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SOLAR HEATING AND COOLING - TASK VII

central solar heating plants with seasonal storage

evaluation of concepts

November 1986

NOTICE

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SOLAR HEATING AND COOLING - TASK VII

Subtask II (b):
Evaluation of System Concepts

central solar heating plants with seasonal storage

evaluation of concepts

November 1986

by:
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PREFACE

INTRODUCTION TO THE INTERNATIONAL ENERGY AGENCY (IEA)

The International Energy Agency was formed in November 1974 to establish cooperation among a number of industrialized countries in the vital area of energy policy. It is an autonomous body within the framework of the Organization for Economic Cooperation and Development (OECD). Twenty-one countries are presently members, with the Commission of the European Communities (CEC) also participating in the work of the IEA under a special arrangement.

One element of the IEA's program involves cooperation in the research and development of alternative energy resources in order to reduce excessive dependence on oil. A number of new and improved energy technologies which have the potential of making significant contributions to global energy needs were identified for collaborative efforts. The IEA Committee on Energy Research and Development (CRD), supported by a small Secretariat staff, is the focus of IEA R & D activities. Four Working Parties (in Conservation, Fossil Fuels, Renewable Energy, and Fusion) are charged with identifying new areas for cooperation and advising the CRD on policy matters in their respective technology areas.

IEA SOLAR HEATING AND COOLING PROGRAM

Solar Heating and Cooling was one of the technologies selected for joint activities. During 1976-77, specific projects were identified in key areas of this field and a formal Implementing Agreement drawn up. The Agreement covers the obligations and rights of the Participants and outlines the scope of each project of "task" in annexes to the document. There are now eighteen signatories to the Agreement:

Australia	Italy
Austria	Japan
Belgium	Netherlands
Canada	New Zealand
Commission of the European Communities	Norway
Denmark	Spain
Federal Republic of Germany	Sweden
Greece	Switzerland
	United Kingdom
	United States

overall program is managed by an Executive Committee, while the management of the individual tasks is the responsibility of Operating Agents. The tasks of the IEA Solar Heating and Cooling Program, their respective Operating Agents, and current status (ongoing or completed) are as follows:

Task I	Investigation of the Performance of Solar Heating and Cooling Systems - Technical University of Denmark (Completed).
Task II	Coordination of Research and Development of Solar Heating and Cooling - Solar Research Laboratory - GIRIN, Japan (Completed).
Task III	Performance Testing of Solar Collectors - University College, Cardiff, U.K. (Ongoing).
Task IV	Development of an Insolation Handbook and Instrument Package - U.S. Department of Energy (Completed).
Task V	Use of Existing Meterological Information for Solar Energy Application - Swedish Meterological and Hydrological Institute (Completed).
Task VI	Performance of Solar Heating, Cooling, and Hot Water Systems using Evacuated Collectors - U. S. Department of Energy (Ongoing).
Task VII	Central Solar Heating Plants with Seasonal Storage - Swedish Council for Building Research (Ongoing).
Task VIII	Passive and Hybrid Solar Low Energy Buildings - U. S. Department of Energy (Ongoing).
Task IX	Solar Radiation and Pyranometry Studies - Canadian Atmospheric Environment Service (Ongoing).
Task X	Materials Research & Testing - Ministry of International Trade and Industry, Japan (Ongoing).
Task XI	Passive and Hybrid Solar Commercial Buildings - Swiss Federal Office of Energy

TASK VII - CENTRAL SOLAR HEATING PLANTS WITH SEASONAL STORAGE: FEASIBILITY STUDY AND DESIGN

In colder climates, solar energy for heating of buildings is least abundant when it is needed most - during the winter. Therefore, seasonal storage is needed to make solar heat gained during warmer months available for later use. From investigations of various storage methods, two observations can be made: (1) The choice of storage method will greatly influence the working conditions for, and the optimal choice of the solar collectors and the heat distribution system; and (2) based on the technology that is available today, the most economic solutions will be found in large applications.

The objectives of Task VII of the IEA Solar Heating and Cooling Program are to determine the technical feasibility and cost effectiveness of large-scale, seasonal storage solar energy systems for the heating of buildings; to evaluate the merits of alternative large-scale system designs for collecting, storing, and using solar energy; and to prepare detailed system designs for specific site parameters.

In the first phase of the Task, which was finished in June 1983, the initial emphasis was on the development and collection of design data, followed by presentation of preliminary designs by each Participant. The Phase I subtasks and lead countries were as follows:

Subtask I(a) - System Studies and Optimization (Canada)
Subtask I(b) - Solar Collector Subsystems (USA)
Subtask I(c) - Heat Storage (Switzerland)
Subtask I(d) - Heat Distribution System (Sweden)
Subtask I(e) - Preliminary Design Study (Sweden).

Phase I was immediately followed by Phase II, which ended in December 1985. The purpose of Phase II was as follows:

- To compare simulation results from the MINSUN program, which was developed during Phase I, with other similar or more detailed tools
- To examine a wide range of system configurations, operational strategies and load/location characteristics
- To recommend which configurations deserve further attention for specific applications, and which configurations are economically least attractive
- To enhance the MINSUN program to cover a wide range of configurations
- To prepare for further cooperative use of data from existing plants to validate the design data from Phase I and II and to evaluate components, systems, control strategies, etc.

The work in Phase II was organized in three subtasks as follows:

Subtask II(a) - MINSUN Enhancement and Support (Canada)
Subtask II(b) - Evaluation of System Concepts (USA)
Subtask II(c) - Exchange of Detailed Engineering Data and Experience with CSHPSS Systems (Netherlands).

This report deals with the system analysis and parametric studies performed under Subtask II(b).

ACKNOWLEDGEMENTS

This report is the combined effort of all participants in Subtask II(b). The participants prepared, ran, analyzed, and plotted results from thousands of computer simulations in order to produce the results it contains. The analytic work was divided among three teams whose efforts were coordinated by J.C. Hadorn of Switzerland, Allan Michaels of the U.S.A., and Heimo Zinko of Sweden. Each team prepared reports which form the basis of this report. In addition, the present study relies heavily upon the work done in the first phase of Task VII, and this report has drawn freely from the material in the reports and working papers from Phase I. It would be impossible to acknowledge all of the important contributions from all the participants, but the author is especially grateful for the contributions of Verne Chant of Canada for development of the optimization methodology and to Heimo Zinko and J.C. Hadorn for their technical leadership and tireless work.

The author's participation in this task was made possible by support from the United States Department of Energy through a contract with the Argonne National Laboratory, under the direction of Dr. Frederick H. Morse at DOE and Dr. Allan I. Michaels at ANL.

All of the participants in Task VII have been incredibly supportive through their comments and suggestions during the study and the preparation of the report. The author and all the participants also wish to acknowledge the support and encouragement of the Task Operating Agent, Arne Boysen of Sweden.

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1.0 INTRODUCTION

1.1 TASK VII - BACKGROUND

The objective of Task VII was to determine the technical feasibility and cost-effectiveness of central solar heating plants with seasonal storage (CSHPSS). It was planned that Participants in this Task would evaluate the merits of various components and system configurations for collecting, storing and distributing energy, and would prepare site-specific designs. The work originally was divided into two phases, preliminary design and detailed design.

1.1.1 Phase I

Phase I was organized in five subtasks:

a) system studies and optimization
b) solar collector subsystems
c) heat storage subsystems
d) heat distribution subsystems
e) preliminary site-specific design.

A substantial effort in Subtask (a) of Phase I was devoted to thermal and economic analytic tool development and preliminary site-specific design [1].* However, since the analytic tool development took longer than anticipated, in many cases the site-specific designs were prepared using analytic tools and design aids developed nationally. These national tools and design aids were often specially developed and suited to the site-specific designs undertaken. At the end of Phase I, the analytic tools developed within the Subtask II(a) (primarily MINSUN, but incorporating specially modified collector subsystem, storage subsystem, and heat pump models) were capable of analyzing a wide range of CSHPSS configurations [2]. Even with this wide range of applicability, however, in some cases these tools were modified by Participants in order to model the specific configurations and operational strategies represented by the preliminary site-specific designs completed in Phase I.

At the end of Phase I, the common analytic tools were evaluated by applying them to several cases of interest by Participants [1]. These application cases were useful for assessing the range of applicability of the tools, the usefulness of the type of information they provided, and the appropriateness of the level of detail of the simulation and economic models. The consensus of the Participants who undertook these application cases was that these analytic tools were appropriate for an extensive evaluation of a wide range of

*Numbers in square brackets refer to references listed at the end of the report.

configurations on a common basis and such an exercise would be worthwhile in a revised Phase II work plan. The experience from the application cases undertaken pointed to several enhancements of the analytic tools which should be made to make that exercise most effective.

The Subtask I(b) effort reviewed the performance, cost engineering data, and operating experience relevant to collector subsystems suitable for CSHPSS. The final report [3] provided the necessary performance models and cost data for flat plate collectors, evacuated collectors, parabolic troughs and central receivers used in the current work. In addition, the final report contained information on the design, installation, operation and maintenance of these collectors.

Three final reports were published as a result of the Subtask I(c) work on thermal energy storage. Reference 4 presents basic engineering information for different concepts that store energy in the form of sensible heat. It describes the heat storage concepts and their applicability to CSHPSS systems, reviews the status of the technologies, and presents a short technical compilation of some of the interesting projects in the participating countries. Reference 5 discusses simulation models that are available throughout the world for the analysis of seasonal sensible heat storage systems. It provides the basis for selection of the computer models that were used in the present study. Reference 6 summarizes the relevant cost data for seasonal, sensible heat storage construction and provides the cost equations that were used in the economic analysis of this work.

The distribution of thermal energy to buildings by means of hot water was reviewed by the Subtask I(d) Participants, and their final report [7] provides basic design and cost data applicable to a variety of systems in each of the participating countries. Information in this report is provided at the subcomponent level, i.e. pipe and fittings, rather than the subsystems level that was needed for the parametric evaluations of this work. The component data will be of great value in detailed designs.

The final subtask of Phase I involved site-specific preliminary design studies of CSHPSS systems in each of the participating countries [8]. The study, research, analysis and evaluation involved in the ten design studies provided a solid basis of expertise from which to launch the present study using a more refined methodology than was available during Phase I.

1.1.2 Phase II

Phase II was intended to continue into detailed site-specific design for those Participants who chose to continue participation in Task VII. However, near the end of Phase I, there was consensus among Participants that the overall Task objective could be better served by revising the work plan for Phase II.

Some Participants had already completed detailed design of a CSHPSS according to their own national priorities and schedules, and other Participants were not planning to proceed with detailed design. Discussions by Participants at the Fifth Experts' Meeting in February, 1983 led to the adoption of work described here as one useful component of Phase II.

Phase II is organized into three subtasks:

a) MINSUN enhancement and support
b) Evaluation of systems concepts
c) Exchange of detailed engineering data and experience with CSHPSS systems.

Only II(b) is discussed in this report.

1.2 OBJECTIVES OF SUBTASK II(b)

The objectives of the revised Phase II Subtask (b) fully complement and support the overall Task objective stated above. These Phase II objectives are:

- To examine a wide range of CSHPSS configurations, components and load/location characteristics in order to compare and rank, from both a thermal and economic perspective, CSHPSS designs and applications

- To recommend configurations, from those examined, that deserve further attention for specific applications and identify those which are unlikely to be economically attractive under specified assumptions

- To make these findings readily available in the form of a major report and the presentation of one or more symposia for both technical audiences and decision-makers

- To identify major uncertainties in analysis, modeling, or data and to coordinate with Subtasks II(a) and II(c) to resolve them or to develop plans for Phase III that will lead to their removal.

1.3 SCOPE

The scope of the Phase I studies was broad, in order to include all configurations and components that made technological or economic sense and so that the information would be applicable to all the climatic and economic conditions in participating countries. This meant that the solar collector study included all collector types from solar ponds to central receivers. The preliminary design concepts incorporated features such as distributed collectors, distributed and central buffer storage, heat pumps for thermal stratification of storage and passive building architecture.

The Phase II(b) evaluation study, although sharing the objective of breadth, was constrained to examine only the options that, in the view of the participating experts, might be technically and economically viable in the participating countries in the mid-1980's. The means by which the set of options was reduced to a workable set are described in Section 2 and the technologies and conditions selected for the reference study are discussed. Briefly the scope of study comprises:

- o Collectors -- restricted to non-tracking types
- o Storage -- earth, rock, and water
- o Energy conditioning -- heat pumps
- o Auxiliary energy -- fuel fired or electric boilers
- o Loads -- building space heating and domestic hot water
- o Configuration -- restricted to those allowed by MINSUN
- o Optimization -- subsystems optimally sized.

1.4 APPROACH

The large number of system configurations and parametric variations that could be considered within the constraints of the MINSUN code demanded that the first step in the project be the narrowing of choices to a manageable set. The second step in the process was the optimization, parametric analysis, and evaluation of the general concepts (or reference cases) for a range of parameters. Optimization is used in the general sense of selecting the most cost-effective combination of components and parameter values for a given concept. The result of this second step is a set of optimized configurations that allows ranking for at least two different climates.

The third step was the localization of the optimization and evaluation to the national climates and economic environments of the participants. Countries selected one or more concepts that appeared to be well-suited to national context and repeated the optimization and evaluation of Step 2 using local weather and parameters. The results of the national evaluations enlarged the data set containing the reference cases and provided a number of variations based on real conditions.

The entire data set was analyzed and summarized to meet the primary objective of the subtask, i.e., the evaluation (on paper) of the technical and economic feasibility of CSHPSS. The analytical results form the basis for further investigation of technical and practical feasibility in the field.

The process just described involved interaction with Subtasks II(a) and II(c) regarding the validity of the methods and models used in the analysis and the need for field experience to confirm the methods used in the analysis. This interaction can be regarded as a systematic process to validate the way in which the project objectives were achieved.

It was anticipated that the results of the subtask efforts would be of major interest and importance to national decision-makers as well as to solar researchers. Therefore, in addition to this summary report, it was decided that a symposium be developed in which the major findings of the Task could be presented to appropriate audiences. Presentations were made at the International Solar Energy Society Meeting in Montreal, Canada in June 1985 and the Third International Conference on Energy Storage in Building Heating and Cooling, Toronto, Canada, September 1985. A number of technical papers have been written, and there have been numerous contacts with builders, utilities, and manufacturers.

1.5 ORGANIZATION OF THE WORK

Eight participating countries in Task VII were involved in Subtask II(b). The calculational effort was divided among three teams, each of which concentrated on a specific storage technology. The subtask organization and national efforts are shown below.

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2.0 METHODOLOGY

The general approach to this evaluation study has been outlined in Section 1.3. This section provides a more detailed description of the methodology and discusses the major analytical tools and resources. Even this presentation, however, is quite limited. Each of the analysis teams performed the many simulations required to produce the results on which this report is based, and many of those calculations involved special methods or treatments. For example, a special procedure had to be devised to set a collector feedwater temperature for aquifer systems because the aquifer model could not provide a return temperature from a second cold water well in a doublet. These details are not included here but can be found in the working reports and documents of the analysis teams [9,10,11].

2.1 SCREENING AND SELECTION OF REFERENCE CASES

The components studied in Phase I of this project can be configured in an enormous number of ways to form systems for central heating plants. Considering only the major subsystems, such as the collector array, the storage unit, the energy conditioning device, the auxiliary energy source, the distribution system, and the load, and only a few of the many possible arrangements of these subsystems, it was estimated that about 500 configurations should be considered. In addition, the economic viability of any of the system configurations depends upon its location (latitude, climate, and geology), its economic environment, and the manner in which the system and the load are controlled. Even a small subset of these variables would require that each system be evaluated under about 50 different sets of conditions, thus bringing the total number of evaluations required to 25,000. Such an undertaking was obviously not feasible within the constraints of the resources available for the subtask work, so the first task was to reduce the number of evaluations required by making use of the knowledge and good judgment of the Task VII participants.

Two methods were used. A formal hierarchical decision tree method in which the decision process is carefully structured, weighted, and evaluated with the aid of a microcomputer was used by half the group. The other half of the group followed a more traditional process of forming discussion groups to rank subsets of options (based on storage technology) and then comparing and combining those rankings in a consensus-finding meeting of the small groups. The results of the two processes were surprisingly similar. The agreement between the finding of the two expert groups strengthens the credibility of the process and the combined judgment.

The configurations that were selected and the components that were included were restricted to those that could be modeled using MINSUN. Although this is somewhat restrictive, the limitations are not considered serious. In addition, some of the promising configurations that could not be modeled with MINSUN were subsequently studied using the more flexible TRNSYS program. The general configuration and set of components chosen for the reference study, along with the evaluation conditions, are shown in Figure 2-1.

Although only stationary collectors were included in this phase of the work, the full set of storage technologies were represented. A theoretical electric heat pump model was the only energy conditioning device included in the study. Ice and chilled water storage for summer air conditioning were not included.

The subtask experts agreed to limit severely the set of general evaluation conditions. Each configuration was to be evaluated with only two climatic databases: Madison, Wisconsin, U.S.A., representing a severe continental climate and Copenhagen, Denmark, representing a northerly maritime climate; three load patterns: 0, 20, and 50 percent DHW; and three annual total loads: 3.6, 36, and 360 TJ (corresponding roughly to 50, 500, and 5,000 residences respectively).

The temperature required in the distribution system to meet the load is an important design consideration or constraint. Lower temperatures provide an increase in efficiency of the collector array and the heat pump, and storage and distribution losses are reduced. However, larger load heat exchangers and auxiliary energy for peaking DHW may be required. Two specifications, a low and high-temperature demand, were assumed for this evaluation, to represent construction and retrofit loads respectively. The distribution supply and return temperatures for the low and high temperature specifications are shown in Figure 2-2. Note that the specified temperatures require a varying mass flow in order to meet the building load.

A complete set of common parameters required for the analysis was selected from the database developed during the first phase of the work. Collector cost and performance parameters were taken from the Subtask I(b) report [3], the storage data from the Subtask I(c) reports [4, 5, 6], the distribution systems data from the Subtask I(d) report [7], the heat pump data from reference 12, and the economic and system control parameters from the Subtask I(a) reports [1, 2]. A complete listing of the common parameter set as required to simulate the performance of any of the system configurations is included as Appendix A.

Parameters specific to each storage technology--for example, the diameter of bore holes in a duct storage system--were determined by the analyst of that technology option and were usually derived through a subsystem optimization procedure.

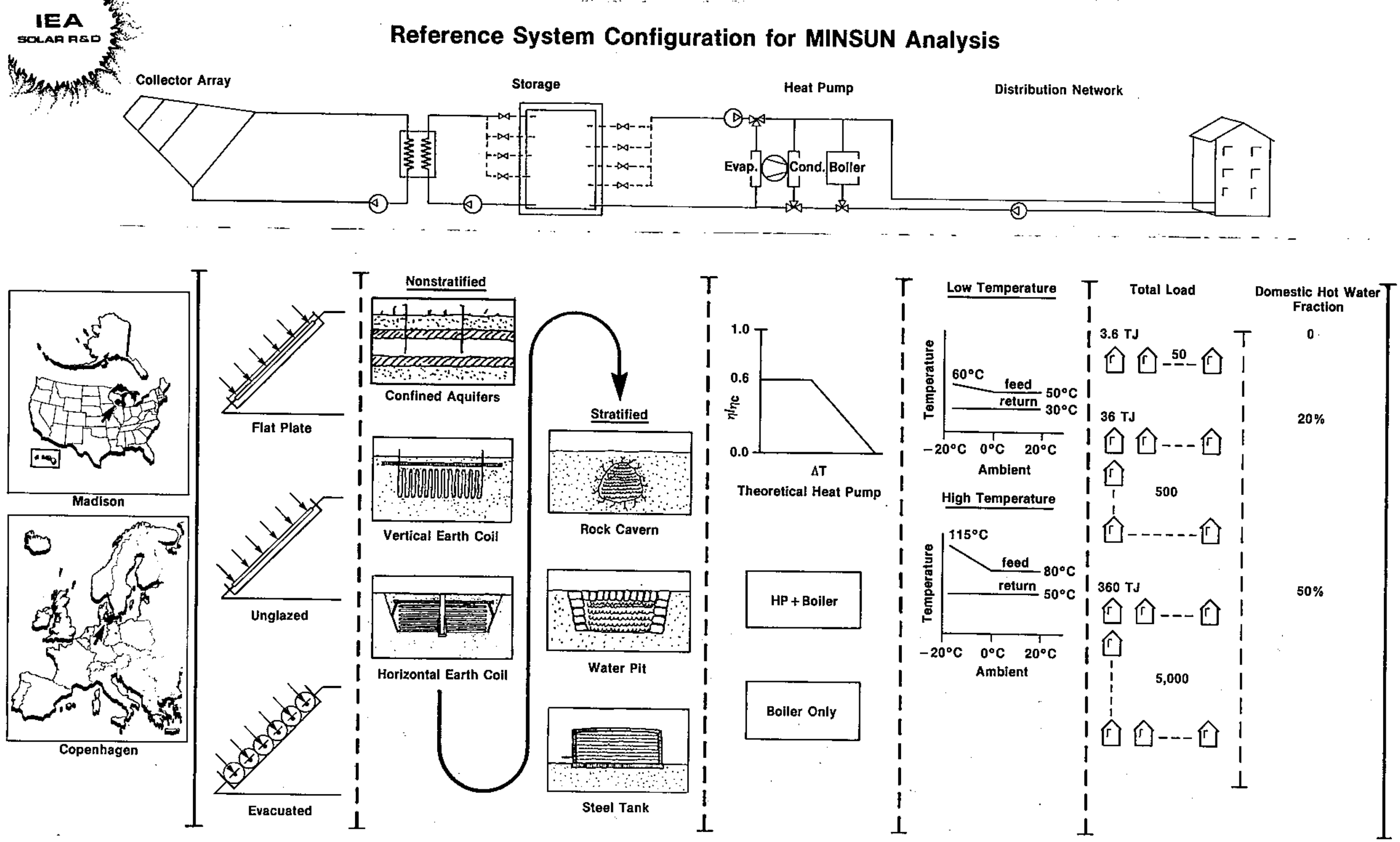
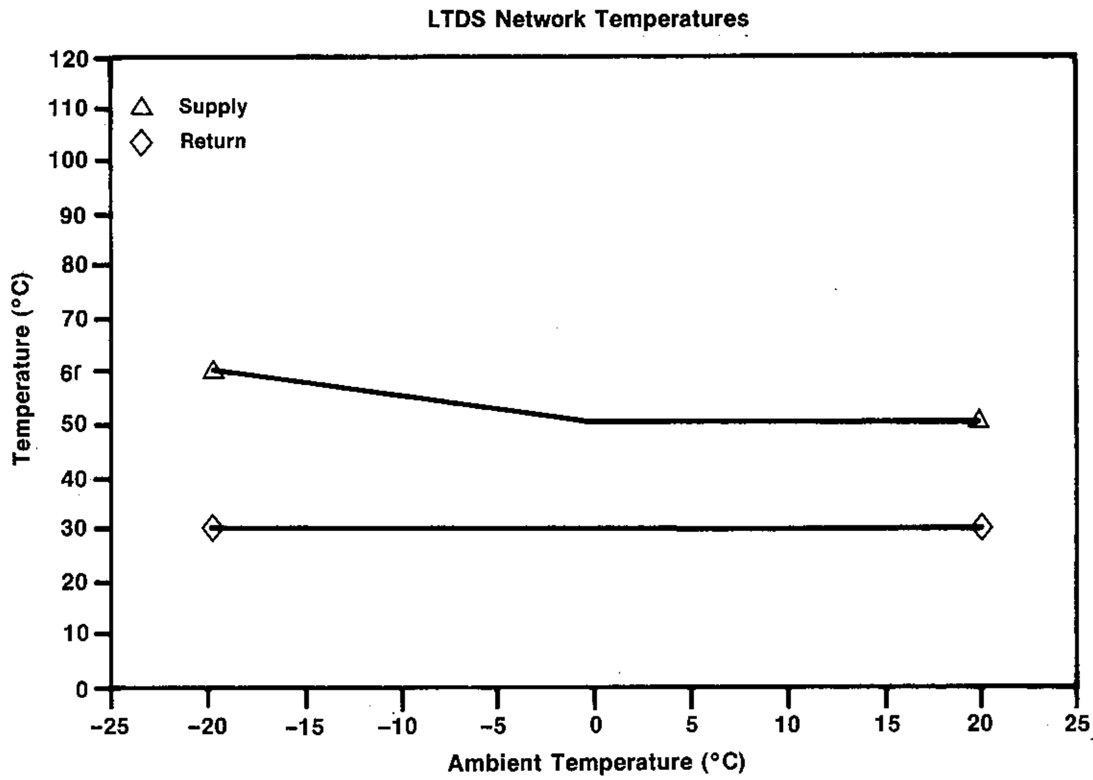


Figure 2-1. Reference System Configuration, Components, and Evaluation Conditions



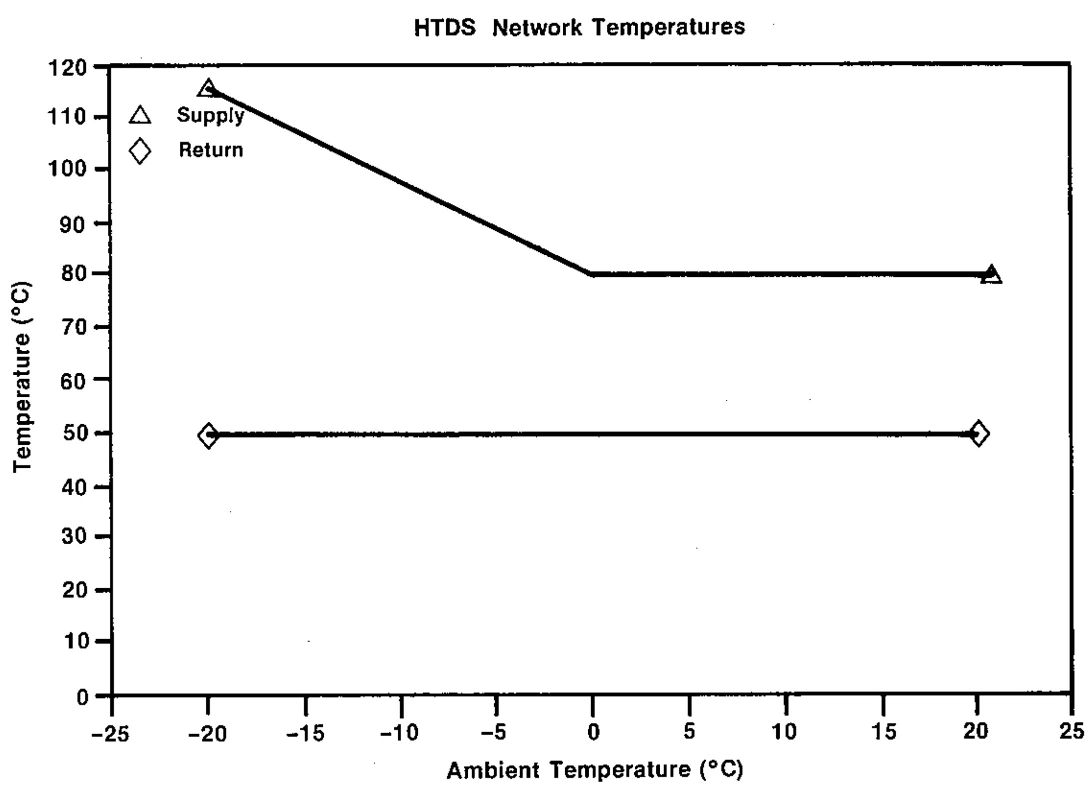


Figure 2-2. Specification of Delivery and Return Temperatures for Low- and High-Temperature Distribution Systems as a Function of Ambient Temperature

2.2 ANALYSIS METHODS

There were two main computer program system models used in Task VII. The MINSUN set of programs, which were originally developed at Studsvik in Sweden, were significantly modified and extended specifically for Task VII. This set of programs provides CSHPSS simulation, economic analysis, and optimization. The second main computer program system model used in Task VII was TRNSYS Version 11.1. This version has a few modifications for Task VII purposes but is essentially the University of Wisconsin program. TRNSYS provided the capability for detailed simulation and subsystem examination. These analytic tools are described below.

2.2.1 MINSUN

The MINSUN [2] solar system simulator is a set of FORTRAN programs that models a central solar energy heating system. The programs provide for system thermal simulation, costing and economic analysis, and algorithmic optimization of selected system parameters. Subsystem capital costs are calculated by cost equations using user-specified parameters. Economic analysis combines capital costs and annual heat pump and auxiliary energy costs into an equivalent levelized annual cost using present value theory. Optimization is based on minimizing this levelized annual cost for a given load. The control strategy employed by the system is built into the MINSUN program.

2.2.1.1 Configuration Options

Each system is made up of several components: solar collectors, thermal storage, heat pumps, auxiliary heaters, a network of connecting pipes, and residential heat load. They are illustrated in Figure 2-3. The collector subsystem types available include flat plate, evacuated collectors (simple and compound parabolic concentrator, CPC), parabolic trough, central receiver, and shallow pond. The seasonal storage subsystem types include insulated water tanks, water storage in underground tanks, caverns, or pits; duct storage systems in earth or rock; and aquifer systems. The auxiliary heat source options are either a boiler or a boiler and a heat pump. The distribution subsystem requires only the specifications of design temperatures, flows, and costs. The system load is calculated for residential space heating taking account of daily average outdoor temperature below a threshold temperature and within a specified "heating season" and indoor setpoint temperature by simple heat-loss and heat-gain equations. DHW load is set as an input parameter.

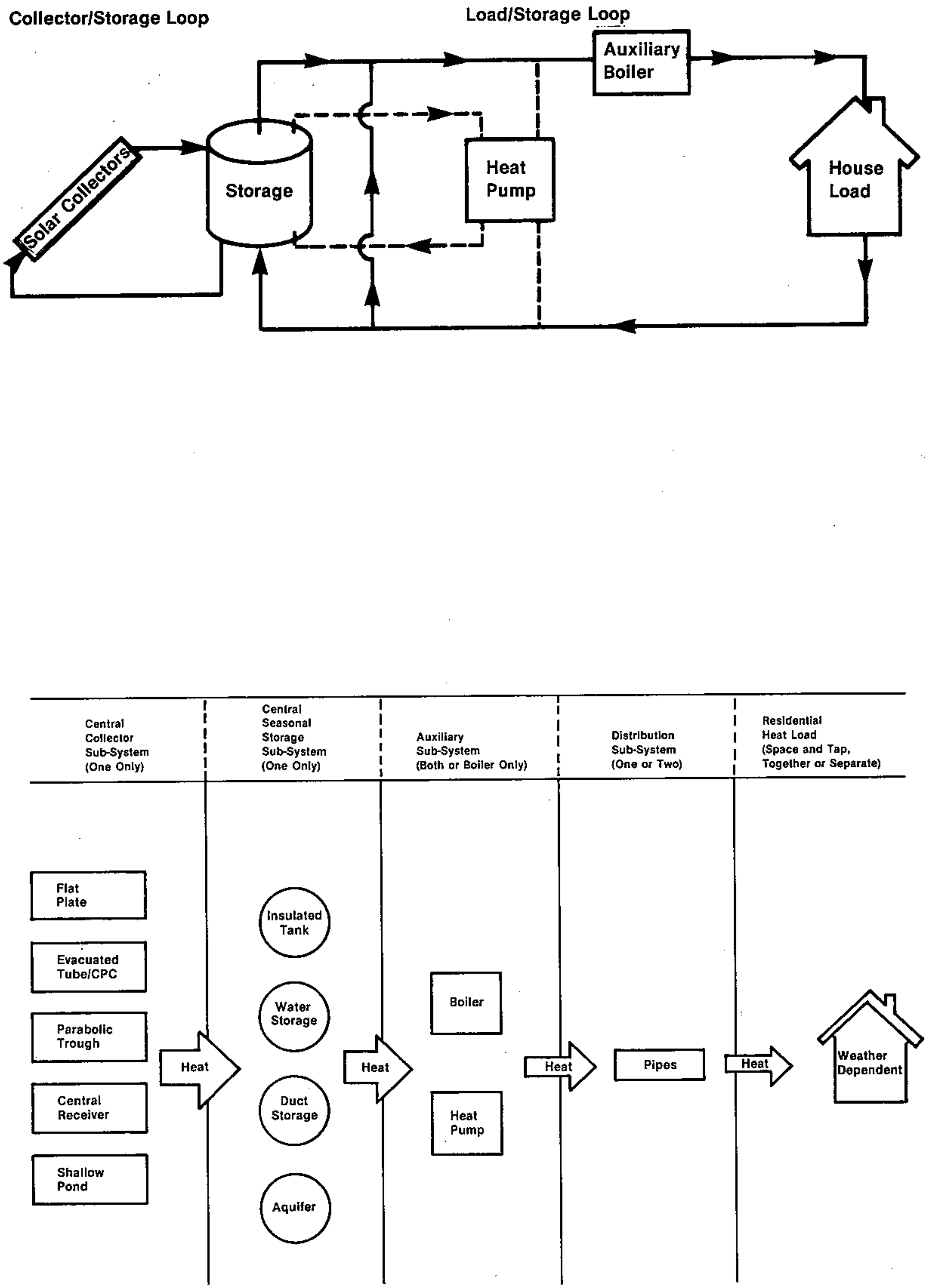


Figure 2-3. Solar System Configuration and Components for MINSUN Simulations

2.2.1.2 Execution Modes

The MINSUN program can be used both to simulate the thermal behavior of a central solar energy system and to determine the optimum size of some of the components in the system. These two MINSUN characteristics provide three different modes for running the program - Single Simulation, Multiple Simulation, and System Optimization. All three modes require the engineering parameters of the system being modeled and weather information to drive the simulator. The three different modes are described below.

Single Simulation of a System Configuration

The simplest use of MINSUN is to perform a thermal simulation for a given, fixed configuration. All parameters of the system are defined. The program simulates the thermal behavior, does the energy balance calculations, cost calculations, and generates output on the thermal and economic characteristics of the system specified. The thermal characteristics include a daily specification of heat flows and temperatures among the major subsystems (from collectors, to and from storage, to load, losses, etc.).

Multiple Simulation of a System Configuration

Any system design parameter can be varied during multiple simulation runs. Key variables usually include collector area, storage volume, storage height to diameter ratio, storage insulation thickness, specific heat transfer of heat pump evaporator in space distribution system, specific heat transfer of heat pump condenser in space heating distribution system, specific heat transfer of heat pump evaporator, number of pipes in a duct storage system, control setpoint temperatures, etc.

Any system design parameter can be varied during multiple simulation mode using the MINREP procedure. Single or iterative changes of key system parameters can be specified in this mode. In addition, any results which appear on the standard detailed Simulation Summary output can be specified for inclusion in the summary output. In order to decrease the number of runs needed to define the optimum, scaling factors can be used to relate variables, e.g., the number of boreholes to the storage volume and collector area.

System Optimization

The MINSUN programs also have the capability of automatically selecting values for certain design variables that minimize overall system levelized annual cost. The program uses a steepest descent algorithm to find the values of the design variables which minimize overall cost. It simulates the thermal behavior and computes the cost of the current system and compares it with that calculated in previous iterations. The program systematically closes in on the values of the design variables that minimize system cost. Up to nine design variables can be selected for the optimization process. However, the computation time increases significantly as the number of variables to be optimized is increased. Once the optimum system, as selected by the optimization algorithm, is determined, the program performs a single simulation and and re-calculation and prepares summary data on the thermal and economic characteristics.

Experience with MINSUN automatic optimization procedures for typical CSHPSS systems showed that reliable determination of the optimum was difficult and time consuming even when the number of variables to be optimized is small. This is due to the relatively flat cost surface near the optimum and to small scale roughness of the surface due to the numerical procedures used in many of the component models. With the automatic search procedure, usually 150 to 200 simulations were required to locate a minimum. With the level of detail of thermal simulation in the storage subsystems, these simulations were costing from two to three dollars each on a typical main frame computer. Since the objective of the study was to examine a broad range of configurations and conditions, a more economical approach was sought. This approach is described in Section 2.5.

2.2.2 <u>TRNSYS</u>

The TRNSYS simulation program was selected by Subtask I(a) for detailed simulation. Version 11.1, made available to Task VII, was modified by the University of Wisconsin (May, 1982) to accommodate collector models of special interest in Task VII. The Lund University storage models which were adopted for Task VII were made compatible with TRNSYS [13, 14, 15]. They could be run separately with the TRNSYS main program but were never integrated into a complete configuration system model.

TRNSYS analyses were used to verify simulations of the MINSUN program and to analyze some configurations that could not be modeled using the restricted MINSUN configuration and control options.

2.3 SUBSYSTEM PERFORMANCE MODELS

The performance models used for the collector, storage, heat pump, and load subsystems were derived from the information collected during the first phase of Task VII. The guideline for developing the models was the performance that could be expected for a system designed in 1984-85. Since most of the data were collected between 1981 and 1983, it was necessary to make some projections from the commercial products of 1981-83 to the anticipated products of 1984-85. This was done by the participants based on their knowledge of the status and intensity of research and development.

2.3.1 Collector Subsystem Models

The basic information for the collectors was developed by Subtask I(b) and implemented as a part of the MINSUN program by Subtask I(a). The procedure employs empirical mathematical relations for the collector module output as a function of the orientation of the collector and the sun (e.g. incident angle modifiers for stationary collectors or single-axis tracking collectors), the temperature of the collector fluid, the ambient temperature, and the irradiance available to the collector (i.e. beam or global on collector plane). Daily energy outputs are calculated from the hourly weather and the irradiance data for each site and stored in tables for a series of collector inlet temperatures spanning the operating range of CSHPSS systems. This calculation was done only once for each site. The MINSUN main program obtains collector output for each day of the simulation by interpolating the table using the mean collector temperature, $(T_{in} + T_{out})/2$, as the independent variable. Since T_{out} depends on the collector output, an iterative loop is necessary. The period of collector operation during the day is found in a similar manner. The performance equation and the FORTRAN implementation for MINSUN may be found in Reference 3. An abbreviated table of collector parameters used in the reference study is shown below.

