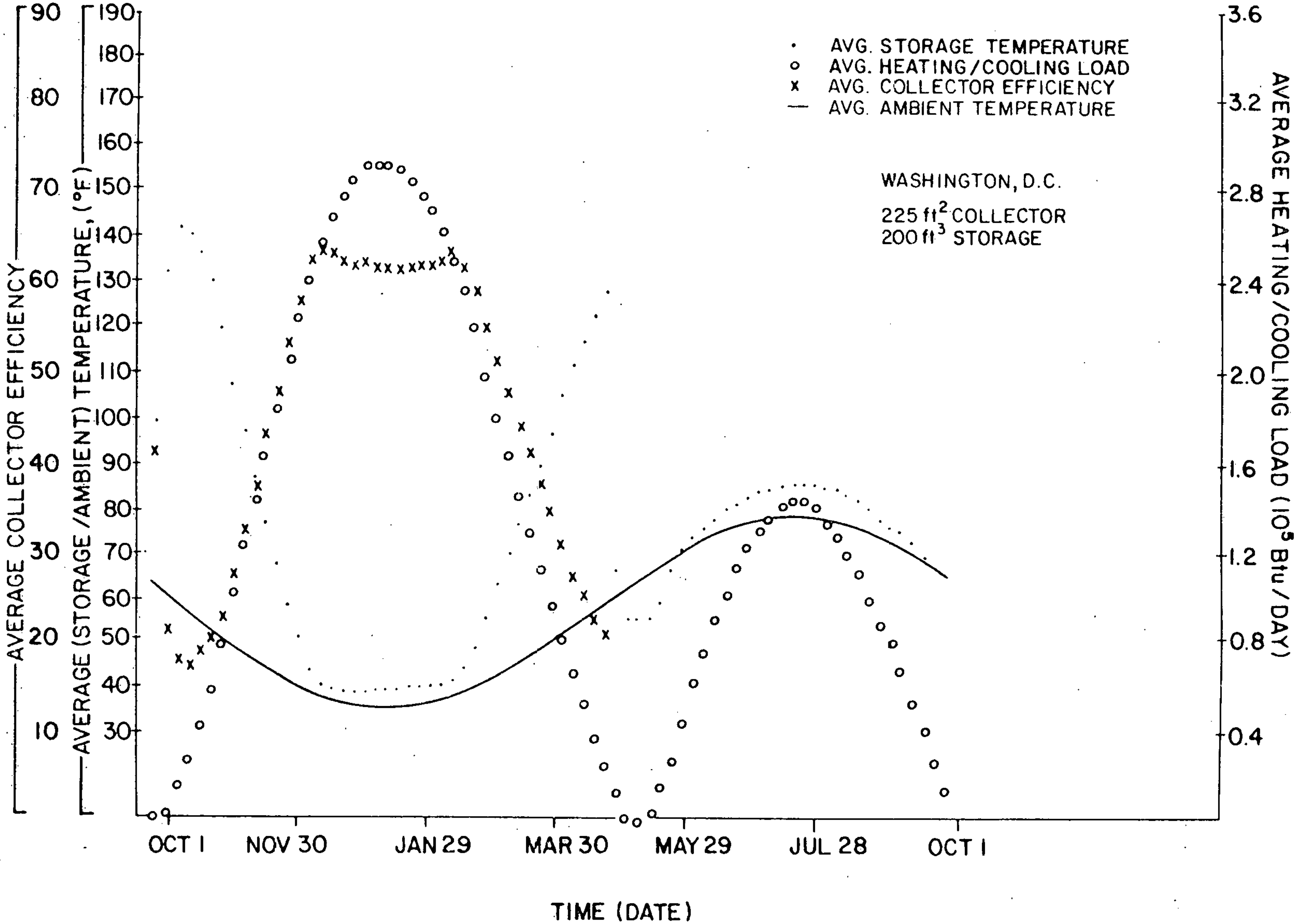

Fig. X-10. Simulation Results for Washington, D.C., 225 ft^2 Collector Area, 200 ft^3 Storage Volume.

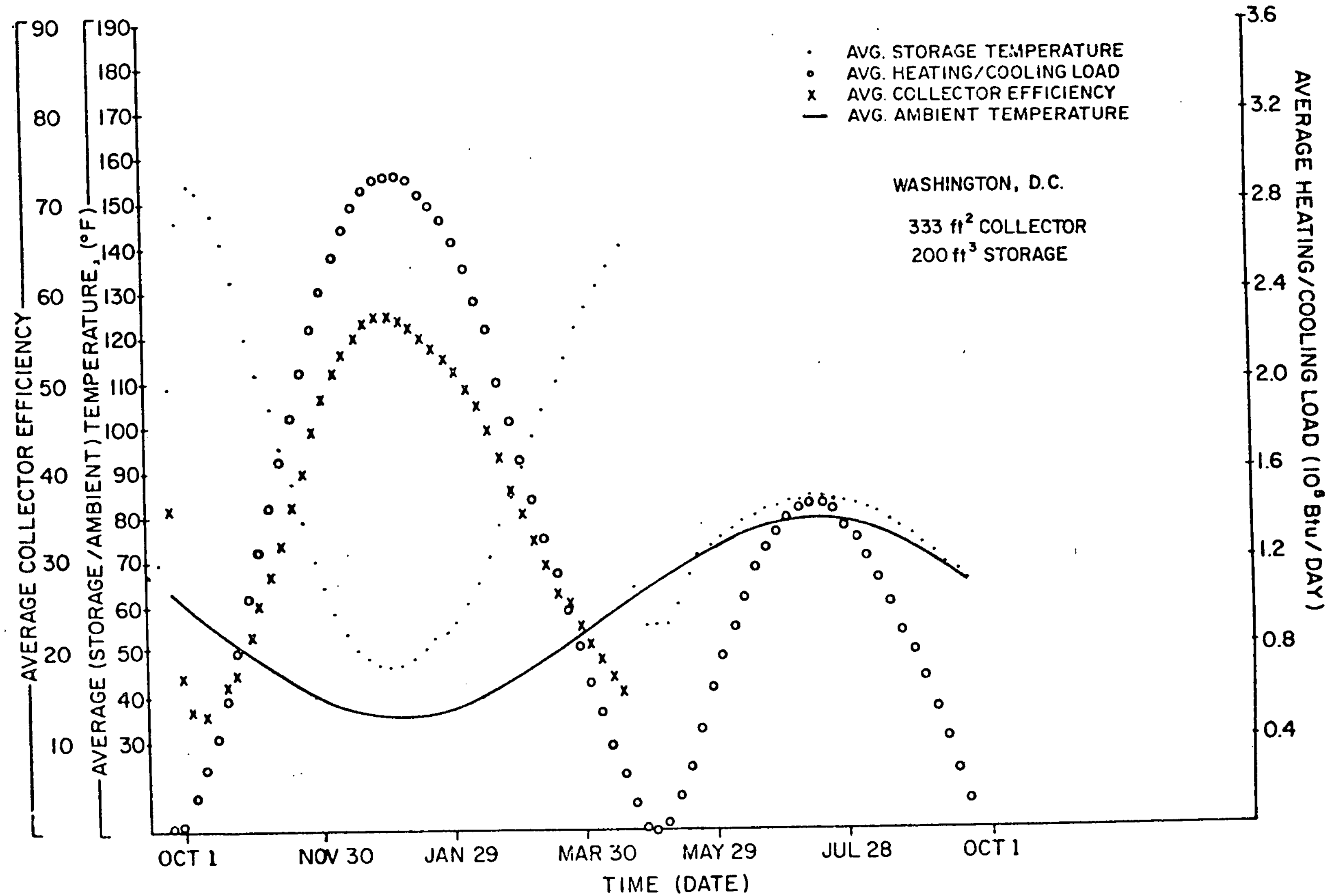

Fig. X-11. Simulation Results for Washington, D.C., 333 ft^2 Collector Area, 200 ft^3 Storage Volume.

