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SIMULATION STUDY OF SOLAR HEAT PUMP SYSTEMS

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ABSTRACT

The performance of several configurations of solar-augmented heat pump systems are evaluated. These systems include both air and water collection systems with either air-air, water-air, or special hybrid heat pumps which can use both stored energy and ambient air as energy sources.

The performance evaluations employ factorial design to determine the effect of the parameters of each individual system. The systems are compared with each other and with conventional solar and conventional heat pump systems. Simulations are done for Madison, Wisconsin; Albuquerque, New Mexico; and Charleston, South Carolina, to investigate climatological effects on solar heat pump performance. The results are used to formulate general guidelines for designing solar-augmented heat pump systems.

INTRODUCTION

The purpose of this paper is to analyze viable types of solar-augmented heat pump systems for residential heating and cooling, and to compare them to "conventional" solar and "conventional" heat pump systems. Our goal is to develop general design guidelines. These are established through computer simulations of the proposed systems using the generalized solar energy simulation program TRNSYS [1].

Solar heat pump systems have recently received widespread attention. Their advocates argue that since solar energy collector performance is relatively low for the temperatures needed in space heating, and since air-air heat pump system performance decreases with the low source temperatures available for winter-time heating, an ideal solution is to combine these two systems to compensate for their weaknesses. Hence, proposed solar-augmented heat pump systems utilize a solar collector and storage combination to provide a moderate temperature energy source for the heat pump, raising its COP and capacity.

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The early analysis by Jordan and Threlkeld [2,3,4] and Löf [5] showed that solar heat pump systems are economically feasible throughout much of the United States, and that evaporator-side storage requires much smaller heat pumps than do condenser-side storage systems. The General Electric Phase 0 report [6] extended the study of solar-augmented heat pumps to include 6 types of buildings in 9 locations with a variety of different heating and cooling modes. A significant advantage was shown for systems with dual evaporators that switched between ambient and storage to select the highest temperature heat source. These principles of operation have been incorporated in most of the systems studied to date.

Proceedings of the NSF-sponsored workshops held at Pennsylvania State University [7], June 12-14, 1975, summarized the state of development of solar energy heat pump systems for the heating and cooling of buildings. Detailed design studies for single buildings are reported by Rittleman, Gilman, Jardine, Bridgers and Dubin, while Drucker and Freeman presented more general feasibility studies. Most of these studies demonstrated that substantial savings can be realized utilizing solar heat pump systems. However, few of these studies made specific recommendations for system design or attempted to show how different solar heat pump system configurations compare to each other and to conventional solar and conventional heat pump systems. Furthermore, in most of the studies, assumptions have been made to reduce the complexity and quantity of the calculations. These often include the use of "average" or "design" weather conditions (radiation and air temperature), constant collector plate temperature, constant storage temperature, constant heat pump COP, and similar simplifications.

Simulations of the detailed dynamic behavior of solar-augmented heat pump systems have been made by Freeman, et al. [8] and Bosio and Suryanarayana [9]. These studies were limited to two locations (Albuquerque and Madison) and considered a limited number of different parameter values and operational modes in order to make economic evaluations. These studies substantiated the significant effect of collector area and storage size on thermal performance. Neither was a very exhaustive study of the configurations, operational modes, or climatological dependence of solar heat pump systems.

From these and other studies a few basic solar heat pump system configurations have emerged as potentially the most economically advantageous. These systems have the following features in common: an "in-line" heat pump to elevate the temperature of solar energy stored on the evaporator side, the capability to utilize ambient air as the heat pump source when storage is depleted, the capability to bypass the heat pump when storage temperatures are high enough, rejection of energy to ambient air in the cooling mode, "second-stage" auxiliary space heating by duct heaters located downstream of the heat pump condenser coil, and a preheater for domestic hot water using "unassisted" solar.

It should be noted that there are a number of more unusual solar-assisted heat pump systems that are receiving special attention. Some of these systems incorporate "off-peak", two-phase, or cold storage, solar Rankine-powered heat pumps, variable-speed heat pumps, multi-condenser heat pumps, and combination collector-evaporators. These appear, for the most part, either technically unfeasible at the present time or prohibitively high in first cost and therefore are not included in this study.

METHOD OF ANALYSIS

The complexity of the thermal analysis of solar-augmented heat pump systems makes the use of computer simulations the only method for adequately determining the system dynamics. A comparative analysis of different systems requires that the interaction of the expected important design parameters be studied. These parameters include the collector area (and construction), storage size, control options, and the weather (i.e. the ambient temperature and solar radiation distributions). Unimportant or unnecessary parameters and options must also be identified. The factorial design method is employed to analyze and compare these systems in an orderly and effective manner with a minimum number of simulations. This approach allows the systematic analysis of the effects of the parameters using two carefully selected levels of each system parameter. Some parameter levels are actual numerical values (e.g. area of collector) while others are indicative of a system design option (e.g. whether or not to use an ambient air source).