COLLECTOR	η_o	$F_R U_L$ W/m^{2}K	b_o	AF	Reference
Unglazed	0.90	15.0	0.00	0.70	[11]
Flat Plate	0.81	4.4	0.10	0.66	[3]
Evacuated	0.70	1.0		0.70	[3]

The energy collected was calculated from the equations below.

$$\eta = \eta_o \kappa_{\alpha\tau} - F_R U_L (T_i - T_A) / I_a$$
$$\kappa_{\alpha\tau} = 1 - b_o [(1/\cos\theta) - 1], \text{ or } f(\theta_{NS}, \theta_{EW})$$
$$Q_C = \eta I_a AF$$

The thermal output of the collector module was adjusted using a set of energy reduction factors to account for the energy losses that result from combining a large number of modules into an array. The factors represent a conservative estimate of the <u>annual</u> energy delivery of an array of the specified module relative to the output of a single module.* These reduction factors were used in lieu of the more accurate methods that would have required knowledge of such array details as row spacing, feeder and header pipe length, diameter and

* For a different approach, see Section 5.3.

insulation, night time fluid volume, etc. Although the use of a single factor cannot be defended on theoretical grounds, an analysis of the results reported at the IEA Workshop on the Design and Performance of Large Solar Thermal Collector Arrays [16] indicated they are in reasonable agreement with operational experience when the temperatures were in the same range.

2.3.2 Storage Subsystem Models

There are four central seasonal storage models that can be used with MINSUN: insulated tank, stratified storage temperature model (SST), duct storage system (DST), and aquifer (AST).

The tank model may be used for a tank or a pit if thermal coupling with the surrounding medium can be ignored. The stratified storage model, which can be used for a tank, pit, or cavern, includes a solution of the heat diffusion equation in the surrounding media and can be used even when the store is in good thermal contact with the ground. Duct and aquifer storage codes were developed especially for those types of storage. The thermal processes simulated are different in each of these models. A brief description of each follows.

2.3.2.1 Insulated Tank Storage

Energy is stored in a water-filled insulated tank, either above or below the ground. The tank has a number of "nodes" evenly spaced between the top and the bottom of storage. Water can be injected or extracted at any of the nodes. The water at the top node will always be the warmest and that at the bottom node the coldest.

The thermal model is very simple. The water is assumed to be in homogenous layers. Mass flow and energy transfer between layers is treated as a one dimensional process. The model also allows for heat transfer through the walls of the tank into the environment. The operational strategy built into this model is indicated in the tank schematic diagram shown in Figure 2-4.

2.3.2.2 Duct Storage Temperature Model - DST

A region of rock or soil is used for heat storage. Heat is injected and extracted via a system of pipes or ducts in which a fluid is circulated. The thermal process in the storage region with its duct system is quite complex. There is a "global" temperature variation from the center of the store out to the boundaries and into the surrounding ground. There is also an important and often intense local heat transfer process around each duct. Finally, there is a variation along the ducts, which is coupled to the heat exchange between fluid and ground and involves the flow pattern of the fluid through the storage. All these processes must be fitted together. The local processes are important in order to obtain the right heat exchange between fluid and ground. But the local process depends on the global temperature level. The global temperature on the other hand is strongly influenced by the local injection/extraction of heat at the ducts.

The temperature in the storage region is represented by three parts: a global temperature, a local solution, and a steady-flux part. The total temperature at a point is obtained by superposition of these three parts. The MINSUN model is a finite difference solution of the coupled heat transfer equations. Details of the solution may be found in Reference 13. For the calculations in this report, the outer diameter of a borehole was assumed to be 0.10 m and the heat resistance between the fluid and surrounding earth was 0.05 m^2K/W.

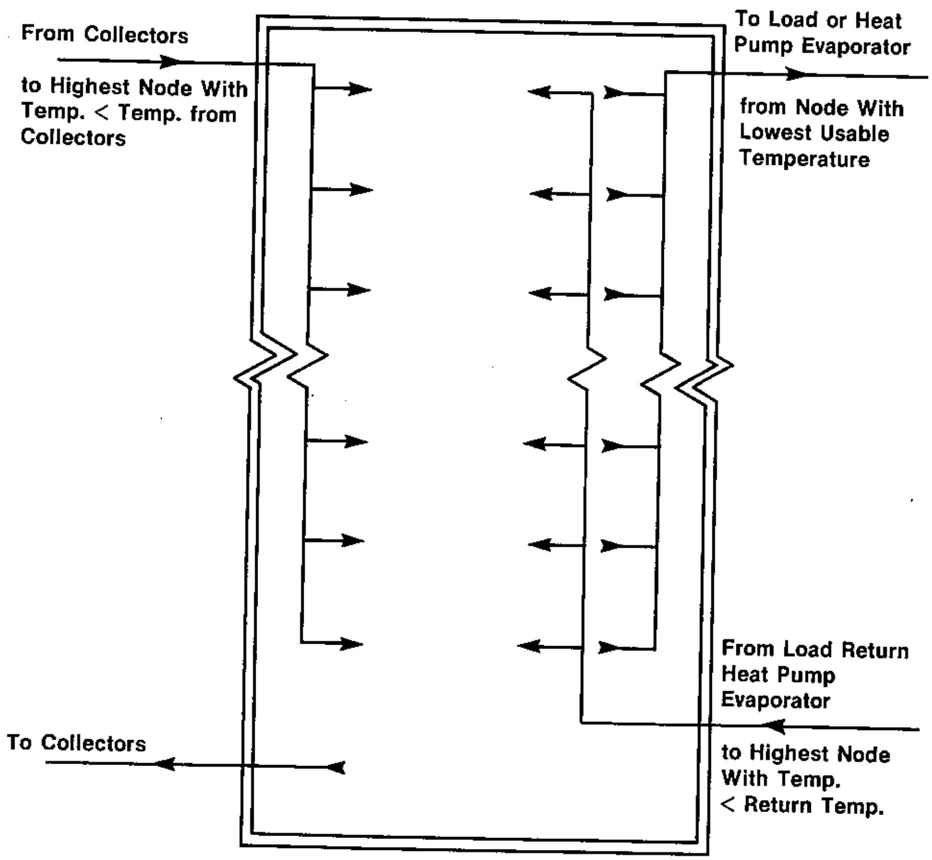


Figure 2-4. A Typical Tank Schematic

2.3.2.3 Stratified Storage Temperature Model - SST

This model simulates heat storage in water-filled tanks, caverns, or pits that are in good thermal contact with the surrounding ground. The storage volume may contain water or a mixture of water and stone. The temperatures in the storage volume are horizontally stratified. There is a vertical one-dimensional diffusive heat transfer process in the storage volume. In the surrounding ground there is three-dimensional diffusive heat flow. The two processes are coupled by the heat flow through the boundaries of the storage volume.

The storage volume is assumed to have the shape of a vertical cylinder. It is divided into horizontal layers of equal thickness. The thermal properties in the ground are given for a number of horizontal strata. The layers are divided into cells by an automatic mesh generator that adjusts according to the thermal properties of the ground, daily changes in load conditions and the duration of the simulation. A finer grid spacing is used where large temperature gradients are expected.

2.3.2.4 Aquifer Storage Model - AST

An aquifer is a porous layer in the ground, generally sand, that is surrounded above and below by impermeable material. MINSUN models a horizontal aquifer stratum of constant thickness. The thickness of the covering soil layers is also constant. The aquifer is assumed to extend a long distance from the well in all directions. The thermal properties of the aquifer are assumed to be uniform, but the caprock (above the aquifer) and bedrock (below the aquifer) can each contain several layers with distinct thermal properties.

The model simulates thermal energy storage in a confined aquifer without buoyancy. A single well is used for injection and extraction of water. The thermal processes in the aquifer and the surrounding ground are computed by a finite difference method.

Normally, an aquifer is accessed by two wells in a doublet. MINSUN does not model the thermal behavior of the second well. The temperature of water drawn from the second well must, therefore, be determined by other means [9].

The height and depth of the aquifer and the expected radius of the thermally active region are given as input values to the model. The region of computation is extended radially outwards and vertically downwards until more or less undisturbed conditions prevail. A finer grid spacing is used in the thermally active aquifer region near the well and at the boundaries between aquifer and caprock/bedrock. The spacing is increased strongly outwards and downwards. A finer grid spacing is used where large temperature gradients are expected.

A special problem with simulation of an aquifer storage system is that the storage volume is unconfined. The thermally active region in the aquifer will vary according to external conditions. It is somewhat difficult to choose a mesh for the numerical calculation which is suitable for all possible evolutions of the thermally active region. MINSUN handles this problem by repeating the simulation if necessary. For the first simulation, the mesh is generated using a specified thermal radius. During the simulation, the true thermal radius is calculated. If this calculated radius is not between 60 percent and 150 percent of the radius originally specified, the simulation is redone using the new calculated radius to generate the mesh.

The aquifer model and operation are characteristically different from the other storage methods. Operation is based on a displacement principle in that the last water injected is the first water to be extracted. There is no gradual heating of a confined volume as in the other storage systems.

For this investigation, the collector control strategy for the aquifer systems was set to provide a constant outlet temperature. Collector inlet flow was assumed to come from a second distant "cool" aquifer well which is not explicitly modeled by MINSUN. The second well also would be used to re-inject the heat pump evaporator outlet water or distribution return water, and would be above ambient aquifer temperature. This average annual cold well outlet temperature was calculated and used as the collector inlet temperature.

2.3.2.5 Thermal Properties of Soil

Throughout this study, the thermal properties of soil have been described by common values agreed upon by all participants and suggested as typical by the Subtask I(c) group. The soil or rock thermal conductivity, k, and specific heat, C_p, is given for each storage type in the following table:

	Tank	Cavern	Pit	Duct	Aquifer
k(W/m K)	(above ground)	3.5	2.0	2.0	2.00 Horizontal 2.75 Vertical
C_p(MJ/m^{3}K)	--	2.0	2.0	2.5	2.5

2.3.2.6 The Solar Collector/Storage Loop

The amount of energy absorbed and retained by the solar collectors depends on the temperature of the water supplied to the collectors.

When Insulated Tank or Stratified Storage Temperature (SST) models are used, this water is always drawn from the bottom (i.e. the coolest) node in the tank. After passing through the collector, the heated water is returned to the highest node which has a temperature lower than that of the incoming water.

With the Aquifer model, water is assumed to be supplied to the collectors from a remote well which is not affected by the operation of the storage. The water is supplied at a constant temperature throughout the year and is always returned to the storage well.

For Duct Storage, the calculation is somewhat more complex since the water from the collectors is circulated through the duct system where it loses some heat and is then fed back to the collectors. Thus the fluid temperature from the store depends on the temperature of the fluid delivered from the solar collectors and the flow rate. An iterative procedure has to be used to calculate the temperature of the water supplied from the storage each time the loading conditions are changed.

2.3.2.7 The Heat Load/Storage Loop

The heat load consists of house heating and tap water heating. If separate space and DHW distribution systems are used, the flow loops for these loads have the same control logic.

The water from the store, if warmer than the return water from the heat load, is supplied to the heat load. It is mixed with the return water from the heat load to yield the right temperature. If the temperature of water from the store is lower than the demand temperature of the heat load, it is boosted to the right temperature by a heat pump or an auxiliary heater.

With the Insulated Tank model and the Stratified Storage Temperature model, the water can be drawn from any node in the storage. It is taken from the node with the lowest temperature above the demand temperature and from the node with the highest temperature below the demand temperature and is mixed in order to supply the correct temperature to the load.

If a heat pump is included in the configuration, it takes water from the node with the lowest usable temperature. Finally, an auxiliary heater is used instead of the heat pump if the heat pump cannot deliver the required temperature.

Water returning from the heat load loop is always returned to the water storage at the highest node with a temperature less than the return flow temperature.

In the Duct and the Aquifer Storage models, there is only one source temperature of water from the storage. For DST, the water outlet temperature depends upon the energy collected and the flow rate. As in the solar collector/storage loop calculation, this is estimated using an iterative procedure. For AST, the water temperature is as extracted from the storage well.

2.3.3 The Electric Heat Pump Model

The central heat pumps are used to transfer heat from a low-temperature source (storage water) to a higher temperature receiver (house heating water). Work done on the system is fully recovered as heat. It usually represents less than a third of the total energy output of the heat pump.

A heat pump is composed of four main sections: evaporator, condenser, compressor, and valve (see Figure 2-5). The condenser and evaporator are heat exchangers; the valve acts as an expansion nozzle. The theoretical model in MINSUN for the electric heat pump employs the equations below. A model for the gas-driven heat pump is described in Section 4.

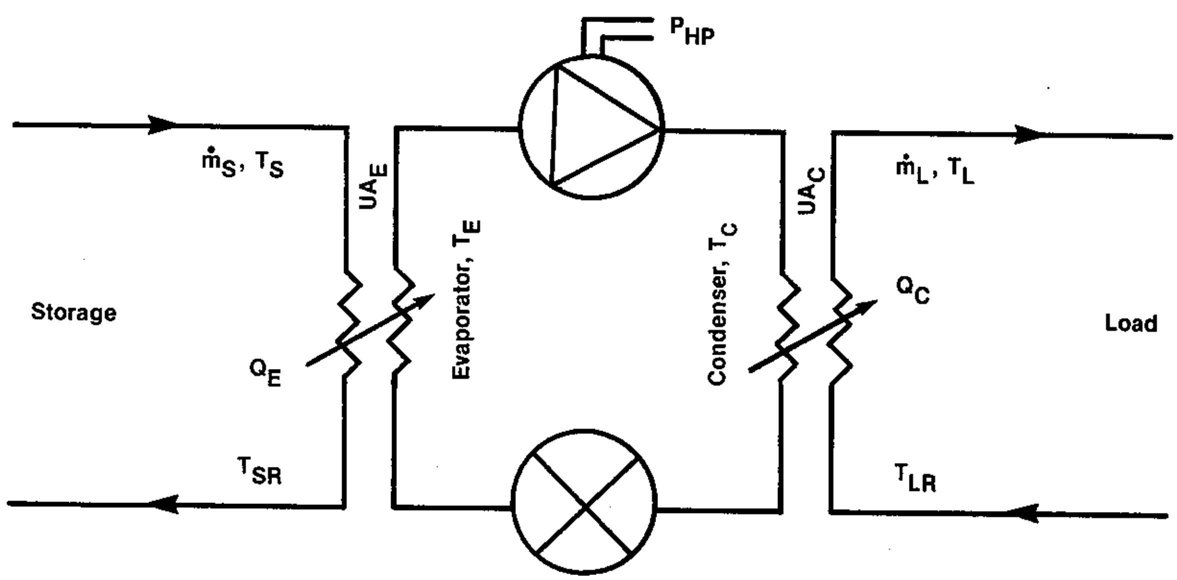


Figure 2-5. Electric Heat Pump Schematic Diagram

The heat transfer from the storage to the evaporator is given by

$$Q_E = UA_E((T_S + T_{SR})/2 - T_E)$$

$$= m_S C_p (T_S - T_{SR})$$

and the heat transfer from the condenser to the load by

$$Q_C = UA_C((T_L + T_{LR})/2 - T_C)$$

$$= m_L C_p (T_L - T_{LR}) ,$$

where UA_E and UA_C are the heat transfer capacities of the evaporator and condensor and the temperature and mass flows are as indicated in Figure 2-5.

The power to the electric motor is

$$P_{HP} = Q_C - Q_E$$

and the coefficient of performance is

$$COP = Q_C/P_{HP}$$

the inefficiency of the motor-compressor unit is accounted for by the equation

$$COP = \eta_c \, T_C/(T_C - T_E) ,$$

where

$$\begin{aligned} \eta_c &= 0.6, \text{ for } (T_C - T_E) < \delta T_B \\ &= 0.6\,((T_C - T_E - \delta T_B)/(\delta T_S - \delta T_B)), \text{ for } \delta T_B < (T_C - T_E) < \delta T_S \\ &= 0.0, \text{ for } (T_C - T_E) < \delta T_S \end{aligned}$$

That is, the heat pump compressor efficiency is assumed to be 60 percent up to a temperature difference δT_B, and then to decrease linearly to zero at δT_S.

According to this simple theory, the heat pump COP approaches infinity as the evaporator temperature approaches the condenser temperature. In practice, however, other parameters such as pressures and flow rates within the compressor limit the COP. For this reason, a maximum COP was sometimes specified. Also, the maximum available condenser temperature depends on the boiling point of the heat pump fluid. If high condenser outlet temperatures are desired, multiple heat pump stages have to be introduced, thus further decreasing the heat pump COP because of additional exergy losses in heat exchangers.

The MINSUN control strategy assumes that, when a heat pump is used, the entire heating load is supplied by the heat pump. Therefore, the heat transferred to the condenser, Q_C, equals the heat load calculated elsewhere in the program.

If the entire load cannot be met with the heat pump, the heat pump is turned off and auxiliary energy is used to boost the temperature of the water supplied from storage to the load demand temperature.

2.4 COMPONENT COSTS

Cost data were derived from the first phase of the Task as were the performance models described in Section 2.3. The guidelines used were that costs should be representative of probable costs used for design in 1984-85 (and, presumably, representative of actual construction costs in 1985-86). Costs were derived by the Phase I participants from data available in the 1981-83 time period and projected to 1984-85 on the basis of cost trends and market expectations. Since the cost database is of international origin, it is also important to point out that there have been significant shifts in the exchange rates since the data were compiled.

The cost data listed in this section were used for the reference case analyses and optimization. The national evaluations described in Section 4 use current (1984-85) national cost figures. These are compared with the reference case data in Section 4.

2.4.1 Collector Cost

The cost of the solar collector subsystem dominates the cost of most CSHPSS systems--especially those with high temperature distribution systems. Therefore, it is important that reliable cost data be used for solar collectors. Details of the cost analysis methods used to estimate the installed collector subsystem costs are given in Reference 3. The procedures employed take account of the manufactured cost of the collector modules, the cost of distribution and the manufacturer's profit as well as the installation cost including site preparation, support structure, piping, insulation, pumps, controls, and wiring. A modular method of cost estimation was used to derive the installed subsystem cost from the cost of components and materials using well-established multipliers for associated materials, direct labor, indirect labor, and other indirect costs. For example, the cost of the flat plate collector subsystem is based on a collector module cost of 130 $\$/m^2$ and associated materials costing 25 $\$/m^2$. The addition of direct and indirect labor for installation and other indirect costs brings the estimated installed cost to the owner to 245 $\$/m^2$.

Most of the collector cost information developed in Reference 1 was based on U.S. experience. There is a significant variation in costs between participating countries. An informal comparison of the 1982-1983 installed cost of flat plate collectors in the nine countries participating in Task VII indicated a range of 155 $/m^2 to 335 $/m^2. The average, however, was 239 $/m^2 which is quite close to the value used in the current study (245).

Unglazed collectors were not included in the study cited in Reference 3. Therefore, the cost figures for unglazed collectors were based on European experience--principally in Sweden and the Netherlands. The cost of evacuated collectors is based on a very small market and, therefore, is more variable and less reliable than either the cost of flat plate or unglazed collectors. Evacuated collector costs are especially volume dependent and could be reduced considerably if a reasonably large market were to develop. If a large, truly competitive market were to develop in CSHPSS or other large-scale, low-temperature applications, the competitive environment would drive all collector costs toward the same cost per unit of energy delivered.

The complete cost of the collector subsystem including all piping, pumps, controls, and installation (ground mounting) are listed below. Land cost is not included.

TYPE	COST
Unglazed	140 $/m^2
Flat Plate	245 $/m^2
Evacuated	350 $/m^2

In countries where large installations have been realized, the cost of collectors appears to be dropping faster than anticipated. In 1985-86 high efficiency flat plate collectors were available for 140 $/m^2 (installed) in Sweden, and high performance, large-scale parabolic trough collectors were available for less than 200 $/m^2 in the U.S. Therefore, the collector subsystem cost estimates used in the present study may be conservative.

An additional cost for connecting the collector array to the storage was considered to be the following, for each load investigated.

LOAD, TJ	ADDITIONAL COST
3.6	$25,000
36	$78,750
360	$250,000

2.4.2 Storage Cost

2.4.2.1 Water Storage

For this type of storage, which includes the above ground tank, cavern, and pit stores, the costs were calculated from:

$$\text{cost} = V\,[C_b + (C_s - C_b)(V_S/V)^a] + (V_i C_i) \ ,$$

where V is the storage volume in m^3

V_i is the volume of insulation around the storage in m^3

V_s is the volume of reference storage for which cost is C_s

C_i is the cost of insulation taken as $100/m^3

C_b is the asymptotic storage cost per unit volume without insulation, land cost, or fixed costs

C_s is unit costs of reference storage system without insulation, land cost, or fixed costs.

The cost parameters used in the reference studies follow.

	STEEL TANK	CAVERN	PIT
C_b ($/m^3)	50	10	20
C_s ($/m^3)	90	48	30
V_s (m^3)	10000	50000	5000
a	0.4	0.7	0.4

2.4.2.2 Duct Storage

The following equation was used to describe the cost of a duct store with vertical boreholes:

$$\text{cost} = V[C_b + (C_s - C_b)(V_S/V)^a] + (V_i C_i) + (N_{bh} C_{bh} Z) \ ,$$

where N_{bh} is the number of boreholes

Z is the depth of the boreholes in m

C_{bh} is the specific cost of drilling and installing 1 m of borehole.

The cost parameters were chosen as:

C_i = 100 \$/m^3	C_{bh} = 30 \$/m
C_b = 0.1 \$/m^3	V_s = 10000 m^3
C_s = 0.2 \$/m^3	a = 0.1

The capital system cost depends primarily on the cost of the boreholes.

2.4.2.3 Aquifer Storage

The estimate for aquifer storage was based on actual costs incurred at the aquifer field trial in Scarborough, Canada. This particular aquifer has four production wells with a capacity equivalent to that of the reference case in this study (500 houses, 10,000 MWh per year). Although equipment sizing, and therefore cost, is dependent on actual flow rates, little information was available on the nature and parameters of this dependence. Since the aquifer cost is a small part of the total system cost (in the cases in this study), we decided to use a single, constant cost for all aquifer cases. This constant amount is $154,000. The aquifer parameters other than the charging and discharging flow were not optimized. The aquifer is assumed to be 20 m thick and covered by 20 m of impervious caprock.

2.4.2.4 Comparison

The cost equations for the four storage technologies are compared in Figure 2-6. The aquifer cost curve shown represents the fixed cost divided by the volume of water injected. For reference the volume of water injected in the no heat pump reference case for Madison climate (500 houses) is 330,000 m^3.

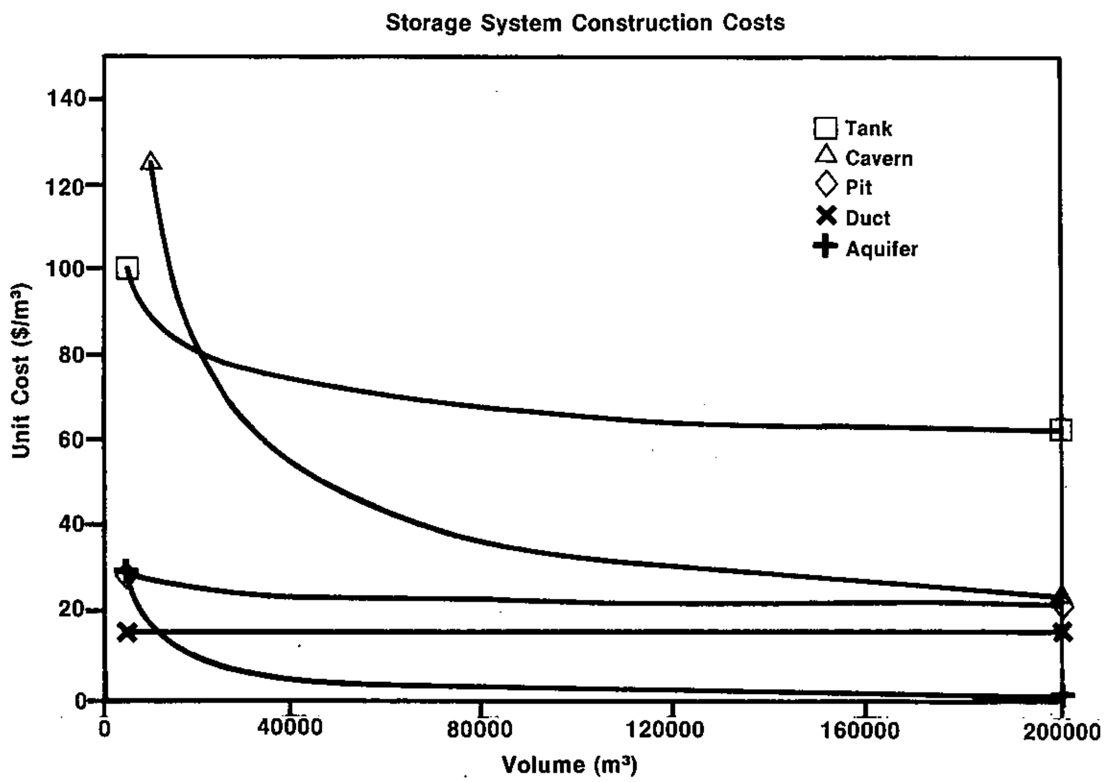


Figure 2-6. Cost Relationships for Reference System Storage Units

2.4.3. **Heat Pump Cost**

The cost of the heat pump is given by the following equation:

$$\text{cost} = [C_C UA_C + C_E UA_E + C_{HP} P_{HP}](P_{Max}/600)^{-0.3} \text{ ,}$$

where UA_C is the condenser heat transfer capacity in kW/K

UA_E is the evaporator heat transfer capacity in kW/K

P_{HP} is the maximum electrical power needed by the heat pump during the simulation in kW

P_{Max} is the maximum condenser power delivered to the load during the simulation in kW

600 is the reference condenser power in kW for which the specific costs are given

C_C is the specific cost of the condensor in \$K/kW

C_E is the specific cost of the evaporator in \$K/kW

C_{HP} is the specific cost of the motor and compressor in \$/kW.

In the reference study, the following specific costs have been chosen, in \$K/kW:

	Low Temperature Distribution Network	High Temperature Distribution Network
C_C	200	300
C_E	200	300
C_{HP}	200	300

Example:

We assume a heat pump with UA_C = 300 kW/K and UA_E = 400 kW/K. For this heat pump in a given system, the MINSUN program finds, for a low-temperature distribution network, P_{HP} = 1000 kW and P_{Max} = 3000 kW. The heat pump cost as computed by MINSUN, is, therefore:

$$\text{cost} = (200 \times 300 + 200 \times 400 + 200 \times 1000) \times (3000/600)^{-0.3} = \$209800$$

2.4.4 Cost of Auxiliary Heater

If auxiliary energy (other than electricity for the heat pump) is needed in a given system during the simulated year, the capital cost of the auxiliary heater is taken to be:

$$\text{cost} = 100 \times P_H$$

where P_H is the maximum installed power needed, as computed by the MINSUN program in kW.

2.4.5 Economic Parameters

In our study, the capital cost annualization is performed with the following basic assumptions:

Depreciation time = Economic life time = 20 years
Real Interest (Discount) Rate = 5% per annum

Operating costs other than auxiliary energy costs and maintenance costs have not been explicitly considered.

No assumptions on the cost of auxiliary energy (electricity or fuel), nor on fuel price escalation rate are necessary, since the optimization is performed in terms of marginal solar cost.

2.5 ECONOMIC ANALYSIS AND OPTIMIZATION PROCEDURES

The methodology used for economic evaluation is the common present value analysis. Costs in real, constant terms are estimated and projected over the life expectancy of a system, and the total present value of these costs is calculated using an appropriate discount rate. Since costs are expressed in real (constant) terms, the discount rate does not include an allowance for inflation. The total present value of all costs, both capital and operating, can be expressed as a normalized annual cost which is constant over the life expectancy of the plant. This constant annual cost can be divided by the annual energy produced by the system to obtain a unit energy cost from the system. This present value analysis methodology is useful and appropriate for the analytic work performed in Phase II. More detailed capital investment and financial analyses would be required before detailed design and construction are considered.

In addition to converting all costs to present value, however, the methodology must specify how costs are to be combined and compared and which costs or cost functions are to be used to optimize components and system configurations. The criteria considered for determining a suitable approach included:

- The applicability of the results to a wide range of fuel and electricity prices which already exist in many countries

- The use of solar-only costs so that sensitivities are not masked by conventional system costs

- The need to rank a large number of system configurations with different cost structures with respect to cost effectiveness

- The need to compare CSHPSS economic performance with conventional systems.

The approach adopted satisfies these criteria. Costs in real, constant terms were estimated and projected over the life expectancy of a system, and the total present value of these costs was calculated using an appropriate discount rate. The following procedures were applied for each reference case for which "optimal" designs were identified.

- A "reasonable" system configuration, based on a series of preliminary model simulations, was specified as a starting point.

- Design parameter values and component definitions were varied over a wide, but appropriate, range.

- Solar component cost and solar system useful heat output were calculated for each simulation run and the results were plotted on a graph of unit solar cost versus solar fraction.

- Those system configurations and design parameters which have the lowest solar cost for each solar fraction were identified.

- Marginal cost analysis was performed to determine the optimal system solar fractions for the range of auxiliary fuel prices of interest.

2.5.1 Solar Cost

Solar capital cost includes the collector subsystem cost, the storage subsystem, the collectors-to-storage transmission pipes, and the heat pump, if any. These capital costs are converted to a levelized annual cost by multiplying by the appropriate annualization factor dependent on life expectancy and discount rate. Operating and maintenance costs or replacement costs may be included in the present value. Auxiliary fuel and electricity costs are not included. The rationale for including heat pump capital cost is that the solar system design is enhanced by having the heat pump in the system (collectors and storage operate at lower temperatures); thus, this cost should be included in the solar component cost. The heat pump energy auxiliary cost is excluded because this energy is delivered to the load (so is not unlike auxiliary fuel cost), and the cost of this energy is dependent on assumptions about energy prices and location. The control strategy is such that load is always served directly from storage if the temperature is high enough. The heat pump is used when the storage temperature is too low for direct use by the load but high enough to allow the heat pump to function at a specified minimum COP. Auxiliary energy is used when the heat pump COP falls below this minimum. Thus the heat pump energy cost does not enter the algorithm for daily operation of the heat pump system.

The unit solar cost is the levelized solar cost of the system divided by the annual solar heat output of the system. The solar heat output is defined as that provided by the storage subsystem to the heat pump or to the load directly.

To determine the least cost solar designs for each load and set of common parameters, the unit solar cost and fractional solar output are plotted on a single graph. By plotting all relevant simulations on one graph, the least cost design for various solar fractions can be identified even though the designs may be for different subsystem types (e.g., different collector types), storage system parameters, or control strategies. Figure 2-7 illustrates the graphical representation for unit solar cost versus solar fraction.

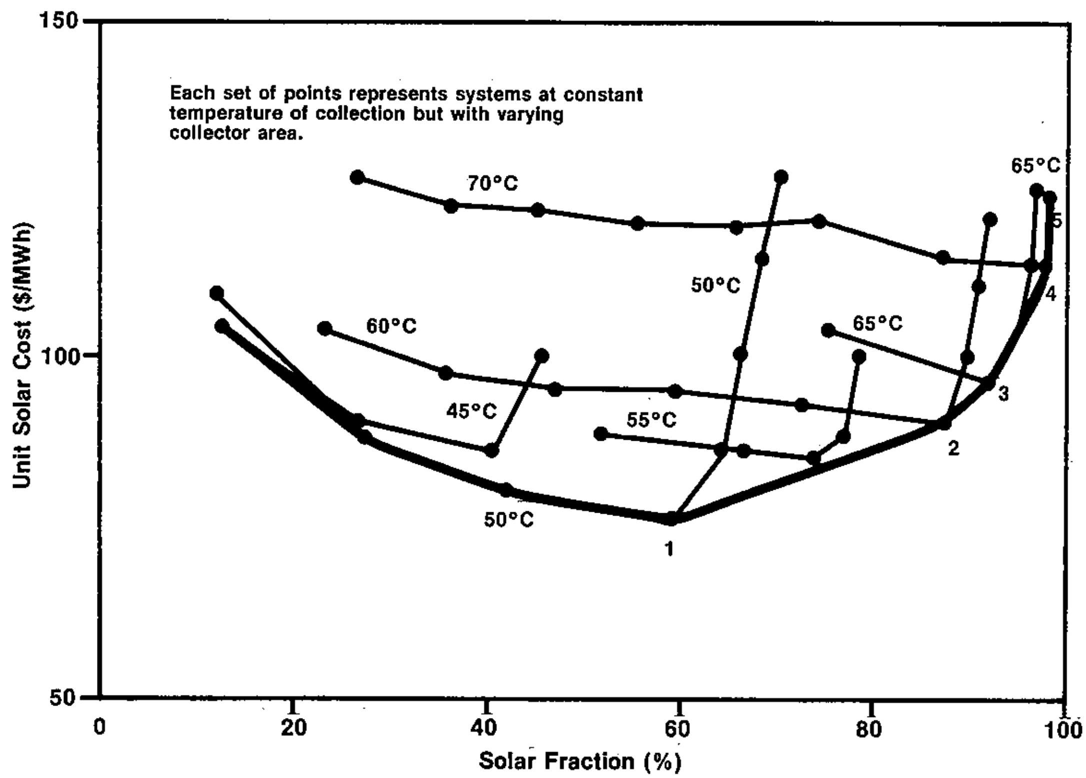


Figure 2-7. Construction of System Expansion Diagram

The envelope of minimum-cost points (numbered 1, 2, 3,...in Figure 2-7) can be used to compare one type of system with another (e.g., to compare aquifer storage systems with duct storage systems). The optimum system solar fraction, and therefore system design and size of subsystems, depends on the cost of auxiliary energy, as indicated by the equation below for system unit energy cost:

$$C_T = sC_s + (1-s)C_A ,$$

where C_T = unit system cost of energy
C_s = unit solar cost of energy
C_A = unit cost of auxiliary energy
s = solar fraction.

2.5.2 Auxiliary Energy Cost

Since auxiliary energy may be either fuel or electricity or some combination, the total unit cost of energy from a particular solar system will depend upon the respective price, efficiencies, and requirements for fuel and electricity. Figure 2-8 illustrates a hypothetical total cost surface for a solar system that requires both auxiliary fuel and electricity in the space (C_f, C_e, C_T). The variation of fuel and electricity prices for a particular country is represented by a domain, D, in the (C_f, C_e) plane. For that country the part of the solar system cost surface of interest would just be that portion above D. Another country might be characterized by a completely different domain in the C_f, C_e plane. In most countries where electricity is generated from fuel, the cost of electricity exceeds the cost of fuel. In some countries, however, there is low cost hydropower or nuclear power and the cost of electricity may actually be less than the cost of fossil fuels. To simplify this comparison and the interpretation of the present economic analysis, we have made the simplifying assumption that $C_a = C_e$ = effective cost of supplying heat from either the fuel or electric source. This is equivalent to reducing the total cost surface to a curve in the vertical plane passing through the line $C_a = C_e$. Systems of interest will now be those for which $C_T < C_a = C_e$, i.e. those that lie to the right of the $C_T = C_a = C_e$ line shown on Figure 2-8. This umption is convenient and simplifies the analysis, presentation, and discussion of the optimization procedure; however, it is not essential. The procedure can be adapted for any set of auxiliary energy costs as discussed in a later section.

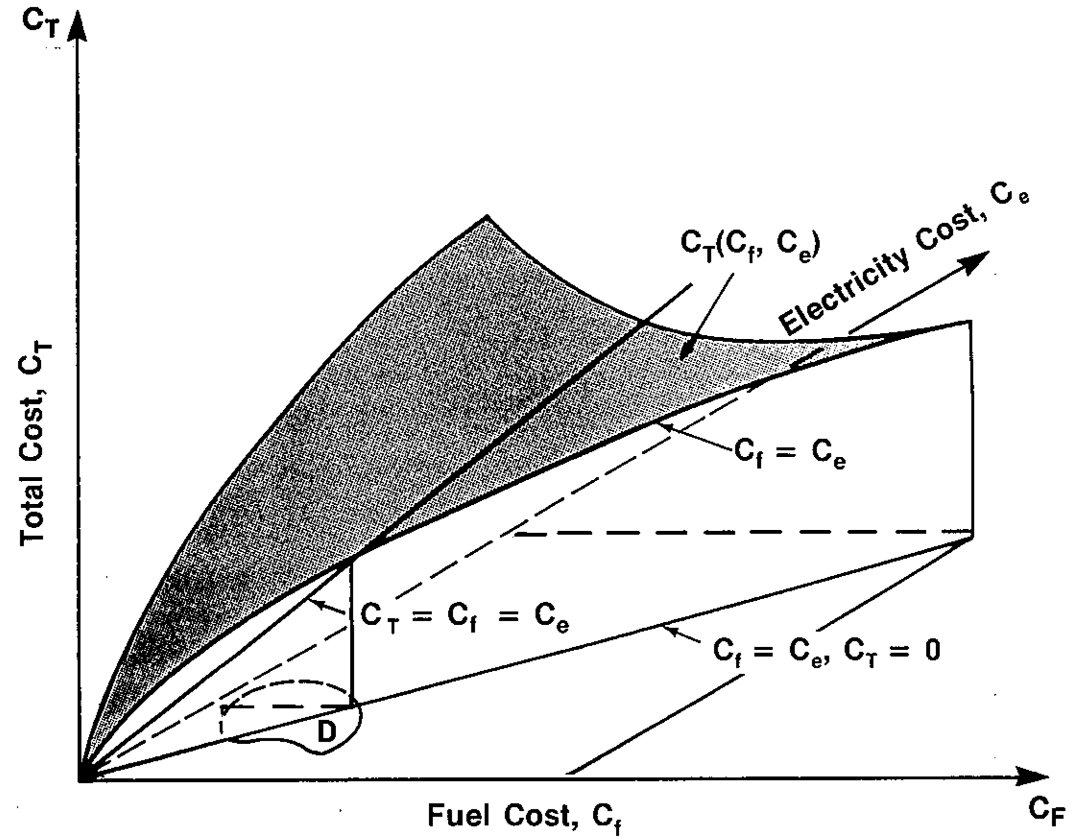


Figure 2-8. Total Cost Surface in (C_f, C_e, C_T) Space

2.5.3 System Optimization

If the minimum unit solar cost is less than the auxiliary fuel cost, the optimum system design will yield a larger solar fraction than that at the minimum unit solar cost. The solar fraction should be increased until the marginal cost of an additional unit of solar output equals the fuel cost per unit of output. This situation is illustrated in a general case in Figure 2-9.