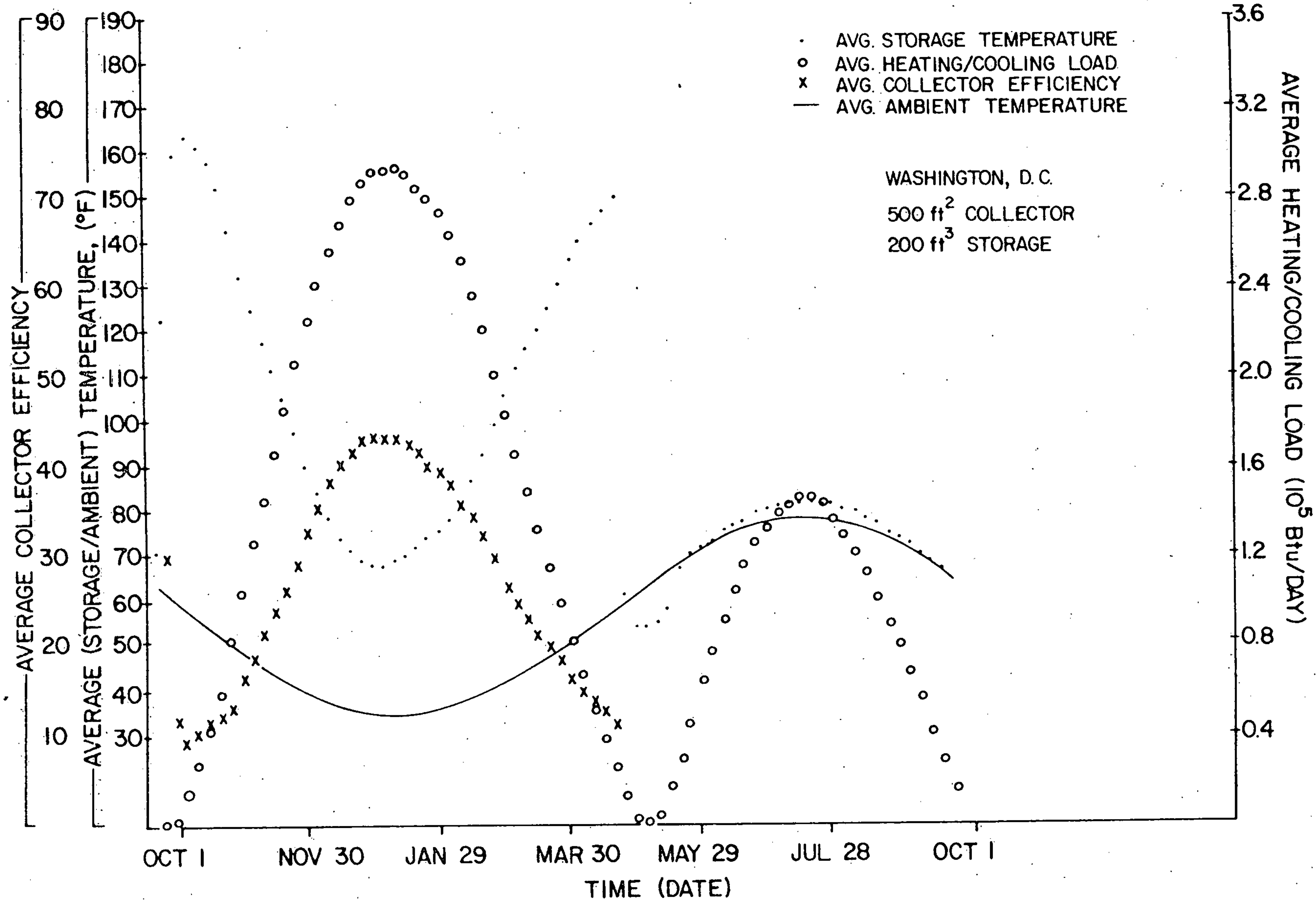

Fig. X-12. Simulation Results for Washington, D.C., 500 ft^2 Collector Area, 200 ft^3 Storage Volume.

in New York, April 19 in Washington) the system is switched from the heating mode to the cooling mode. The remaining heat is dissipated from storage and the system is prepared to do cooling. The tank temperature drops rapidly to about 55°F and remains steady for 15 to 20 days until the cooling load becomes significant. Then the average tank temperature rises because for part of the day heat rejected by the heat pump is stored therein.

In addition to these pictures of the system's operating conditions, the simulations provided data from which system performance can be calculated. Table X-7 shows the results used in arriving at the seasonal performance factors for the system in the heating mode, while Table X-8 shows the cooling mode. The system seasonal performance factor (SPF) is the desired heating or cooling effect delivered over the entire season, divided by the total electric energy input, <u>including</u> auxiliary parasitic energy requirements. The pump and fan power auxiliary to solar collector operation was assumed to be ½ horsepower for a 500 ft^2 collector, based on preliminary calculations, and was prorated for other collector sizes. In calculating electric power requirements, motor efficiency was taken to be 85%. With these requirements, auxiliary collector power averaged 5% of the energy delivered to the tank. In addition to the power needed to run the vapor compression cycle, it was assumed that the heat pump would require a blower to move the air through the space to be heated or cooled. It was assumed that a ½ horsepower fan would suffice, with an 85% efficient motor. Finally, when the direct-heating coil was operating, it was assumed that the blower motor consumed electrical energy equal to 5% of the delivered heat.

In comparing our results with those of others, it is important to ask whether all of these auxiliary power requirements have been taken into account.