Bi-level factorial design requires 2^n simulations where n is the number of variables being evaluated. A pertinent number indicative of system performance is then generated for each simulation. For this study it is the fraction of the total load met by auxiliary energy, subsequently referred to as F. Main effects for each parameter, defined as the average change in system performance caused by changing the parameter from its (-) level to its (+) level, can be calculated. For an n = 4 factorial design, the main effect of, for example, collector area, A, is found by subtracting the mean value of F of the eight runs where A = (-) from the mean of the eight runs where A = (+). This gives a numerical value for the average change in F due to the increase in collector area. Interactive effects are also calculated to show how the effect of changing one variable is affected by the level of another. To assess the interaction of collector area, A, with the storage volume to collector area ratio, V/A (denoted as A x V/A), the mean of the F's from runs where A is (+) and V/A is (-), plus runs where A is (-) and V/A is (+) is subtracted from the mean of the F's from runs where both are (+) or both are (-). If A x V/A is large, then increasing A at high V/A changes F faster than increasing A at low V/A. Each combination of parameter and control option levels in the factorial design are simulated for a "design year" in one or all of three locations (Madison, Albuquerque, and Charleston).

The general simulation program, TRNSYS, is composed of a library of subroutines which model individual pieces of hardware (e.g. collectors, tanks, heat pumps, load), and an executive routine which links these component models and solves the resulting system of algebraic and differential equations. The simulation calculations are performed with a 15-minute computational time-step to allow consideration of the transient effects and short-term interactions of components.

The heat pump model used in these simulations is quasi steady-state in nature. The "transient" behavior of the heat pump is determined empirically by interpolating performance data which has been supplied to the model from data adapted from manufacturers' specifications [8]. The performance characteristics for the "standard" three-ton unit used in these simulations is shown graphically in Figs. 1 and 2. The data represents the heat pump capacity and the work input in the heating and cooling modes of both the air and liquid source heat pumps as well as special "hybrid" heat pumps having both an air and liquid source evaporator. Since actual performance data is lacking for such heat pumps, it is assumed that their performance at a given source inlet temperature is identical in either the air or liquid source modes. This is equivalent to assuming that the evaporator heat exchangers have been designed to have equal heat transfer effectivenesses.

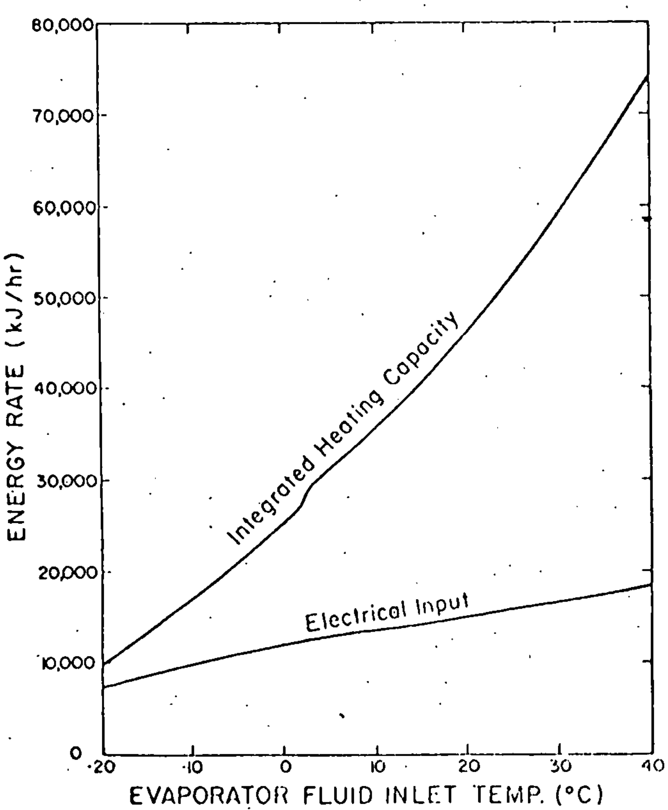

Fig. 1 Heat Pump Heating Performance

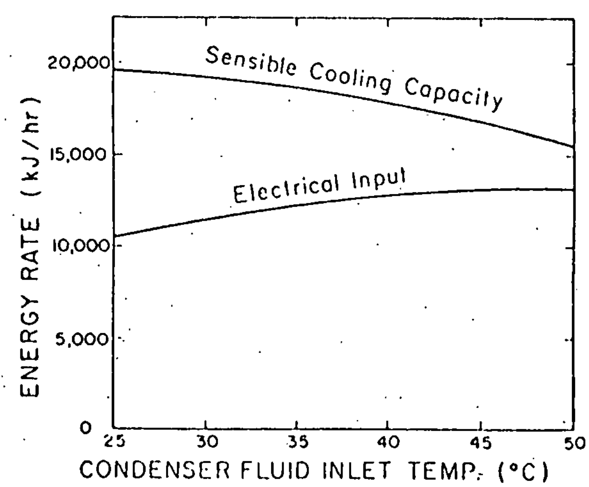

Fig. 2 Heat Pump Cooling Performance

The collector model is a simplified adaptation of the Hottel and Woertz [10] model with a constant heat removal factor, (F_R), transmittance-absorptance product ($\tau\alpha$), and loss coefficient (U_L). The values used in these simulations are listed in Table 1, and are typical values chosen by the methods outlined by Duffie and Beckman [11].

Collector Type	F_R	U_L	$(\tau\alpha)$
single-glazed, air	0.75	30.0	0.76
zero-glazed, water	0.90	108.0	0.90
single-glazed, water	0.90	30.0	0.76
double-glazed, water	0.90	20.0	0.69

TABLE 1

Collector Parameters

Hourly values of total radiation on a horizontal surface are separated into beam and diffuse components via the method of Liu and Jordan [12]. The beam component is corrected for incidence angle on the collector surface while the diffuse component is assumed to be evenly distributed within the collector-to-sky view factor.