In Figure 2-9, the marginal cost (MC) curve is the cost of an additional unit of solar output at each point. This curve is the derivative of the total solar cost curve, but it can also be calculated from the unit solar cost curve, C_S as follows:

$$\begin{aligned} MC &= d(TC)/dS \\ &= d(C_S \cdot S)/dS \\ &= C_S + S \cdot dC_S/dS \, , \end{aligned}$$

where S is the annual useful solar output.

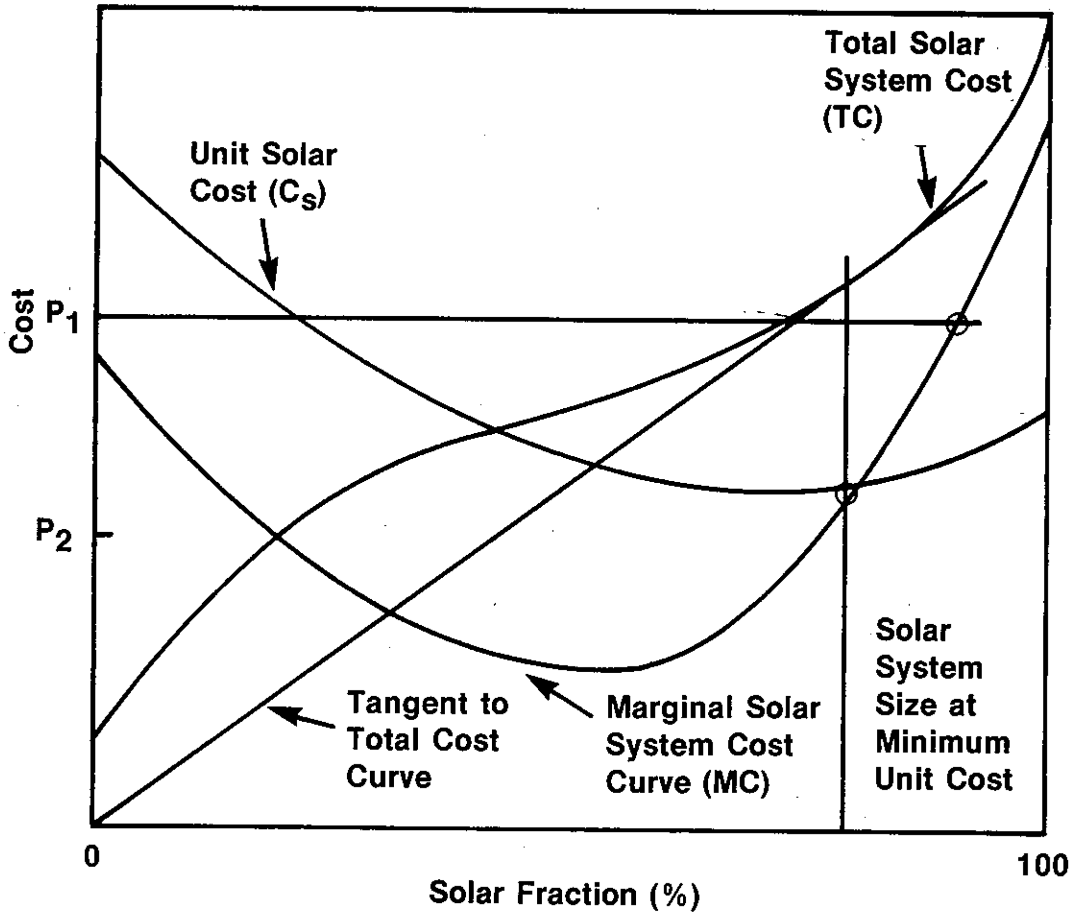


Figure 2-9. Cost Curves in the General Case

If the effective cost of auxiliary energy is P_1 as shown in the figure, the optimum system size is determined at the point where the marginal cost curve is equal to P_1. As illustrated, this point is for a larger solar fraction than where the unit solar cost curve is minimum. The system size should be increased from the minimum unit cost point so long as the marginal cost of each extra unit of solar output is less than the cost of the auxiliary that it replaces.

For an auxiliary energy cost of P_2 as illustrated in the figure, the overall minimum cost system would have no solar contribution; this is evident from the fact that at each level of solar output the unit solar cost is greater than P_2. If a solar plant were to be designed and built for reasons other than economics, the design could be "optimized" at the minimum average cost point or at the point where the marginal cost rises to P_2. The minimum average cost point would be the system producing solar output at minimum unit cost--a design that is independent of auxiliary fuel cost. The premium paid for using solar energy to displace auxiliary energy per unit displaced may also be of interest:

$$C_P = (C_s - C_a)/s .$$

If systems that are not economically viable are to be built in order to save renewable energy resources, the system with the minimum solar premium would yield the greatest energy savings per dollar invested. When electricity and fuel costs are the same, it is easy to show that the minimum solar premium occurs at the minimum unit solar cost. However, when electricity and fuel costs are different, or if other forms of energy are used, the solar premium is a useful concept. In practice, the best design would likely be determined by considering the reasons for building such a plant, the shape of the curves (sensitivity), and other factors.

2.5.4 System Unit Energy Cost

It is useful to examine how the system (i.e. solar plus auxiliary) unit energy cost varies with the auxiliary energy cost. As outlined above, the optimum system design and size can be identified given the cost of auxiliary fuel. Generally, this design will be for less than 100 percent solar. The central plant unit energy cost is the annual levelized solar system cost plus the levelized auxiliary energy cost divided by total annual load.

For all values of auxiliary energy cost below a certain point, the optimal system designs would not be solar. Central plant energy costs are less than fuel costs at auxiliary energy cost levels above a certain point. That point is determined by the minimum of the unit solar cost curve. The relationship between central plant energy cost and auxiliary cost can be determined as described below.

The minimum unit solar cost system may be identified as System 1 illustrated by point 1 of Figure 2-7. The systems on the expansion path may be identified as Systems 2, 3, 4, etc. for increasing solar fraction (and increasing cost). System 1 determines the minimum solar fraction that is economically attractive. Let the annual solar output for that system be X_1 and the solar fraction be s_1 (s_1 is X_1 divided by the annual load). If the annual solar cost is T_1, then the average (unit) solar cost, a_1, is T_1/X_1. The marginal cost, M_{12}, following the expansion path from System 1 to System 2 can be expressed as:

$$M_{12} = (T_2 - T_1)/(X_2 - X_1) ,$$

$$= (a_2X_2 - a_1X_1)/(X_2 - X_1) ,$$

where variables with subscript 2 apply to System 2. Since, by assumption, marginal cost M_{12} is greater than System 1 unit cost a_1, the optimum central plant design is System 1 for auxiliary energy costs between a_1 and M_{12}.

For auxiliary energy costs greater than M_{12}, the optimum central plant design is System 2, until auxiliary energy costs reach M_{23} defined analogously to the definition for M_{12} above. Figure 2-10 illustrates the resulting relationship between central plant energy cost and auxiliary energy cost. Note that for large values of auxiliary energy cost, the central plant energy cost is constant at the level of a 100 percent solar supply system.

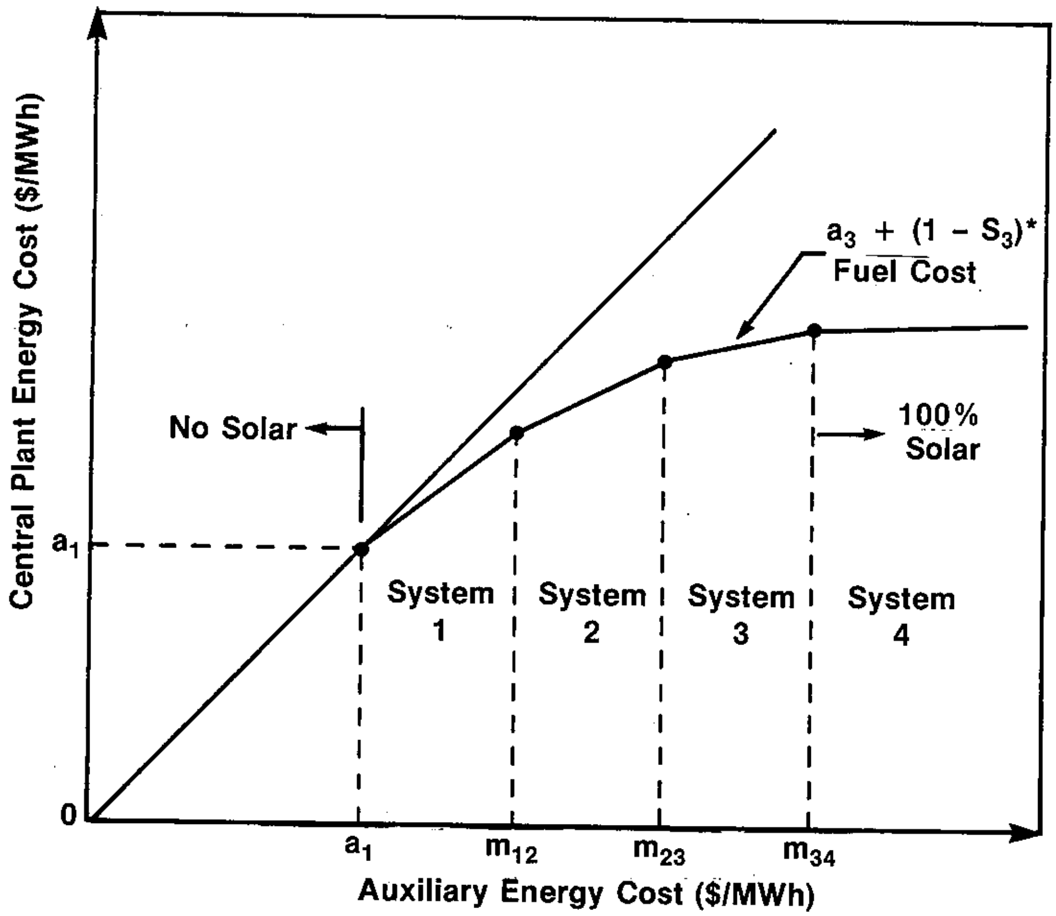


Figure 2-10. Relationship Between Central Plant Energy Cost and Auxiliary Energy Cost

The procedure just described uniquely identifies minimum total cost systems when the unit solar cost versus solar fraction curve is strictly convex. In our application, these curves do not always meet this requirement. In most of our applications, however, the curves are piecewise convex. To apply the above procedure in these cases, each convex section of the curve must be analyzed to determine the overall minimum total cost systems for each value of auxiliary energy cost.

In comparing total solar system costs with the cost of conventional auxiliary energy, annualized cost of energy actually delivered to the thermal store (if the alternative source is also seasonal or off peak) or load should be used. If the investment is to be evaluated over the assumed 20-year amortization period of the plant, the cost of auxiliary energy should be annualized over that period using the appropriate fuel escalation rates, and the cost should be adjusted for burner efficiency or other losses in the auxiliary energy system. Note the specification of the auxiliary energy cost is the final step in the optimization process. It was not necessary to specify the time horizon. The optimization procedure always yields the most effective design for any given auxiliary energy cost regardless of when or where it may occur.

2.6 GENERALIZATION OF THE OPTIMIZATION METHOD

The analytical method described in the section above applies in cases where there is only one other (non-solar) source of heat (usually auxiliary fuel or electricity for a heat pump) or where both electricity and auxiliary sources have the same effective cost. If there are two non-solar sources of heat with different effective costs, then the method described above must be modified. In this section, two methods of analyzing these multi-source configurations are presented:

- o Fixing the electric (or auxiliary) cost and determining optimal systems as a function of the other source cost

- o Varying one source cost parametrically and determining optimal systems as a function of the other source cost.

Before describing these two methods, a very simple example will illustrate the basic analytic problem when there is more than one non-solar source of heat. The following table illustrates two hypothetical systems, either of which could be the least cost system overall depending on the relative costs of electricity and auxiliary fuel. System A is assumed to include a heat pump such that the load is met 80% by solar heat and 20% by the electricity used for the heat pump. System B uses no heat pump but still meets 80% of the load by solar and uses auxiliary fuel for the remaining 20%. As is shown in the table, the least cost system overall depends on the effective costs of the two non-solar heat sources.

UNIT COST OF SOLAR ($/MWh)	UNIT COST OF ELECTRICITY ($/MWh)	UNIT COST OF FUEL ($/MWh)	UNIT COST OF SYSTEM A ($/MWh)	UNIT COST OF SYSTEM B ($/MWh)
10	10	10	10	10
10	5	10	9	10
10	10	5	10	9

2.6.1 Fixed Electric Cost Method

The analytic problem arises when there is more than one non-solar source of heat in the system configuration. Usually, electricity serves as a source for a heat pump, and fossil fuel as a source for auxiliary heat. It can be argued that the cost of fossil fuel in the long term is more uncertain than that of electricity. Under this kind of future cost regime, it may be satisfactory to assume a fixed cost or escalation rate for electricity and to analyze the optimal system designs as a function of auxiliary fuel cost. This method of analysis is analogous to that described above for sources with the same effective costs except that the solar unit cost curves are replaced with solar plus electricity unit cost curves.

The basic step in this method is to recalculate unit cost points for every configuration (whether on the expansion path or not) including the heat pump electricity cost and heat contribution to the load. This new solar plus heat pump system unit cost can be plotted versus the fraction that solar plus heat pump supplies to the load. Any remaining load requirements must then be met with auxiliary fuel. With this unit cost versus fraction plot, the procedures described in section 2 above can be applied to determine optimal system configurations as a function of auxiliary fuel cost.

Figure 2-11 illustrates the basic plot that must be prepared as the first step of applying this method. This plot shows the unit supply cost for the solar and heat pump part of many of the system configurations which were modelled by the aquifer storage team within the IEA Task VII [9]. All these systems included a heat pump with electricity as the energy source. The balance of the specified 10,000 MWh per year load which was not met by the combined solar plus heat pump systems was met by auxiliary fuel. The fraction shown as the abscissa in this plot, therefore, is that part of the load served by the solar plus heat pump. (In contrast, the solar fraction defined in the previous section did not include the heat pump input energy as part of the solar contribution.) The unit cost shown as the ordinate in this plot is the annualized cost of the solar (including storage and heat pump capital costs) plus the cost of the heat pump input electrical energy for a fixed effective cost of $100 per MWh, divided by the energy delivered to the load from the solar plus heat pump system. Since auxiliary energy must be used to meet any remaining load, optimal overall system design can be calculated as a function of effective fuel cost in the same manner as previously described. The plot in Figure 2-11 is used to select those systems with minimum unit cost at each value of solar plus heat pump fraction. An envelope of least cost systems is selected based on this plot. In practice, this plot can be enhanced with additional thermal simulations for systems with characteristics close to those that are on or near this least cost envelope.

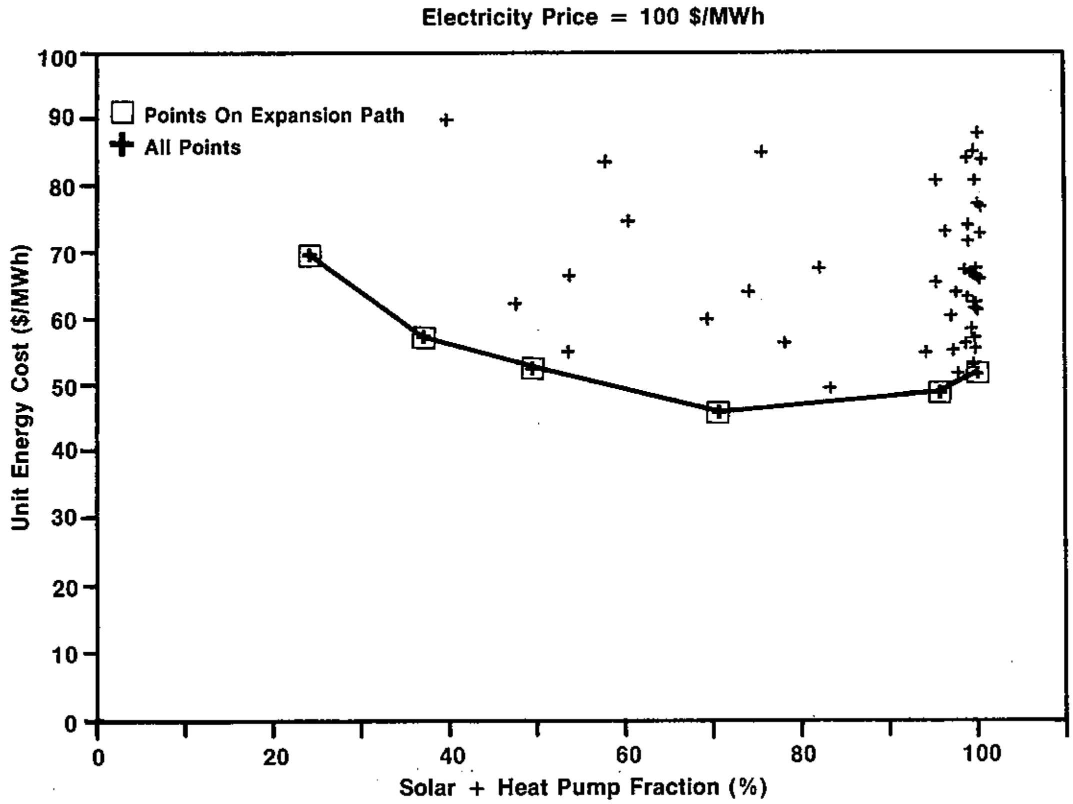


Figure 2-11. Cost Versus Fraction Plot for Solar Plus Heat Pump Energy Supply

2.6.2 Parametric Electric Cost Method

This method yields the optimal system design for any value of either electricity or auxiliary effective cost. It is a simple extension of the method described above. This method involves the repeated application of fixed electricity cost method for several values of electricity cost while solving for optimal system designs as a function of auxiliary cost for each value of electricity cost. The results can be displayed parametrically for each value of electricity cost or the total system cost can be represented as a cost surface defined over the Cartesian plane of electricity and auxiliary costs.

Figure 2-12 illustrates the least cost envelopes for the same aquifer-based systems with heat pump that were represented in Figure 2-11. In addition to the envelope resulting from those points shown in Figure 2-11, Figure 2-12 includes envelopes for electricity costs of 0, 50, 100 and 150 $/MWh. Each envelope would now have to be analyzed to determine marginal cost in order to complete the identification of optimal overall least cost systems as a function of auxiliary costs.

For the systems illustrated in Figure 2-12, the same system designs appear on all the least cost envelopes for the range of electricity costs used (zero to $150 per MWh). In general, this would not be the case. For this example, it was determined that electricity cost would have to be greater than $500 per MWh before other designs would replace those represented in Figure 2-12.

The question of whether system designs on the least cost envelope change as the electricity cost is varied, however, is sensitive to the number of systems used in the analysis and how close these systems are to each other in a plot such as that of Figure 2-11. In practice, one would probably use more system designs than are represented in these illustrative figures.

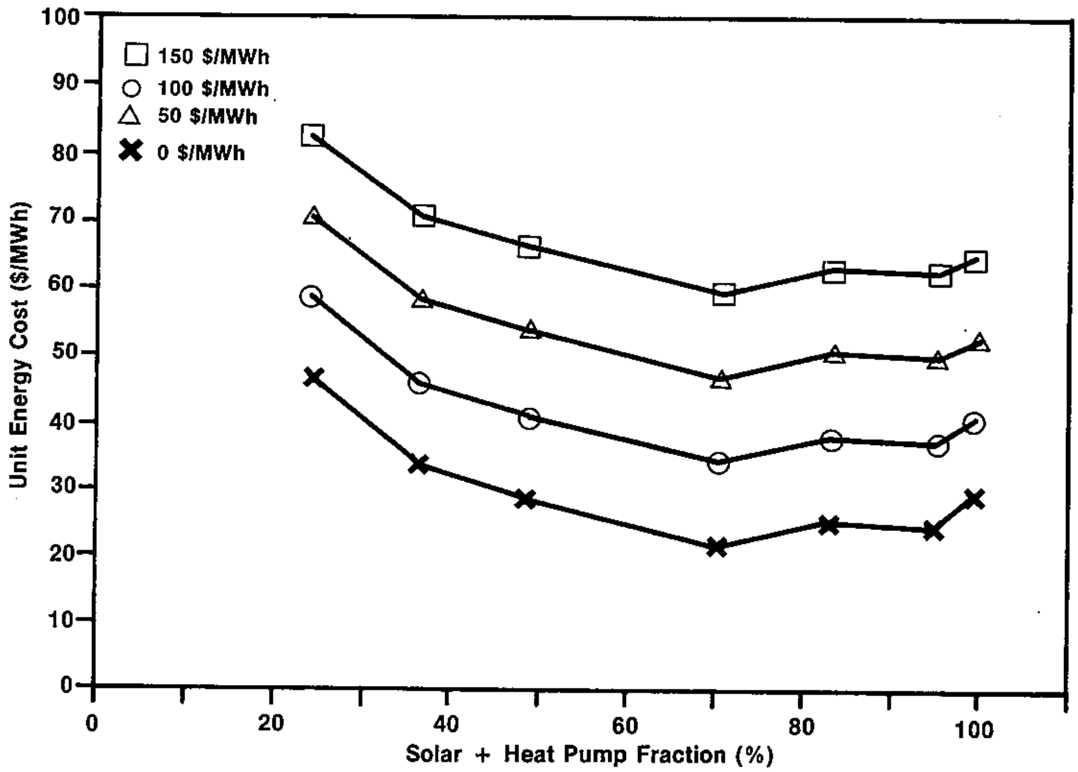
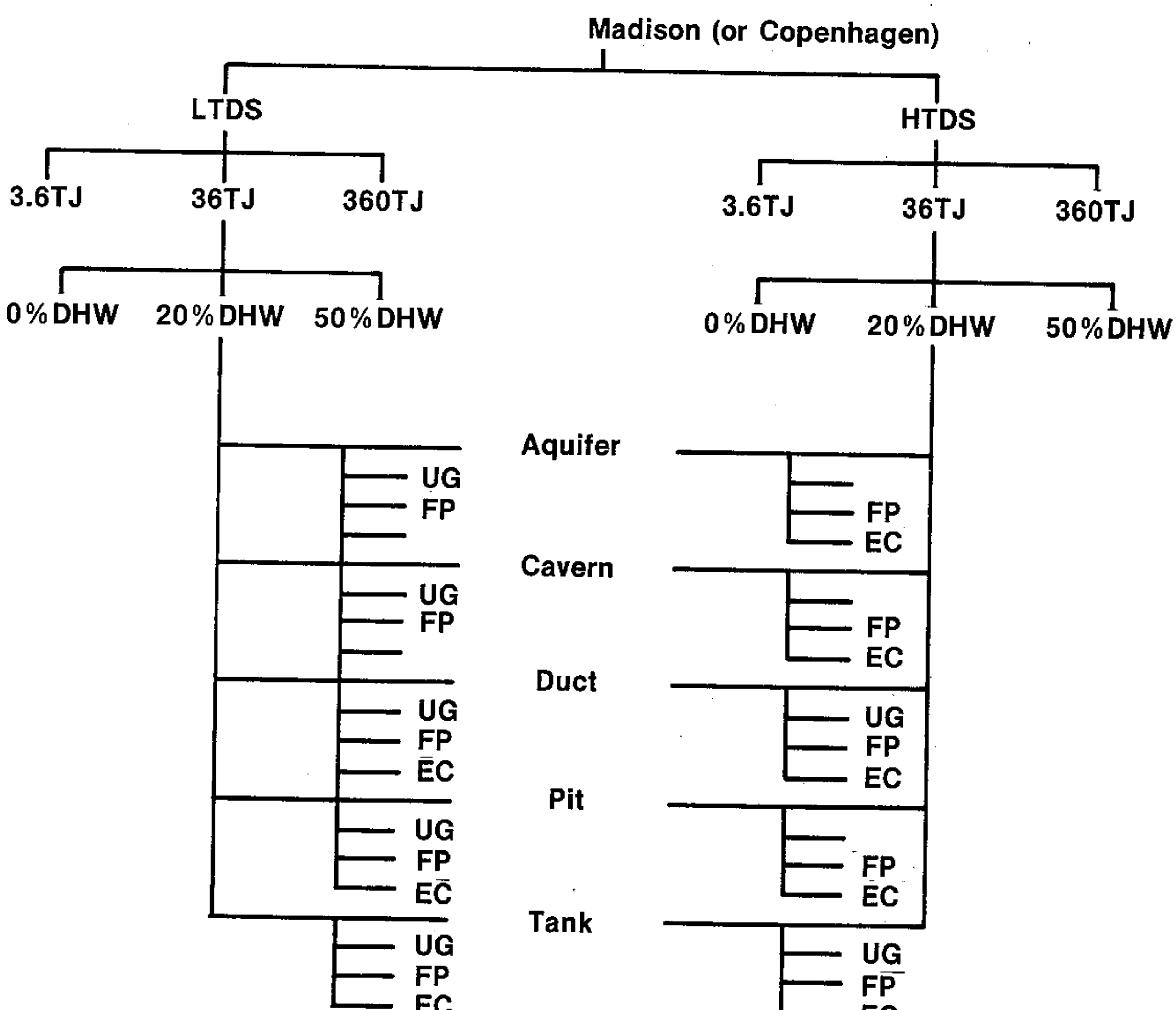


Figure 2-12. Least Cost Envelopes for Solar Plus Heat Pump Supply for Several Values of Electricity Cost

3.0 REFERENCE CASES - RESULTS

This section summarizes the main results of the reference case studies performed by the three teams of analysts as described in Section 1.5. Figure 3-1 shows the hierarchy of reference cases that were analyzed as part of the reference study. The first bifurcation of the tree is on the basis of location--Madison or Copenhagen. Each of these branches subsequently divides into low and high-temperature distribution systems. The study focused most of its attention on the cases depicted in the figure and the corresponding cases on the Copenhagen branch. An expansion diagram was developed for each case (leaf). The side branches, i.e., the 3.6 and 360 TJ loads with 0 or 50% DHW were generally analyzed as sensitivity studies. Even the sensitivity calculations, however, involved system re-optimization.



UG = Unglazed Collectors
EC = Evacuated Collectors
FP = Flat Plate Collectors

Figure 3-1. Hierarchy of Reference Cases. Each System Was Considered With and Without a Heat Pump.

This section contains four major divisions corresponding to the two locations (Madison and Copenhagen) and two distribution systems (LTDS and HTDS). Each subsection contains the system expansion diagrams for each storage technology and a total cost comparison for all the technologies with the restriction that auxiliary fuel energy cost equals electrical energy cost. In addition, sensitivity curves and tables are presented. More detail, including tabulated results may be found in the analysis team reports [9,10,11].

3.1 MADISON, LTDS

Expansion diagrams for aquifer, duct, tank, cavern and pit storage technologies are presented in Figures 3-2 through 3-6 for the low-temperature distribution systems with Madison weather and 500 houses (36 TJ annual load) with 20% DHW. The diagrams were constructed by superimposing expansion paths generated as described in Section 2.4 for several generic configurations, i.e., systems with and without heat pumps and with arrays of various collector types. The least expensive system at each solar fraction is emphasized by a heavy line, which represents the expansion path for all systems employing a particular storage technology. The collector type and the presence of a heat pump in the system are indicated in the diagrams.

A common feature of the expansion paths is the relative insensitivity of the cost to solar fraction over a fairly broad range. Between 30 and 70 percent solar, the unit cost of most of the systems varies little--especially those with heat pumps. Outside the broad range, the shape of the cost curves is more dependent on the type of storage, the distribution temperature, and the presence of a heat pump.

The characteristics of the aquifer storage expansion paths in Figure 3-2 differ from the others due to the collector control strategy of constant outlet temperature. There is a point at which additional collector area operating with the same outlet temperature will not increase the solar fraction of the aquifer over the annual cycle. The maximum solar fraction is then limited by the heat pump operating between the aquifer return temperature and the load demand temperature. Attempting to increase this limit by adding collector area and heat pump capacity only results in increased cost with little or no improvement in solar fraction. If the collector outlet temperature is increased above the demand temperature (plus a few degrees for heat losses and heat exchange inefficiencies), the aquifer storage can meet the load directly and the heat pump is off for all except peak load conditions. At even higher outlet temperatures, the heat pump system operates identically to the no heat pump case. Analysis by the aquifer team shows the rather abrupt switch from heat pump to no heat pump operation as the collector temperature is increased, resulting in the unusual shape of the expansion path. It is possible that more effective strategies for charging the aquifer could be developed to match better the aquifer return temperature to the load demand temperature.

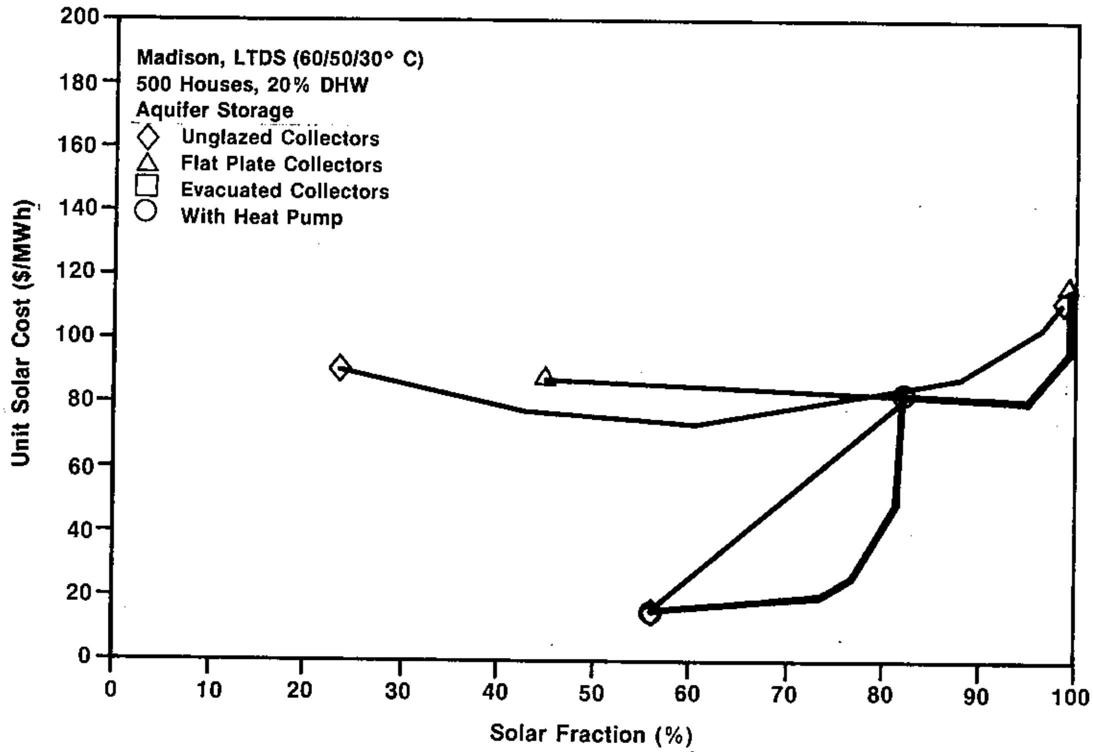


Figure 3-2. Aquifer System Expansion Diagram - Madison, LTDS, 500 Houses, 20% DHW

Similarly explained is the sharp increase in solar unit cost for no heat pump aquifer systems as a 100 percent solar fraction is approached. The last few percent solar fraction are covered by meeting the peak demands on the coldest days which require the maximum distribution temperatures. For the aquifer system to meet these peaks with the present strategy, the collectors would have to operate above the required peak temperature throughout the year. This requirement would substantially reduce collector efficiency and a larger array would be needed to inject sufficient energy and volume of water into the aquifer. To meet peak load conditions with solar, therefore, becomes more expensive.

It also takes a substantial extra amount of energy to warm up the region surrounding the aquifer active storage volume, and this must be supplied over several annual cycles. Calculations were carried out for five years to determine the significance of the warmup. These results are discussed in a Section 3.5.

The duct storage system results shown in Figure 3-3 bear a similarity to the aquifer results. Systems with heat pump and unglazed collectors deliver the lowest unit solar costs at low solar fractions but have a limited ability to meet the full load. Again, this behavior is due to the nature of the energy charge and discharge from the store, the characteristics of the unglazed collectors, and the limited COP of the heat pump. The method of extracting heat from storage draws energy fairly uniformly from the storage volume. As a consequence, the highest available temperature decreases continuously from its peak in late October. By the end of January, the store is nearly depleted and the temperature available is very low--so low that the heat pump may not be able to meet cold day demands and the auxiliary boiler will be required. When more efficient collectors are used, the temperature level of the entire storage is increased and the system can meet a greater fraction of the load. When evacuated collectors are used, the temperature of storage is maintained at a level sufficient to meet essentially the entire load without heat pumps. Achieving 100 percent solar, however, may require that the storage temperature never drop below the maximum distribution delivery temperature (60°C for LTDS) and this becomes costly.

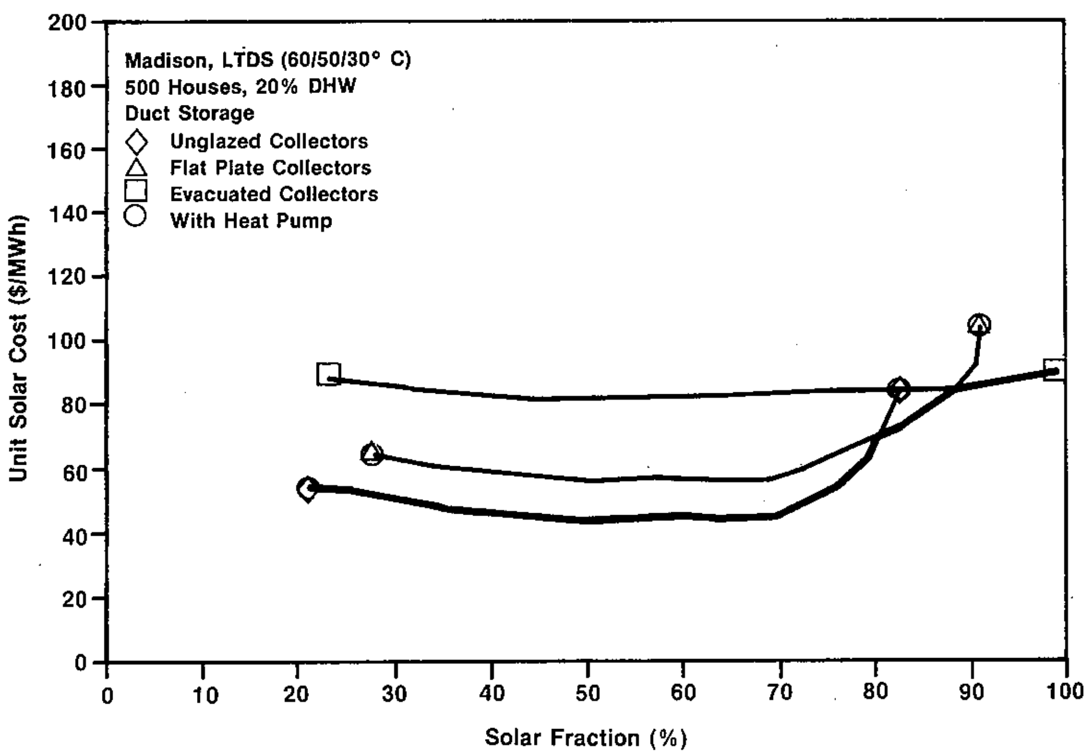


Figure 3-3. Duct System Expansion Diagram - Madison, LTDS, 500 Houses, 20% DHW

The tank, cavern and pit results shown in Figures 3-4 through 3-6 also show a cost advantage for the heat pump systems with unglazed collectors at low solar fractions. The advantage is much less pronounced than for the aquifer and duct system, however, because of the low cost of the aquifer. The most notable distinction between the water storage results and the previous systems is that the thermally stratified storage systems can achieve 100 percent solar fraction with little or no cost penalty. This is possible because the stratified system allows high temperatures to be maintained in the storage volume even when most of the stored energy is depleted. Stratification is also responsible for the smaller difference between glazed and unglazed collector cost effectiveness. In the stratified systems, the collector feedwater is always drawn from the bottom of the volume which may remain low throughout the year.

Figures 3-2 through 3-6 indicate that unglazed collectors are the most cost-effective when used with a heat pump, even in the severe Madison climate, when the solar fraction is relatively low. The unglazed collectors, however, are not suitable for high solar fractions. We also noted that the model used for the unglazed collectors in the reference study was considered by some participants to be inadequate for predicting performance of systems that frequently operate at subambient temperatures and somewhat optimistic in predicting performance in windy areas. Evacuated collectors are generally the best choice for 100 percent solar systems. None of the teams included tracking collectors in this phase of the work.