Table X-7

Seasonal Performance Factors (SPF) in Heating Mode
(All Energies in Millions of Btu)

City Collector Area, Storage Volume	(1) Heating Load	(2) Direct Heating	(3) Heat Pump Heating	(4) Aux. Resistance Heating	(5) Electric Energy to Compressor	(6) Electric Energy to Collector Pump and Fan	(7) Electric Energy to Forced-Air Blower	Heat Pump SPF $\frac{(3)}{(5)}$	System SPF $\frac{(1)}{(4)+(5)+(6)+(7)}$
NY, 333 ft^2, 200 ft^3	45.84	1.86	37.64	6.32	9.08	1.62	2.16	4.15	2.39
NY, 333 ft^2, 450 ft^3	45.84	1.45	40.06	4.33	9.79	1.76	2.30	4.09	2.52
NY, 500 ft^2, 200 ft^3	45.81	3.45	41.42	0.94	8.92	2.20	2.34	4.64	3.18
NY, 500 ft^2, 450 ft^3	45.81	2.74	43.07	0.	9.18	2.29	2.39	4.69	3.31
NY, 750 ft^2, 200 ft^3	45.78	6.03	39.74	0.	6.73	2.93	2.35	5.90	3.81
NY, 750 ft^2, 450 ft^3	45.78	4.87	40.90	0.	6.69	3.00	2.32	6.11	3.81
WASH, 225 ft^2, 200 ft^3	34.68	1.95	29.66	3.00	7.09	1.11	1.73	4.18	2.68
WASH, 333 ft^2, 200 ft^3	34.58	3.70	30.88	0.	6.21	1.45	1.75	4.97	3.67
WASH, 500 ft^2, 200 ft^3	34.61	7.05	27.49	0.	3.94	1.93	1.84	6.98	4.49

Table X-8

Seasonal Performance Factors (SPF) in Cooling Mode
(All Energies in Millions of Btu)

City, Collector Area, Storage Volume	(1) Cooling Load	(2) Electric Energy to Compressor	(3) Electric Energy to Dissipation Pump, Fan	(4) Electric Energy to Forced Air Blower	Heat Pump SPF $\frac{(1)}{(2)}$	System SPF $\frac{(1)}{(2)+(3)+(4)}$
NY, 333 ft^2, 200 ft^3	7.60	1.06	0.57	0.42	7.17	3.71
NY, 333 ft^2, 450 ft^3	7.60	1.05	0.47	0.42	7.24	3.92
NY, 500 ft^2, 200 ft^3	7.60	1.02	0.62	0.41	7.45	3.71
NY, 500 ft^2, 450 ft^3	7.60	1.02	0.57	0.41	7.45	3.80
NY, 750 ft^2, 200 ft^3	7.60	1.00	0.61	0.40	7.60	3.78
NY, 750 ft^2, 450 ft^3	7.60	1.00	0.62	0.40	7.60	3.76
WASH, 225 ft^2, 200 ft^3	12.37	1.98	0.80	0.81	6.25	3.45
WASH, 333 ft^2, 200 ft^3	12.37	1.86	0.93	0.76	6.65	3.48
WASH, 500 ft^2, 200 ft^3	12.37	1.76	1.02	0.71	7.03	3.54

From Table X-7 we see that the seasonal performance factor for heating increases with increasing collector area, as one might expect. The economic optimum collector area turns out to be that which reduces auxiliary resistance heating requirements to a low level. Additional collector area beyond that results in further improvement but with a diminishing rate of return.

The economic optimum system in New York (500 ft^2 collector, 200 ft^3 storage) had a heating seasonal performance factor of 3.18. For Washington (333 ft^2 collector, 200 ft^3 storage) the SPF was 3.67. For comparison, in the Westinghouse-EPRI study[6] of air-to-air heat pump performance, the seasonal performance factors found for Boston, the nearest city to New York in climate of those studied, ranged from 1.34 to 1.75.

System performance was not very much different for the two storage volumes studied in the case of New York. Because it is our intent to study ground-coupled storage in the future, we were content to leave the storage volume somewhat open-ended. In the economic comparisons which follow, the 200 ft^3 storage volume is used.

One of the most evident results is the importance of keeping fan and pump power low. This requirement is one of the prime motivations behind our system with its short duct run.

In the cooling mode, the seasonal performance factor of the system averaged approximately 3.8 in New York and 3.5 in Washington, and was not a strong function of storage volume or collector area. Although the collectors are not used in the cooling mode, the fan/coil unit was sized for the collector, and its size could affect cooling performance. At least for the sizes we have simulated, the smaller unit running a larger number of hours nearly equaled the performance of the larger unit running for shorter periods of time.

The performance in the cooling mode is especially worthy of note. In most areas of the country, electricity is becoming more expensive in summer than in winter, so good performance in cooling will greatly increase the cost-effectiveness of a system. From the utilities' point of view, the system should look especially attractive, since not only is the cooling done efficiently, but part of the energy is used at night. Figured on daytime electricity use only, the average coefficient of performance of the system is 5.3 in New York and 5.0 in Washington.

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6. H. S. Kirschbaum and S. E. Veyo, An Investigation of Methods to Improve Heat Pump Performance and Reliability in a Northern Climate, Electric Power Research Institute Report EPRI EM-319 (Palo Alto, Calif., 1977), pp. 2.1-42.

XI. ECONOMIC COMPARISONS

We have used the results of our simulation study to compare the cost-effectiveness of the series solar assisted heat pump system with four competing systems:

1. a parallel solar/heat pump system;
2. a stand-alone air-to-air heat pump;
3. electric resistance heating with electric air conditioning;
4. oil-fired heating with electric air conditioning.

Domestic hot water was included in each system cost estimate. Table XI-1 shows how each system heats, cools, and makes hot water.

Any economic analysis must make a number of assumptions about the future. The results should be viewed in light of these assumptions as general expectations. Nevertheless, such preliminary estimates of the economic viability of a system are vital, for they serve to weed out systems that are clearly uneconomic and identify those systems that are potentially strong competitors. We have attempted to state clearly all of our assumptions so that the reader may properly evaluate the results.

In order to compare the cost-effectiveness of the five systems, the following general information, applicable to all the systems, was needed:

1. heating, cooling, and domestic hot water loads
2. cost of energy for each form used
3. general inflation rate
4. maintenance cost escalation rate
5. energy cost escalation rate
6. mortgage term and interest rate
7. tax consequences of investment in each of the competing systems

Table XI-1

Systems for Heating, Cooling, and Hot Water Used in Economic Comparison

	System	Heating	Cooling	Hot Water
1.	Series Solar Assisted Heat Pump	Solar Collectors Feed Storage Heat Pump Delivers Heat to Space at Higher Temperature	Heat Pump Rejects Heat to Storage Nocturnal Dissipation from Storage to Ambient	Summer-Solar Collectors Winter-Electric Resistance
2.	Parallel Solar Assisted Heat Pump	Solar System Supplies 50% of Heating Load Heat Pump Supplies Other 50% from Outside Ambient	Heat Pump Rejects Heat Directly to Ambient	Solar System Supplies 50% of Domestic Hot Water Electric Resistance Supplies the Remainder
3.	Stand-Alone Heat Pump	Heat Pump Supplies Heat from Ambient	Heat Pump Rejects Heat Directly to Ambient	Electric Resistance
4.	Electric Resistance	Electric Resistance	Electric Air Conditioner	Electric Resistance
5.	Oil Furnace	Oil Furnace	Electric Air Conditioner	Oil Furnace

In addition to this general information, applicable to all the systems, the following system-specific information was required:

1. system first cost
2. system seasonal performance factors
3. initial annual maintenance cost
4. initial annual energy cost

The period of the comparison was taken to begin in 1981, which is the estimated year in which the series solar assisted heat pump will become available on the market.