The liquid storage tanks are modeled as being fully mixed. The rock beds are multi-node "infinite NTU" models (i.e. infinite heat transfer coefficient from rock surface-to-air) which adequately predict stratification effects [13]. The rock bed model is representative of approximately cube-shaped beds of 3 cm. diameter rocks. Both the liquid tanks and the rock beds have loss coefficients of 1 kJ/hr-°C, with storage losses assumed to be uncontrolled heat gains to the structure.

The building used in each of the simulations is the same single-family residence of approximately 120 m^2 floor area which is well insulated and weather stripped. The thermal capacitance of the walls and roofs are modeled using a finite difference representation. Internal generation and solar heat gains are included. The load profiles generated are representative of a house with an overall loss coefficient of 615 kJ/hr-°C. The service hot water load in these simulations is represented by a profile of 21.5 kg/hr of 60°C water drawn uniformly during the hours from 6:00 A.M. to 7:00 P.M. every day. Main water temperature is 10°C. This results in an annual service hot water energy requirement of 21.37 GJ.

SYSTEM DESCRIPTIONS

Three base configurations of solar-heat pump systems are identified.

These incorporate the features previously discussed to optimize performance. Since a complete and general investigation of all combinations of systems having these features is impractical, several compromises in the study have been made as described below.

The preheat coils for the service hot water (SHW) system have been located on the solar side of the system as shown in Figs. 3, 4 and 5. These are positioned to maximize heat transfer to the SHW tank, and sized in keeping with standard commercial practice. The solar system can deliver energy to the SHW system when there is no space heating load. In the systems with small collector areas, very little service hot water preheating is achieved in the winter months since the collector-storage circuit is maintained at a low temperature by the heat pump. The use of the heat pump for preheating service hot water is not considered in this study.

The heat pump employed in each of these simulations is a "standard" 3-ton unit. Freeman et al. [8] showed that the size of the heat pump was not critical as long as it was large enough to meet the design load. Conventional air-air heat pump installations are generally sized for the cooling load to insure proper dehumidification, but when heating is the prime concern, the unit should be oversized so that electrical resistance auxiliary heat is not often required. Excessive compressor cycling sometimes caused by oversizing can be avoided by widening the thermostat deadband range and by utilizing direct heating from the collectors or from storage when temperatures are sufficiently high.

Since the "optimum" systems discussed here incorporate the utilization of ambient air as a heat pump source, the air source evaporator is presumed to be available as an air-sink condenser in the cooling mode. This precludes the possibility of "night-sky radiation" of heat rejected for summer-time cooling by circulating stored cooling water through the collectors at night. These systems require another storage tank and are probably prohibitively complex and expensive. The cooling mode of each solar-heat pump system simulated here yields the same results (slight changes are caused by differences in storage tank losses to the house) for a given location since the heat pump is completely uncoupled from the solar part of the system. The solar collectors and storage still operate in the summer to supply the service hot water load.

Each of the three base systems is described briefly below. System I, shown in Fig. 3, is a liquid-solar heating system utilizing an ethylene glycol solution in the collector loop. Energy is transferred into the water storage tank across a heat exchanger in the tank. The space heating load can be met in one of four possible ways: 1) If the storage temperature is higher than a pre-set level, T_{set}, the tank water is pumped through a fan-coil unit to heat the house directly (heat pump bypass mode). 2) If the tank temperature is less than T_{set} but greater than the ambient air temperature, T_{amb}, and above a minimum temperature, T_{min}, the heat pump uses the tank fluid