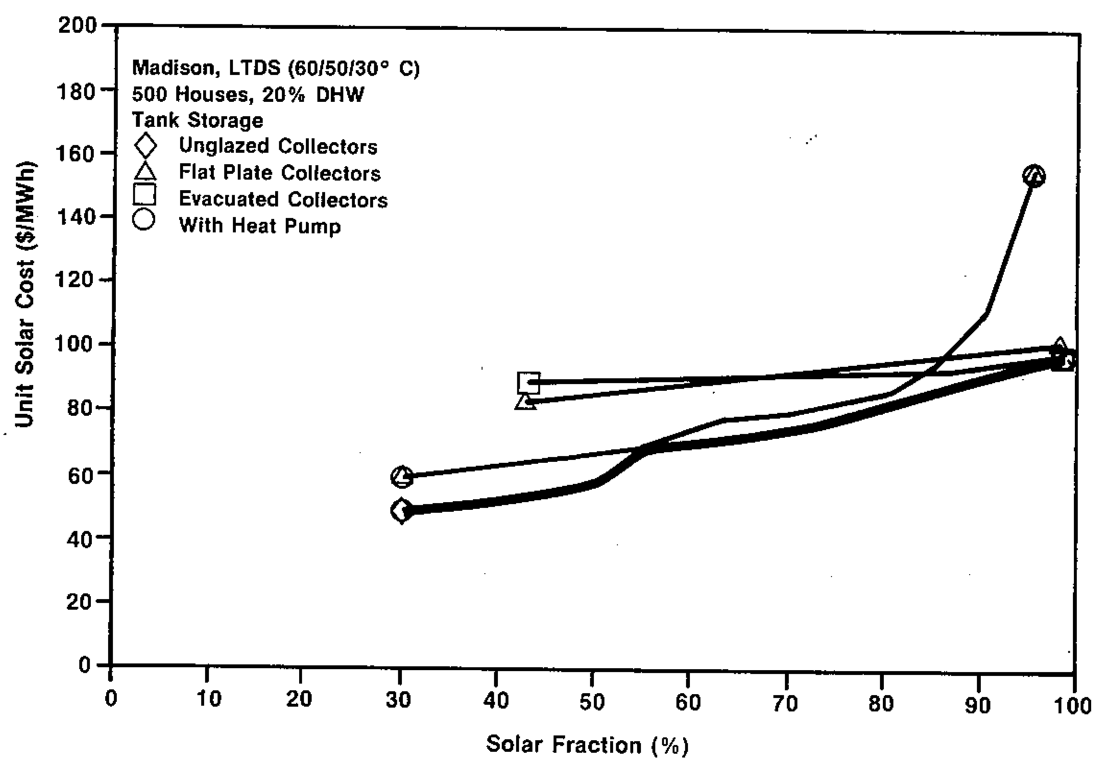


Figure 3-4. Tank System Expansion Diagram - Madison, LTDS, 500 Houses, 20% DHW

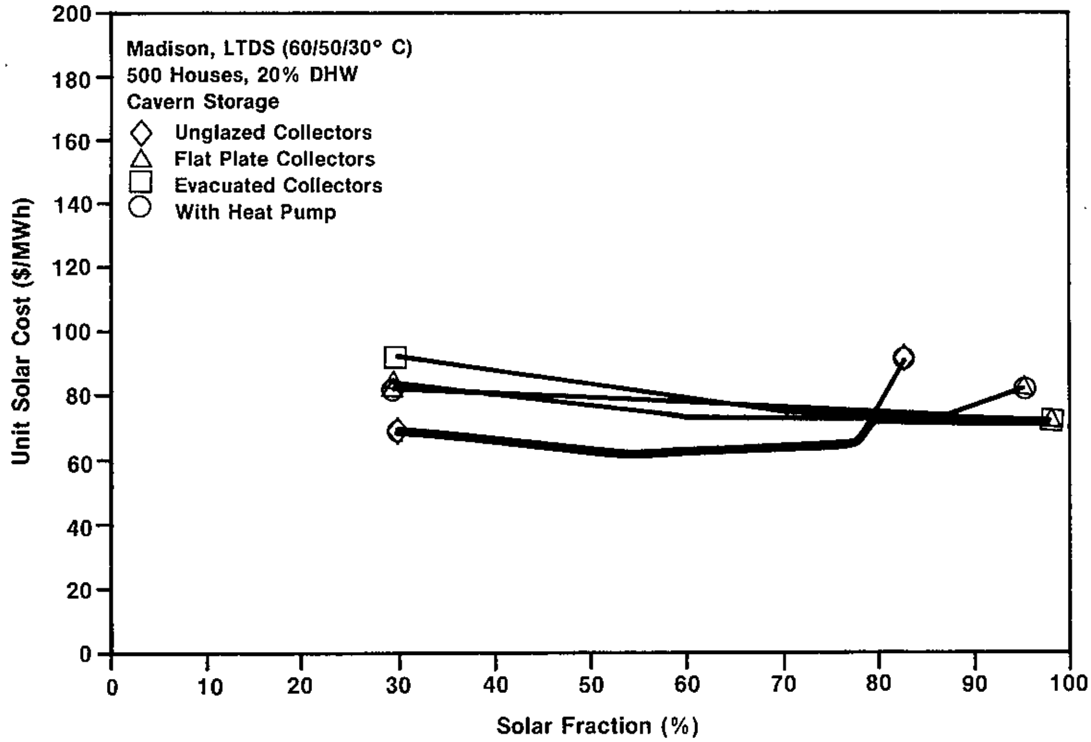


Figure 3-5. Cavern System Expansion Diagram - Madison, LTDS, 500 Houses, 20% DHW

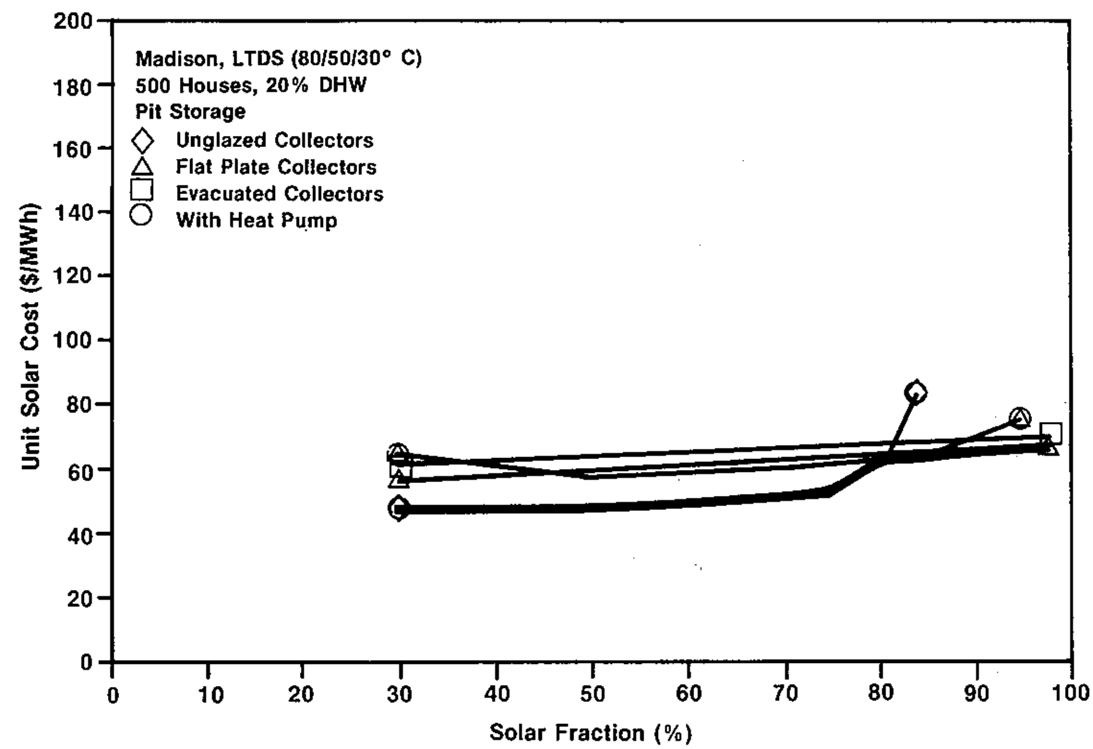


Figure 3-6. Pit System Expansion Diagram - Madison, LTDS, 500 Houses, 20% DHW

A composite expansion diagram showing the expansion paths for the best systems for each storage technology is presented in Figure 3-7. Rankings are discussed in a subsequent section; however, one may conclude that aquifer, duct or pit systems with heat pumps offer the lowest solar costs at low solar fractions, but that to achieve high solar fractions, where electricity is expensive or where the highest degree of energy independence is desired, it is best to construct a pit or cavern storage system without a heat pump.

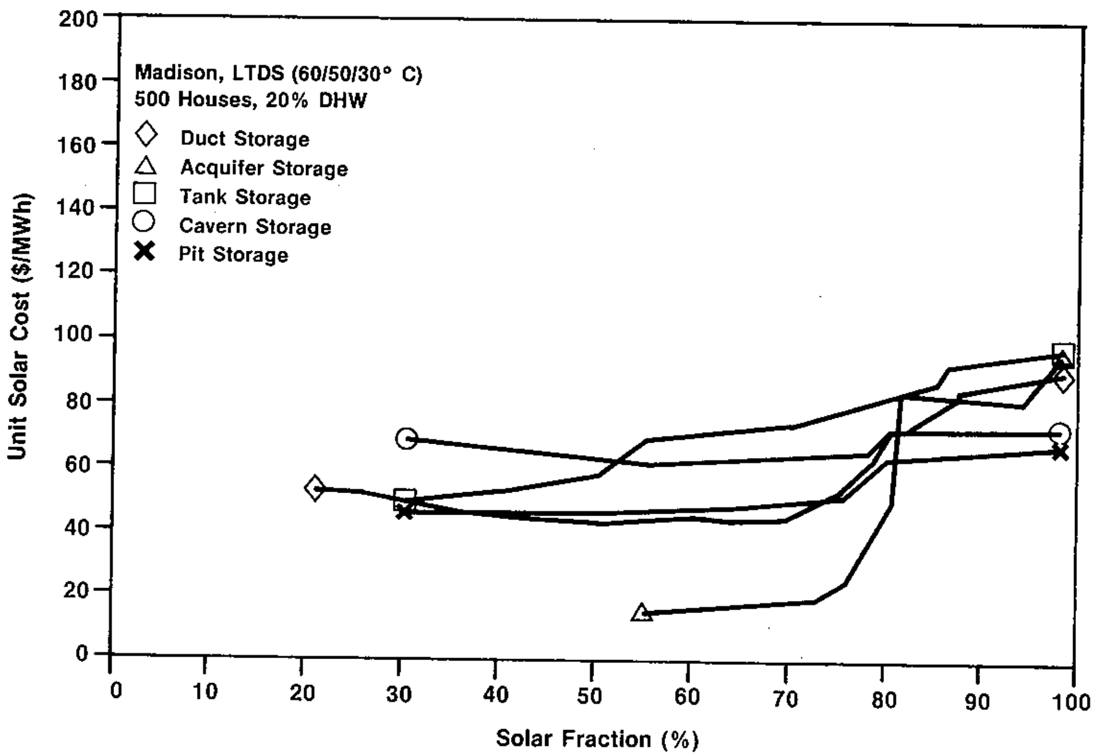


Figure 3-7. Composite Expansion Diagram - Madison, LTDS, 500 Houses, 20% DHW

The optimal system selection depends upon cost of fuel and electricity as already discussed in Section 2.4. To simplify presentation and interpretation of the reference case results, we have compared all systems in the plane of equal fuel and electrical energy costs. The solar unit costs shown in Figure 3-7 are converted to system unit costs in Figure 3-8. The relation is

$$C_T = sC_S + (1-s)C_A$$

where s and C_S are the solar fractions and solar cost at which the marginal solar cost is equal to the equivalent auxiliary energy cost C_A. If C_A is greater than the minimum solar cost, s and C_S are evaluated at the minimum point; however, under those conditions, the combined cost C_T of the system would be greater than C_A.

The cost of operating a plant with no solar energy is shown as a reference. This is the straight line of unit slope and zero intercept. Each of the solar system curves is terminated at the point where it intersects the non-solar system line, since the solar system would not be economically attractive if the auxiliary energy cost less. Horizontal sections are those for which the solar fraction is 100 percent.

As shown in Figure 3-8, if the equivalent cost of auxiliary energy is above 16 \$/MWh, a solar system employing aquifer thermal energy storage would be economically attractive. Duct storage systems become competitive with conventional sources at about 45 \$/MWh, and above 80 \$/MWh all CSHPSS systems can meet the load at costs lower than conventional energy. Data from the U.S. Department of Energy's Active Program Research Requirements Project [17] adjusted to 1985 indicate that current cost of thermal energy from fuel (natural gas) burned with an efficiency of 70% and electricity in Madison should be about 35 and 64 \$/MWh respectively. Annualization of these costs at a real fuel escalation rate of 2 percent and a real discount rate of 5 percent would increase the current values by about 18 percent. It is probably reasonable to expect the cost of natural gas to increase more than 2 percent relative to general inflation.

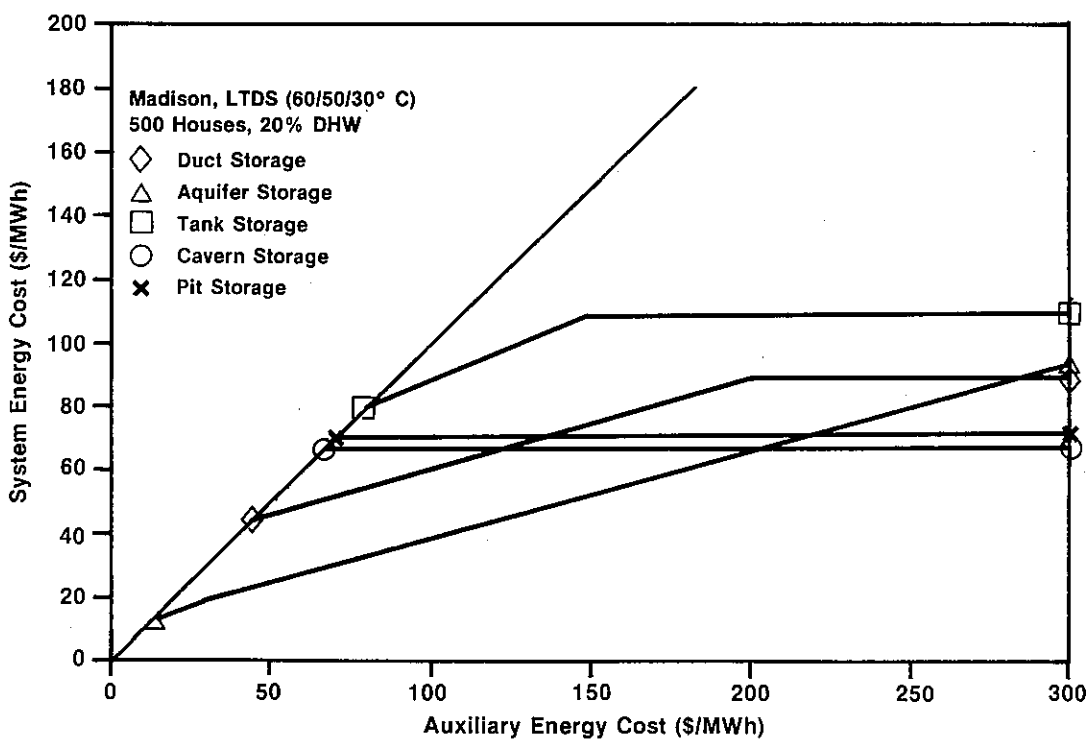


Figure 3-8. System Unit Energy Costs - Madison, LTDS

Costs shown in Figure 3-8 do not include distribution costs and are therefore a comparison of the economic viability of energy sources for district heating systems or for sites requiring a minimal amount of distribution piping. Comparison with diurnal storage solar systems will be discussed in a subsequent section.

3.2 MADISON, HTDS

Expansion diagrams for aquifer, duct, tank, cavern, and pit storage technologies are presented in Figures 3-9 through 3-13 for the high-temperature distribution system using the Madison weather and a total load equivalent to 500 houses with a 20 percent DHW fraction. The notation is identical to that of Section 3.1.

The general features of Figures 3-9 through 3-13 are similar to those of Figures 3-1 through 3-5, and the same comments apply. We note in addition that for high-temperature distribution systems, all unit costs are somewhat higher; the cross-over from heat pump systems to systems without heat pumps occurs at lower solar fractions; unglazed collectors are generally not viable; and evacuated collectors are most attractive for high solar fractions.

The composite expansion diagram shown in Figure 3-14 indicates that pit and cavern systems without heat pumps are attractive over the system size range. Aquifer systems are the most economical only at solar fractions below 40 percent.

The system unit energy cost diagram for high-temperature distribution systems shown in Figure 3-15 indicates that these CSHPSS systems become competitive with conventional energy at substantially higher auxiliary equivalent energy costs than the low-temperature distribution systems. Aquifer and pit storage systems become competitive at auxiliary energy costs below 80 \$/MWh and systems with high solar fractions are viable at auxiliary costs near 80 \$/MWh. Although the low-temperature systems become competitive at much lower energy costs (i.e., 16 \$/MWh), there is little cost penalty for the high-temperature distribution for systems that deliver 100 percent solar fraction (80\$/MWh vs 70 \$/MWh).

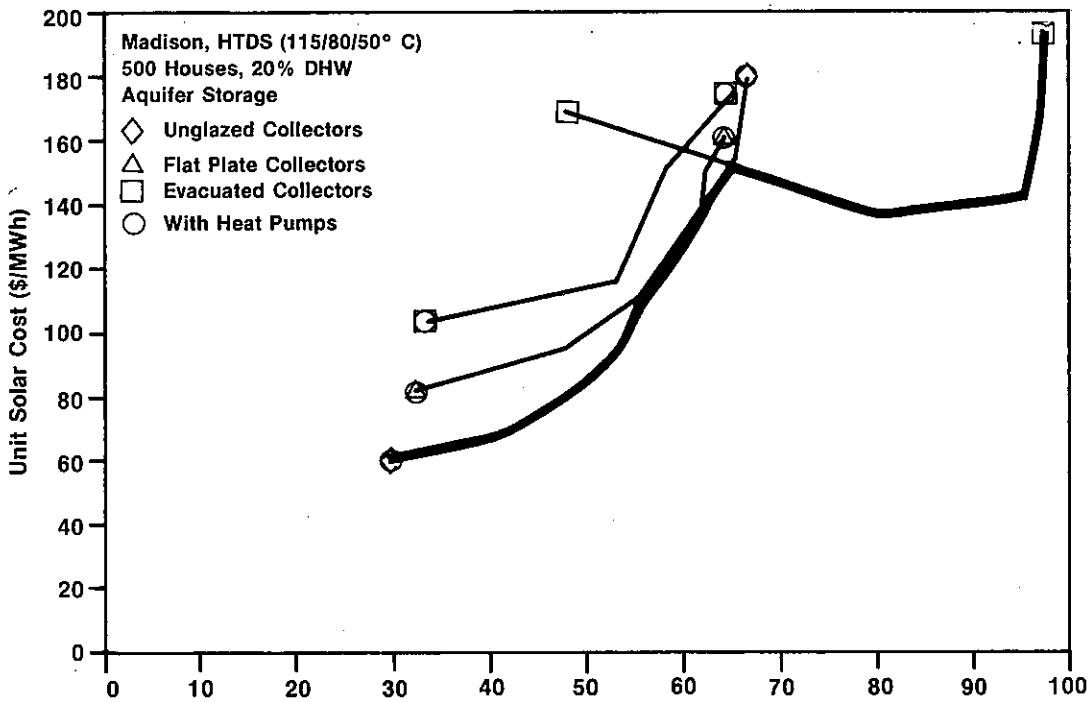


Figure 3-9. Aquifer System Expansion Diagrams - Madison, HTDS, 500 Houses, 20% DHW

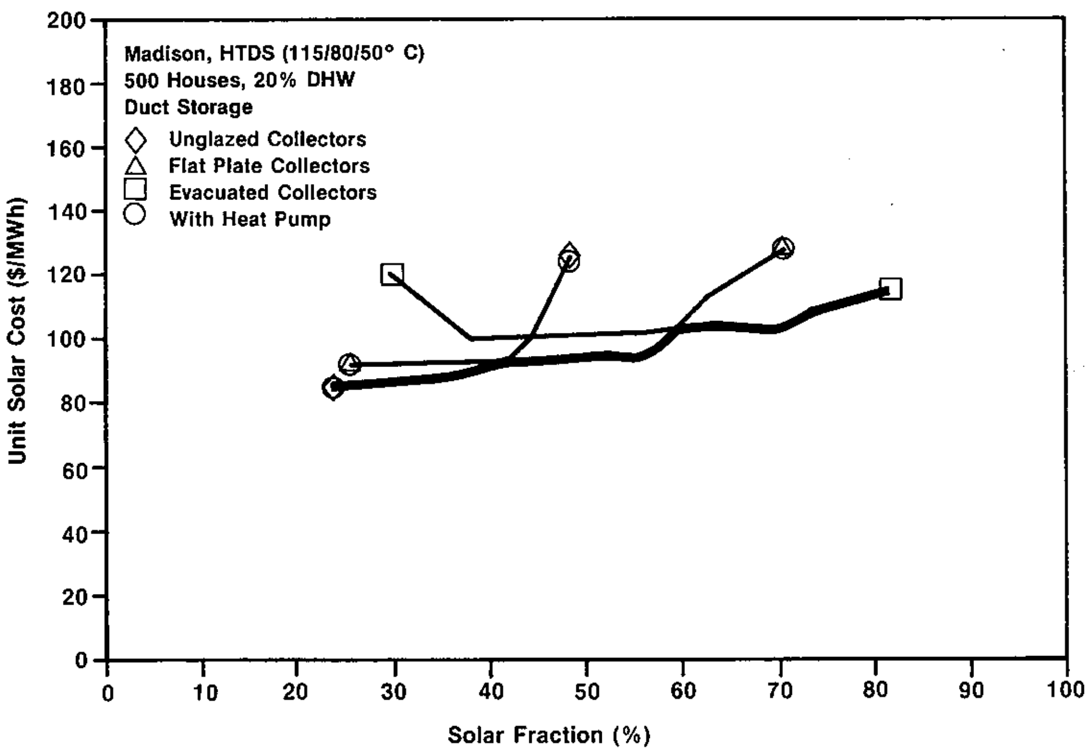


Figure 3-10. Duct System Expansion Diagrams - Madison, HTDS, 500 Houses, 20% DHW

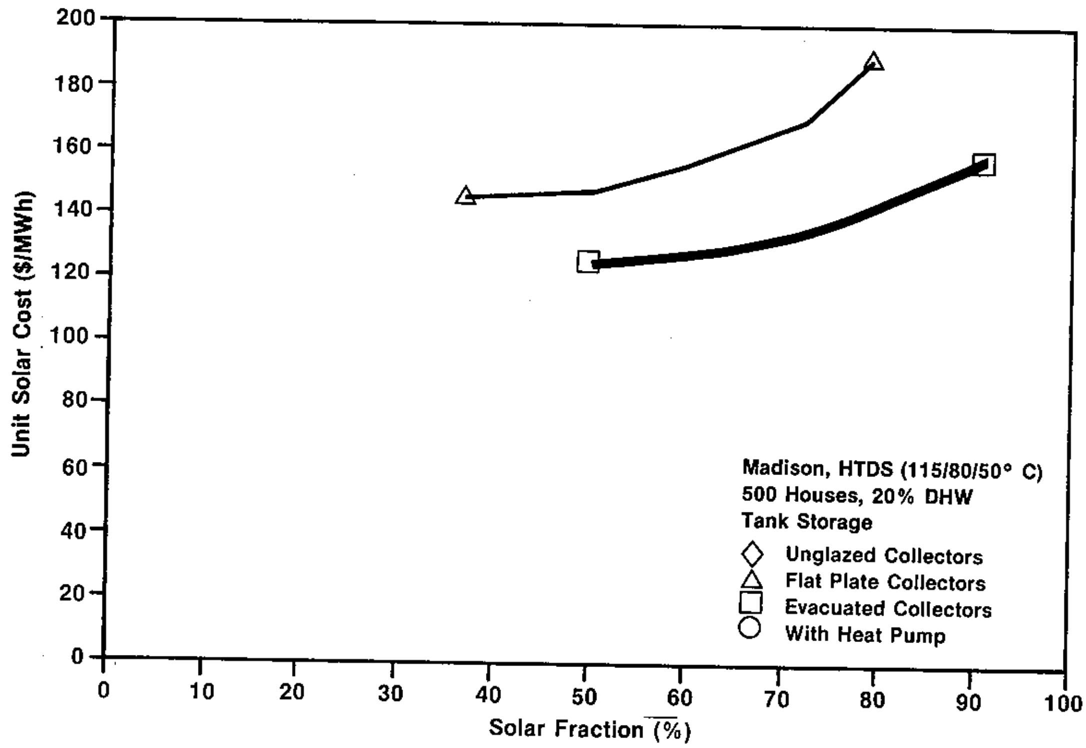


Figure 3-11. Tank System Expansion Diagrams - Madison, HTDS, 500 Houses, 20% DHW

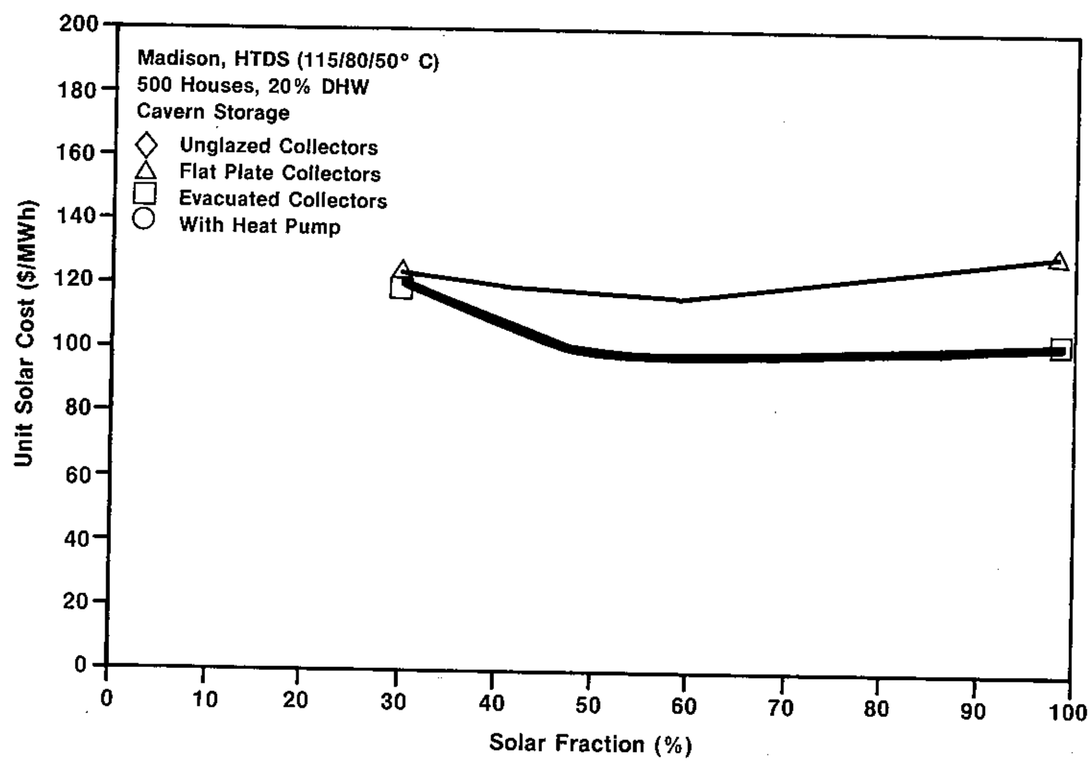


Figure 3-12. Cavern System Expansion Diagram - Madison, HTDS, 500 Houses, 20% DHW

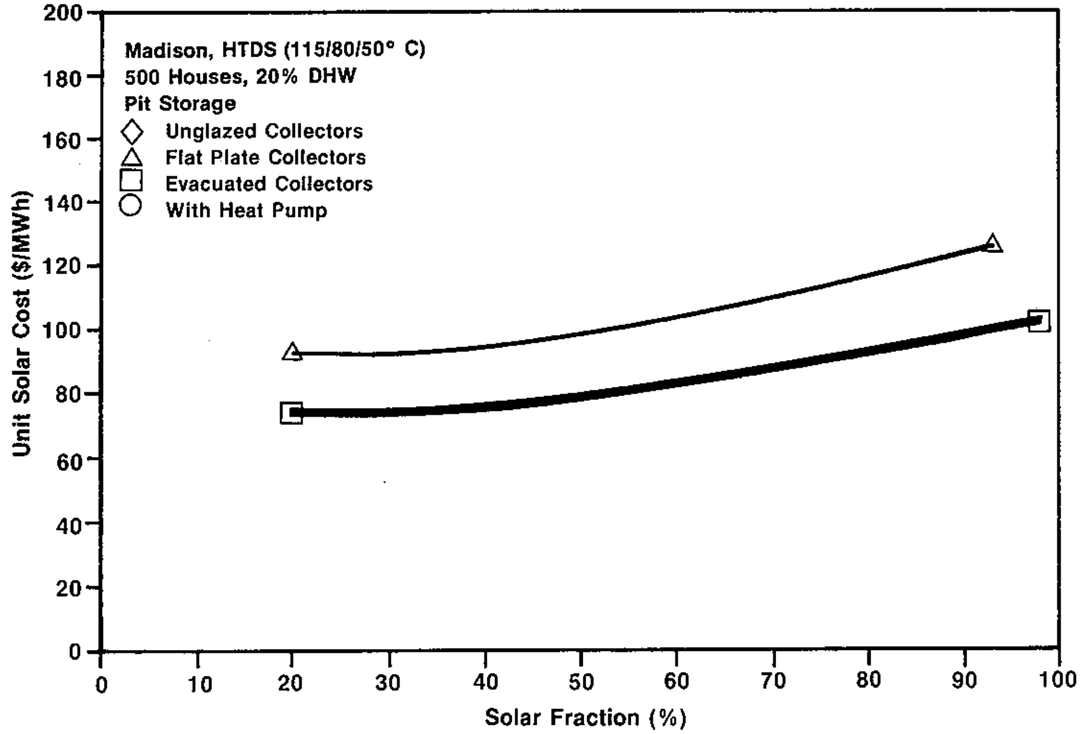


Figure 3-13. Pit System Expansion Diagram - Madison, HTDS, 500 Houses, 20% DHW

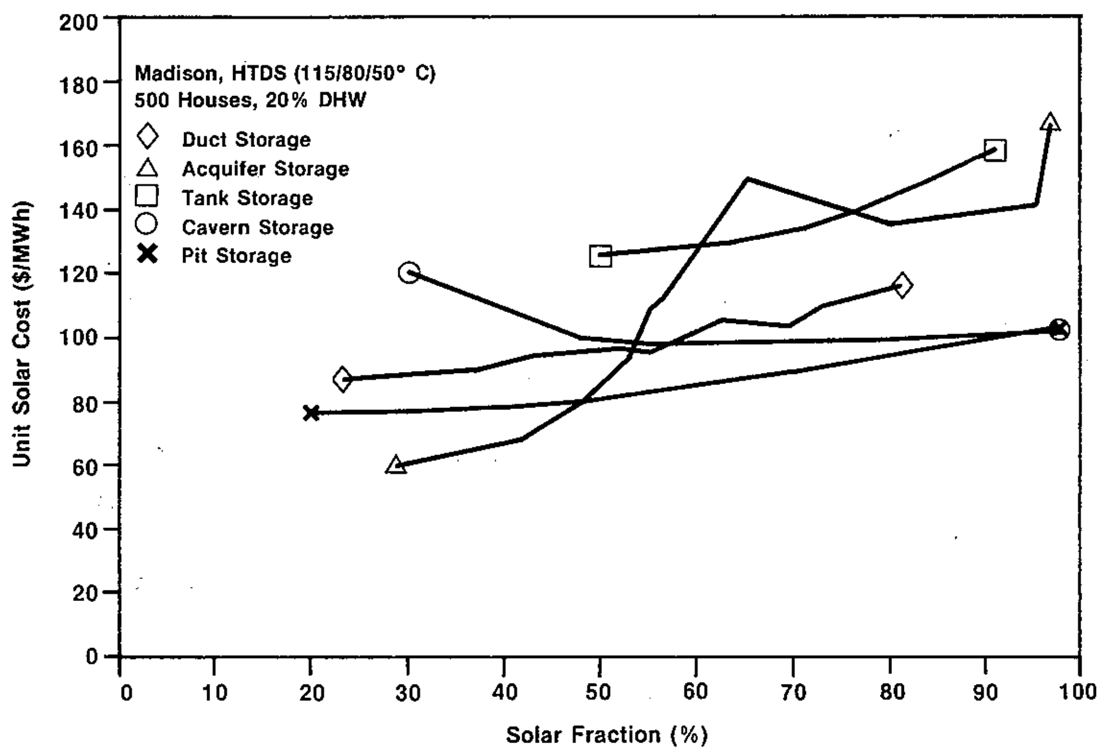


Figure 3-14. Composite System Expansion Diagram - Madison, HTDS, 500 Houses, 20% DHW

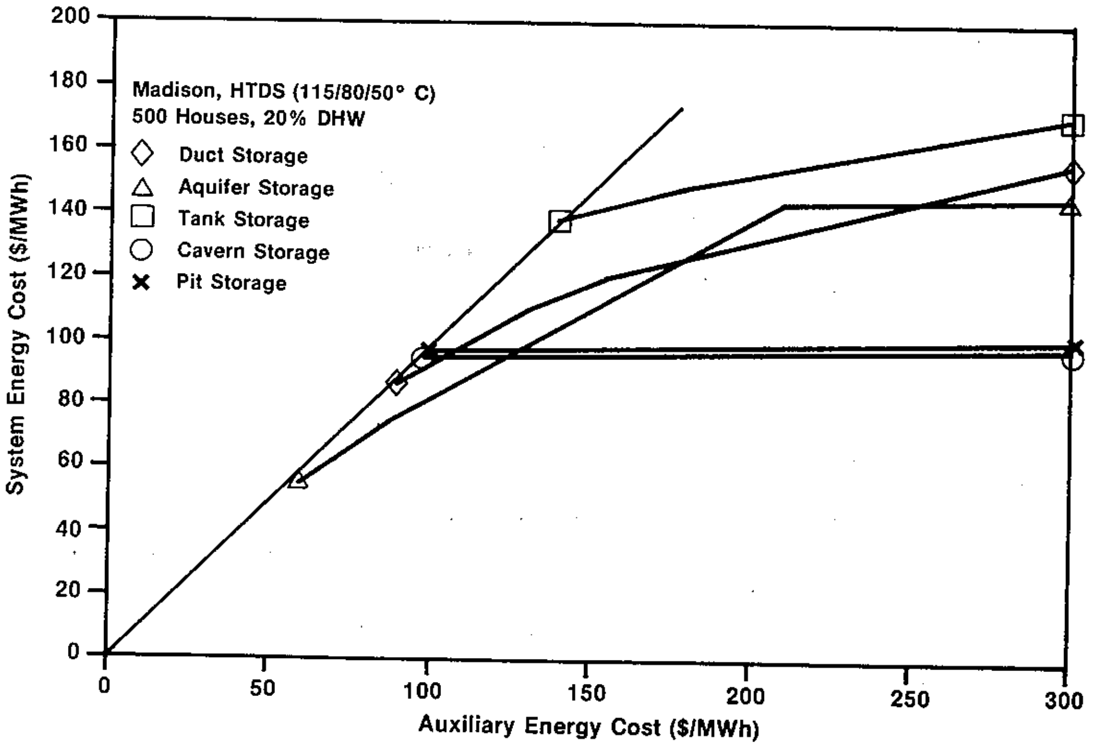


Figure 3-15. System Unit Energy Costs - Madison, HTDS

3.3 COPENHAGEN, LTDS

The composite expansion diagram in Figure 3-16 presents the expansion diagrams for aquifer, duct, tank, and cavern, and pit storage technologies for the Copenhagen reference case with the low-temperature distribution system. These curves exhibit the same features as the results for comparable systems in Madison, but are slightly more expensive for the same solar fraction. The relative ranking of the systems using the five different storage technologies are the same as in Madison.

The system unit energy graph for this reference case, shown as Figure 3-17, indicates that the cost of heat pump systems is relatively insensitive to location whereas the unit cost of energy from the non heat pump systems is higher in Copenhagen than in Madison--presumably because the total insolation is substantially lower.

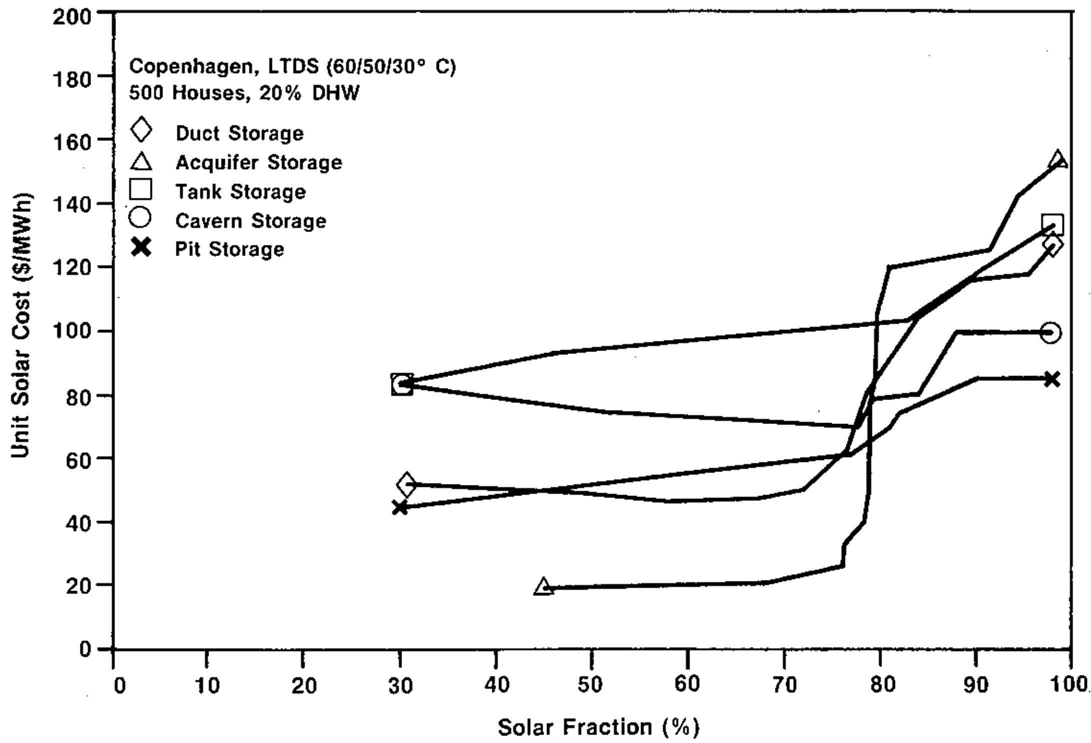


Figure 3-16. Composite System Expansion Diagram for Copenhagen, LTDS, 500 Houses, 20% DHW

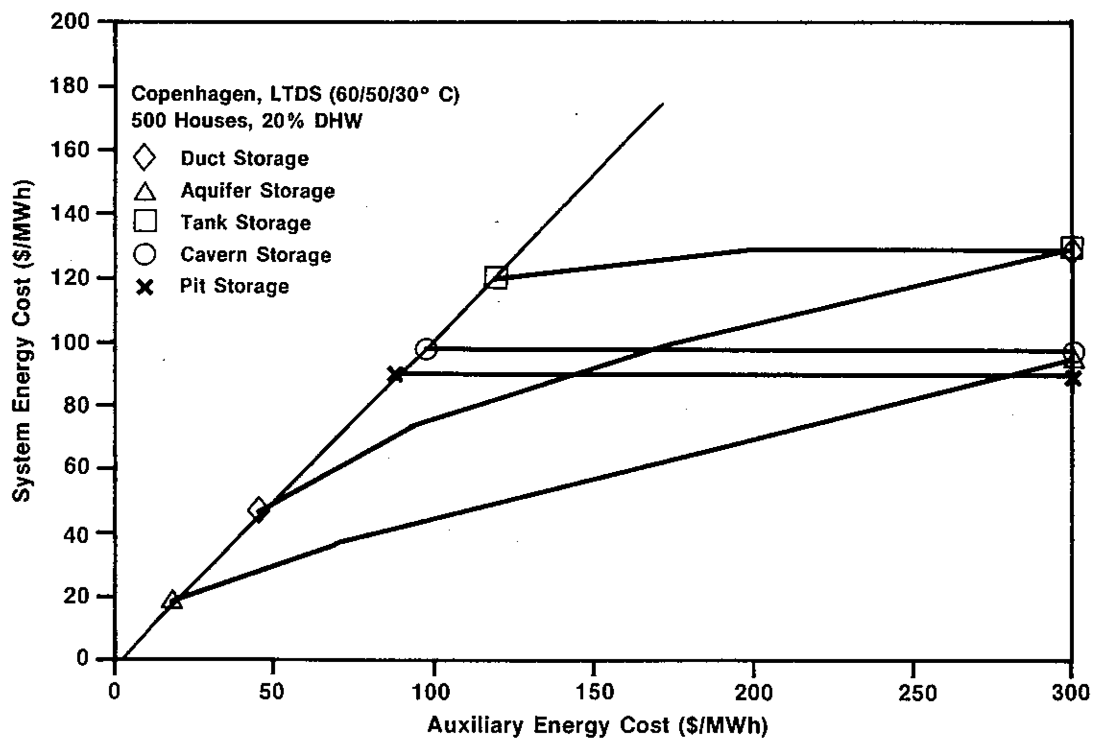


Figure 3-17. System Unit Energy Costs - Copenhagen, LTDS

3.4 COPENHAGEN, HTDS

The composite system expansion diagram for the aquifer, duct, tank, and pit and cavern storage technologies for the high-temperature distribution system reference case in Copenhagen is shown in Figure 3-18. Again, we note that the unit costs of the heat pump systems are not sensitive to location, (at least at low solar fractions), but the unit cost of non-heat pump systems is substantially higher than for comparable systems in Madison. The system rankings in the system unit cost diagram shown in Figure 3-19 are the same as in Madison, but the costs are higher because of the reduced radiation available.