The heating load used in this analysis was 45.8 million Btu per season for New York and 34.6 million Btu for Washington. A domestic hot water load of 10 million Btu was assumed in all cases. This would provide approximately 50 gallons per day of 130°F water.

The current cost of electric energy in Washington, D.C. was determined from published rates of the Potomac Electric Power Company. Winter-summer price differentials were included, as was the fuel cost adjustment. For New York, the area served by Long Island Lighting Company was selected. LILCO's rates are closer to the average for New York and New Jersey than are those of Consolidated Edison Company, which serves most of New York City. Table X1-2 shows the winter and summer rates for these cities. It was assumed that the same electric rates would be in effect for all systems, in the belief that electric resistance heating and all heat pumps will be treated on an equal basis by 1981 as part of a national energy conservation policy.

The price of oil in each city was set at 48 cents per gallon in 1977. At an energy content of 138,000 Btu/gallon, this is equivalent to a price of \$3.48 per million Btu.

Table XI-2

Electric Energy Prices for Selected Locations

New York (LILCO)	Basic Rate ¢/KWh	Fuel Cost Adjustment ¢/KWh	Total ¢/KWh	Total (1977) $\$/10^6$ Btu	Total (1981) $\$/10^6$ Btu
Summer (4 months)	5.12	0.98	6.10	17.89	26.19
Winter (8 months)	2.92	0.82	3.74	10.97	16.06
Hot Water*				11.45	16.76
Washington (PEPCO)					
Summer (5 months)	3.67	1.83	5.50	16.13	23.62
Winter (7 months)	1.82	1.26	3.08	9.03	13.22
Hot Water*				9.73	14.25

*Based on year round use of electric resistance for hot-water heating.

The general inflation rate was taken as 6% annually. Maintenance costs were assumed to escalate at the same 6% rate, while the energy price escalation rate was set at 10% annually. The last column of Table XI-2 reflects this escalation. The price of oil, $3.48 per million Btu in 1977, increases to $5.10 per million Btu in 1981. The initial annual maintenance cost of each system was assumed to equal 2% of that system's first cost. These assumptions, exhibited in Table XI-3, are the same as those used by the U.S. Energy Research and Development Administration in a recent report.[1]

We took the rate of interest on a loan to cover the first cost of each system as 9% annually, and we considered both 10 and 20 year repayment terms.

The tax consequences of investment in a heating and cooling system stem from two sources. First, the interest on the loan taken out to finance any system is tax-deductible. We assumed that the taxpayer was in a 30% tax bracket in determining this benefit. The second source is the proposed tax credit on solar systems of 30% of the first $2000 and 20% of the next $8000. We assumed that this benefit will be available by 1981. We have assumed that legislation exempting solar systems from real estate taxation will be in effect.

A. Costs of the Series Solar Assisted Heat Pump System

The first cost of the series system has been estimated and is summarized by component in Table XI-4. The system for distributing thermal energy through the house was assumed to be the same for all systems, and this common first cost was not included in the analysis.

The price of the needed advanced concept heat pump was estimated by adding 30% to the current price of a two-ton water chiller, and moving the price to 1981 at a 6% inflation rate. The 30% was included to account for additional manufacturing costs incurred in producing this heat pump. Development costs

Table XI-3

Economic Assumptions in SAHP Analysis

General Inflation Rate	6%
Maintenance Cost Escalation Rate	6%
Energy Cost Escalation Rate	10%
Mortgage Interest Rate	9%
Initial Annual Maintenance Cost as Precent of System First Cost	2%
Amortization Terms	10 and 20 years
Consumer Tax Bracket	30%
Solar Incentive Tax Credit	30% of first \$2000 20% of next \$8000

Table XI-4

1981 Component Cost Estimates for
Series Solar Assisted Heat Pump System

		New York (500 ft^2 collector)	Washington (333 ft^2 collector)
1.	Heat Pump	\$2625	\$2625
2.	Collector	2500	1665
3.	Storage (1600 gal)	505	505
4.	Other*	1010	912
	Total First Cost	6640	5707
	Tax Credit	1528	1341
	Net First Cost	5112	4366

*Collector duct run, fan/coil unit, controller, pump, piping.

will have been paid largely by the Federal Government. The resulting estimate for the 1981 price of this advanced heat pump is \$2625, as shown in the table.

The low-temperature site-built collector has been priced at \$5/ft^2 installed in 1981, as discussed in section VII.

The storage element will eventually be of the uninsulated, in-ground variety, and we have estimated that the price of this type of storage may approach as low a value as 10¢/gallon (1977 prices), as discussed in section VIII. This estimate is based on the use of technology now employed in the cesspool industry. We have used 25¢/gallon in this analysis, so 200 ft^3 (1600 gallons) of storage is priced at \$400 in 1977 (\$505 in 1981).

The remaining equipment includes the fan/coil unit, the short length of collector ductwork with two dampers, the controller pump, valves, and connecting piping. These were estimated to cost \$800 installed (\$1010 in 1981) based on the 500 square feet of collector found to be optimum in New York. For other collector areas this cost was adjusted to take into account the variation in some component sizes with collector area. The resulting total first cost of the series system sized for New York is then \$6640 in 1981. The net first cost, after the tax credit, is \$5112. For Washington, D.C., the economic optimum collector area was found to be 333 ft^2, and the resulting first cost was \$5707 before the tax credit and \$4366 after, in 1981 dollars.

Energy costs for heating and cooling were estimated using the data from Tables X-7 and X-8. Hot water was added with a supplementary calculation, assuming that the system as constructed would provide 80% of the hot water during the cooling season and during the beginning and end of the heating season but was not available for this use when the storage tank temperature dropped below 110°F. During the peak of the heating season, all of the collector was needed for heating. The use of the heat pump to preheat hot water would increase the total efficiency of the system, but as this would draw down the storage temperature and thereby affect the simulation, we have not included it in this analysis but have left it for future work. These assumptions resulted in the solar system providing 50% of the hot water in Washington and 46% in New York. The remainder of hot water was provided by electric resistance.

Tables XI-5 and XI-6 show the calculations performed in arriving at the annual energy costs for the system. The other systems are also presented in these tables and are discussed below.

B. Costs of the Parallel Solar Assisted Heat Pump System

The solar portion of the parallel system was sized to satisfy 50% of the heating load, according to the method of Balcomb and Hedstrom.[2] For New York, the collector area required was 278 ft^2 and for Washington, 192 ft^2. We used a solar system cost of $30/$ft^2$ in 1981 dollars. In section II we used the Mitre Report[4] to estimate the installed cost of this system as $34/$ft^2$. A major contribution to this system cost is the collectors. The reader should note that the collectors appropriate for the parallel system are of a very different type than those discussed in section VII in connection with the series system. Our estimate represents a greater than 8% annual real decline in the cost of conventional solar systems over the four-year period 1977-1981. The first cost of the solar system is then $8340 in New York and $5760 in Washington.