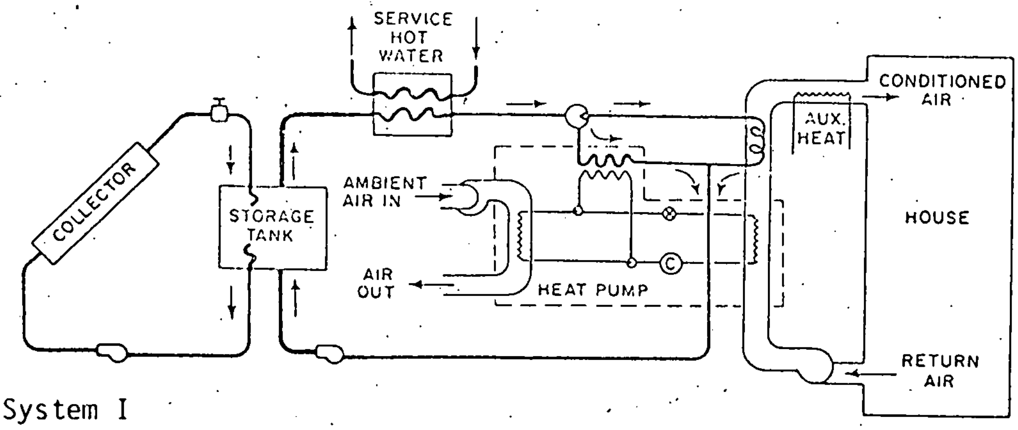

Fig. 3 System I

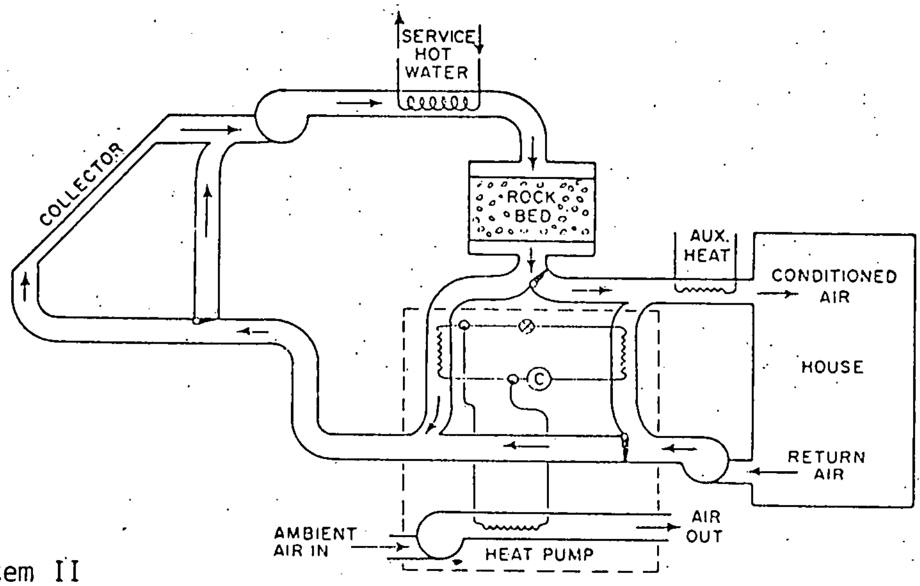

Fig. 4 System II

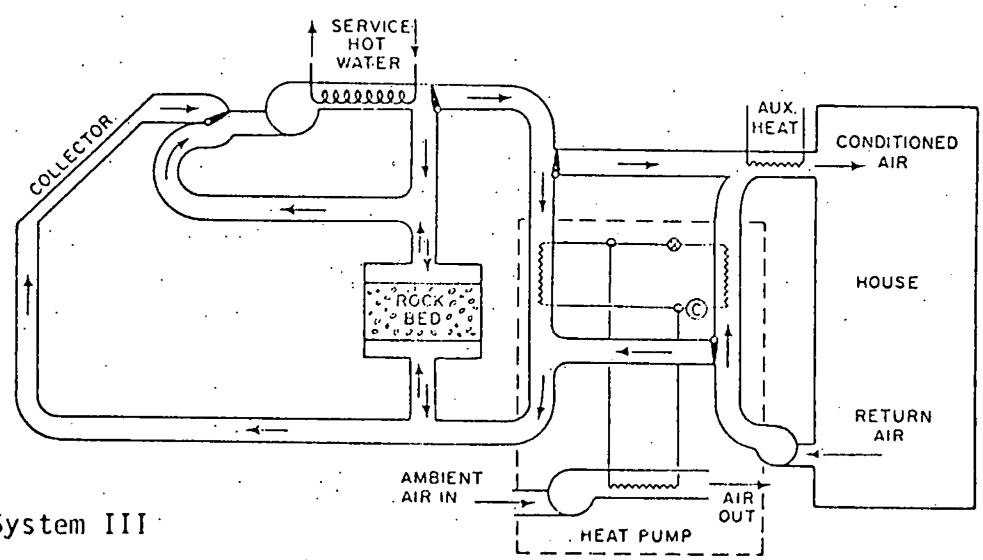

Fig. 5 System III

as its heat source. 3) If the tank temperature is less than either T_{min} or T_{amb}, the heat pump uses ambient air as its heat source. If, in modes 2 or 3, the heat pump capacity is insufficient to meet the load, auxiliary is used to supplement the heat pump output. 4) If both source temperatures are too low (the tank temperature less than T_{min}, and the ambient less than the cut-off temperature (-20°C)), the load is met entirely by auxiliary.

System II, shown in Fig. 4, is an air-heating solar system utilizing rock bed thermal storage and an air-to-air heat pump. Air is circulated through the collectors, service hot water heat exchanger, and rock bed whenever solar energy can be collected. When a load has to be met and no solar is available, a bypass is used around the collectors. The space heating load can be met in the same four modes as in System I.

System III, shown in Fig. 5, differs from System II in that it takes advantage of the effects of temperature stratification in the rock bed by reversing the flow direction during charging and discharging. However, the design is inherently more complex and is unable to simultaneously collect solar energy and supply the heat pump from the solar source.

These three base systems have been simulated with an ordered series of combinations of important system parameters. A summary of the runs in the factorial design are presented in Tables 2 and 3. Additional simulations not shown in Table 3 were also performed to investigate options not considered in the factorial design.

Parameter or Option	(+) Value	(-) Value
N - Collector Glazings	2	1
SRC - Source Capability	Solar & Amb. (Dual)	Solar Only (Single)
BP - Bypass Capability	Max. Bypass	Never Bypass
A - Collector Area	30 m^2	10 m^2
V/A - Volume/Area: Water	0.33 m^3/m^2	0.075 m^3/m^2
Air	1.11 m^3/m^2	0.25 m^3/m^2

Table 2: Factorial Design Bilevel Values

System	Location	N	SRC	BP	A	V/A	Number of Runs
I	Madison	+	-	+-	+-	+-	8
		+	+	+-	+-	+-	8
		-	-	+	+-	+-	4
		-	+	+	+-	+-	4
	Albuquerque	-	-	-	+-	+-	4
		-	+	-	+-	+-	4
	Charleston	-	-	-	+-	+-	4
		-	+	-	+-	+-	4
II	Madison	-	-	-	+-	+-	4
		-	+	-	+-	+-	4
III	Madison	-	-	-	+-	+-	4
		-	+	-	+-	+-	4