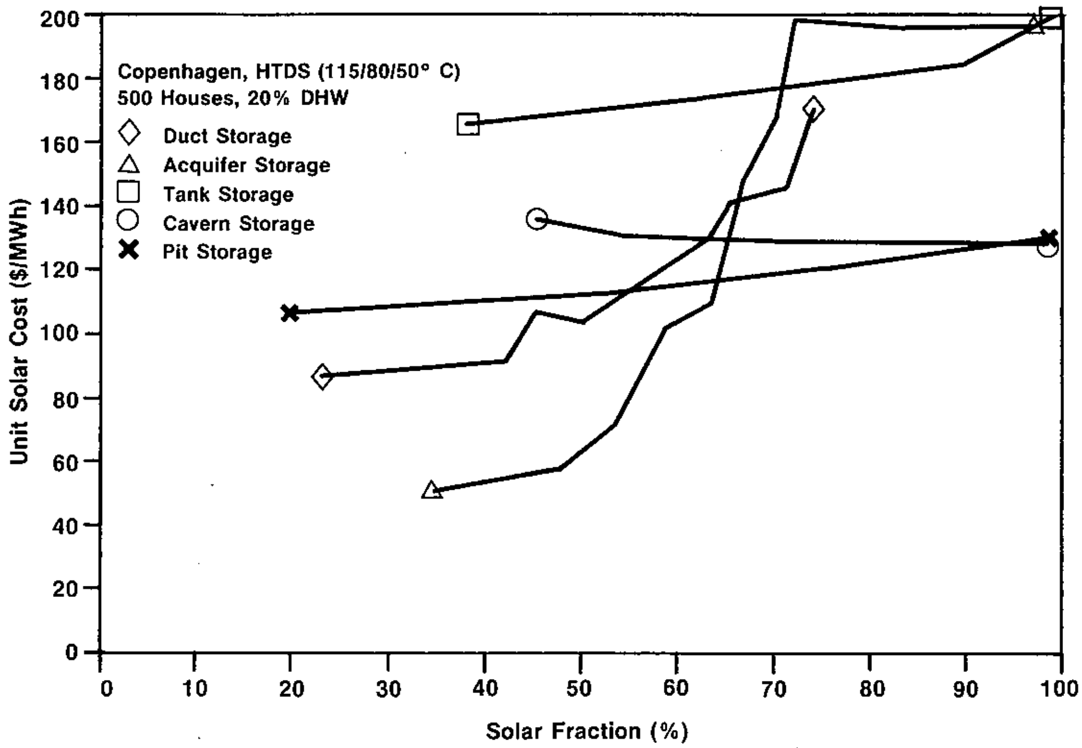


Figure 3-18. Composite System Expansion Diagram for Copenhagen, HTDS, 500 Houses, 20% DHW

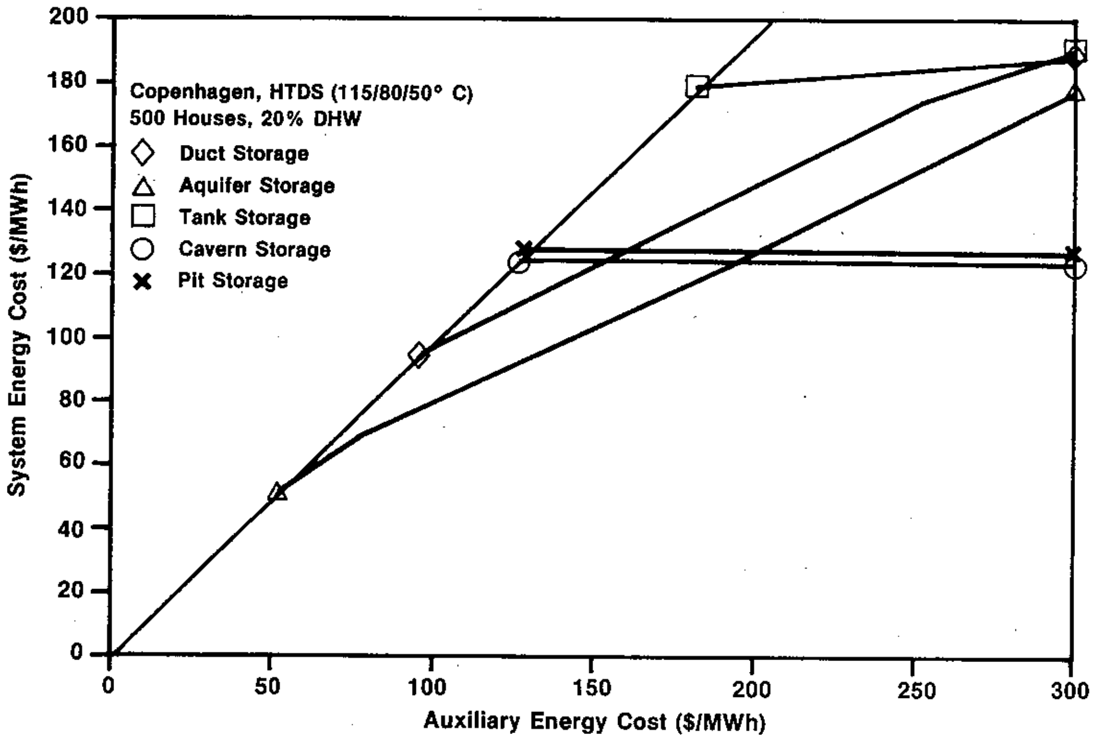


Figure 3-19. System Unit Energy Costs - Copenhagen, HTDS

3.5 SYSTEM RANKINGS

The ranking of the various systems in the four basic reference cases are summarized in Table 3-1. As we have already noted, heat pump systems are generally least expensive for low solar fractions and low-temperature distribution. The cost comparison curves for low-temperature distribution systems with $C_f = C_e$ indicate that the heat pump systems are generally more advantageous until the cost of auxiliary energy exceeds 150 $/MWh.

For the high-temperature distribution systems, the heat pump systems have the advantage at an energy cost of about 100 $/MWh and therefore the non-heat pump systems will generally be preferred for applications where conventional distribution temperatures are required. This conclusion is qualified, however, since there are many locations where the cost of electrical energy exceeds the cost of fossil energy by a factor of two or more.

We also note that although none of the systems is power independent, a heat pump system requires a great amount of power in the winter when storage energy is low and the load is high. If there is no control over the rate or timing of heat pump electricity purchase, the system may be subject to severe penalties in the form of peak demand charges or peak period rates. The rankings also show a prevalence of unglazed collectors for the high-temperature systems and evacuated collectors for the high-temperature systems.

Table 3-1. RANKING OF REFERENCE SYSTEMS

Low Cost Auxiliary Energy: C_a <150 \$/MWh

Rank	Madison		Copenhagen	
	LTDS	HTDS	LTDS	HTDS
1	Aquifer,HP,UG	Cavern , ,EC	Aquifer,HP,UG	Aquifer,HP,FP
2	Duct ,HP,UG	Pit , ,EC	Duct ,HP,UG	Duct ,HP,FP
3	Pit , ,FP	Aquifer,HP,FP	Pit , ,EC	Cavern , ,EC
4	Cavern , ,FP	Duct ,HP,FP	Cavern , ,EC	Pit , ,EC
5	Tank, ,HP,FP	Fuel	Tank , ,EC	Fuel

High Cost Auxiliary Energy: C_a <150 \$/MWh

Rank	Madison		Copenhagen	
	LTDS	HTDS	LTDS	HTDS
1	Pit , ,EC	Cavern , ,EC	Aquifer,HP,FP	Cavern , ,EC
2	Cavern , ,EC	Pit , ,EC	Duct ,HP,UG	Pit , ,EC
3	Aquifer,HP,UG	Aquifer, ,EC	Pit , ,EC	Aquifer, ,EC
4	Duct ,HP,FP	Duct , ,EC	Cavern , ,EC	Duct , ,EC
5	Tank , ,EC	Tank , ,EC	Tank , ,EC	Tank , ,EC

UG = Unglazed Collectors
EC = Evacuated Collectors
FP = Flat Plate Collectors

3.6 LOAD AND PARAMETER SENSITIVITY STUDIES

The majority of the reference study results were obtained for those reference cases that constitute the four main branches of the hierarchical tree depicted in Figure 3-1, i.e., Madison, LTDS, 36 TJ, 20%DHW; Madison, HTDS, 36 TJ, 20% DHW; Copenhagen, LTDS, 36 TJ, 20% DHW; Copenhagen, HTDS, 36 TJ, 20% DHW. However, after locating the optimal configurations and subsystem sizes for all the cases along these main branches, a few additional calculations were performed to determine the influence of major changes in the size and distribution of the heating load. Generally the additional reference case results were obtained by using the optimal configuration found along one of the main branches as a starting point (with appropriate scaling where total load changes were involved) and re-optimizing by performing a few additional variations of major parameters, e.g., collector area and storage volume.

Table 3-2 shows the results of this type of sensitivity analysis for the cavern storage system in a high-temperature distribution system in Madison. The row in each set of variations marked R is the "reference" system result at the main branch conditions. In this set of calculations the collector area, collector unit cost, storage volume, and relative storage cost also were varied to obtain additional sensitivities. The load variation results are shown in Figures 3-20 and 3-21. The unit solar cost decreases both with total load and with the DHW fraction. Note that the solar fraction at which the least cost systems operate also increases with load and DHW fraction. Thus, the larger systems, and those with less seasonal load profiles not only cost less per unit of energy, they also displace more non-renewable energy and hence offer greater savings than indicated by the decrease in unit cost alone.

Table 3-3 summarizes the unit solar cost sensitivity results for all the Madison reference cases. Note that the total load sensitivity is modest for all storage technologies except the rock cavern, and that the reductions in unit cost of systems larger than 36TJ (500 houses) are small for all storage systems. Therefore, it appears that while large systems are desirable, systems as small as 50 house equivalent load are feasible for all but rock cavern storage systems.

The cost sensitivity curves for the high-temperature rock cavern/system for variations in collector and storage size and unit cost are shown in Figure 3-22. This figure illustrates that the reference system is indeed optimal with respect to collector and storage sizes and that the sensitivity to collector unit cost is nearly twice as great as the sensitivity to storage unit costs. The corresponding solar fraction sensitivity results are shown in Figure 3-23.

Collector costs dominate all but the low-temperature systems that employ unglazed collectors and heat pumps. Heat pumps are a significant but not dominating element of the cost of those systems that employ them (about 15%). For most collector-dominated systems, a collector cost reduction of 20 percent would result in a total system cost of about 10 percent.

Table 3-2. SENSITIVITY ANALYSIS RESULTS FOR ROCK CAVERN REFERENCE SYSTEM WITH EVACUATED COLECTOR, LTDS, AND NO HEAT PUMP IN MADISON

CASE		Auxil./ Load*100 %	COP -	Solar Fraction %	Solar Unit Cost $/MWh	Total Unit Cost $/MWh
Collector Area	m^2					
	10 000	64.3		35.7	153.9	109.
	15 000	39.1		60.9	112.4	102.5
R *	20 000	18.7		81.3	100.5	98.
	25 000	13.6		86.4	110.	108.
	30 000	10.5		89.4	121.4	112.2
Storage Volume	m^3					
	56 250	32.9		67.1	110.2	102.9
	84 375	24.9		75.1	103.9	101.7
R	112 500	18.7		81.3	100.5	98.
	140 625	15.5		84.5	100.8	102.4
	168 750	15.1		84.9	104.1	106.4
Load Variation						
	50	55.9		44.1	359.5	199.5
R	500	18.7		81.3	100.5	98.
	5 000	16.7		83.3	79.2	79.3
Domestic Hot Water	%					
	0	25.1		74.9	109.6	101.5
R	20	18.7		81.3	100.5	98.
	50	16.9		83.1	98.1	95.8
Collector Costs	$/$m^2$					
	175	23.1		76.9	71.5	73.6
	262	18.7		81.3	83.9	84.7
R	350	18.7		81.3	100.5	98.0
Storage Costs	%					
	50	18.7		81.3	83.6	84.6
	75				92.	91.3
R	100				100.5	98.
	125				108.6	104.7
	150				117.	111.4

* Reference or optimum configuration

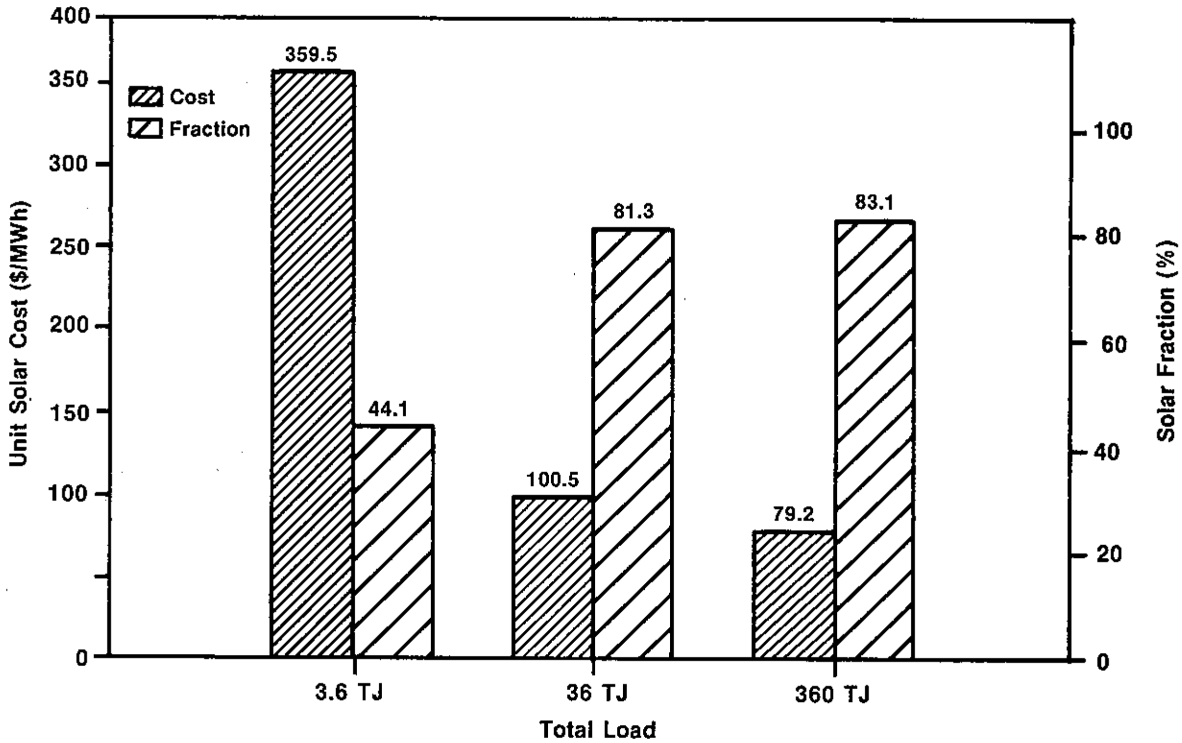


Figure 3-20. Effect of Total Load Variation on Unit Solar Cost and Solar Fraction of High-Temperature Rock Cavern Systems

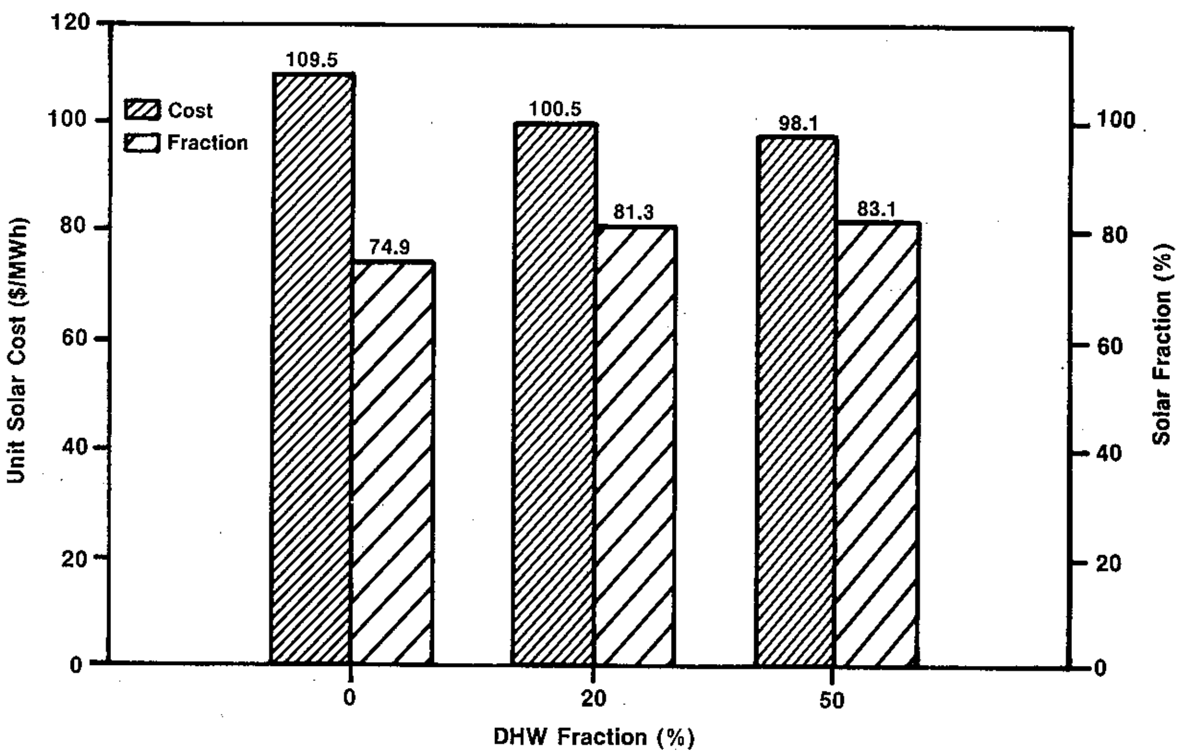


Figure 3-21. Effect of Domestic Hot Water Fraction Variation on the Unit Solar Cost and Solar Fraction of High-Temperature Rock Cavern Systems

Table 3-3. LOAD SENSITIVITIES FOR LOW- AND HIGH-TEMPERATURE DISTRIBUTION SYSTEMS IN MADISON FOR 50, 500, AND 5000 HOUSE LOADS

System		Unit Energy Cost, $/MWh			Cost Ratios C/C(500)		
		50	500	5000	50	500	5000
Cavern	EC, HTDS	199.5	98.0	79.3	2.04	1.0	0.81
	EC, LTDS	150.6	83.2	66.8	1.81	1.0	0.80
	FP, LTDS	187.7	88.6	73.2	2.13	1.0	0.83
Pit	EC, HTDS	117.5	94.2	88.1	1.25	1.0	0.94
	EC, LTDS	84.9	74.5	70.8	1.14	1.0	0.95
	FP, LTDS	99.9	79.6	78.3	1.25	1.0	0.98
Tank	EC, HTDS	143.5	116.5	106.7	1.23	1.0	0.92
	EC, LTDS	111.3	93.5	86.6	1.19	1.0	0.93
	EC, LTDS, NPH	119.4	96.0	87.8	1.24	1.0	0.91
	FP, LTDS, HP	98.8	80.9	73.4	1.22	1.0	0.90
Duct	EC, HTDS	124.7	102.8	98.5	1.21	1.0	0.96
	EC, LTDS	99.8	85.7	81.4	1.16	1.0	0.95
	FP, HTDS, HP	115.1	95.5	85.7	1.20	1.0	0.90
	UC, LTDS, HP	67.8	57.5	55.9	1.18	1.0	0.97
Aquifer	EC, HTDS, NHP	201.0	139.0	139.0	1.45	1.0	1.00
	FP, LTDS, NHP	127.0	82.0	77.0	1.56	1.0	0.94
	EC, HT, HP	130.0	102.0	95.0	1.27	1.0	0.91
	FP, LTDS, HP	58.0	41.0	38.0	1.41	1.0	0.91

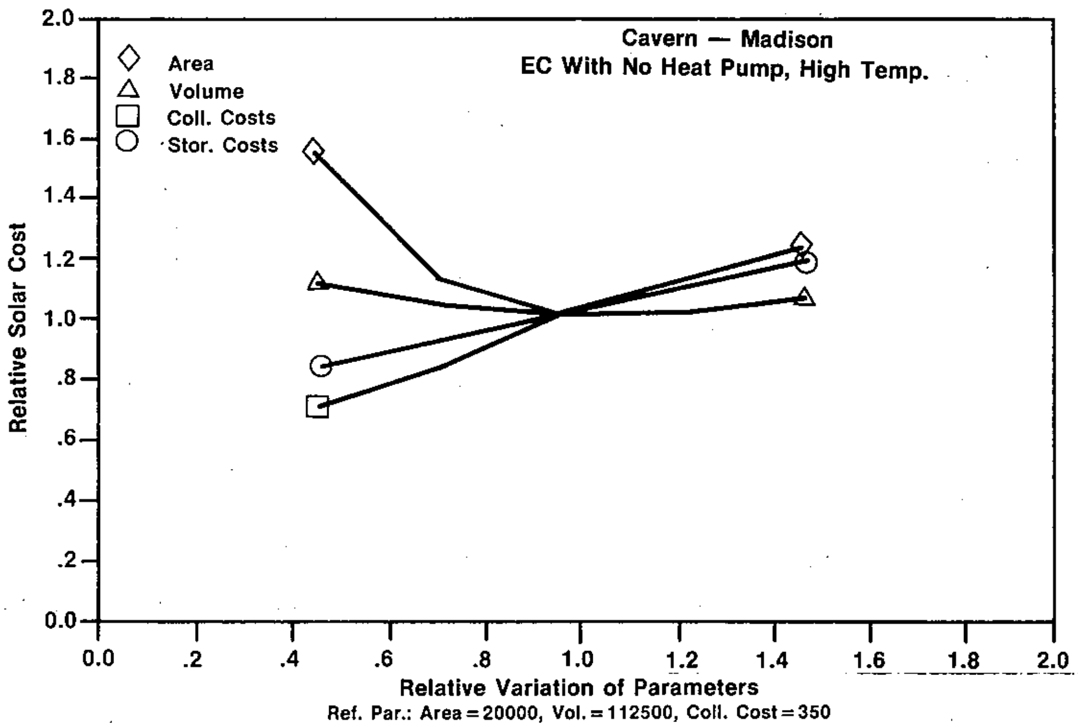


Figure 3-22. Cost Sensitivities for High-Temperature Rock Cavern System to Collector and Storage Size and Cost Variations

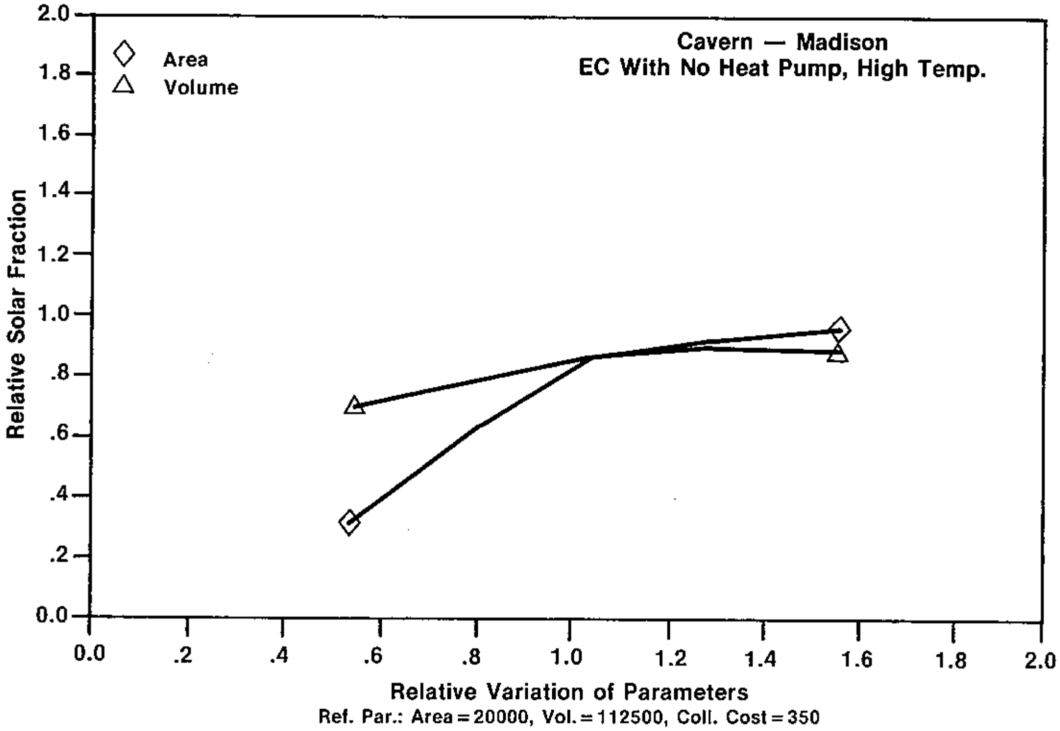


Figure 3-23. Solar Fraction Sensitivities for Rock Cavern System to Collector and Storage Size and Cost Variations

The general findings for the cost sensitivities of systems based on the various storage options are summarized in Table 3-4 which indicates the relative sensitivities for the various major subsystems as high (H), medium (M) or low (L).

Table 3-4. SUMMARY OF COST SENSITIVITIES FOR REFERENCE SYSTEMS

	LOAD	% DHW	A_c	V_s	HP	STORAGE PARAMETERS
Aquifer	M	M	H	M	-	L
Duct	L	M	H	L	-	M
Tank	L	M	H	M	L	M
Pit	L	M	H	L	L	L
Cavern	H	M	H	H	L	L

The reports of the analysis teams [9,10,11] contain results and additional data that permit the determination of cost and performance sensitivities of many additional variables. Sensitivities to design variables such as insulation thickness, diameter and spacing of boreholes, aquifer depth, heat pump heat exchanger capacity, were generally small (usually less than 10 percent), but are still important in the design of actual plants. For example, the sensitivities to doubling or halving the heat transfer capacity of heat pumps is less than 1 percent for cavern and pit systems and less than 5 percent for duct systems. The parameters of the duct storage are more important: -50 to +100 percent deviations from the optimum insulation thickness, duct number, spacing, and borehole depth can lead to cost increases of 10 to 20 percent.

3.7 SYSTEM START-UP EFFECTS

Most of the energy storage technologies covered in this study involve periodic heating of a large mass of water, earth, or rock that is thermally coupled to the surrounding earth. After a few cycles of heating the storage mass, the heat loss to the surrounding becomes relatively constant and is normally small compared to the energy stored. During the initial cycle, and perhaps several subsequent cycles, however, a substantial amount of energy is required to warm the surroundings in the immediate vicinity of the storage mass. Thus, the "heat loss" may be much higher during the first few storage cycles than it will be during most of the economic life of the system. To account for this, the MINSUN program provides a "pre-heat" feature in which the storage model is run through several cycles of sinusoidal variation in temperature before the actual simulation is begun. All of the calculations discussed in this section employed the "pre-heat" feature except the aquifer calculations.

To investigate the importance of the start-up heat losses on performance of aquifer systems, a five-year simulation was run for a direct-coupled (no heat pump) LTDS in Madison [9]. The system was sized so that the first year solar fraction was 94 percent. The results are shown in Figures 3-24 and 3-25. The temperature available to the system during the final weeks of the year is much higher after five years of operation, and the auxiliary energy required is greatly reduced. The five year average for solar fraction is 98 percent compared with 94 percent initially; average auxiliary energy required is 212 MWh c.f. 600 MWh; and average solar energy costs 78 $/MWh c.f. 82 $/MWh. Thus in comparing the aquifer results with the other storage technologies, one must remember that we are essentially comparing first year aquifer performance with second to fifth year performance of the other technologies. The start-up energy will be relatively unimportant for the low solar fraction heat pump results since the energy storage occurs at temperatures that are not much above the surrounding rock temperature. The high solar fraction results, however, especially those for the HTDS, will be quite different after a few years of service. Therefore, the rapid rise in cost of aquifer systems as the solar fraction approaches unity is partly due to using first year performance rather than a long term average in computing the unit cost.

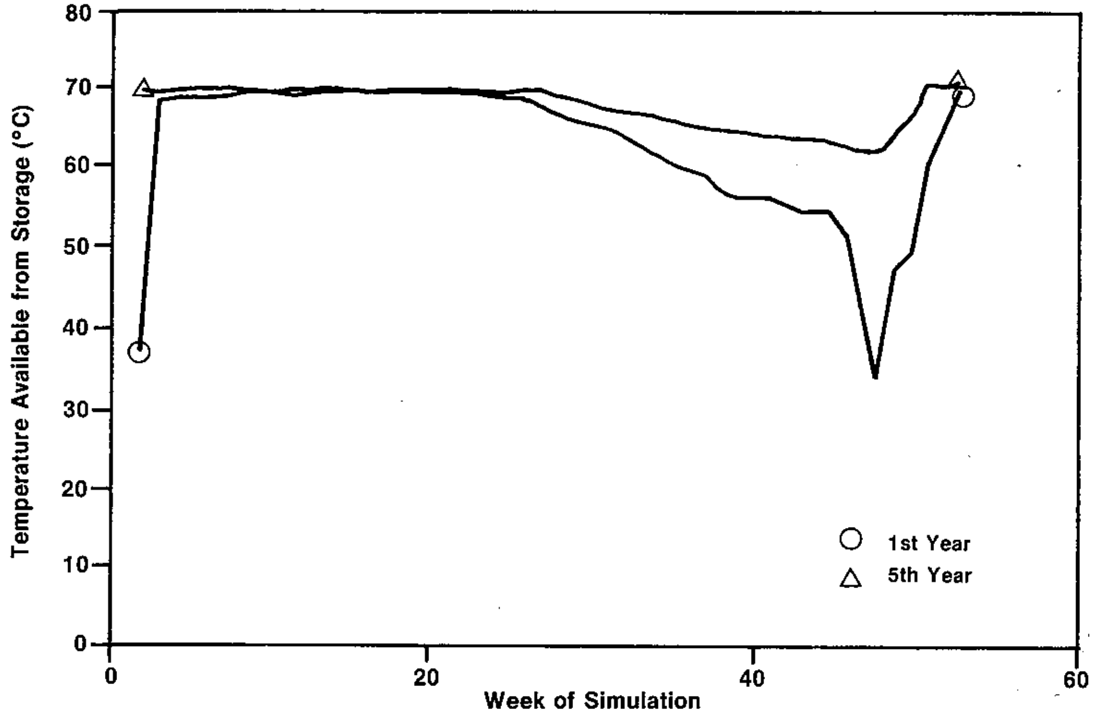


Figure 3-24. Aquifer Output Temperatures from Multi-Year Simulation

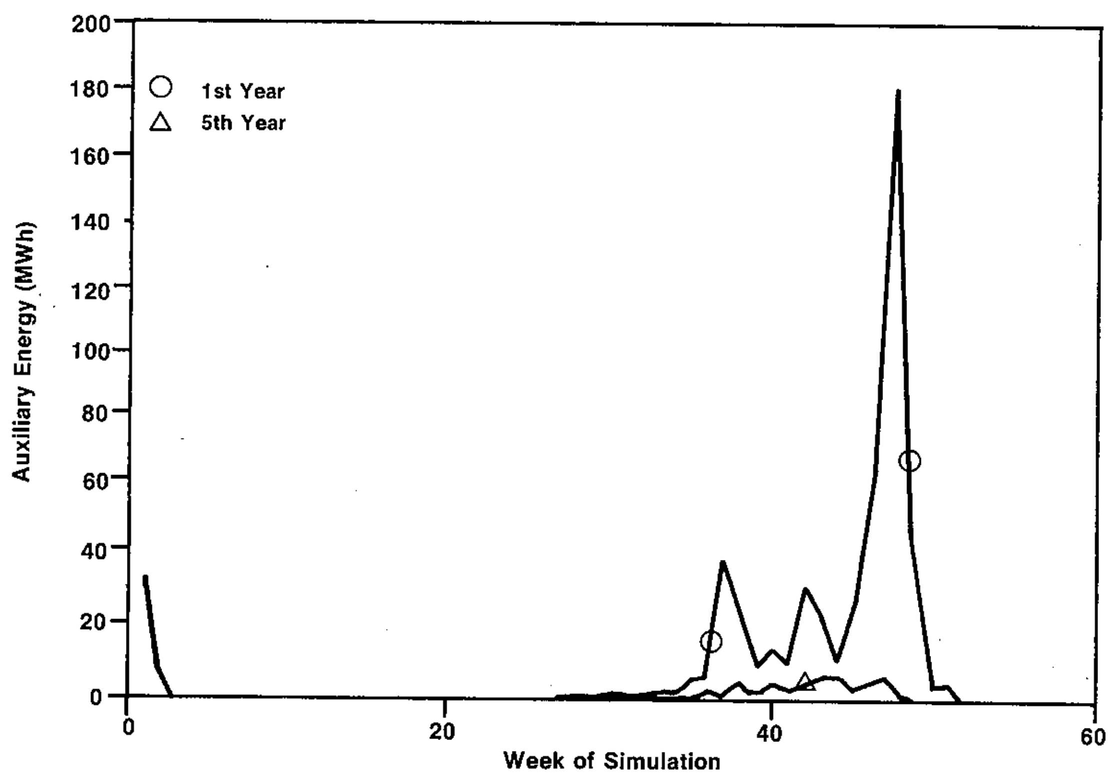


Figure 3-25. Auxiliary Energy Requirement of Aquifer CSHPSS System from Multi-Year Simulation

A similar study was performed in Sweden using the MINSUN model of the Lyckebo rock cavern storage system [18]. The calculations indicate an increase of about 5° C in the maximum temperature at the top of the cavern over a three year period. The thermal energy storage losses drop dramatically after the first year and more slowly after the second year as shown in Figure 3-26. The expansion diagram calculations were all run with the MINSUN five year pre-heat feature, and from the Lyckebo simulation five years appears to be more than adequate to establish an equilibrium annual heat loss.

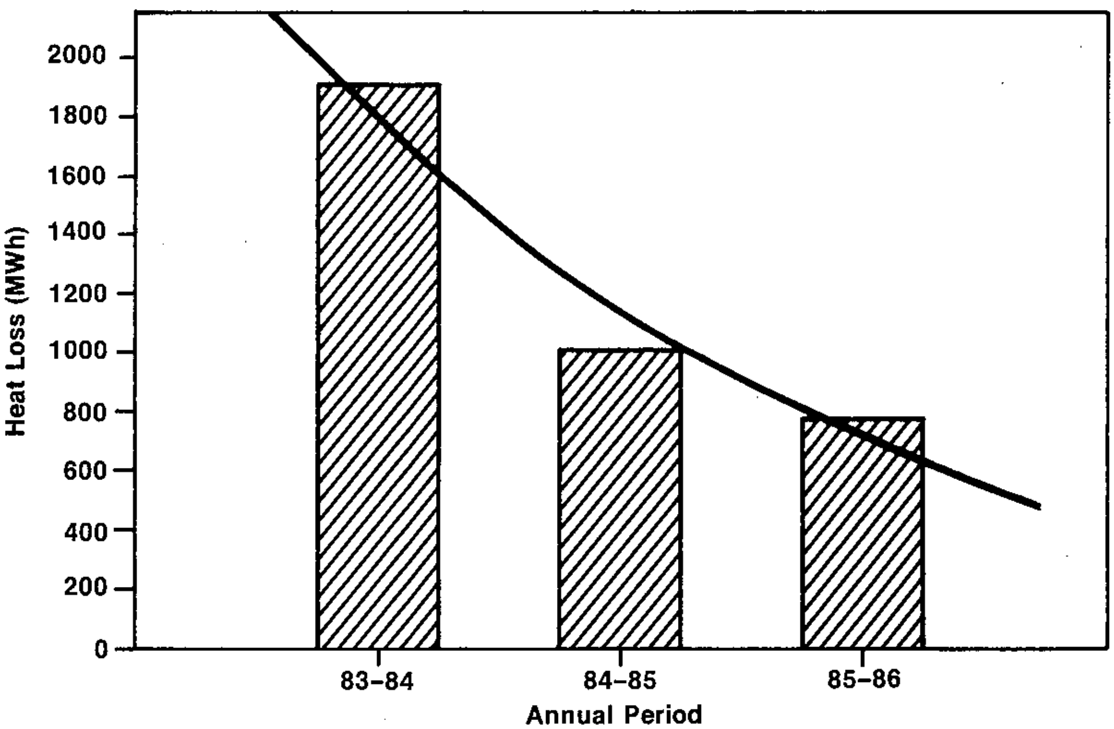


Figure 3-26. Simulated Annual Heat Losses from Rock Cavern Storage of Lyckebo Plant

3.8 ECONOMICS OF REFERENCE SYSTEMS

Although reference case evaluations were not intended to assess the viability of CSHPSS in relation to competing technologies, we can make a few general remarks under this heading.