The cost of the air-to-air heat pump was estimated from the Westinghouse-EPRI heat pump study.[3] The Lennox SHP9-265 unit was selected from three possibilities because its two-ton capacity was most appropriate for the load and because it had the best seasonal performance of the three. Its 1975 price was given as $1312, which becomes $1861 in 1981.

The total 1981 first cost of the parallel system is then $10,201 in New York and $7621 in Washington. After the tax credit on the solar portion of the system, these prices become $8333 and $6269, respectively.

The seasonal performance factors for this heat pump in the heating mode in New York and Washington were estimated by plotting SPF's given in the Westinghouse-EPRI heat pump study for Albany, Boston, and Denver against average annual degree days for those cities. A straight line drawn through these points was extrapolated to the degree-day values for New York and Washington. The seasonal performance factors of 1.8 and 1.9, respectively, include auxiliary electric power requirements. The seasonal performance factor for cooling was quoted as 2.17 in the reference for all three cities, and this value was used for New York and Washington. The solar system was assumed to use 5% of the energy it delivered, for a seasonal performance factor of 20.

It was assumed that the solar system could supply half the hot water in both cities, largely in the summer months, the remainder being supplied by electric resistance. As with the series system, this fraction could be increased by increasing the collector area, but with questionable cost effectiveness.

The annual maintenance costs for the system in 1981, at 2% of the system first cost, are $204 in New York and $152 in Washington. The energy cost computations for 1981 are shown in Tables XI-5 and XI-6.

C. Costs of the Stand-Alone Heat Pump System

The first cost of the stand-alone heat pump, taken from reference 3 and translated into 1981 dollars, was $1861, as discussed above. The annual maintenance cost for 1981, at 2% of the first cost, is $37. The seasonal performance factors for the heat pump are the same as those discussed above. Hot water is obtained from electric resistance. See Tables XI-5 and XI-6 for the

Table XI-5

System Energy Costs
(Long Island, New York)

System	Load Type	Load Type (10^6 Btu)	Seasonal Performance Factor	Energy Used (10^6 Btu)	Unit Energy Cost (1981 $/ 10^6 Btu)	Annual Energy Cost (1981 $)
Series Solar Assisted Heat Pump	H	45.8	3.18*	14.4(e)	16.06	231
	C	7.6	3.71*	2.0(e)	26.19	52
	HW(Solar)	4.6	20.††	0.2(e)	26.19	5
	HW(I^2R)	5.4	0.9††	6.0(e)	16.06	96
	TOTALS	63.4		22.6(e)		384
Parallel Solar Assisted Heat Pump	H(Solar)	22.9	20††	1.1(e)	16.06	18
	H(HP)	22.9	1.8†	12.7(e)	16.06	204
	C	7.6	2.17†	3.5(e)	26.19	92
	HW(Solar)	5.0	20††	0.2(e)	26.19	5
	HW(I^2R)	10.0	0.9††	5.6(e)	16.06	90
	TOTALS	63.4		23.1(e)		409
Stand-Alone Heat Pump	H	45.8	1.8†	25.4(e)	16.06	408
	C	7.6	2.17†	3.5(e)	26.19	92
	HW(I^2R)	10.0	0.9††	11.1(e)	16.76	186
	TOTALS	63.4		40.0(e)		686
Electric Resistance	H	45.8	1.0†	45.8(e)	16.06	736
	C	7.6	2.05†	3.7(e)	26.19	97
	HW	10.0	0.9††	11.1(e)	16.76	186
	TOTALS	63.4		60.6(e)		1019
Oil Furnace	H	45.8	0.6†	76.3(t)	5.14	392
	C	7.6	2.05†	3.7(e)	26.19	97
	HW	10.0	0.6†	16.7(t)	5.14	86
	TOTALS	63.4				575

*Computer simulation results
†References 6b and 7
††Assumed for this analysis

Note: e - electrical
t - thermal

Table XI-6

System Energy Costs
(Washington, D. C.)

System	Load Type	Load Type (10^6 Btu)	Seasonal Performance Factor	Energy Used (10^6 Btu)	Unit Energy Cost (1981 $/ 10^6 Btu)	Annual Energy Cost (1981 $)
Series Solar-Assisted Heat Pump	H	34.6	3.67*	9.4(e)	13.22	124
	C	12.4	3.48*	3.6(e)	23.62	85
	HW(Solar)	5.0	20.††	0.2(e)	23.62	5
	HW(I^2R)	5.0	0.9††	5.6(e)	13.22	74
	TOTALS	57.0		18.8(e)		288
Parallel Solar-Assisted Heat Pump	H(Solar)	17.3	20.††	0.9(e)	13.22	12
	H(HP)	17.3	1.9†	9.1(e)	13.22	120
	C	12.4	2.17†	5.7(e)	23.62	135
	HW(Solar)	5.0	20.††	0.2(e)	23.62	5
	HW(I^2R)	5.0	0.9††	5.6(e)	13.22	74
	TOTALS	57.0		21.5(e)		346
Stand-Alone Heat Pump	H	34.6	1.9†	18.2(e)	13.22	241
	C	12.4	2.17†	5.7(e)	23.62	135
	HW(I^2R)	10.0	0.9††	11.1(e)	14.25	158
	TOTALS	57.0		35.0(e)		534
Electric Resistance	H	34.6	1.0†	34.6(e)	13.22	457
	C	12.4	2.05†	6.0(e)	23.62	142
	HW	10.0	0.9††	11.1(e)	14.25	158
	TOTALS	57.0		51.7(e)		757
Oil Furnace	H	34.6	0.6†	57.7(t)	5.14	297
	C	12.4	2.05†	6.0(e)	23.62	142
	HW	10.0	0.6†	16.7(t)	5.14	86
	TOTALS	57.0				525

*Computer simulation results
†References 6b and 7
††Assumed for this analysis

Note: e - electrical
t - thermal

energy cost results for this system.

D. Costs of the Electric Resistance System

This system uses electric resistance heating with central air conditioning. The first cost of the system from reference 3 was $854 in 1975, which becomes $1211 in 1981. The 1981 maintenance cost is $24 at the same 2% rate used previously. The heating seasonal performance factor is 1.0, while that for cooling was 2.05, taken from reference 3. Hot water was heated by electric resistance. The energy cost calculations are shown in Tables XI-5 and XI-6.

E. Costs for the Oil Furnace System

This system uses an oil burner for heat and hot water and electric air conditioning. Reference 3 gave the first cost of this system as $1109 in 1975, which becomes $1573 in 1981. The 1981 maintenance cost is $31 on the same basis as the other systems. We have taken the average furnace efficiency to be 60%, which is the higher of the values listed in references 1 and 3. The same electric air conditioner is used in this as in the preceding system. Tables XI-5 and XI-6 show the energy cost calculations for the system.