Table 3 Factorial Design Run Summary

SIMULATION RESULTS

The simulation results of interest in this study are long-term integrated energy quantities. These include the total heat gain of the solar collectors (Q_{SOL}), the total heat load (Q_{LOAD}), the total auxiliary energy added by the furnace and service hot water heater (Q_{AUX}), the total energy removed by the heat pump from the ambient air in the heating mode (Q_{AIR}), and the total heat pump electrical input for compressor and pumps or fans (W_{HP}). For a heating season the system energy balance (assuming negligible change in stored energy) is

$$Q_{SOL} + Q_{AIR} + Q_{AUX} + W_{HP} = Q_{LOAD} \quad (1)$$

The single most informative indicator of the performance of a solar heat pump system is the percentage of the load carried by "conventional" fuels (F), defined as

$$F = (W_{HP} + Q_{AUX})/Q_{LOAD} \quad (2)$$

Since practical solar heat pump systems will probably utilize electric resistance air and water heaters, combining the heat pump electrical input with the auxiliary furnace and water heater inputs is reasonable.

SYSTEM I - Madison

The initial set of runs in the factorial design procedure were chosen to study the effects of collector area, storage volume/collector area ratio, ambient energy source use, and the extent to which "bypass" (direct solar) heating is utilized. Madison, Wisconsin, was chosen as the site for the first set of simulations which is comprised of the first set of 16 runs in Table 3. The total space heating and service hot water load is 85.4 GJ. The main and interaction effects of bypass heating were found to be negligible. Using direct solar heating wherever possible results in saving less than 1% of the total annual heating energy requirement. Therefore the parameter BP was deleted in subsequent factorial design simulations and maximum bypass mode was used.

One of the suggested advantages of solar heat pump systems is that lower quality collectors can be effectively used. This effect was introduced to the factorial design procedure with a new set of 8 runs using single-glazed collectors to compare to the first 8 using double glazing and bypass heating. These runs now comprise a new sixteen-run factorial design where the four variables are the number of collector covers (N), collector area (A), storage volume/collector area ratio V/A, and the use of ambient air as an energy source (SRC). Main and first-order interaction effects on F are shown in Table 4.

Main Effects	Interaction Effects	Interaction Effects
$\overline{A}$ = -22.8%	A x V/A = 0.9%	V/A x SRC = 2.5%
$\overline{V/A}$ = - 1.7%	A x SRC = 6.9%	V/A x N = 0.7%
$\overline{SRC}$ = -11.6%	A x N = 1.5%	SRC x N = 2.9%
$\overline{N}$ = - 3.8%		

Table 4

Factorial Design Results for System I in Madison

Main Effects	Interactive Effects
$\overline{A}$ = -28.7%	A x V/A = 0.1%
V/A = - 1.8%	A x SRC = 2.2%
SRC = - 2.7%	V/A x SRC = 1.6%

Table 5

Factorial Design Results for System I in Albuquerque

The main effect of using two glazings is a relatively small average increase in F. The interaction effects show that the second glazing saves more energy for the systems with large collector areas not equipped with dual-source evaporators than it does for systems with small collector areas and dual-source evaporators. From these results it does not appear advantageous to use two-cover collectors for solar-assisted heat pump systems. All ensuing simulations are therefore performed with single-cover collectors (except for some miscellaneous simulations of coverless collector systems).

The main effect of increasing the storage volume to collector area ratio is very small, but the magnitude of the interaction V/A x SRC shows that increasing the volume/area ratio has a much more beneficial effect on single-source (SRC = -) systems than it does on dual-source (SRC = +) systems; in fact, increased storage volume actually has a small negative effect on thermal performance in dual-source systems with small collectors.

The option of single or dual-energy sources (SRC) is reflected in the large main effect -11.6%. The interactions A x SRC and V/A x SRC reveal that systems with small collector areas reduce F by much more than systems with large collector areas. This means that the energy savings resulting from using ambient air as an energy source diminish as collector area increases. Collector area, as expected, has a strong effect on F; tripling the area reduces F by 22.8% The strong interactions involving A and SRC show that the addition of collector area results in much larger energy savings for single-source systems than for dual-source systems.

In summary, the results from the study of the first five system parameters establish the following. The extent to which bypass is used has very little effect on F. The use of less expensive, single-glazed collectors does not seem to penalize thermal performance enough to warrant the use of two covers. The use of large storage volumes is warranted only for single-source systems or for dual-source systems with large collector areas; the latter actually operate much like single-source systems because the storage energy is seldom depleted. Also each additional increment of collector added to a single-source system saves more energy than if added to a dual-source system. Additional simulations of "conventional" (no heat pump) solar systems and an air-air heat pump system were performed. The results of all simulations are shown in Fig. 6, and illustrate the aforementioned effects of A, V/A, and ambient energy use on F. Comparisons can also be made between the solar-assisted heat pump, the conventional solar systems, and the air-air heat pumps. (Note that an air-air heat pump is the equivalent of a dual-source system with zero collector area.)

Several additional simulations were performed for System I with A = 20 m^2, V/A = 0.075, and dual sources to investigate the effect of using antifreeze in the storage tank to allow the use of storage as a source down to a much lower temperature. The water tank system has F = 0.427, the antifreeze tank system has F = 0.431. Clearly there is no advantage in using antifreeze in the tank. System I was also simulated with coverless collectors, and resulted in F = 0.623, which is only slightly better than a conventional air-air heat pump system with no solar collectors at all.