The first observation is that solar costs from the better solar systems are already low enough to be competitive in areas where auxiliary energy equivalent cost is high--say 50 to 100 $/MWh. There are many areas in the United States and the rest of the world where energy is already that expensive. Whether those are also areas where the climatic, geological and institutional conditions favor CSHPSS remains to be seen. The appeal of the lowest cost CSHPSS systems is predicated upon the acceptance of low temperature distribution technology. This has been successful in Sweden [19].

Since the distribution network cost was not included in this analysis, it is only legitimate to compare the unit energy costs derived from the base case studies with other energy sources available to central plants. These sources include gas, oil and electricity or waste heat from industry or power plants. Rates charged to the largest users or contract buyers for energy from these sources may be substantially lower than the residential or commercial rates that are most familiar to consumers.

It is much more difficult to determine whether CSHPSS offer economic advantages over distributed conventional heating systems. This would require a detailed assessment of the cost of the district heating system and knowledge of the mix of buildings, the energy density distribution, the type of excavation required for pipes, the existing building heating equipment (if any), and the cost of distributing the conventional energy. If a CSHPSS is being considered for a new community or development, the problem is somewhat easier because only the net differences between district heating and distributed heating costs must be estimated. In Section 5, based on Swedish results, we estimate that the distribution network would add about 10 $/MWh to the cost of delivered energy.

It is clear from the sensitivity studies that a reasonably high DHW fraction, or other fixed load, is quite advantageous. This would imply that a full community including commercial and light (low-temperature) industrial loads might be a better target market for solar plants than a strictly residential application. Although the total load sensitivity is not great for any of the systems other than the rock caverns, there is a clear economy of scale in all systems. Systems for loads of 36 TJ or more are large enough to warrant such cost, labor, and energy-saving features as fully automated (smart digital) control systems, efficient maintenance and repair schedules, large purchase power, thorough design, construction and acceptance procedures, etc. The chief difficulties with large installations would probably be siting the collectors and storage near the load centroid, and the institutional problems of initiating a new technology on a large scale. Non-heat pump systems smaller than 3.6 TJ, probably do not make sense because of high heat losses from storage. Larger systems, say 360 TJ or more, would probably have advantages in addition to the five to ten percent cost reduction found here when operation and maintenance costs are included.

3.9 FURTHER ANALYSIS

There are questions raised by some of the results of the reference studies that need to be answered. The relative performance of systems with and without heat pump appears to be dependent more strongly on the type of storage than one would expect.

The apparent limits to solar fraction in heat pump systems using aquifers seems to be due partly to the limited range of temperatures used in the simulations and start up heat losses in the aquifer. It would be worth removing these limitations from the analysis so that the system expansion paths could be extended. The single well restriction of the aquifer model also should be removed. It introduces a degree of artificiality and uncertainty into the analysis that is hard to assess.

One of the advantages of a plant employing an aquifer is its flexibility. One can imagine that a very attractive scenario for introducing CSHPSS at an early date is to build a plant with a heat pump and operate it at the current economic optimum point--maybe 50 to 60 percent solar--but as the price of electricity increases, the plant could be operated at progressively higher solar fractions simply by increasing the temperature of the aquifer. Perhaps no physical changes would be necessary, or it might be necessary to add collectors.

Another possibility that has not been explored in the present work is to combine unglazed and high performance (evacuated or tracking) collectors in series for use with high-temperature distribution systems. This would be especially effective with highly stratified storage or aquifers without heat pumps in which the maximum temperature difference exists in the storage most of the time.

Sensitivity results for the economic parameters are needed. Although we have removed the largest source of uncertainty (fuel escalation rate) by optimizing on solar cost, there are still questions about the effects of plant lifetime and discount rate. Operation and maintenance costs and land values should be included explicitly rather than including those costs implicitly in the initial capital costs.

There are variations and alternative configurations that a system designer or owner would want to consider. It is probable that some of those variations could improve cost effectiveness by perhaps 10 to 20 percent. It is equally probable that some of the simplifications and assumptions necessary to make the analysis practical could lead to results that are optimistic to a similar degree.

4.0 NATIONAL EVALUATIONS

The purpose of the national evaluations was to examine the potential economic viability of CSHPSS systems in the context of specific geographic, climatic, and economic environments. Although the reference system studies indicate the relative performance of different configurations in typical environments and indicate that CSHPSS are likely to be viable in favorable locations, only the site specific evaluation afforded by the national studies can provide a comparison of the economics of CSHPSS with conventional alternatives that is meaningful in terms of implementation of projects.

The national evaluations described in this section were conducted by the participants using the methods previously described in Section 2. National or site-specific data were substituted for reference data where possible, and the configurations selected were those deemed most appropriate for the particular site on the basis of reference system rankings and local conditions.

4.1 CANADA *

4.1.1 Introduction

The Canadian National System Study examined an Aquifer Thermal Energy Storage (ATES) concept together with low cost solar collectors for residential heating purposes. Three locations, Toronto, Winnipeg, and Fredericton, were analyzed using economic parameters appropriate for Canada and for these locations. The results indicate that the solar unit costs would be between 47 and 55 CAN$/MWh (1985 Canadian dollars). Winnipeg, with its severe climate but favourable insolation, has the lowest solar unit energy cost. Winnipeg also has the lowest fuel and electricity costs--15.9 and 19.0 CAN$/MWh respectively.

4.1.2 Conditions and Systems

In the Canadian Evaluation an attempt was made to maintain the common parameter values and assumptions employed in the analysis reported in Section 3. Differences between these analyses and the Canadian Evaluation reflect Canada's particular situation. Changes were made in the following areas: weather, costs, tapwater load, and collector parameters.

The differences are summarized in Table 4-1.

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*Material for this section was provided by Verne C. Chant, James F. Hickling Management Consultants, Ltd, Sixth Floor, 350 Sparks Street, Ottawa, Ontario K1R 7S8, CANADA. Please contact Dr. Chant directly for additional details.

Table 4-1. COMPARISON OF II(b) REFERENCE CASE AND CANADIAN NATIONAL EVALUATION PARAMETERS

	II(b) Reference Case	Canadian National Evaluation	
Location:	Madison	Toronto Fredericton Winnipeg	
Load:			
Houses	500		500
Heat Loss Coefficient (W/m^2K)	0.583		0.49
Tap Water (W/house)	460		600
Total Load (MWh)	10072	Toronto	10047
		Fredericton	10947
		Winnipeg	13524
Collector:			
Type	Flat Plate	Canadian Flat Plate	
Area	Optimized for each location		
Storage:			
Type	Aquifer		Aquifer
Supply Temperature (^oC)	10		8
Storage (collector outlet) Temperature (^oC)	Optimized for each location		
Heat Pump:			
Heat Exchange Capacity (kW/K)			
Evaporator	Optimized for each location		
Condenser	250	Toronto	225
		Fredericton	200
		Winnipeg	250
Component Capital Costs:	**US$**		**CAN$**
Solar Collectors ($/m^2$)	245		300
Piping ($/m)	250		225
Aquifer ($)	154000		250000
Heat Pump ($)	201500	Toronto	175000
		Fredericton	217000
		Winnipeg	305000
Auxiliary Heater ($/W)	0.1		0.1
Energy Costs:	**US$**		**CAN$**
Heat Pump Electricity ($/kWh) (Industrial Sector Prices)	0.07	Toronto	0.0367
		Fredericton	0.0421
		Winnipeg	0.0190
Auxiliary Fuel ($/kWh) (Industrial Sector Prices Divided by Efficiency)	0.07	Toro. (Oil)	0.0201
		Fred. (Oil)	0.0245
		Winn. (Gas)	0.0159

4.1.2.1 Weather

Canada has many different climatic conditions. Three locations were chosen for this study: Toronto, Winnipeg, and Fredericton. Each represents a different geographic area and is considered likely to contain suitable aquifer formations. The locations are illustrated in Figure 4-1, and the important weather indicators are summarized in Table 4-2.

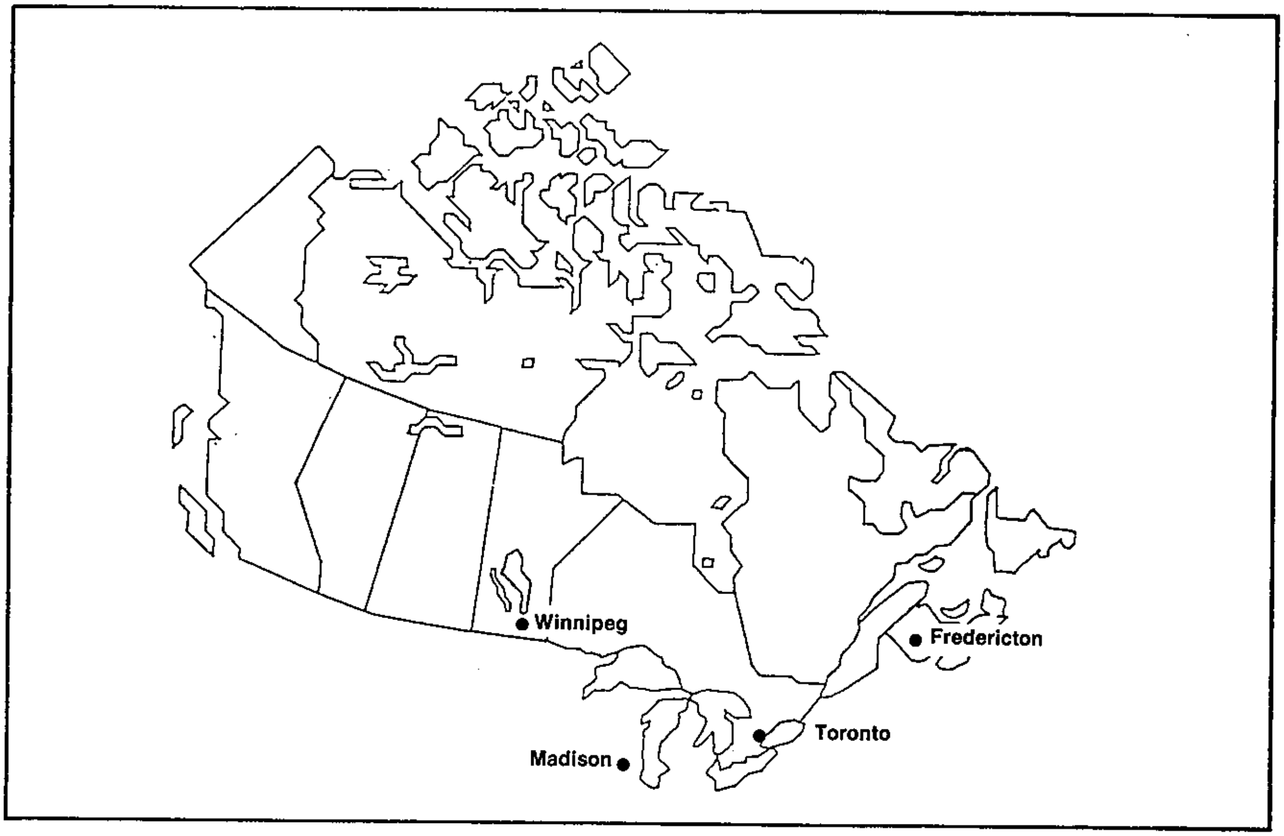

Figure 4-1. Canadian National Evaluation City Locations

Winnipeg is situated in the prairies and experiences continental weather patterns. There is already extensive use of aquifers to supply ground water for cooling.

The weather near Toronto is strongly influenced by the Great Lakes. Aquifer energy storage is currently being implemented at two sites in this area.

Fredericton is in the Maritime provinces; however, it does not have a true maritime climate due to its distance inland. Fredericton is one of the few sites in the Maritime region which may contain aquifers suitable for thermal storage.

TABLE 4-2 CANADIAN WEATHER DATA

	JAN	FEB	MAR	APR	MAY	JUNE	JULY	AUG	SEPT	OCT	NOV	DEC	TOTAL
Hours Bright Sunshine													
Toronto	92	112	145	182	233	253	281	252	192	149	81	75	2045
Fredericton	106	122	140	154	200	206	233	221	167	138	95	93	1878
Winnipeg	121	124	176	220	226	276	216	283	185	152	91	93	2321
Mean Daily Temperature													
Toronto	-5.3	-4.9	-0.3	6.2	11.9	17.6	20.6	19.8	16.1	10.0	3.9	-2.5	7.8
Fredericton	-9.1	-8.3	-2.5	4.0	10.6	16.0	19.1	18.1	13.4	7.7	1.7	-6.4	5.4
Winnipeg	-19.0	-15.1	-7.8	3.6	11.5	17.2	20.2	18.8	12.8	6.6	-4.2	-13.7	2.6
Degree-Days Below 18													
Toronto	701	618	536	314	150	31	3	7	67	220	393	608	3646
Fredericton	843	745	633	419	229	79	20	37	153	326	498	758	4740
Winnipeg	1148	938	800	432	211	62	17	39	169	356	665	982	5819

4.1.2.2 Costs

All costs are in 1984 Canadian dollars. Cost parameters which were adequately represented by the reference system values were left as found in the II(b) reference case analysis. The following cost parameters were altered to reflect current Canadian values: collectors, aquifer, piping, and fuel. Heat pump, auxiliary heater, and economic parameters (depreciation time, real interest rate, fuel escalation rate) were not changed.

The collector cost of 300 CAN$/m^2 represents an all-inclusive installed cost for Canadian-made collectors. This is based on extrapolation from the installation at the McLaren pulp and paper plant, Canada's largest at 2,200 m^2 gross area.

The aquifer cost parameters were calculated to obtain a total cost of CAN $250,000. This is based on experience obtained at the Scarborough Government of Canada Building, which has an aquifer available for seasonal cooling energy storage. Cost figures were adjusted to reflect the aquifer formation modeled in the Canadian Evaluation.

The piping cost parameters were calculated to obtain a total cost of CAN $2,250,000 for the distribution system. The figure is based on a district heating study done for the Ontario Ministry of Energy.

Fuel costs for the Canadian System Study were based on a 1981 Energy, Mines, and Resources (EMR) energy price forecast. More recent energy price level estimates from EMR and Statistics Canada indicate that those prices adequately represent current costs.

The heat pump was assumed to be electrically driven. The auxiliary heater was assumed to use the cheapest available fuel in each city. Industrial sector prices were used since a central heating plant would be a bulk purchaser of energy. Prices were adjusted to reflect the relative efficiencies of combusion of each fuel type in an industrial application (electricity 100%, oil 90%, natural gas 85%).

When comparing the unit solar cost with the cost of conventional energy, the annualized cost of conventional energy was used. The cost of energy was annualized using a fuel escalation rate of 2% per year and a 20 year amortization period.

The following table summarizes the energy costs used in the Canadian System Study. Both cost and annualized cost are shown in the table.

CANADIAN ENERGY COSTS

	Cost (CAN $/MWh)		Annualized Cost (CAN $/MWh)	
	Electricity	Auxiliary	Electricity	Auxiliary
Toronto	36.70	20.10 (oil)	43.20	23.70
Winnipeg	19.00	15.90 (gas)	22.40	18.70
Fredericton	42.10	24.50 (oil)	49.50	28.80

A survey of heat pump and auxiliary heater prices did not indicate that Canadian prices would vary significantly from the Subtask II(b) reference case values. These values were 0.1 CAN$/W for the auxiliary heater and approximately 0.2 CAN$/W for the heat pump motor compressor combination.

4.1.2.3 Tapwater Load

The typical domestic hot water load is higher in Canada than in Europe. Therefore, the tap water load was increased from 460 W/house to 600 W/house for the Canadian Evaluation. This figure is based on energy demand estimates developed for Energy, Mines, and Resources, Canada.

4.1.2.4 Collector Parameters

Collector parameters used in the Canadian Evaluation are based on a Canadian built collector array currently in use. The collectors are flat plate, single glazed with selective coatings and back insulation. The heat transfer medium is a 60/40 water and glycol mixture with a maximum flow of 0.02 kg/s m^2.

The efficiency parameters of the Canadian and Subtask II(b) collectors are compared below:

	η_o	$F_R U_L$ W/m^2K	b_o
Canadian Collectors	0.753	5.25	0.1
II(b) Collectors	0.808	4.40	0.1

4.1.3 Methodology

The analysis approach for the Canadian Evaluation follows closely the method used for the heat pump cases of the aquifer reference case analysis.

MINREP runs were performed for several collector outlet temperature/collector area combinations by varying the values of the heat pump heat transfer capacity. The optimal size of the tap water system evaporator and condenser and the house heating system condenser were thus determined.

4.1.4 Results

Calculations for many combinations of collector inlet/outlet temperatures and collector sizes were then made by varying the evaporator heat transfer capacity in the house heating system. The results of these runs, plotted as solar cost versus solar fraction, are contained in Figure 4-2 and show the result for Winnipeg.

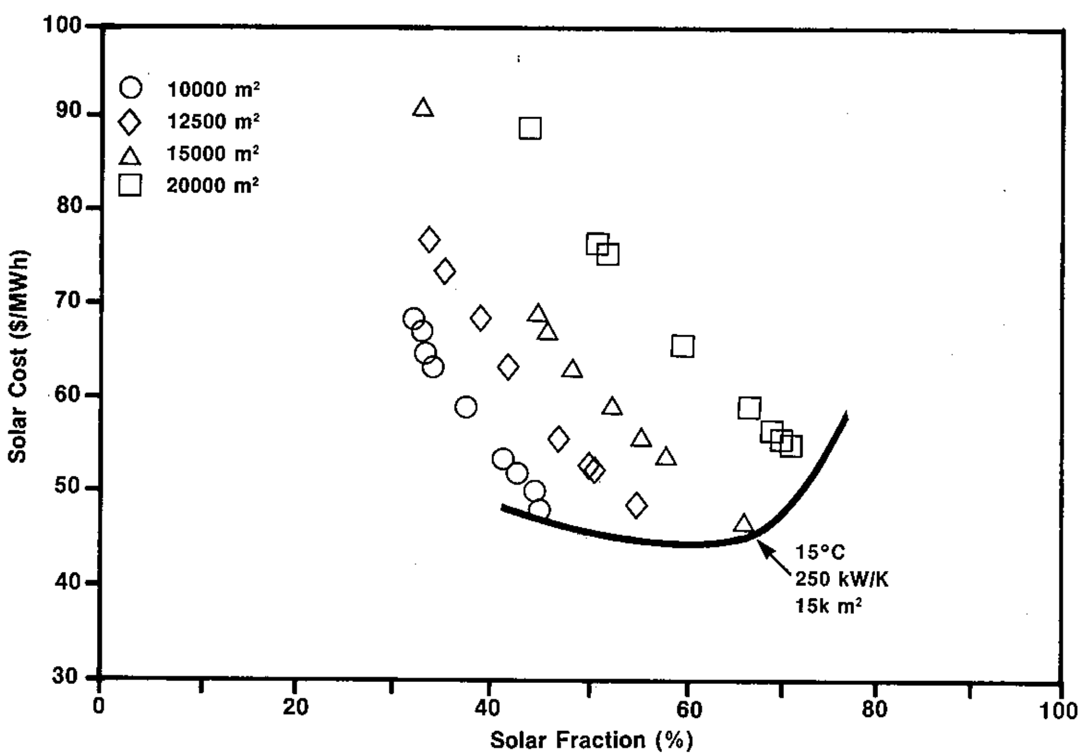

Figure 4-2. Solar Cost vs Solar Fraction - Canadian National Evaluation - Winnipeg

The reference system was defined as the "knee" in the solar cost/solar fraction envelope. These systems were as follows:

	COLLECTOR AREA (m^2)	EVAPORATOR SIZE (kW/K)	OUTLET TEMPERATURE (°C)	SOLAR FRACTION (%)	SOLAR COST ($/MWh)
Toronto	12,500	200	15	68	65
Fredericton	15,000	225	15	69	55
Winnipeg	15,000	250	15	66	47

While the design of these reference systems varies, the solar fraction and solar cost values are approximately the same in each of the three cities at 68% and $51/MWh respectively. These three reference systems form the basis of all further results in this report. These reference systems for the three Canadian cities and the reference case of Subtask II(b) at Madison are compared in Table 4-3. Figure 4-3 shows the energy usage components for Toronto on a weekly basis over the year.

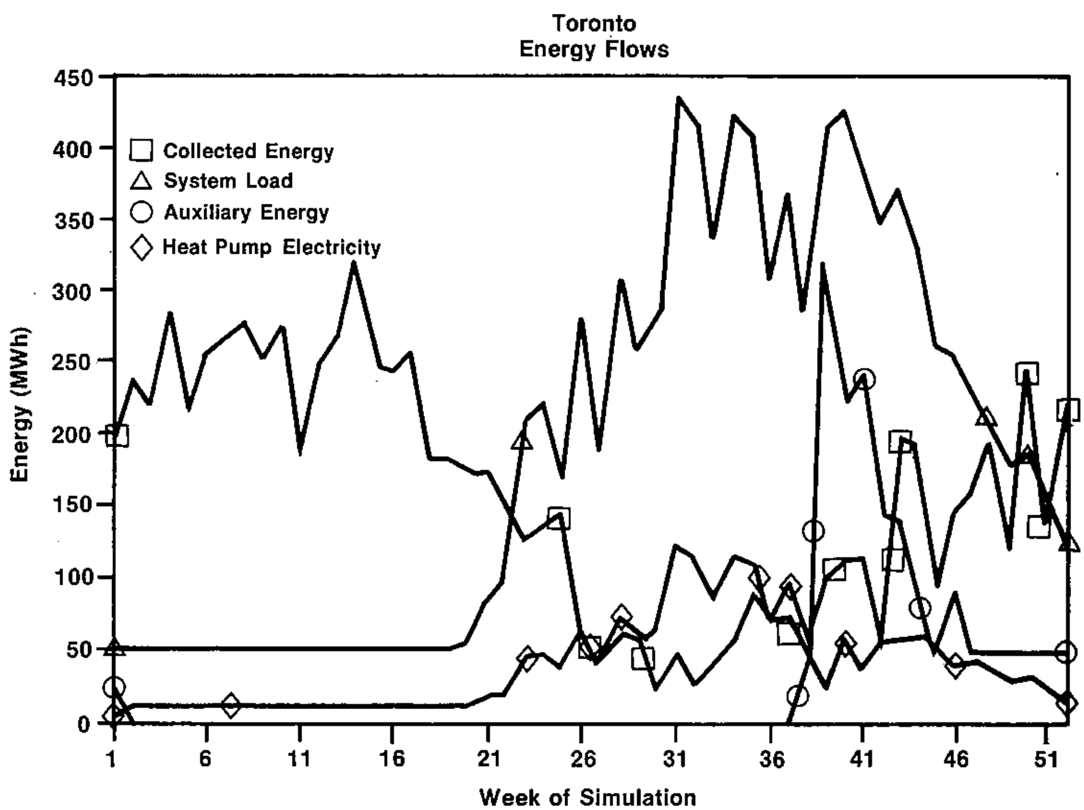

Figure 4-3. Energy Supply and Demand Profiles

TABLE 4-3. COMPARISON OF OPTIMIZED CSHPSS SYSTEMS

		Canadian National Evaluation			II(b)
		Toronto	Fredericton	Winnipeg	Madison
ad:	Houses	500	500	500	500
	Energy (MWh/yr)	10047	10947	13524	10072
lar	Area (m^2)	12500	15000	15000	10000
llectors:	Temperature Out (°C)	15	15	15	15
	Energy:				
	Incident (kWh/m^2 yr)	1446	1421	1662	--
	Collected (MWh/yr)	8346	9619	10716	7336
	(kWh/m^2 yr)	668	641	714	734
uifer:	Thickness (m)	20	20	20	20
	Depth Below Ground (m)	20	20	20	20
	Extracted Energy (MWh/yr)	6812	7526	8901	7205
	Efficiency (%)	82	78	83	98
at Pumps:	Tapwater:				
	Max. Cond. Power (MW)	0.3	0.3	0.3	0.23
	Electric Energy (MWh/yr)	559	561	591	497
	COP	4.0	4.0	3.8	3.9
	House Heat:				
	Max. Cond. Power (MW)	2.81	3.43	4.50	3.43
	Electric Energy (MWh/yr)	1794	2150	3342	1897
	COP	3.9	3.7	3.2	4.1
xiliary:	Tapwater:				
	Max Power (MW)	0.3	0.3	0.3	0.23
	Energy (MWh/yr)	385	404	406	77
	House Heat:				
	Max. Power (MW)	3.00	3.43	3.21	4.2
	Energy (MWh/yr)	498	307	284	397
lar Fraction (%)		68	69	66	72
		CAN$	CAN$	CAN$	US$
lar Cost ($/MWh) xcludes Distribution osts in Canadian Study)		52	55	47	31
pital Costs:	Solar (k$)	3750	4500	4500	2450
	Aquifer (k$)	250	250	250	154
	Heatpump (k$)	175	217	305	202
	Auxiliary (k$)	300	364	321	420
	Piping (k$)	2724	2724	2724	--
	Total Per House (k$)	14.1	16.1	16.2	--
velized Operating sts:	Heatpump (k$)	101	134	88	--
	Auxiliary (k$)	21	20	13	--
ualized System Cost (k$) apital and Operating)		673	770	720	--
	Per House ($)	1346	1540	1440	--

Figure 4-4 illustrates the cost breakdown for the three Canadian locations. Capital costs are equal at Fredericton and Winnipeg, and both greater than at Toronto. This is because of the larger collector arrays required by the two cities. Operating costs clearly indicate the differences in fuel prices. Winnipeg, which uses the most energy, has the lowest operating costs. Total cost is highest for Fredericton, followed by Winnipeg and then Toronto.

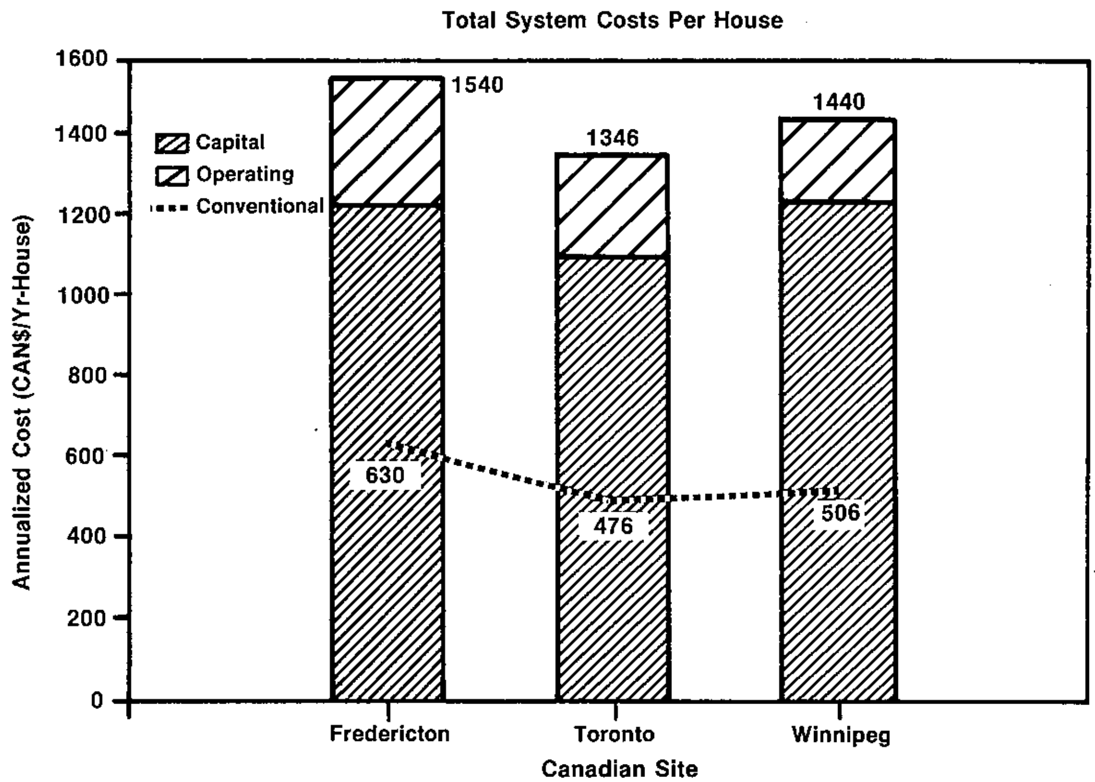

Figure 4-4. Comparison of System Costs for Fredericton, Toronto, and Winnipeg

4.1.5 Conclusions

Present energy costs in Canada are so low that it is difficult for any solar or renewable energy technology to compete on a strictly economic basis. The Aquifer Thermal Energy Storage concept is probably the most competitive active solar heating system that has been proposed.

4.2 COMMISSION OF EUROPEAN COMMUNITIES (CEC)*

4.2.1 Introduction

The CEC National study analyzed the performance of a solar heating system with a duct storage in earth, located in Northern Italy. The results show that costs of the energy delivered by such a system will be in the range of 50-60 US $/MWh.

4.2.2 Conditions and Systems

The system selected for analyses consists of:

- Single glazed, flat plate collectors with an optical efficiency of 0.808 and a heat loss coefficient of 4.4 W/m^2 C^o
- Duct storage in earth without top insulation
- A heat pump
- A load of approximately 500 houses, with a total load 10,000 MWh/year, consisting of 20% DHW and 80% space heating distributed by a low-temperature piping system.

All the other input parameters are those used in the reference case analyses of the duct team.

The weather data used are from Ispra, which is located in Northern Italy, about 60 km NW of Milan.

- Latitude 45° 48'
- Longitude 8° 37'
- Altitude 220 m above sea level
- Shift in solar time +25' 32" .

Ispra has a mild but humid climate. As a consequence an important part of the solar irradiation is diffuse. Some climatic parameters are listed below:

- Global horizontal radiation 1178 kWh/y (Copenhagen = 1018)
- Average ambient temperature in January 1.2°C (Copenhagen = -0.6°C)
- Energy collected at 20°C, 273 kWh/y (Copenhagen = 232)
- Number of degree day: 2500 (base = 18.3°C).

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*Material for this section was provided by Dolf van Hattem, Commission of the European Communities, Joint Research Center, I-21020 Ispra, Italy. Please contact him directly for further information.

4.2.3 Methodology

The system was optimized by varying systematically the value of the most important system parameters and simulating the yearly system performance for each case. The following parameters were varied:

- Collector area: 1000 to 25000 m^2
- Storage volume: 50000 to 375000 m^3
- Number of boreholes: 500 to 6500
- Depth of storage: 20 to 60 m.

The parameters were varied simultaneously. The condensor and evaporator surfaces of the heat pump were held constant since their influence was found to be of minor importance.

4.2.4 Results

The results obtained so far are given in Figures 4-5 to 4-7. Figure 4-5 shows the solar costs vs. solar fraction. This figure is based on about 900 runs. The high COP's obtained (up to 6.5) result in high solar fractions. Results agree with those for Copenhagen in the Duct Team Report [10].

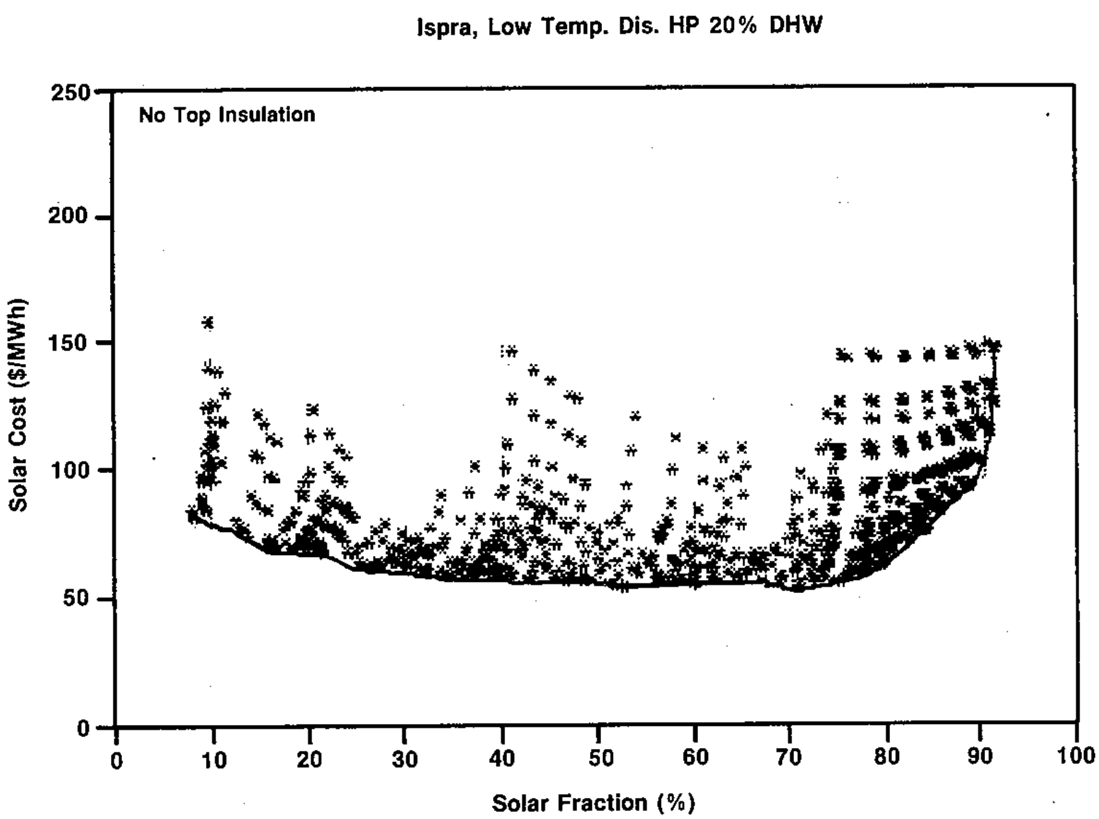

Figure 4-5. Solar Costs as Function of the Solar Fraction

Figure 4-6 shows the marginal solar costs vs. solar fractions. The marginal costs are almost constant between 15 and 70%. The dip between 60 and 70% is due to imperfect optimization. Figure 4-7 shows the total system costs vs. costs of auxiliary fuel. The curve shows that for the cost data used and climate considered, this kind of system will become economically attractive when the cost of auxiliary energy is slightly over 50 $/MWh. However, in this analysis, the distribution costs have been neglected, and the costs of electricity and fuel have the same value.

The results of the sensitivity analyses are shown in the "spider diagrams" of Figures 4-8 to 4-11. The central point of these diagrams corresponds to the <u>reference case</u> which was chosen as follows:

- o Collector area: 10,000 m^2
- o Storage volume: 325,000 m^3
- o Depth of storage: 30 m
- o Borehole radius: 0.05 m
- o Top insulation: 0.0 m.

In Figure 4-8 and Figure 4-7 the variation of the solar costs with some parameters is given. Remarkable is the fact an increase of the load results in an increase of solar costs. This can be explained by the fact that the size of the heat pump is proportional to the number of houses and that the investment costs for the heat pump are considered as solar costs.

The use of top insulation on the storage is not cost-effective as can be seen from Figure 4-8 (note that the scale on the horizontal axis in cm for this case, since the value for the reference case was zero) though the solar fraction increases slightly with the use of top insulation (Figure 4-9).

From Figure 4-10 it can be seen that the number of boreholes chosen for the reference case is not the optimum one. A smaller number of boreholes would have a lower solar cost, but not necessarily a lower system cost (see Figure 4-11). Figure 4-11 also shows that a slightly larger collector array would give lower total system costs.

However, one should be careful with the interpretation of the results. For example, one could conclude from Figure 4-10 and Figure 4-11 that an increase of storage volume would always give a decrease of costs. But one should realize that for this analysis the land costs and the costs due to an increased distance between the boreholes are not taken into account.