F. Cash Flow Analysis

We chose to compare the five systems on the basis of annual cash flow, amortizing first costs of each system in 10 years in one analysis, and in 20 years in a second analysis. The annual cash flow analysis gives the net amount the homeowner must pay each year to own and operate each of the systems.

For each year, the following cash outlays were calculated.

1. The mortgage payment on a 9% loan of 10 or 20 year term in the amount of the system first cost (after tax credit in the case of the solar systems). This outlay remains constant over the life of the mortgage.
2. The maintenance cost, starting with the 1981 value as determined

previously and increasing at a 6% annual rate.

3. The energy cost, starting with the 1981 value as determined previously and increasing at a 10% annual rate.

Against these charges there will be a credit due to the tax-deductibility of the interest on the mortgage. We have assumed the taxpayer to be in a 30% tax bracket, and therefore have deducted 30% of the interest charge on the mortgage each year. This amount declines from year to year since the fraction of the mortgage payment that is interest declines with time.

Tables XI-7 and XI-8 summarize the first cost, initial (1981) maintenance cost, and initial energy cost for each system and city. The annual cash flows for the systems are shown in Figures XI-1 and XI-2 (New York) and XI-3 and XI-4 (Washington).

It appears that in Washington at least the series solar assisted heat pump is a winner, and that it is a strong competitor on Long Island. Research and development effort is required to meet some of the assumed input to this analysis and much of this work is underway. Nevertheless, it is clear that there is a potential path to cost-effectiveness relative to conventional energy sources in the near term for this system. With the attainment of cost-effectiveness, eager and rapid acceptance of this solar system is assured.

Although these results are promising as they stand, it should be pointed out that there are a number of possible refinements in the system which were not included in this first-effort simulation but which will significantly improve the performance of the system over that reported here. The first of these is ground coupling, discussed in section IX, which will raise the average storage temperature, guard the system against extended periods of cloudy weather, and reduce the necessary collector area. A second area in which improvement is possible is

Table XI-7

Input Data for Cash Flow Analysis
(Long Island, New York)

System	Net First Cost (1981)	Maintenance Cost (1981)	Energy Cost (1981)	Energy Cost (1991)
Series Solar Assisted Heat Pump	$5112	$133	$ 384	$ 996
Parallel Solar Assisted Heat Pump	8333	204	409	1061
Stand-Alone Heat Pump	1861	37	686	1779
Electric Resistance	1211	24	1019	2643
Oil Furnace	1573	31	575	1491

Table XI-8

Input Data for Cash Flow Analysis
(Washington, D.C.)

System	Net First Cost (1981)	Maintenance Cost (1981)	Energy Cost (1981)	Energy Cost (1991)
Series Solar Assisted Heat Pump	$4366	$114	$ 288	$ 747
Parallel Solar Assisted Heat Pump	6269	152	346	897
Stand-Alone Heat Pump	1861	37	534	1385
Electric Resistance	1211	24	757	1963
Oil Furnace	1573	31	525	1362

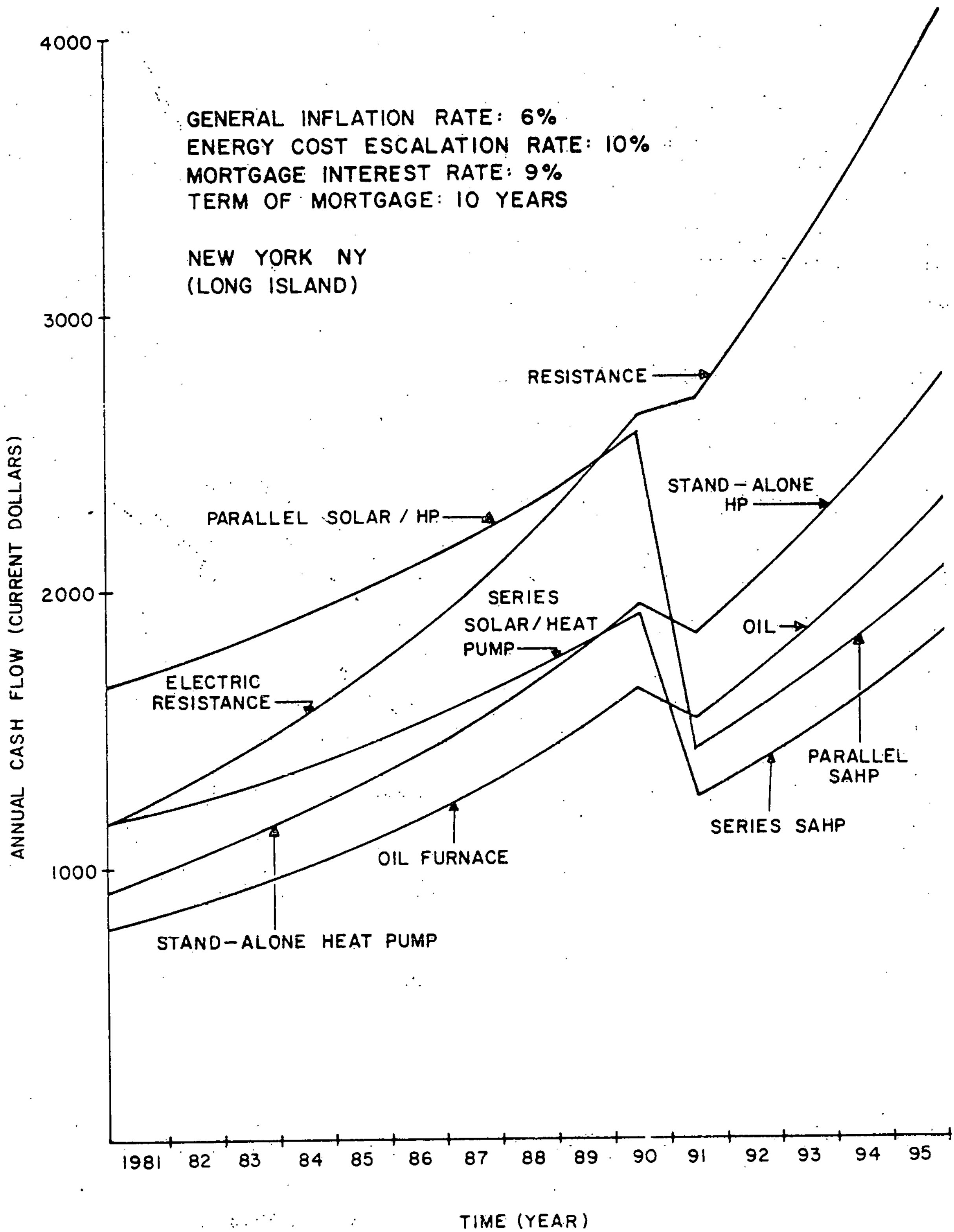

Fig. XI-1. Annual Cash Payout by Homeowner to Own and Operate Each of Five Competing Systems on Long Island, New York. Ten-Year Amortization of System First Cost.