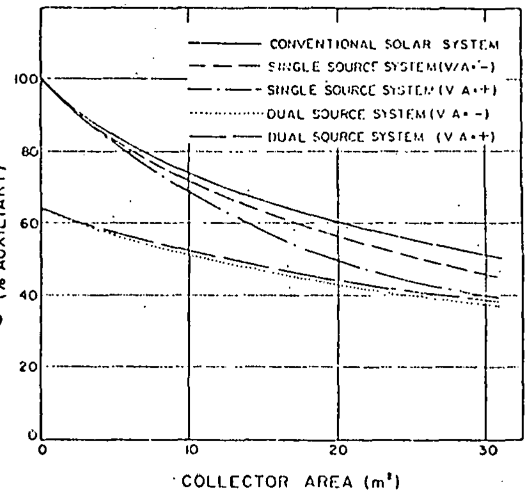

Fig. 6 System I - Madison

Albuquerque, New Mexico

Double glazing is probably unnecessary for solar-assisted heat pump systems in Albuquerque, based on the results in Madison where winter-time temperatures are colder. Bypass heating is recommended for the slight performance improvement and for minimizing compressor start-ups and lengthening compressor life. A factorial design using the same combinations of the three remaining parameters as used in Madison was performed (see Table 3). The total space and service hot water load is 53 GJ. Main effects and first-order interaction effects are shown in Table 5.

The magnitudes of the effects are somewhat different from those in Madison, but the general trends are very similar. Once again the conventional solar systems and the air-air heat pump system are simulated and shown on Fig. 7. The results for Albuquerque show what most likely would have resulted had some systems with collector areas larger than 30 m^2 been simulated in Madison. At small collector areas, substantial energy savings are realized by using dual-evaporator systems; as collector area increases, single and dual-source system performance converges because the ambient air source is used less. Also, a single-source heat pump system with the same storage volume used in the conventional solar system affords very little energy savings.

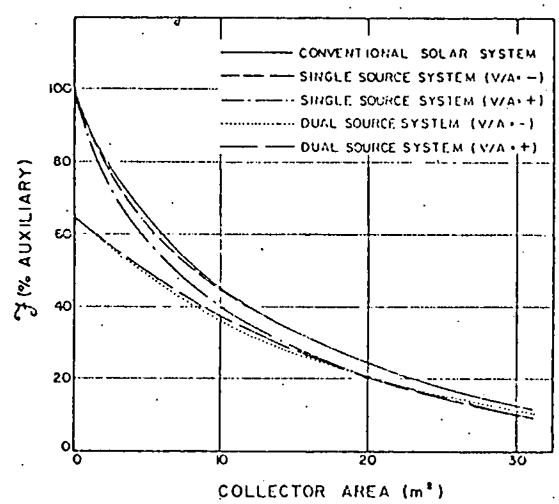

Fig. 7 System I - Albuquerque

Additional systems were again studied. A 20 m^2 coverless collector system with $V/A = 0.75$ and dual sources results in $F = 0.600$. As in Madison, this represents a very small savings over a conventional air-air heat pump system. A system using "seasonal" storage and no ambient source was simulated. A 200 m^3 storage tank was used which is roughly equivalent to a basement full of water. The heat pump was operated so that it rejects energy to the storage tank instead of to ambient during the cooling mode. This system has an F of 0.443, which is identical to the F for the system with 10 m^2 collector area, 2.25 m^3 tank volume and single source.

SYSTEM I - Charleston

The 8 runs generated for the factorial design in Albuquerque were repeated for Charleston, South Carolina with the same parameter levels. The total heating and service hot water load for the year is 37 GJ, over half of which is for service hot water heating. This causes a higher average collector delivery temperature and helps explain departures from the trends in Madison and Albuquerque. The main and interactive effects are shown in Table 6.

Main Effects	Interaction Effects
A = -26.8%	A x V/A = 0.6%
V/A = - 2.5%	A x SRC = 2.0%
SRC = - 1.2%	SRC x V/A = 1.0%

Table 6

Factorial Design Results for System I in Charleston

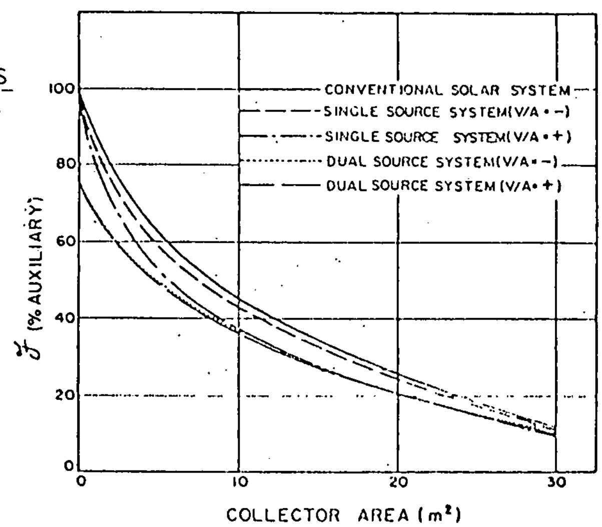

Fig. 8 System I - Charleston

Simulation of a conventional air-air heat pump system in Charleston yields F = 0.755, which is significantly higher than for Madison or Albuquerque. This is due to the SHW load, which is nearly 60% of the total heating load, being met entirely by auxiliary. Conventional solar systems were also simulated and are shown on Fig. 8. These results show that when the SHW load is a large part of the total load, relatively little energy savings result from using an ambient air source evaporator, and the effect of storage volume is not significant. If the space heating load had been the major component of the total load, results more like those in Albuquerque would have been obtained. This indicates that an important system parameter for the solar heat pump system configurations investigated here is the fraction of the total load contributed by SHW heating.