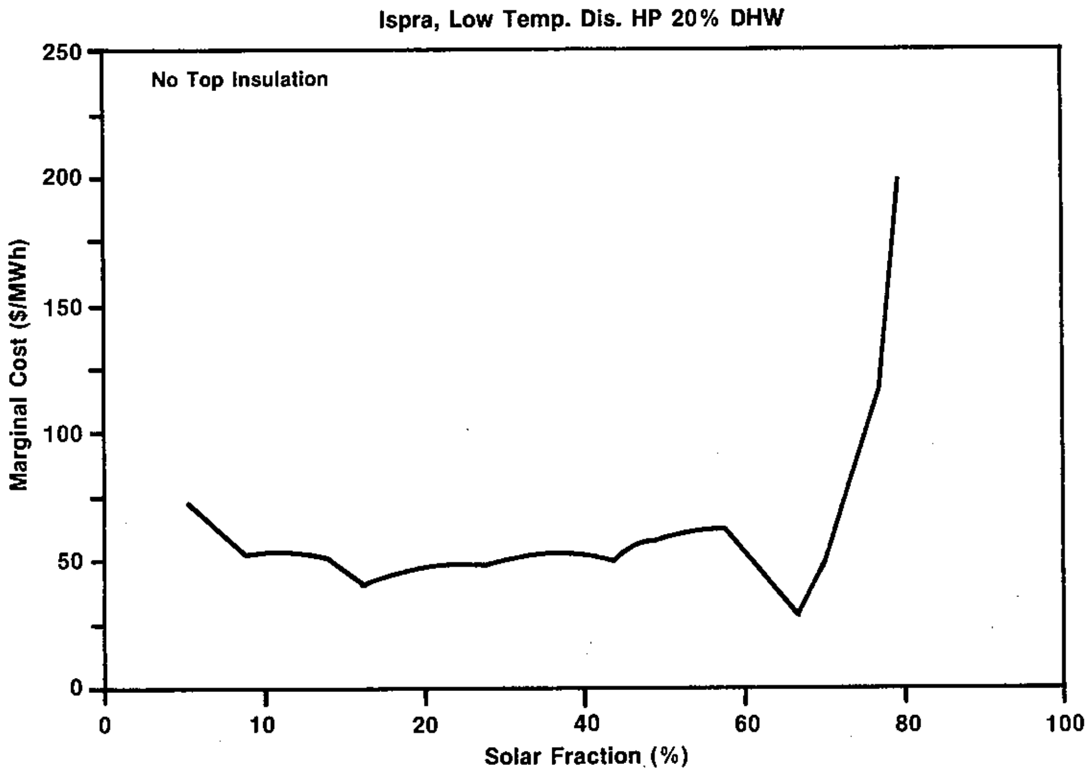

Figure 4-6. Marginal Solar Costs as a Function of Solar Fraction

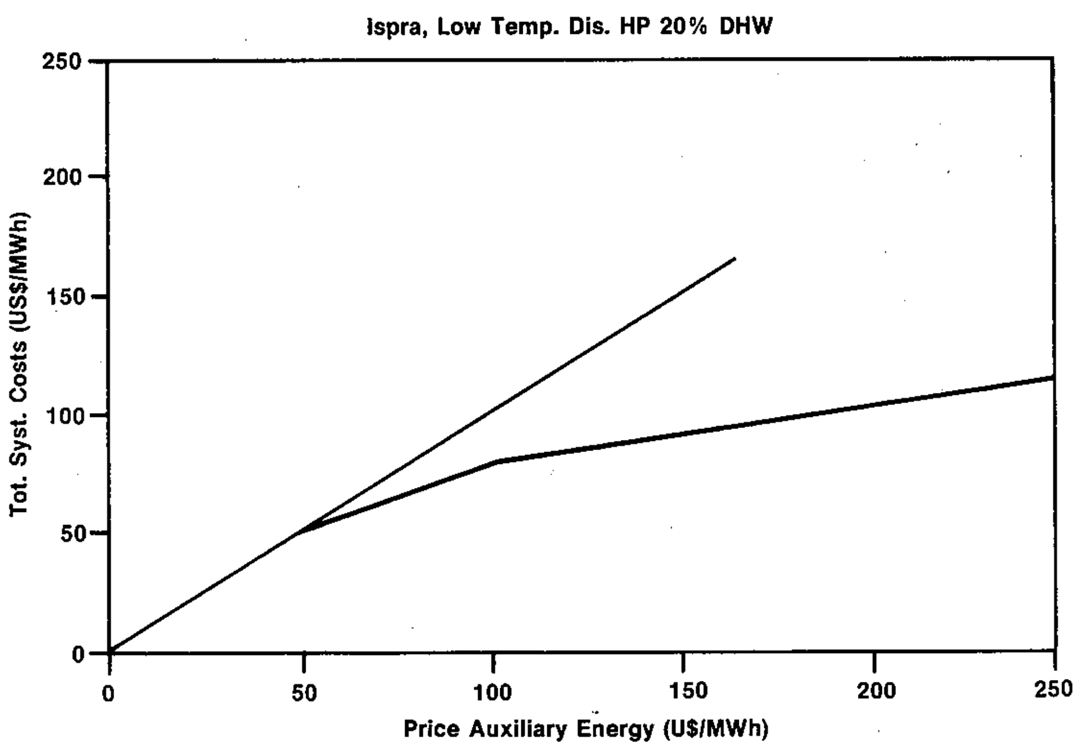

Figure 4-7. Total System Costs Versus Cost of Auxiliary Fuel

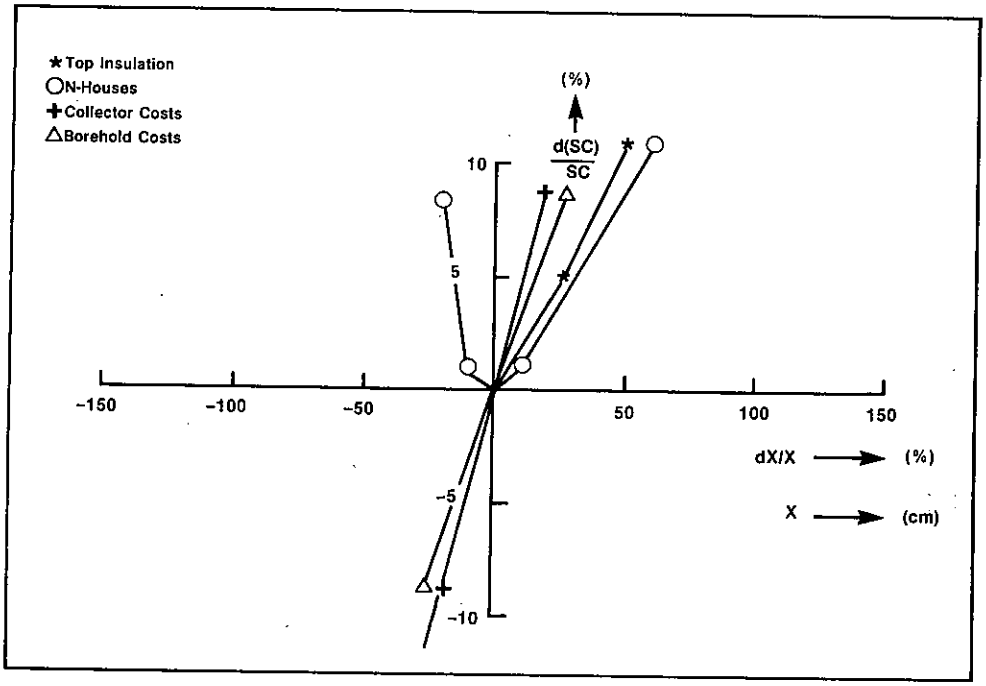

Figure 4-8. Variation of the Solar Costs With Some System Parameters

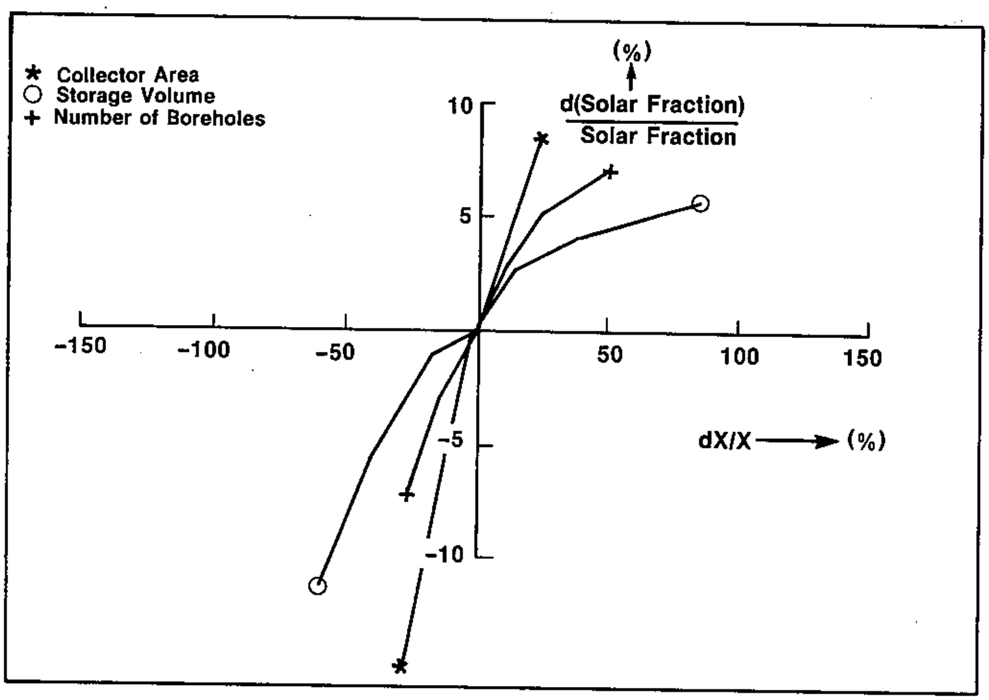

Figure 4-9a. Variation of Solar Fraction With Some System Parameters

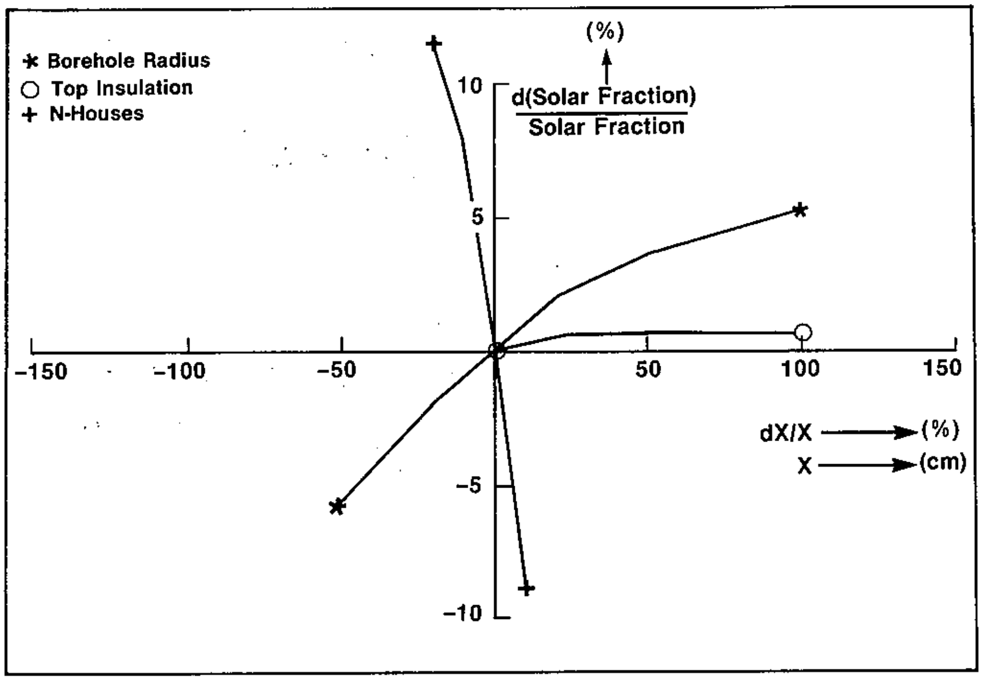

Figure 4-9b. Variation of Solar Fraction With Some System Parameters

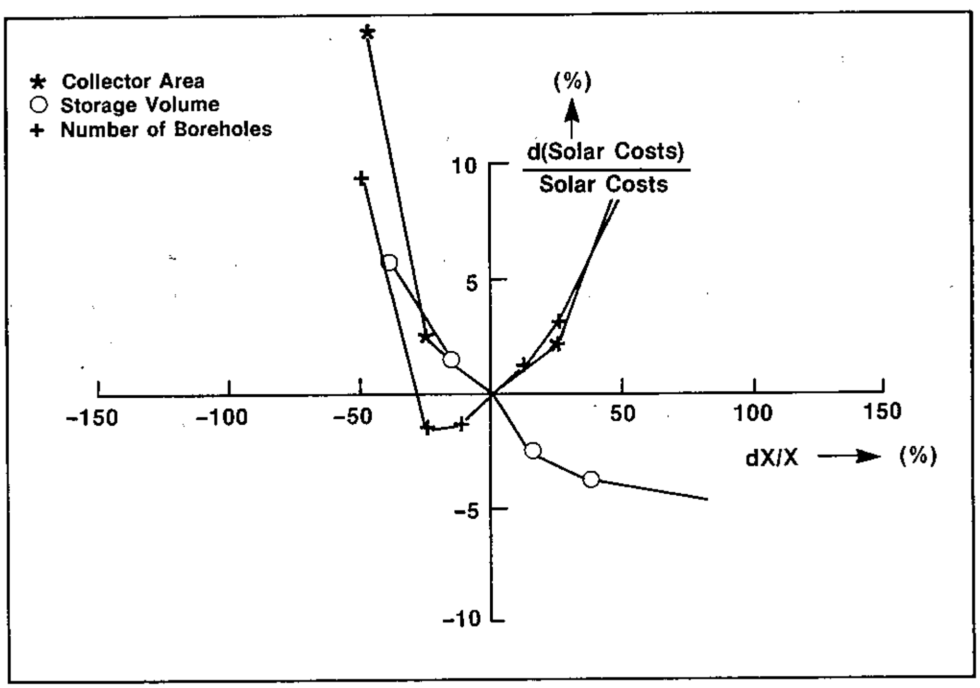

Figure 4-10. Variation of Solar Costs With Some System Parameters

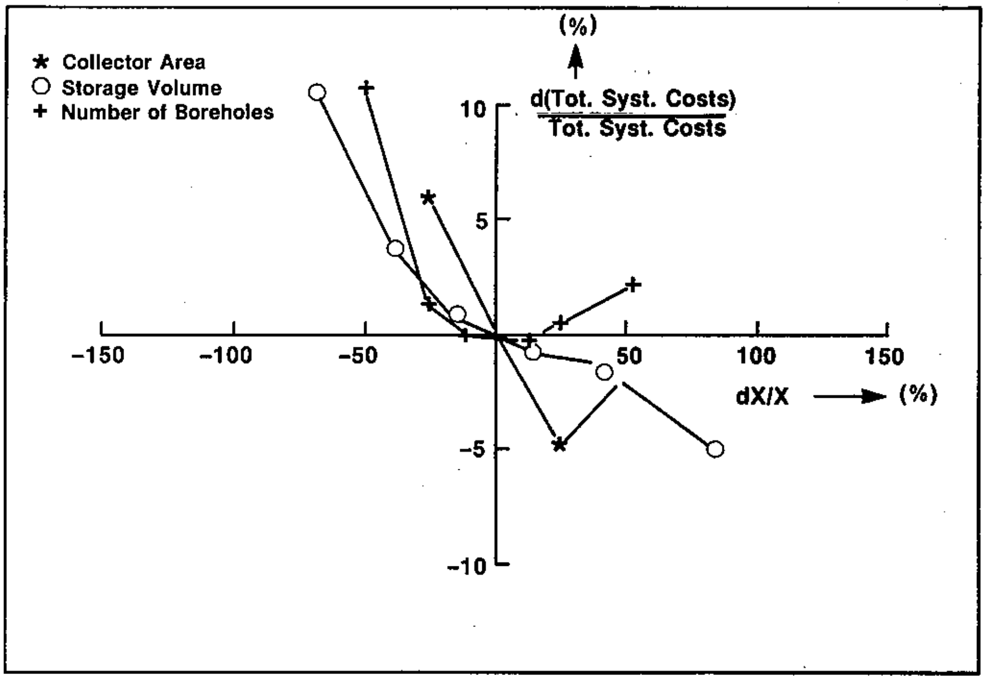

Figure 4-11. Variation of Total System Costs With Some System Parameters

4.2.5 Conclusions

The optimization technique used in Task VII seems to work reasonably well. The spider-diagrams confirm the results obtained.

As expected, the solar costs in Ispra are lower than in Copenhagen (about 14%), which agrees with the fact that the yearly global irradiation in Ispra is about 17% higher.

Within certain limits, the solar costs are rather insensitive to the solar fraction.

4.3 GERMANY*

4.3.1 Introduction

In Phase I of Task VII the project design for Wolfsburg-Glockenberg {1}** was used as the German contribution to Subtask I(e). The design calculations were based on a load of 23 single-family houses for a solar system with tank storage and evacuated tube collectors. In a parallel effort, a pit storage of 10,000 m^3 was designed in two different versions, and the competitive procurement procedure worked out in order to obtain realistic cost data {2}.

As a continuation of these efforts, a comparison was made of the energy and economic characteristics of two CSHPSS versions based on different pit storage concepts:

- o A well-insulated pit supplied by an array of high-efficiency evacuated tube collectors. The pit delivers solar energy directly to the load.

- o A "low quality level" pit without extensive insulation at the bottom or side walls supplied by a field of low-efficiency unglazed solar collectors. The pit serves as the heat source for a heat pump between the storage and the load.

This comparison is based on German weather and economic conditions. Unfortunately, the comparison cannot be related to any German projects which have been much smaller in size or have not fulfilled all the criteria of a CSHPSS.

Prices for the components as well as for the storage as a whole have been taken from the work mentioned above {2} or our experience from similar projects (See {3} for heat pump prices and {5} for prices of so-called steel segment storages). Other values are identical with those from the reference case of the II(b) work {4}.

4.3.2 Conditions and Systems

Table 4-4 presents the FRG-specific parameter values for a MINSUN run compared to the reference case of the water team work in Phase II(b). Only those parameters that are different from the reference case are enumerated {4}. Table 4-5 shows the UMSORT parameters for the two collector types. Weather data were the same as in Reference 1. The geographic and climatic conditions are shown in Figures 4-12 through 4-14 {1}.

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*Material for this section was provided by Detlef Krischel, Interatom GmbH, Friedrich-Ebert-Str., 5060 Bergisch-Gladbach. Please contact Mr. Krischel directly.

**Numbers in {} refer to references listed at the end of this section.

Table 4-4. MINSUN INPUT PARAMETERS FOR SST MODEL

Parameter	Reference Case	FRG Evaluation
Collector		
Minimum flow	0.005	0.003
Maximum flow	0.1	0.016
Storage		
Number of segments without HP	3	5
Insulation thickness		
o with HP	0.2 0.2 0.2 0.2	0.56 0.33 0.16
o without HP	variable	0.56 0.33 0.16
Environment option	variable	2
Houseload		
k-value	0.707 (Copenhagen)	0.7 (Wolfsburg)
LT	50 0.5 0 30	50 0.5 0 30
Heat Pump		
IPAR	1 or 2	1
Costs (US$)		
Asymptotic storage	20	45
Specific for small storage	30	110
Volume of small storage	2000	20000
Insulation	100	90
Condenser	0.2	0.6
Evaporator	0.2	0.6
Electric motor installed	0.2	0.6
Depreciation time	20	15
Unglazed collector	140	93
Evacuated collectors	350	228

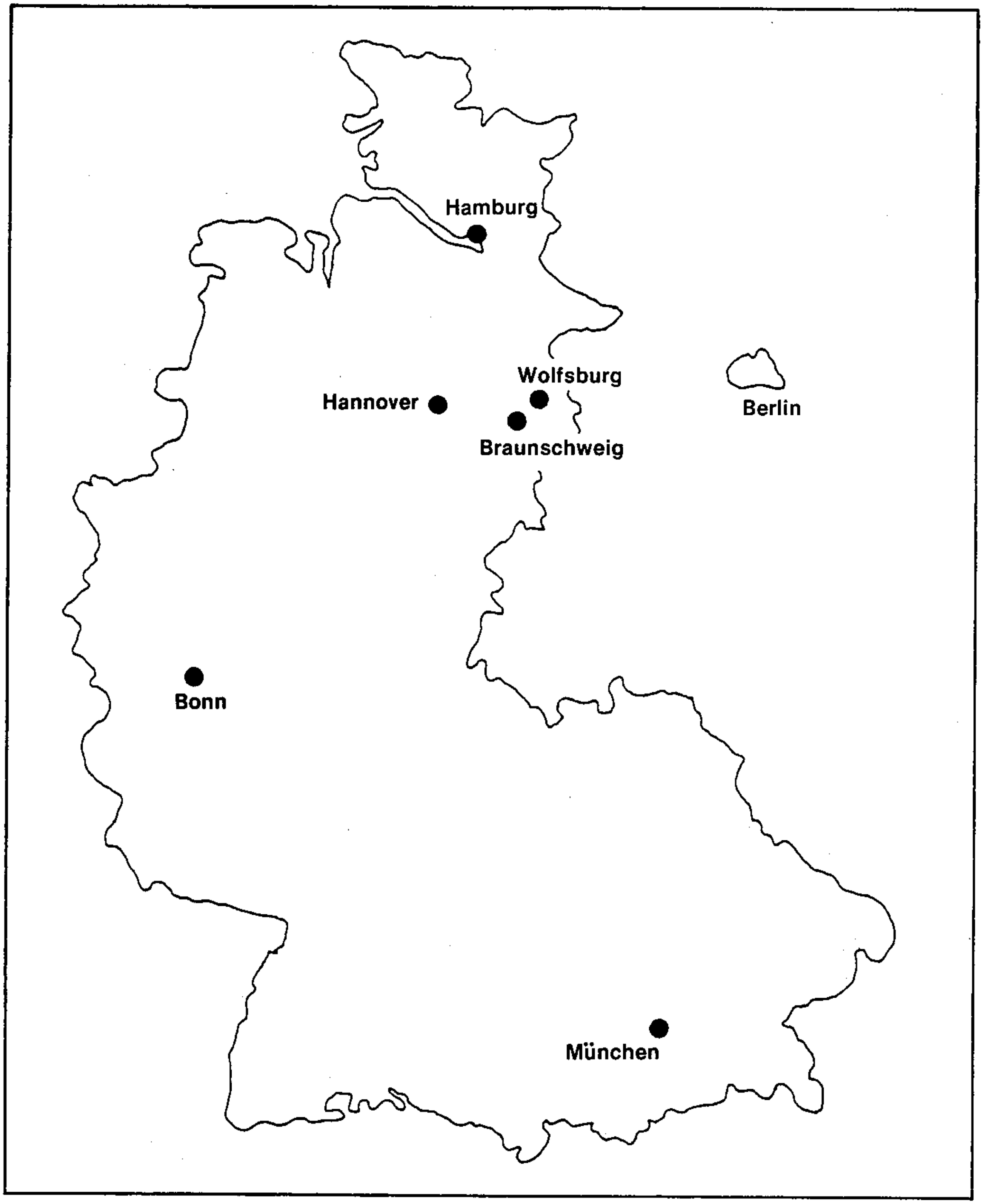

Figure 4-12. The Geographical Position of Wolfsburg

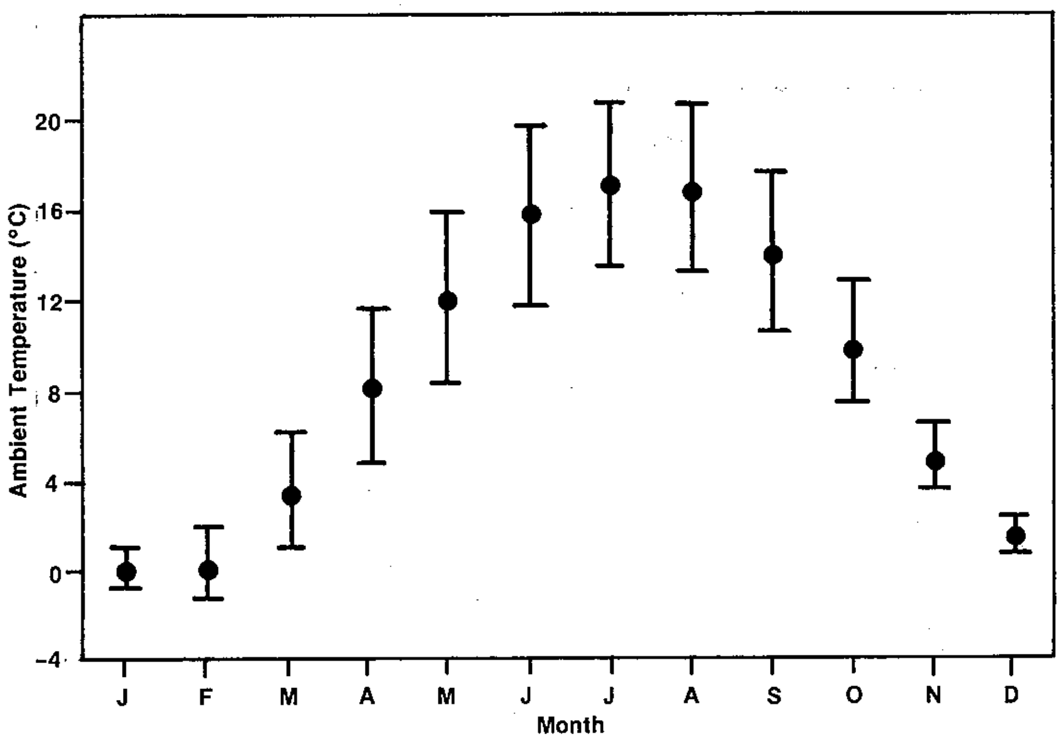

Figure 4-13. Monthly Average Temperatures During the Day and Scatterband of the Hourly Average Temperature Over the Month (1951-70)

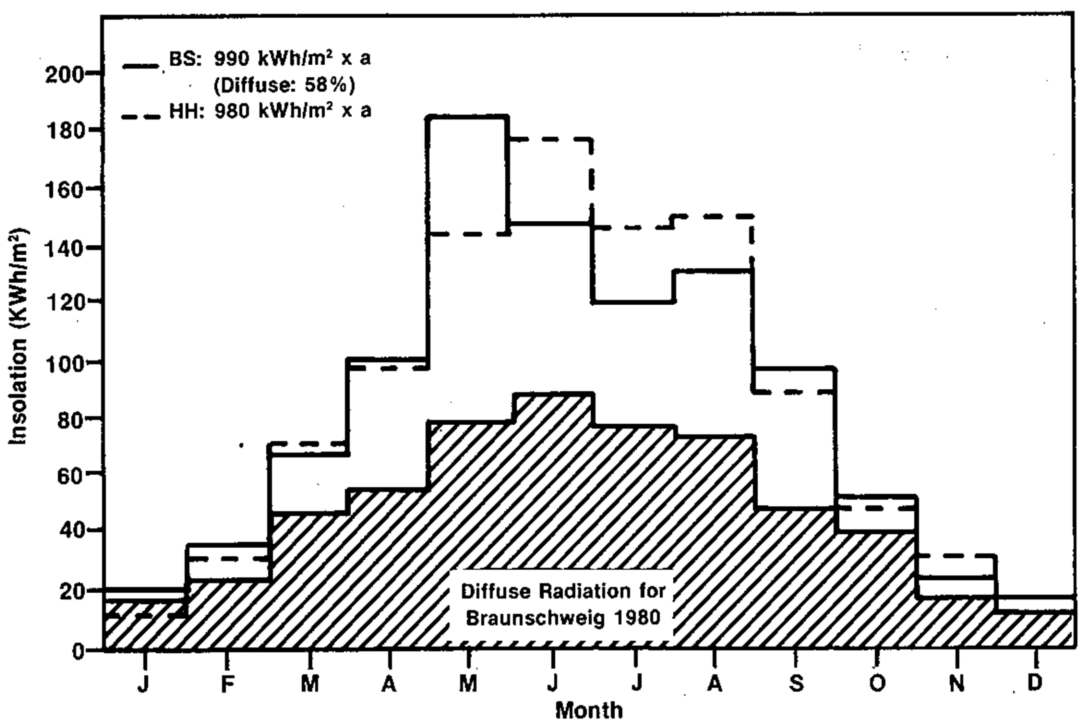

Figure 4-14. Monthly Radiation on a Horizontal Surface for Braunschweig (BS) 1980 and Hamburg (HH) 1973. The Diffuse Radiation for Braunschweig 1980 is Additionally Marked

Table 4-5. UMSORT PARAMETERS

UMSORT Parameters	EC	UG
Array factor	0.7	0.7
Incidence Angle		
Modifier	0.05	0.1
Inclination	45	45
Ground Reflectance	0	0
Linear Unit Loss Coefficient (W/m^2K)	1	15
Eta zero	0.7	0.808
BOES	0	0

No new competitive procurement procedure was used to obtain firm prices. Instead the lowest, most optimistic prices were taken or scaled down from current prices for smaller systems or component production volumes.

Two storage systems prices were derived (see Table 4-6):

- o A basic tank storage price of 200 DM/m^3 for the steel segment tanks built above ground, taken from {5}. The price for highest volumes was taken from the Wolfsburg price scale {2}.
- o A pit storage price consisting of the steel tank price plus ground excavation costs.

Insulation material prices can be reduced tremendously for the "steel tank" pit-version compared with Wolfsburg [1] because the material must not be pressure-proof against water loads but just protected against ground water. Also, compared with 1983, prices for insulation materials have dropped tremendously.

Table 4-6. STORAGE COSTS

Type	Size of Small Storage $10^3 m^3$	Specific Cost for Small Storage DM/m^3	Asymptotic Storage DM/m^3
Basic Steel Segment Tank	10000	200	90
"Pit," In-Ground Tank	20000	220	90
Insulation for Above-Ground	-	120	120
Insulation for In-Ground	-	180	180

The steel segment tank concept offers the chance to design different storage types based on the same (tank) structure:

- o an above-ground, well-insulated <u>tank</u>
- o an in-ground storage in thermal contact to the ground
  - via thermal insulation material <u>insulated pit</u>
  - directly, just protected against corrosion caused by ground water (<u>"low quality" pit)</u>.

4.3.3 Methodology

The first storage version was simulated with the tank model in MINSUN. The other concepts (which may be called pit-versions) were investigated with the stratified storage (SST) model, which divides the storage volume into 10 thermal layers and includes the dynamic heat exchange between storage and surrounding layers.

4.3.4 Results

4.3.4.1 Evacuated Tube Collector (EC), No Heat Pump (NOHP)

An overview of the most promising systems with an evacuated tube collector array and without a heat pump is given in Figure 4-15 for the above-ground tank version, showing total solar costs (TSC) depending on solar fraction. The characteristic of the curves is the same for all storage volumes except that approaching solar fraction f = 1 results in a steep increase of costs. For f = 0.7 no system could be found which offers unit solar costs lower than 312 DM/MWh.

Total solar coverage would be possible with TSC = 350 DM/MWh.

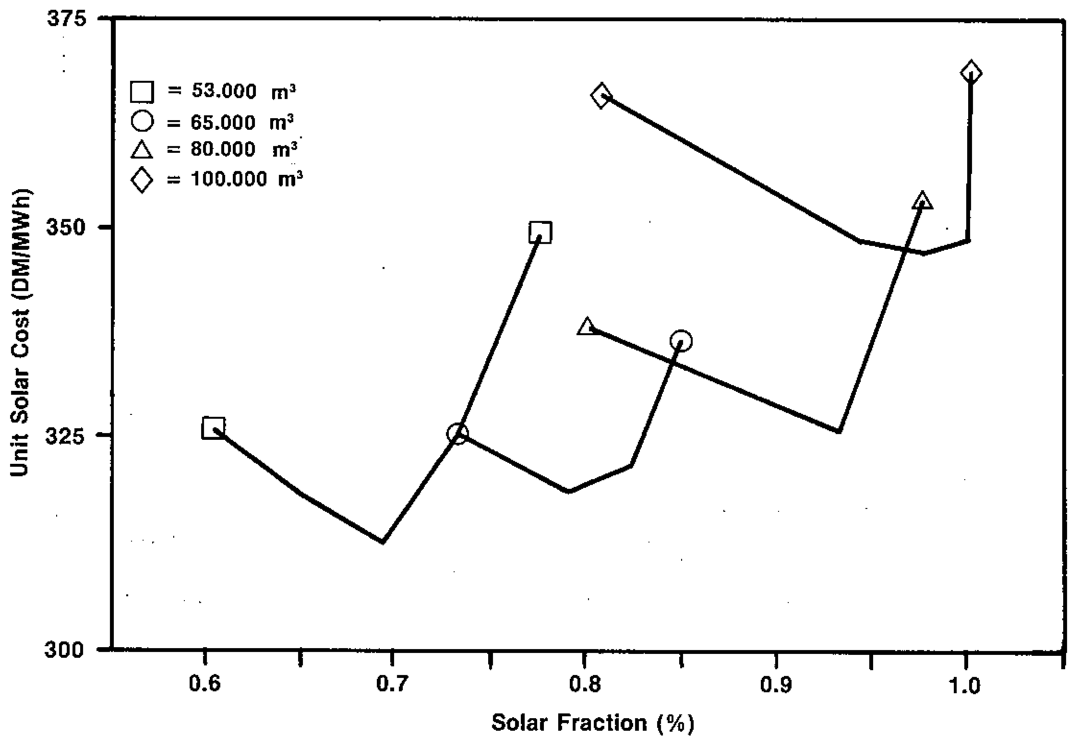

Figure 4-15. Solar Cost vs Solar Fraction - Germany. Evacuated Tube Collector (EC), No Heat Pump (NOHP), Above Ground Tank Storage, Completely Insulated

In Figure 4-16 the envelopes of least unit total solar cost for different storage volumes are compared for the well-insulated, above-ground tank storage and also the well-insulated in-ground "pit" version, calculated with the SST-model. For all storage volumes the minima are found at the same collector areas for both tank and "pit" models, except for 100000. With identical collector areas the in-ground storage systems offer slightly higher solar fractions (2-3 percent).

The resulting least cost curve of the "pit" version, however, is shifted to about 10 DM/MWh higher than the above-ground tank version, corresponding to about a 3 percent shift. Both least cost envelopes show a gradient d(TSC)/df = 60 DM/MWh for $f < 0.92$. For higher solar fractions the gradients are d(TSC)/df = 1430 DM/MWh for the "pit" and d(TSC)/df = 680 DM/MWh for the tank, respectively.

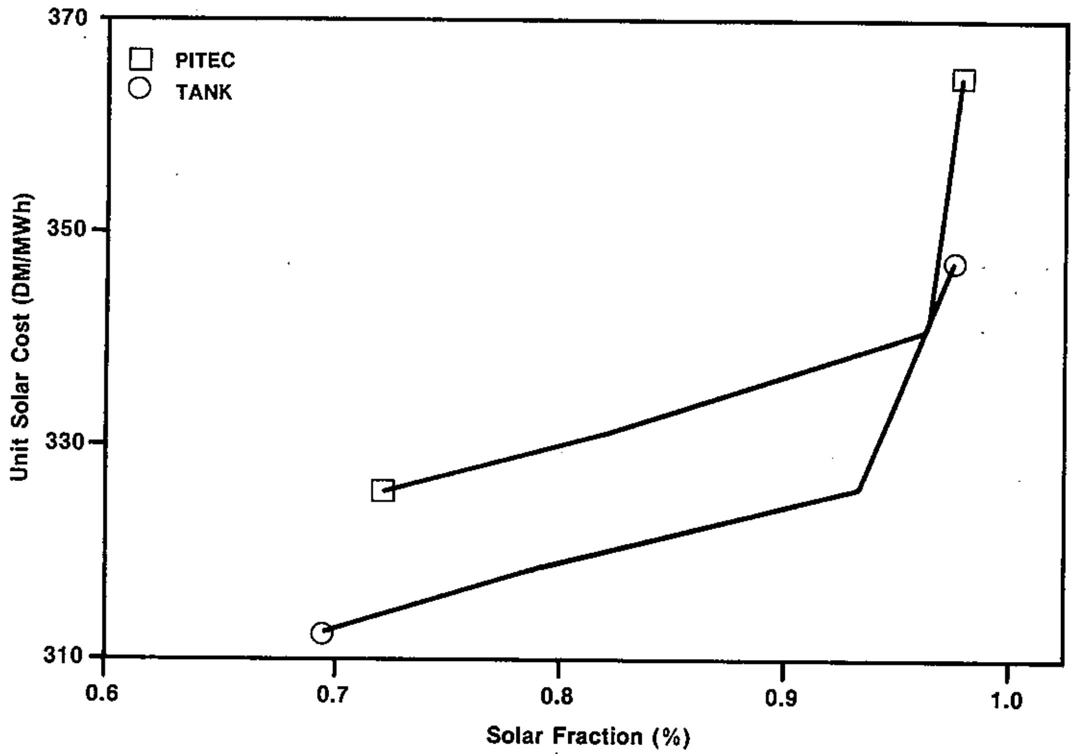

Figure 4-16. Envelope of Cost Minima for Different Collector Areas and Storage Volumes. Partially Insulated In-Ground Pit Compared With Completely Insulated Above Ground Tank With Evacuated Tube Collector and No Heat Pump

4.3.4.2 Unglazed Collectors (UG) With Heat Pump (HP)

The two pit versions supplied by unglazed collectors are distinguished according to the non-insulated interfaces with the ground as

UG, HP, PIT_{b+w} with b = bottom, w = wall and
UG, HP, PIT_b with b = bottom.

Figure 4-17 presents the results for PIT $_{b+w}$ systems. For volumes less than $10^5 m^3$ the dependence on collector area is nearly identical for all storage volumes compared. This results in a relatively broad minimum in total solar cost.

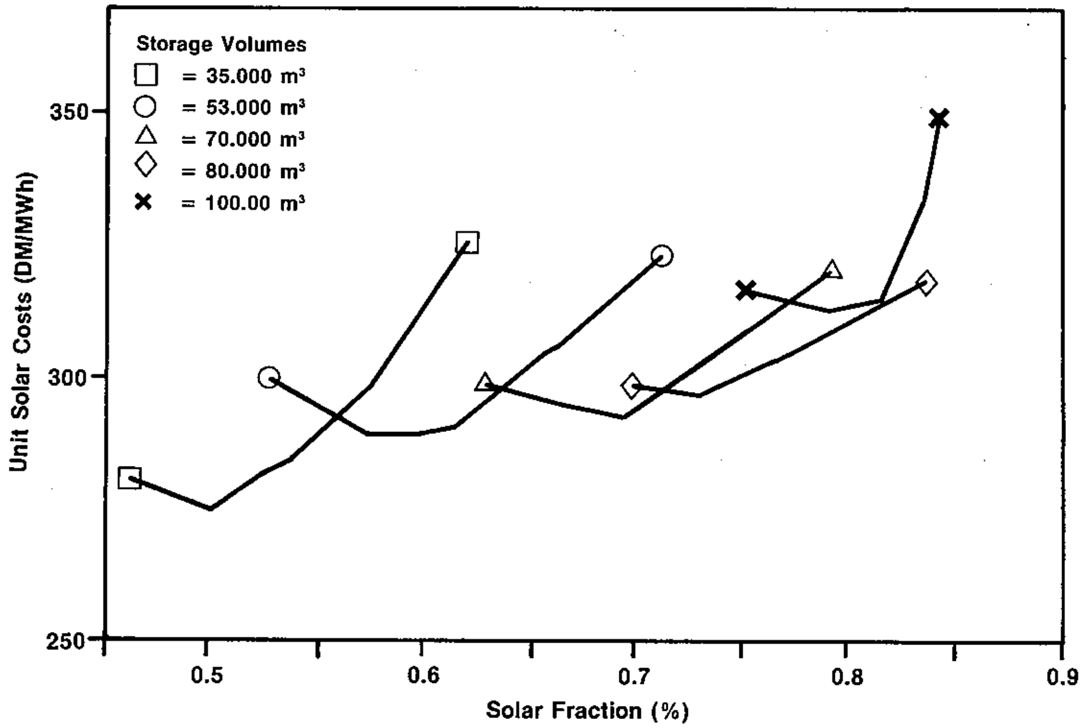

Figure 4-17. Solar Cost vs Solar Fraction - Germany. Partially Insulated (Top Only) Pit With Unglazed Collectors and Heat Pumps

In Figure 4-18 the envelopes of minima for PIT_{b+w} systems are compared with those of PIT_b systems. For solar fractions between about $0.58 < f < 0.73$ the curves have a kind of plateau with only minor cost increase whereas outside of the solar fraction range higher gradients of d(TSC)/df can be found. Whereas for $f < 0.73$ the curves are nearly parallel, there is a crossing of both curves at about $f = 0.76$. For very large systems with $V > 90{,}000\ m^3$ the PIT_b version seems to become economically more favorable.