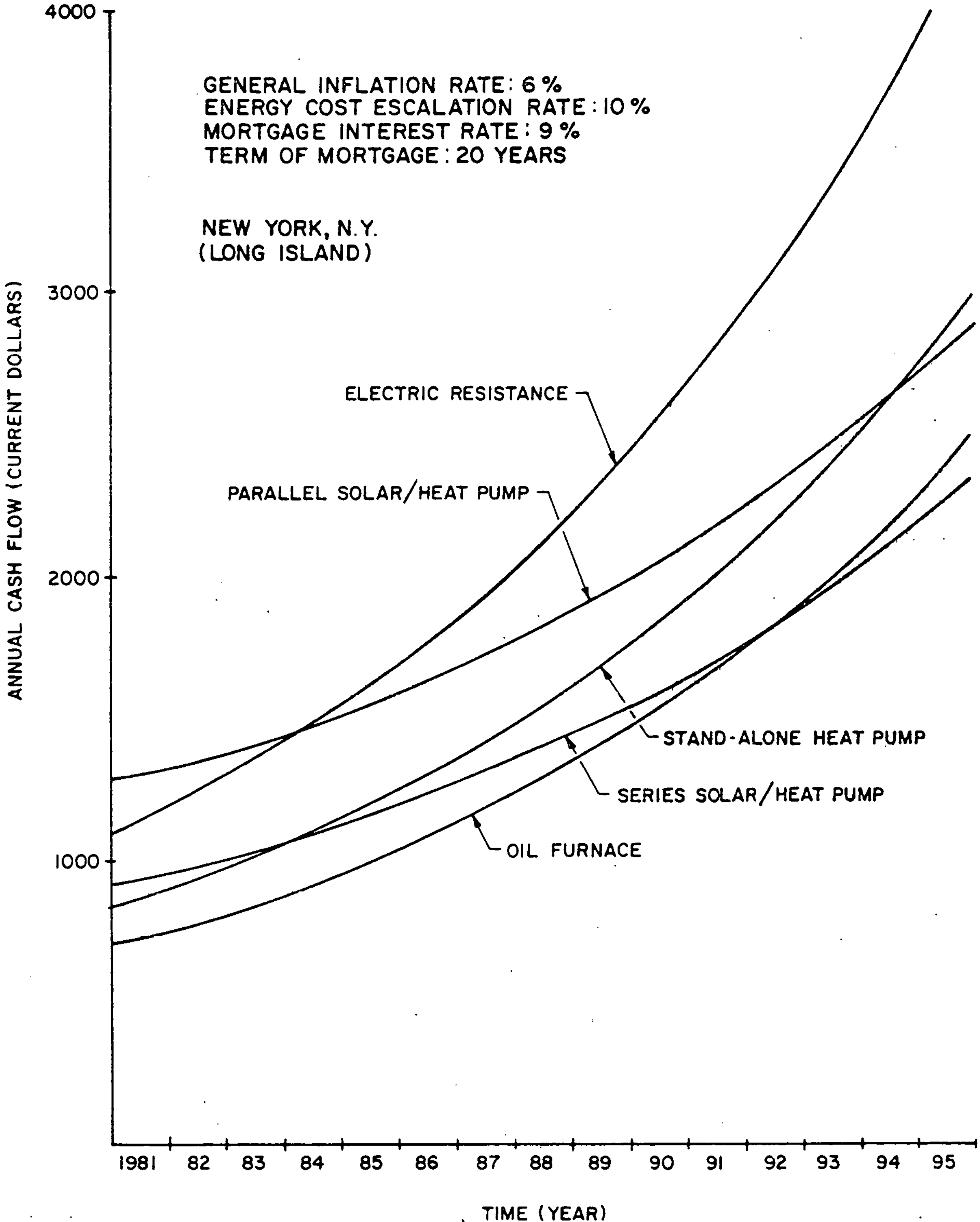

Fig. XI-2. Annual Cash Payout by Homeowner to Own and Operate Each of Five Competing Systems on Long Island, New York. Twenty-Year Amortization of System First Cost.