The primary goals of studying Systems II and III is 1) to compare them with each other to identify the system with the better performance, and 2) to compare them with the water systems. Because of computation cost, systems II and III are simulated in Madison only.

SYSTEM II - Madison

System II analysis again involves selection of high and low levels of the parameters to be studied. Past experience has shown that air system performance is not unlike that of water systems so the same three system parameters that were the most important for System I (A, V/A and SRC) are again investigated here. (The V/A ratios for the rock beds are the thermal equivalent of those used for the water systems.) The main and interaction effects are presented in Table 7, and the results are shown graphically in Fig. 9.

Main Effects	Interaction Effects
A = -19.8%	A x V/A = 0.3%
V/A = - 0.2%	A x SRC = 4.6%
SRC = - 5.6%	V/A x SRC = 0.2%

Table 7

Factorial Design Results for System II in Madison

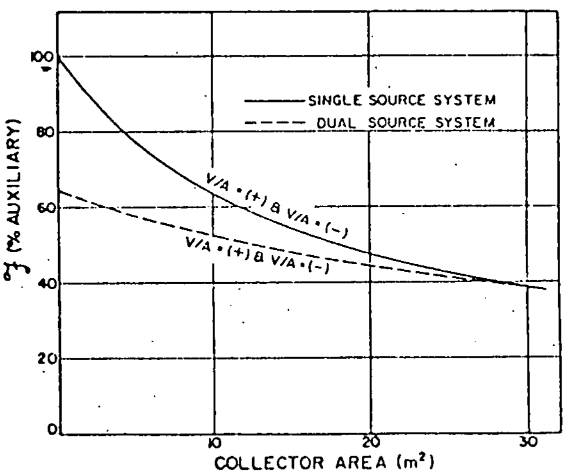

Fig. 9 System II - Madison

It is seen that the main effect of collector area is quite similar to that for System I, but the effect of SRC is much less. Volume effects are also negligible for System II. In comparing these results to those for the water system, it should be remembered that the collector efficiency factor, F_R, is lower for air collectors than for water collectors (0.75 vs. 0.90). Countering this is the more advantageous location of the heat pump evaporator downstream of the rock bed storage in the collector-storage loop which tends to lower collector inlet temperatures and results in better collection efficiency.

SYSTEM III - Madison

An 8-run factorial analysis was generated with system parameters, A, V/A, and SRC again. The main and interactive effects are listed in Table 8 and the results are plotted in Fig. 10.

Main Effects	Interaction Effects
A = -19.9%	A x V/A = 0.1%
V/A = - 0.1%	A x SRC = -5.0%
SRC = - 6.5%	V/A x SRC = 1.3%

Table 8

Factorial Design Results for System III in Madison

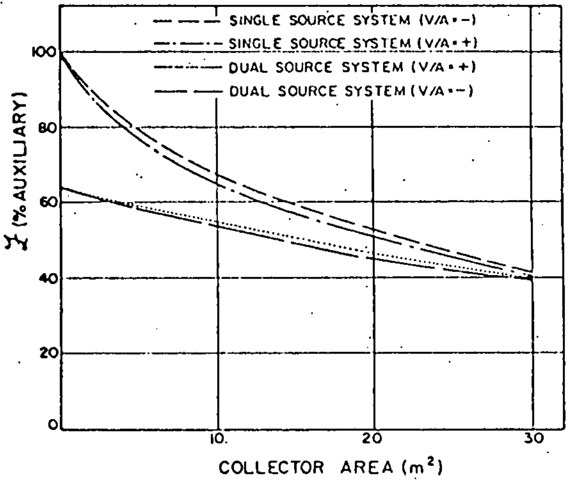

Fig. 10 System III- Madison

Storage volume has a greater effect than it did for System II but overall thermal performance is slightly lower than for System II which is a less complex configuration. System III therefore appears to have no advantage over any systems already considered. This is a particularly interesting result in light of the fact that flow reversal in the rock bed is generally beneficial to system performance in conventional solar air heating systems.

SAMPLE ECONOMIC EVALUATION

An economic study to determine generally optimum systems is not presented here because the results are influenced so greatly by the costs assumed for electricity, collectors, heat pumps, etc. By biased selection of these numbers almost any system can be made to appear economically attractive. A sample economic analysis for a particular house in Madison, Wisconsin, is intended to illustrate the method and help explain economic trends rather than to make quantitative judgements.

The following values are assumed:

Annual space heating and SHW load	85 GJ
Electrical rates	\$12/GJ

Equipment costs for solar system not collector related	\$1000
Cost of collector and storage	
for small V/A	\$90/m^2
for large V/A	\$100/m^2
Cost of three-ton heat pump*	\$2000
Equipment costs for solar HP system not collector related	\$300
Yearly heat pump maintenance cost	\$60
Annual cost factor (20 years, 7.75%)	.10

The basis for all "savings" assumes that electrical resistance heating is the alternative, with an annual cost of \$1020. The savings for the various systems are plotted in Fig. 11. For this particular set of costs, it can be seen that the economic optimum is a 26 m^2 conventional solar heating system which saves \$138 per year. Of the solar heat pump systems, the optimum is a dual source, 15 m^2 system saving \$115 per year. Increases in electrical costs or heat pump costs would make the conventional solar more competitive while increased collector costs would make the conventional air-air heat pump appear better.