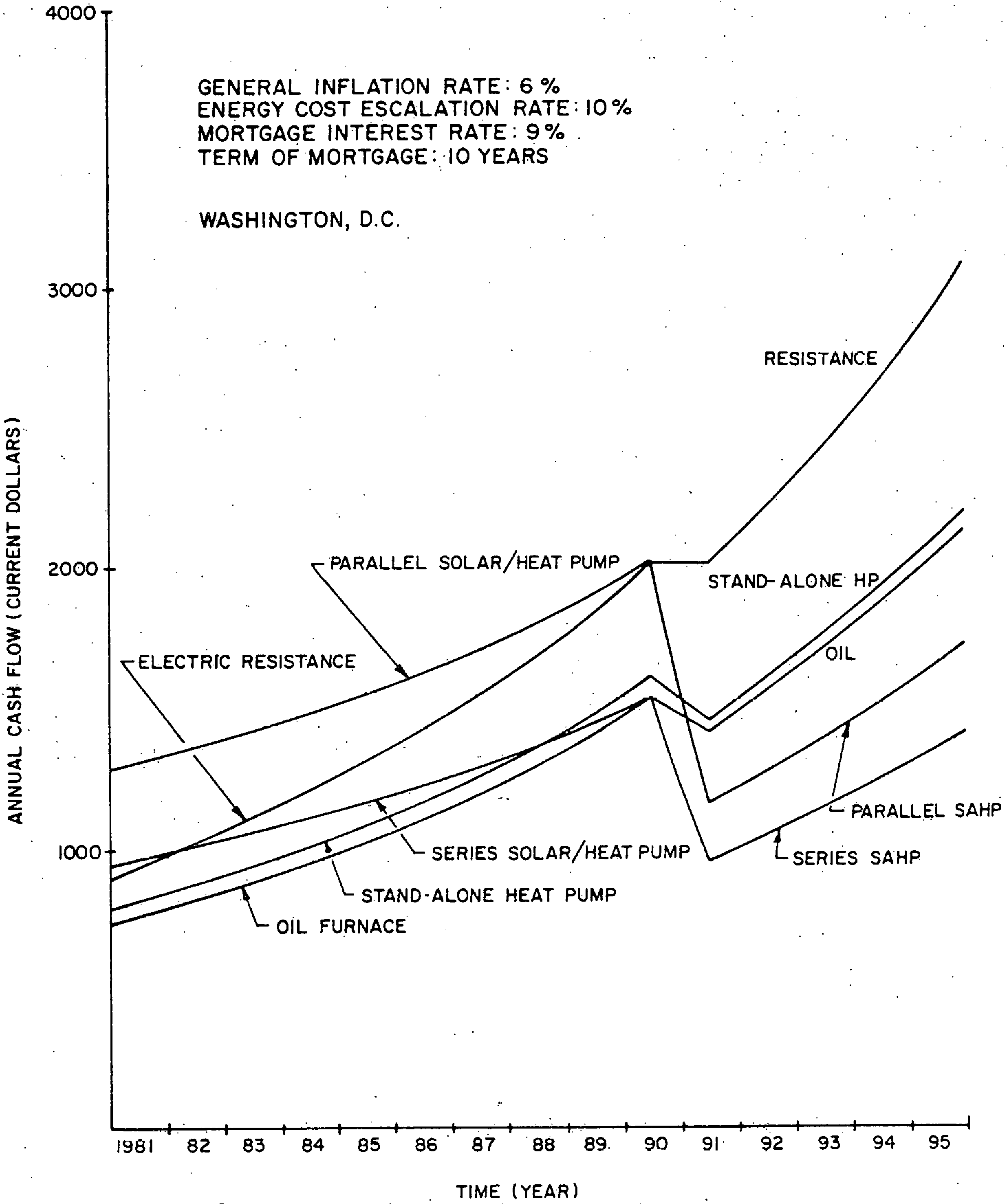

Fig. XI-3. Annual Cash Payout by Homeowner to Own and Operate Each of Five Competing Systems in Washington, D.C. Ten-Year Amortization of System First Cost.

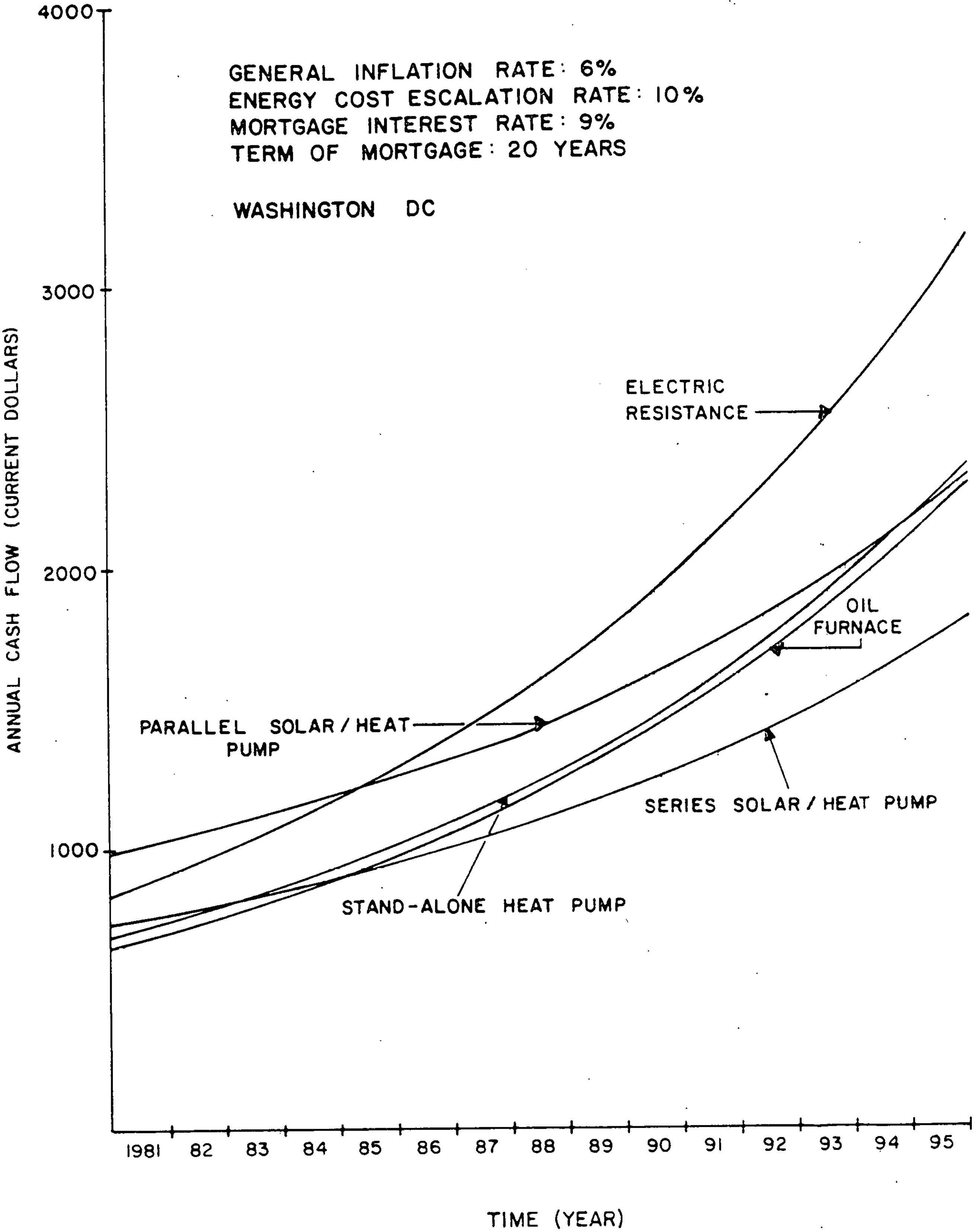

Fig. XI-4. Annual Cash Payout by Homeowner to Own and Operate Each of Five Competing Systems in Washington, D.C. Twenty-Year Amortization of System First Cost.

control strategy. We used a very simple strategy with no attempt at optimization. A third area where improvement is possible is the use of the heat pump in winter for hot water preheating. This should be much more efficient than the use of electric resistance. All of these possibilities will be examined in future work.

It should also be pointed out that the residential system using solar energy is not the only application for the special heat pump. Large commercial buildings having simultaneous heating and cooling loads can use the special heat pump to advantage in moving thermal energy from the core to the perimeter. Industrial applications using waste heat should be numerous. In these contexts, solar energy should be viewed as one of several sources of thermal input to the heat pump. It will be used when other sources of low-grade heat are insufficient to meet the load. This will almost always be the case in single family residences in moderate or cool climates, and this is the reason we have emphasized this application. But it is important not to lose sight of the potential for wide application of a thermodynamically respectable heat pump, a machine able to move heat to higher temperatures at reasonable Second Law efficiencies.

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XII. CONCLUSION

We conclude that the series solar assisted heat pump system provides a clear path to cost effectiveness and energy savings. In order to achieve these objectives, two developments must occur:

1. The design, fabrication, testing, and marketing of a special heat pump whose coefficient of performance rises with increasing temperature in the range appropriate for solar assist, in line with the possibilities suggested by the Second Law of Thermodynamics and within the present capability of vapor-compression science.
2. The concurrent development of low-cost energy collection and storage subsystems. The drive to low cost in these subsystems is made possible by the low-temperature operating regime of the collectors and storage in the series system.

The future will very probably see a variety of means of energy collection and storage involving various combinations of solar energy, reclaimed heat, and thermal energy from the ground, all serving in conjunction with the special heat pumps being developed under the Department of Energy-funded research and development programs. These systems will play an important role in the effort to make effective use of the solar resource and thereby to conserve dwindling supplies of nonrenewable fuels.

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