CONCLUSIONS

In order to provide some insight into system operation, the sources of the energy used for space and SHW heating for System I in Madison are shown in Fig. 12. The dual source system operates like a single-source system with a separate air-air heat pump as an auxiliary energy source. It appears that the storage tank temperature is always higher than ambient since nearly identical amounts of solar energy are utilized by both systems. The extra W_{HP} and the Q_{AIR} for the dual source system exactly balance the amount of Q_{AUX} eliminated, so the amount of energy that the ambient source option saves is equal to Q_{AIR}.

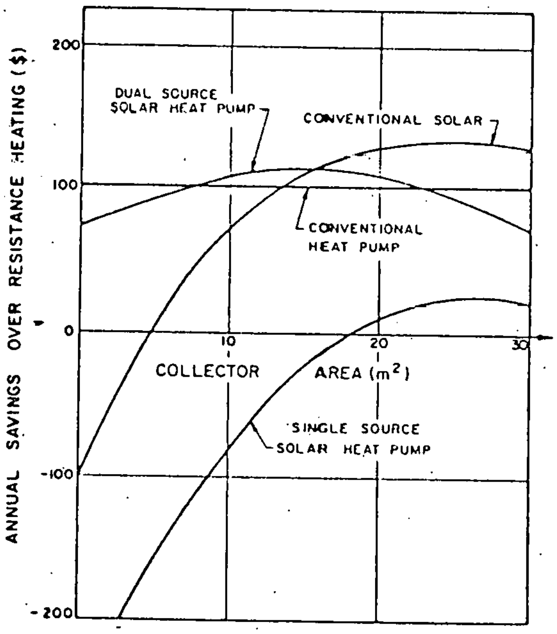

Fig. 11 Sample Economic Results

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*Cost of the heat pump should be reduced if an air conditioning system would have been purchased otherwise. In Madison, air conditioning is not necessary so the full \$2000 should be amortized for heating purposes.

Solar-augmented heat pump systems always result in energy savings compared to electric resistance heating (F = 1.00). However, when compared to air-air heat pumps or conventional solar heating systems the savings are much less striking. In general, solar heat pump systems operate at low capacity and low COP during the coldest months when the majority of the heating load occurs. This can be seen from the performance results, Figs. 6-10. More energy savings can be realized in Madison than in Albuquerque or Charleston, primarily because the annual heating load is larger. A large service hot water load significantly reduces the thermal performance of the combined system.

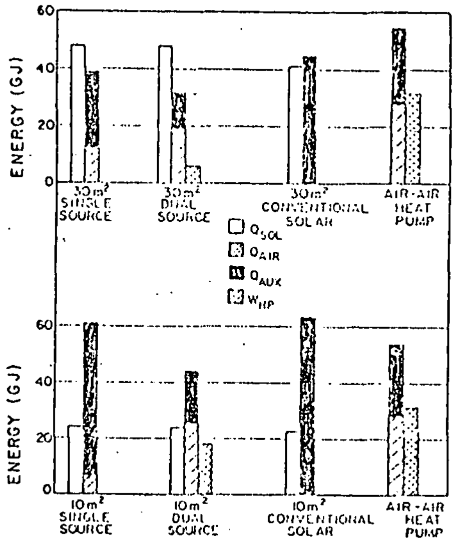

Fig. 12 Annual Heating Energy Sources System I - Madison

The dual-source heat pump system has the best thermal performance of any system. However, economically, it must compete with both the conventional heat pump and conventional solar system. It appears that the dual-source system will be most feasible at relatively small collector areas.

The advantage of single-source systems over conventional solar systems is that the collector efficiency is slightly higher due to lowered collector temperature. However, as in conventional solar systems, the thermal storage is still frequently depleted (too low in temperature to be used as the heat pump source) and the system must rely on pure auxiliary. Thus the thermal performance of these systems is only slightly better, and may not be sufficient to justify the additional cost of the heat pump. In order to show an advantage over a conventional heat pump system, the collector area must be large enough so that the combined system performance is better than that of the conventional heat pump system. Clearly, economic considerations are very important for this configuration.

The possibility of direct solar heating increases performance for all systems, and would be relatively simple to incorporate. Bypass heating would also reduce heat pump operating time, and may increase compressor life.

Of the various systems studied, only the single-source water system shows significant advantages in using larger storage volumes than suggested for conventional solar systems. This is particularly true for the dual-source system, where large volume-to-area ratios reduce performance. Seasonal storage appears to have possibilities, but may not be economically feasible.

Collector design is also important for system performance. The coverless collector systems do not appear feasible in any of the climates tested due to high-collector losses. Air systems require higher flow rates for the heat pump heat exchangers than desirable for collectors, which may necessitate special collectors.

It is evident that optimum solar heat pump systems require significantly less collector area than optimum conventional solar heating systems. While it may be possible to save using a single-source heat pump system, under nearly all possible cost conditions more is saved using a dual-source solar heat pump system, conventional solar system, or an air-air heat pump. Higher electrical costs or heat pump costs favor conventional solar while higher collector costs make air-air and dual-source heat pump systems more attractive. From the standpoint of energy savings the dual-source solar heat pump systems are best but economics will generally dictate the choice. In climates where a cooling system capital expenditure is required, part of the heat pump investment can be credited to cooling and the annual cost of solar heat pump systems will be lowered. In that case there appears to be a small economic advantage for their use. In any case, the findings of this study indicate that there is not a clear-cut incentive for the general utilization of solar heat pump systems.

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