Again as for the EC/NOHP systems the minima in TSC appear for a given volume at the same collector area independent of where the insulation is placed. The solar fraction for which d(TSC)/df exceeds TSC absolute [d(TSC)/df])/TSC > 1, is reached for PIT_{b+w} at $f = 0.76$ and for PIT_b at $f = 0.80$.

4.3.4.3 Comparison of EC/NOHP with UG/HP Systems

Figure 4-19 compares the minima in total solar costs of the two system types, EC with well-insulated storage and UG with HP and less insulated storage. The HP systems appear to show cost advantages over the solar fraction range where both results (UG/HP and EC/NOHP) overlap. The differences, however, do not exceed TSC = 25 DM/MWh corresponding to about 8 percent at $f = 0.50$ and become even smaller for higher solar fractions.

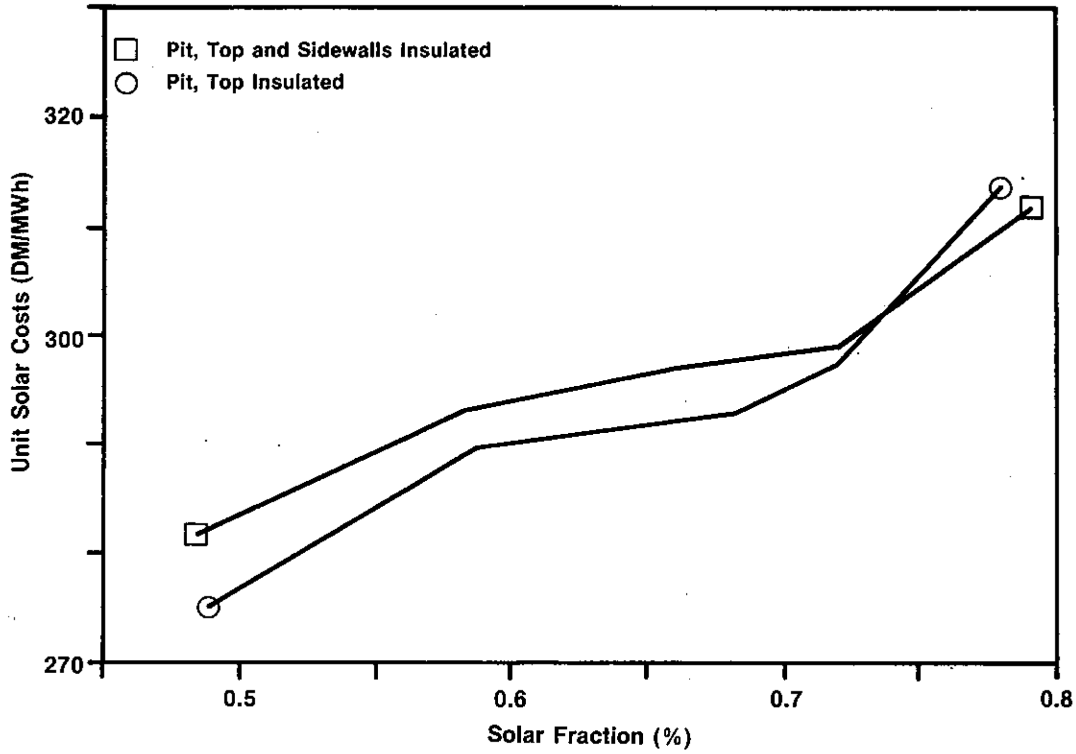

Figure 4-18. Envelope of Cost Minima for Different Collector Areas and Storage Volumes. Partially Insulated on Top or Top and Side Walls

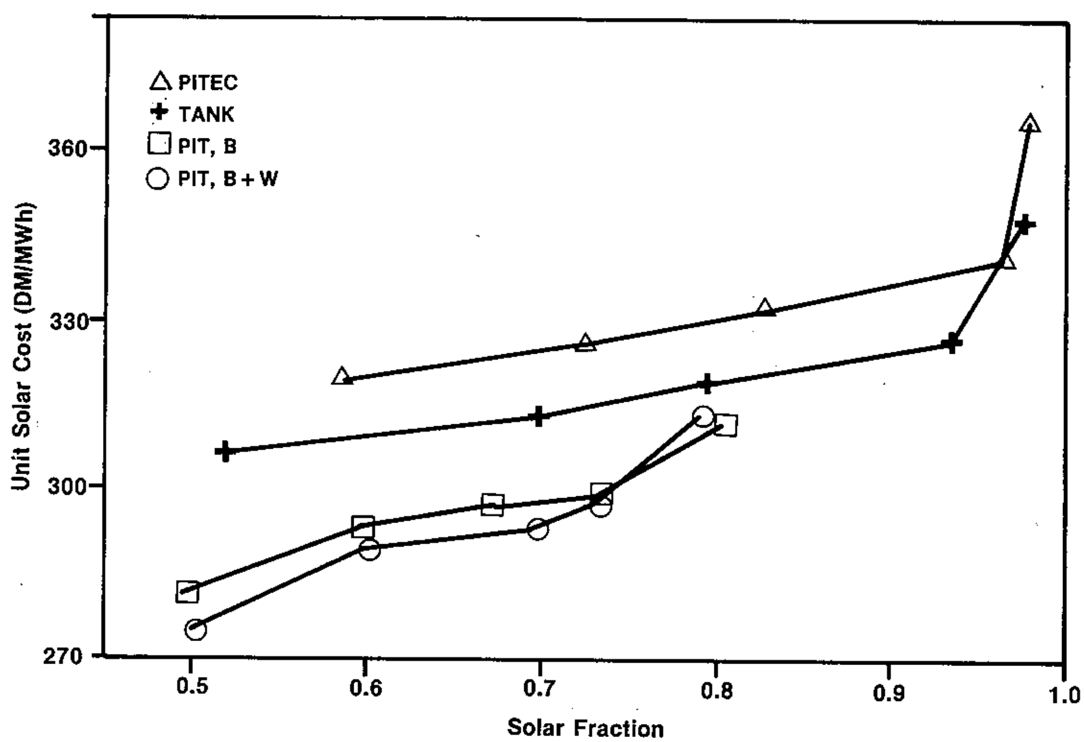

Figure 4-19. Envelopes of Cost Minima of All Systems Investigated PITEC: EC, NOHP Complete Insulation, In-Ground; TANK: EC, NOHP, Complete Insulation, Above-Ground; PIT, B: UG, HP, B = Bottom Not Insulated; PIT, B+W: UG, HP, B+W = Bottom and Side Walls Not Insulated

4.3.5 Discussion

From the absolute cost figures it is evident that none of the simulated systems is competitive with current German costs for conventional primary energy, such as oil or gas, which are around 80 DM/MWh.

Looking at the gradients d(TSC)/df one realizes that, over wide ranges of solar fraction, the extension of a system to higher storage volumes and/or larger collector areas seems reasonable when related to absolute total solar costs. For the EC/NOHP systems [d(TSC)/df 60 DM/MWh] this is true even compared to conventional energy costs. After having accepted that a solar system is not cost-effective because of high basic investment costs, it seems reasonable to choose the system as large as possible up to solar fractions of about $f = 0.92$ to 0.95.

For UG/HP systems the same considerations are true for the plateau in solar costs within the solar fraction range from $f = 0.58$ to $f = 0.7$.

The relative comparison of EC/NOHP systems with completely insulated tanks shows that the marginal thermal advantages of in-ground storage are not sufficient to cover the additional costs of excavation, ground water protection, etc., compared to the above-ground tank version.

For the pit versions combined with UG/HP it seems of minor importance whether the insulation is omitted at the bottom or at the bottom <u>and</u> side walls; only for very large systems does it seem disadvantageous to use the surrounding ground as an additional storage volume. Attention should be drawn to the fact that over a wide range of storage volumes, $V > 70 \cdot 10^3 m^3$, a collector area of $23 \cdot 10^3 m^2$ seems to be an economic limit.

Taking the minima of all system types together, a recommendation would be to take a UG/HP system with $A = 23000\ m^2$ and $V = 70000\ m^3$ pit storage with insulation only on top for solar fractions $f < 0.7$ and an EC/NOHP, or an above- ground fully insulated tank system with $A = 28000\ m^2$ and $V = 80000\ m^3$ if one wishes to reach higher solar fractions.

4.3.6 Conclusion

For German weather and economic conditions a "low quality" pit, insulated only at the top cover and combined with an unglazed collector and a heat pump, respectively, proves to be the most cost-effective of the simulated CSHPSS concepts for $f < 0.7$. None of the systems investigated is cost competitive compared with actual German fuel prices; solar costs for CSHPSS systems should decrease at least by a factor of 3.6 for UG/HP systems and 4.0 for EC/NOHP systems. If one accepts the economics of CSHPSS on the given level, one finds that, up to some high solar fraction, the larger the system the more reasonable its cost becomes.

4.3.7. References

1. H. Riemer, V. Lottner, F. Scholtz, "Central Solar Heating Plats with Seasonal Storage, Project: Wolfsburg-Glockenberg," Subtask I(e) Report, May 1983. Julich FRG

2. Forschungsgesellschaft Wolfsburg mbH, "Lanzeitwaermespeicher Prototyp Wolfsburg," Wolfsburg, July 1983

3. H. Lange, INTERATOM, Private Communication

4. Task VII, Central Solar Heating Plants with Seasonal Storage, Minutes of Subtask II(b) Meeting, Studsvik, February 1984

5. H. Damjakob, Untersuchung der Machbarkeit eines Grobwaermespeichers in Form eines Schalen oder Segmentspeichers, Thermische Energiespeicherung BMFT Statusbericht 1983, V. Lottner, Private Communication.

4.4 THE NETHERLANDS*

4.4.1 Introduction

In this chapter the duct system as seasonal heat store is evaluated for Dutch conditions. The evaluation includes the following total system concepts:

- o Gas-driven heat pump with unglazed solar collectors
- o Electric heat pump with unglazed solar collectors
- o Evacuated tubular solar collectors without heat pump.

The evaluation is done for a total load of 4.4 TJ which represents the Groningen system: 96 solar houses with seasonal heat storage in the soil. The DHW-fraction is 16% (0.7 TJ).

It should be noted that in this study a revised definition for unit solar costs is used: the annualized system capital costs plus the heat pump operation costs are divided by the total amount of energy delivered by the solar energy system and the heat pump.

4.4.2 Dutch Conditions

Practical information (parameter values, costs) from the Groningen project are used as much as possible. The gas-driven heat pump, which has not yet been subject for study in this IEA-Subtask II(b), is important for the Netherlands because of the big difference in energy costs: electricity = 0.24 Dfl/kWh (0.069 $/kWh), gas = 0.08 Dfl/kWh (0.023 $/kWh).**

4.4.2.1 Climatic Conditions

The Dutch climate has a strong maritime tendency with a moderate character (rather low temperatures in summer and relatively high temperatures in winter, except for a few weeks in which northeastern and eastern winds for the continent and polar region may occur).

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*Material for this section was provided by A.J.Th.M. Wijsman of the Institute of Applied Physics, Technisch Physische Dienst, TNO/TH, P. O. Box 155, 2600 Ad Delft, The Netherlands. Please contact Mr. Wijsman directly for additional details.

**Conversion to U.S. dollars is arbitrarily made at a rate of 3.5 Dfl/$.

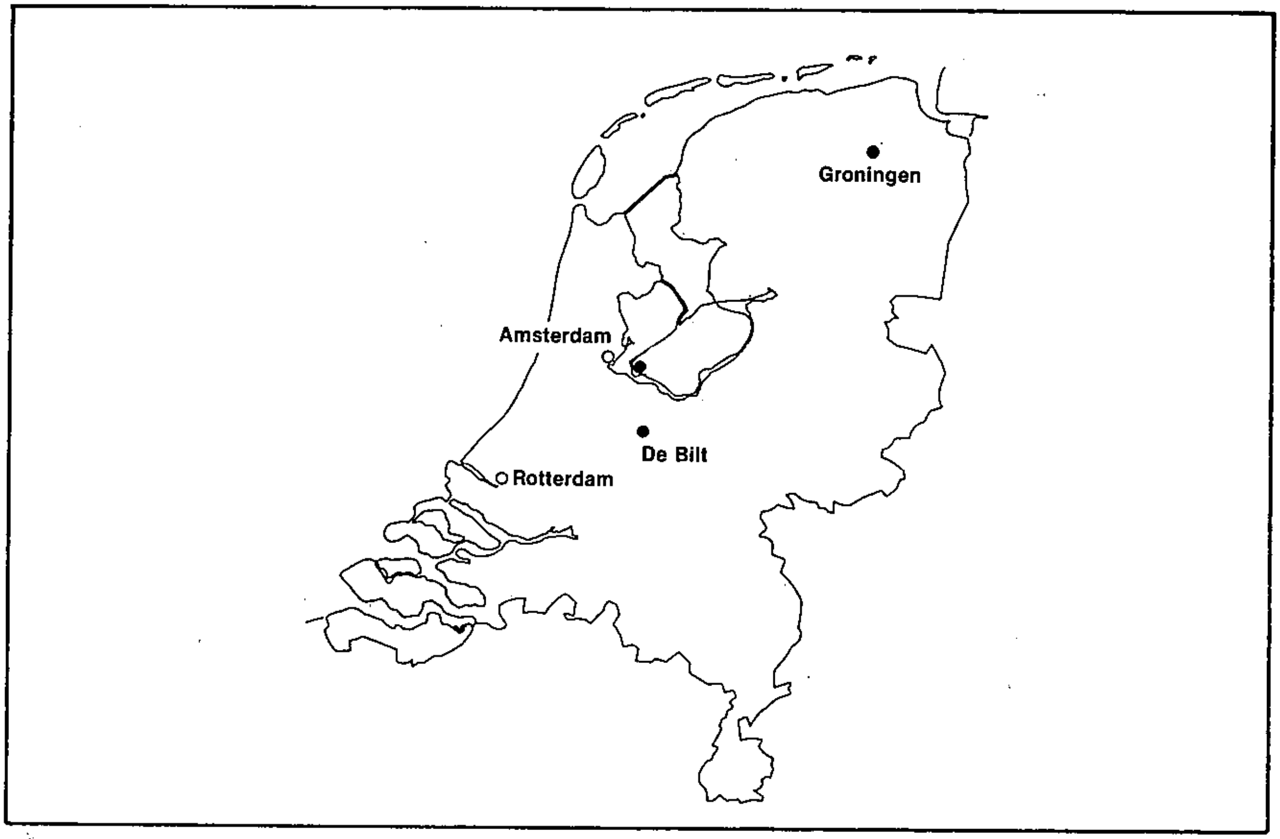

Figure 4-20. Locations of De Bilt and Groningen

Table 4-7. ANNUAL AVERAGES FOR DE BILT (LATITUDE 52° NORTH)

Global irradiation on horizontal plane	3510 MJ/m^2 (975 kWh/m^2)
Diffuse proportion	62%
Sunshine	1400 hours per year
Ambient temperature (Annual)	8.4° C
Average temperature (July/August)	17.0°C
Average temperature (January)	+1.7°C
Degree days (°C) (Base temperature 18°C)	3530 degree days
Prevailing wind direction	SW
Average wind speed	3.3 m/s
Total Precipitation	723 mm
Relative humidity	86%

4.4.2.2 System Data

In the Groningen system the solar collectors are mounted onto the roofs of the houses and coupled to the seasonal heat store by a distribution network. Heat from the seasonal heat store is used for space heating and for domestic water heating. To reach the required supply temperature (with or without heat pump) an auxiliary gas boiler can be used.

Subsystem Data used in this study are as follows:

- <u>Solar collectors</u>

  Unglazed solar collectors (in combination with a heat pump)
  - Optical efficiency $\eta_o = 0.90$
  - Heat loss factor $U_L = 15.0 + 1.6^{*} = 16.6$ W/m^{2}K
  - Array factor AF = 0.83 **

  Evacuated solar collectors (in system without heat pump)
  - Optical efficiency $\eta_o = 0.66$
  - Heat loss factor $U_L = 1.2 + 0.3^{*} = 1.5$ W/m^{2}K
  - Array factor AF = 0.83

- <u>The seasonal heat store</u>

  The seasonal heat store consists of a layer of soil with a heat exchanger in it. Only the top of the store is insulated: 5 cm in a low-temperature system and 20 cm in a high-temperature system. The heat exchanger consists of plastic tubes which are inserted to a depth of 20 m. The soil type is water saturated sand with a heat capacity of 2.72 MJ/m^{3}K and a heat conductivity of is 1.9 W/m K.

- <u>Heat pump and auxiliary heater</u>

  The auxiliary heater is a gas-fired boiler with a total capacity of 600 kW so that, if necessary, the total heating power at design conditions (-10^{o}C) can be delivered. The heat pump can be an electric heat pump or a gas-driven heat pump. The maximum heat to be delivered by either heat pump is fixed at 600 kW. The two options are shown schematically below.

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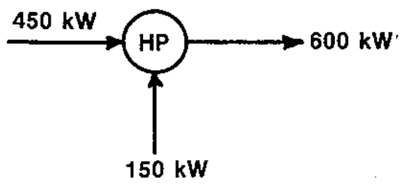
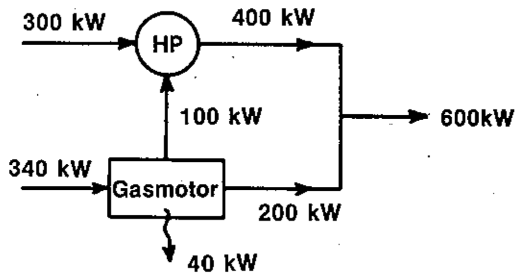
*Increase of U_L for the piping losses in the array (insulated or not).

**Correction for the piping losses in the array are excluded.

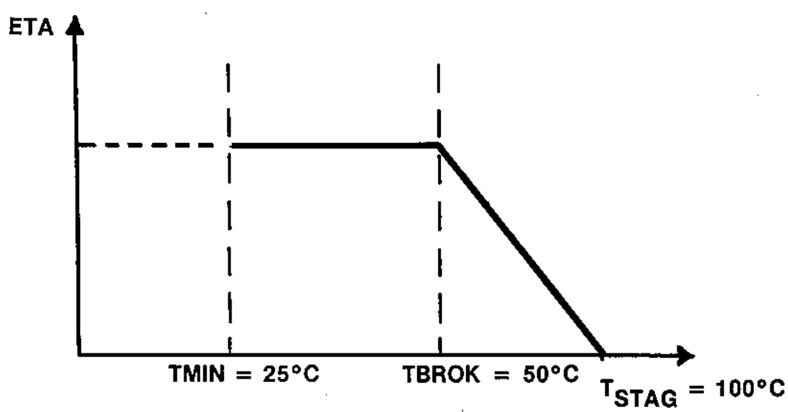


Electrical Heat Pump
COP = 600/150 = 4.0



Gas-Driven Heat Pump
COP = 600/340 = 1.8

The heat pump model uses the efficiency curve depicted in the sketch.



Heat Pump Efficiency Curve

Other specifications for the heat pump are:

o Electric heat pump: evaporator power = 65.0 kW/K
condensor power = 65.0 kW/K
minimum COP = 2.7 --

o Gas-driven heat pump: evaporator power = 47.0 kW/K
condensor power = 47.0 kW/K
minimum COP = 0.9 --

The efficiency of the heat pump is assumed to be independent of power demand.

- Load

The maximum power required for space heating load is 6 kW and 0.23 kW for domestic hot water. For a system with heat pump, the required delivery temperature to the house is:

$$T_{delivery} = 55 + 0.5 \max((0.0 - T_{amb}), 0.0) ,$$

which gives 60° C at design conditions of $T_{amb} = -10°$ C. For a system without heat pump, the required delivery temperature to the house is:

$$T_{delivery} = 20 + 0.75 \max((20.0 - T_{amb}), 0.0) ,$$

which gives only 42.5° C at design conditions. In this system without heat pump, a separate tap water network (at 55°C) is provided.

- Distribution networks

Both networks (between solar collectors and heat store and between heat store and load) have a length of 1000 m. The nominal flow in the collector circuit is 0.010 kg/s m^2 for unglazed solar collectors and 0.006 kg/s m^2 for evacuated tubular solar collectors. The flow in the load circuit is 15 kg/s.

4.4.2.3 Cost Data

- Solar collectors
  - Unglazed solar collectors : 300 Dfl/m^2 (86 \$/m^2)
  - Evacuated tubular collectors: 800 Dfl/m^2 collector (230 \$/m^2)

- Seasonal heat store

A distinction has been made between costs for storage volume, costs for insulation, and costs for heat exchanger tubing.

  - Volume costs: $C_{vol} = 6 + 6(V/23000)^{0.1}$ Dfl/(m^3 soil)
  - Top insulation costs: $C_{ins} = 350$ Dfl/(m^3 insulation)
  - Heat exchanger costs: $C_{hex} = 15$ Dfl/(m tubing)

- Heat pump and auxiliary heater
  - The costs for the gas boiler of 600 kW is: Dfl 120,000 (\$34,000)
  - The costs for either gas or electric heat pump of 600 kW is: Dfl 250,000 (\$70,000)
  - The fuel costs are 0.08 Dfl/kW (0.023 \$/kWh) for gas and 0.24 Dfl/kWh (0.069 \$/kWh) for electricity.

- o Load
    - No costs for the heat delivery system are taken into account.

- o Distribution network between collectors and heat store
    - The cost for the distribution network: 75 Dfl/m piping and 80 Dfl/m^3 insulation material.

4.4.2.4 Economic conditions

- o Depreciation time :20 years
- o Real interest rate :4%
- o Fuel inflation rate (above normal inflation) :2%

4.4.3 Methodology

The methods used in the Subtask II(b) reference studies were used here except that the unit solar cost and solar fraction were redefined to include the heat pump operating cost and delivered energy. This modification was made in order to accommodate the great difference in gas and electricity prices. With these new definitions, the system expansion paths were determined as in the reference study.

To reduce the number of computer runs, some scaling was applied; the storage volume and the number of boreholes (heat exchanger pipes) are proportional to the collector area.

The computer calculations were carried out according to the following procedure:

- o A reasonable system (solar fraction 25-40%) was selected based on experience.

- o Collector area was kept constant while the storage volume and number of boreholes was varied and the "storage volume - number of boreholes" combination with the lowest solar costs were found.

- o The collector area was then varied while scaling storage volume and number of boreholes according to the lowest cost system under 2. Lowest solar cost system and the solar fraction belonging to that system were found.

- o This lowest solar cost system was checked by varying storage volume and number of boreholes.

The advantage of this procedure is that it requires relatively few computer runs per system concept (about 20-30).

Before performing the above-mentioned calculations, an attempt was made to improve the assumptions used in the MINSUN model.

- <u>The solar collector heat gain calculation:</u>
  - The IR-heat losses to the cold sky were taken into account for unglazed solar collectors (QIR = 40 W/m^2). Further, the switch-off criterion for pump in the collector circuit was changed ($Q_{col} < 0.0$).
  - The array factor used to correct for the thermal losses was modified to exclude the thermal losses. These losses were included by increasing the effective heat loss coefficient of the collector, U_L.

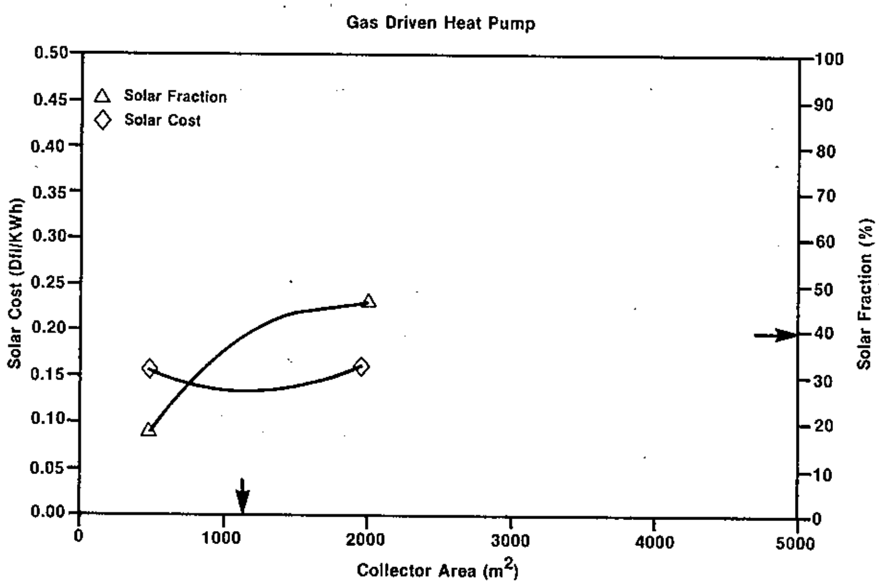
- <u>The heat pump operation:</u>
  - A new heat pump model for a gas-driven heat pump was developed. Since in practice heat pumps cannot operate if

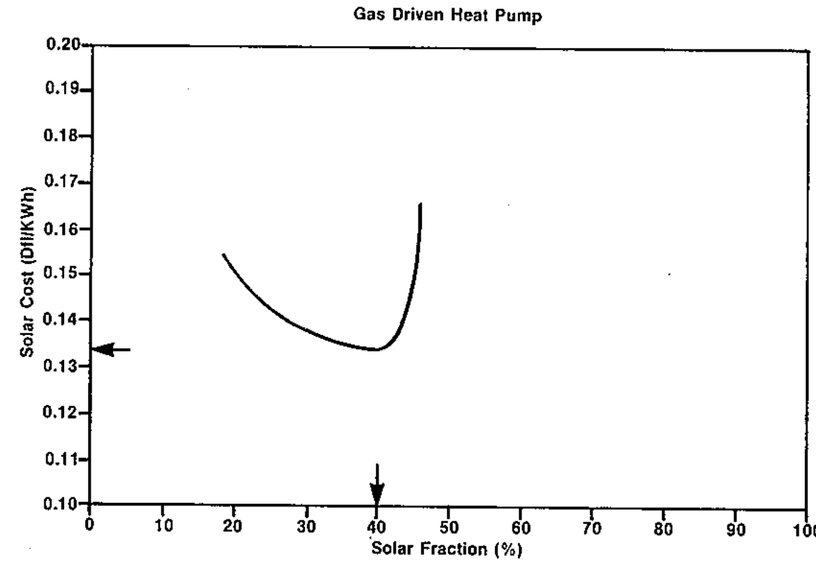
$$(TTC-TTF) < 15\text{-}25^{\circ}C$$

    both models (electric and gas-driven) were modified so that the temperature difference between condenser and evaporator (TTC-TTF) must always be higher than 25°C. With this restriction the model does not calculate COPs greater than 8.
  - The heat pump capacity was chosen to deliver the maximum heating power. This was deemed necessary because of the heat pump control in MINSUN: "if the entire load cannot be met with the heat pump, the heat pump is turned off and the auxiliary delivers the heat." A smaller heat pump capacity might limit the number of running hours too much.

4.4.4. <u>Results</u>

The results of the computer calculations with MINSUN are given in this section. The results are presented in graphs of "solar cost" and "solar fraction" versus collector area and graphs of "solar cost versus solar fraction." From these figures the lowest solar system cost is derived. For this system the heat balance in the total system and the cost breakdown are given. These results are presented for each of the three system configurations in Figures 4-21 through 4-23.





The lowest solar cost system is:

- collector area : 1150 m^2
- storage volume : 38000 m^3
- number of boreholes : 820

For this system the solar costs are 0.13 Dfl/kWh; the solar fraction of the system is 40%. The annual heat pump COP is 1.73 and the heat flows are as shown in the following sketch.

	Investment (Dfl./MWh/y)	Capital (Dfl./MWh)	Operation (Dfl./MWh)	Total (Dfl./MWh)
Solar Collectors	260	18.4	0.0	18.4
Coll. Pipework	107	7.6	0.0	7.6
Storage	569	40.3	0.0	40.3
Heat Pump	188	13.2	54.8	68.0
Total Solar System	1124	79.5	54.8	134.3

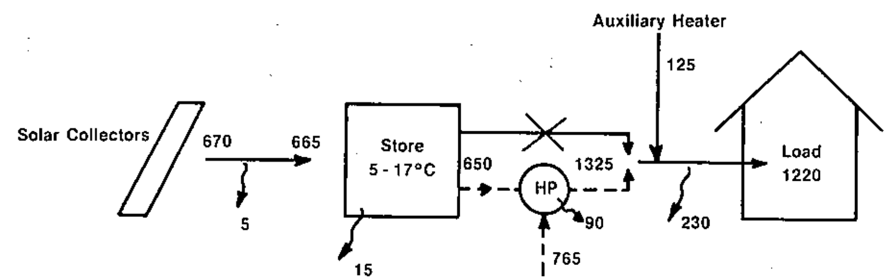
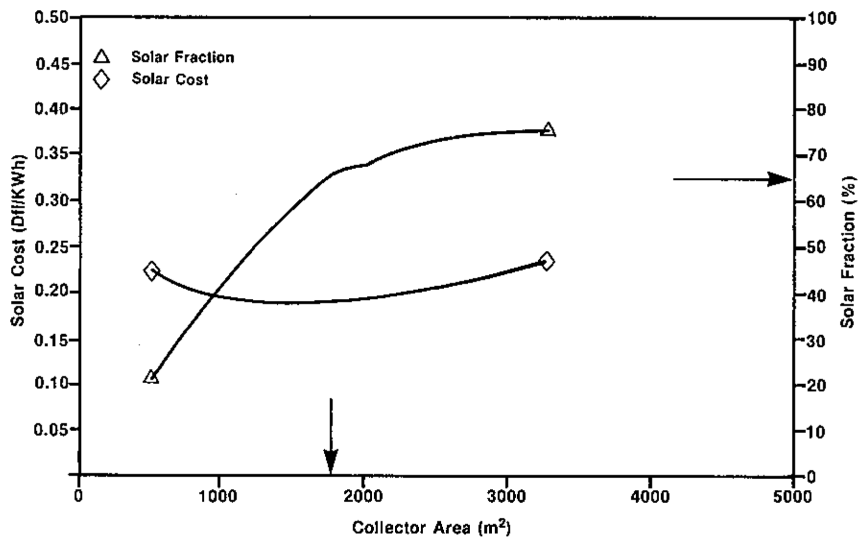
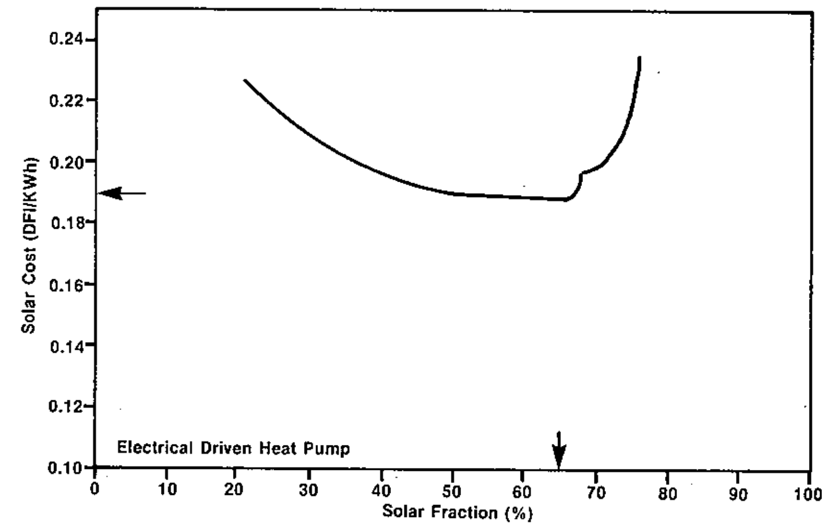


Figure 4-21. Results for Systems With Gas-Driven Heat Pumps and Unglazed Collectors





The lowest solar cost system is:

- collector area : 1750m²
- storage volume : 58000 m³
- number of boreholes : 1250

For this system the solar costs are 0.19 Dfl/kWh; the solar fraction for this system is 65%. The annual COP is 3.5

The heat flows are as shown in the following sketch.

	Investment (Dfl./MWh/y)	Capital (Dfl./MWh)	Operation (Dfl./MWh)	Total (Dfl./MWh)
Solar Collectors	393	27.8	0.0	27.8
Coll. Pipework	107	7.5	0.0	7.5
Storage	845	59.8	0.0	59.8
Heat Pump	185	13.1	81.4	94.5
Total Solar System	1530	108.2	81.4	189.6

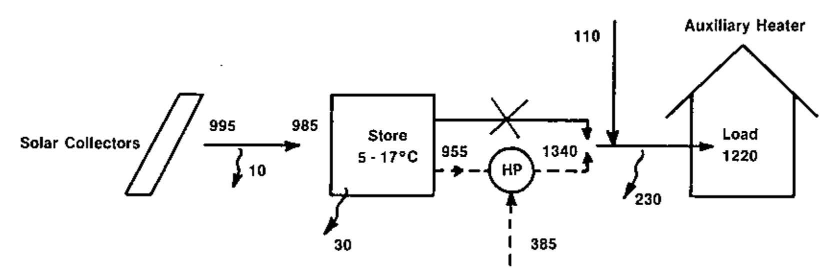
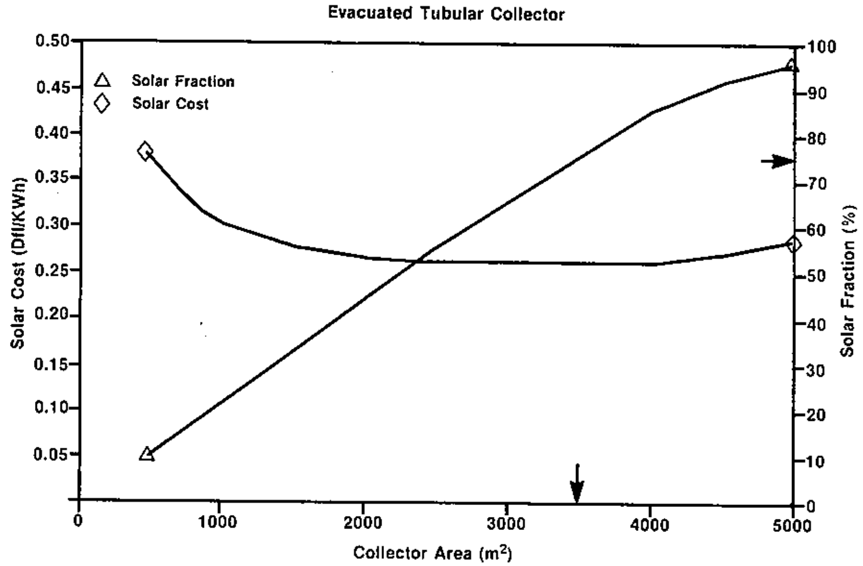
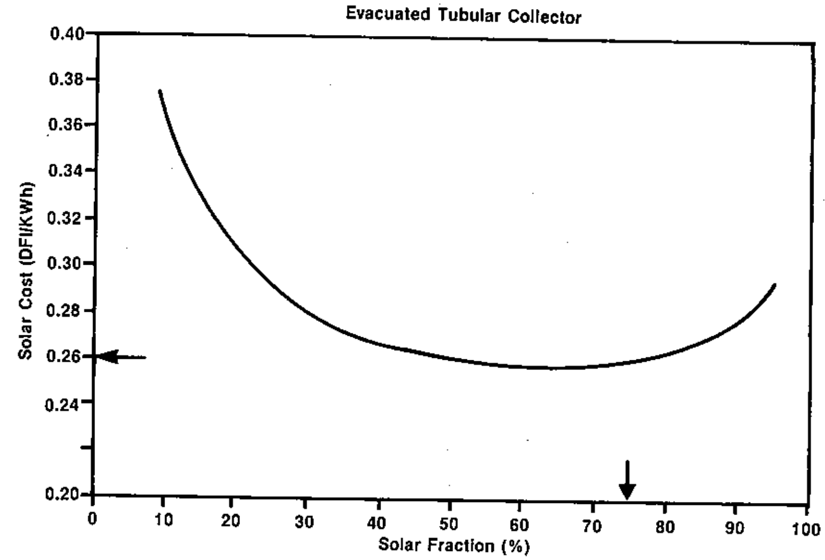


Figure 4-22. Results for Systems With Electric Heat Pumps and Unglazed Collectors

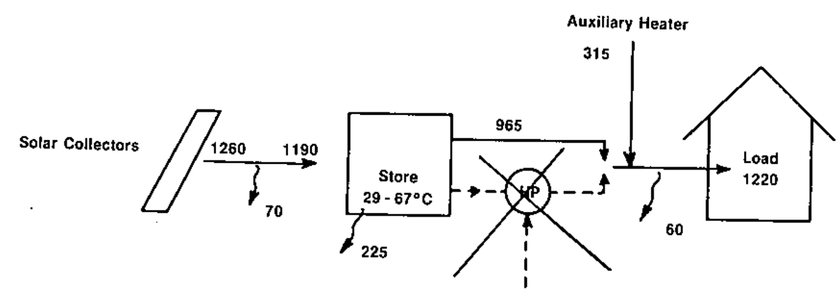




The lowest solar cost system is:
- collector area : 3500 m²
- storage volume : 16800 m³
- number of boreholes : 1050

For this system the solar costs are 0.26 Dfl/kWh; the solar fraction for this sytem is 75%.

The heat flows are as shown in the following sketch.



	Investment (Dfl./MWh/y)	Capital (Dfl./MWh)	Operation (Dfl./MWh)	Total (Dfl./MWh)
Solar Collectors	2892	204.7	0.0	204.7
Coll. Pipework	153	10.8	0.0	10.8
Storage	665	47.0	0.0	47.0
Heat Pump	0	0.0	0.0	0.0
Total Solar System	3710	262.5	0.0	262.5

Figure 4-23. Results for Systems With Evacuated Collectors and No Heat Pump

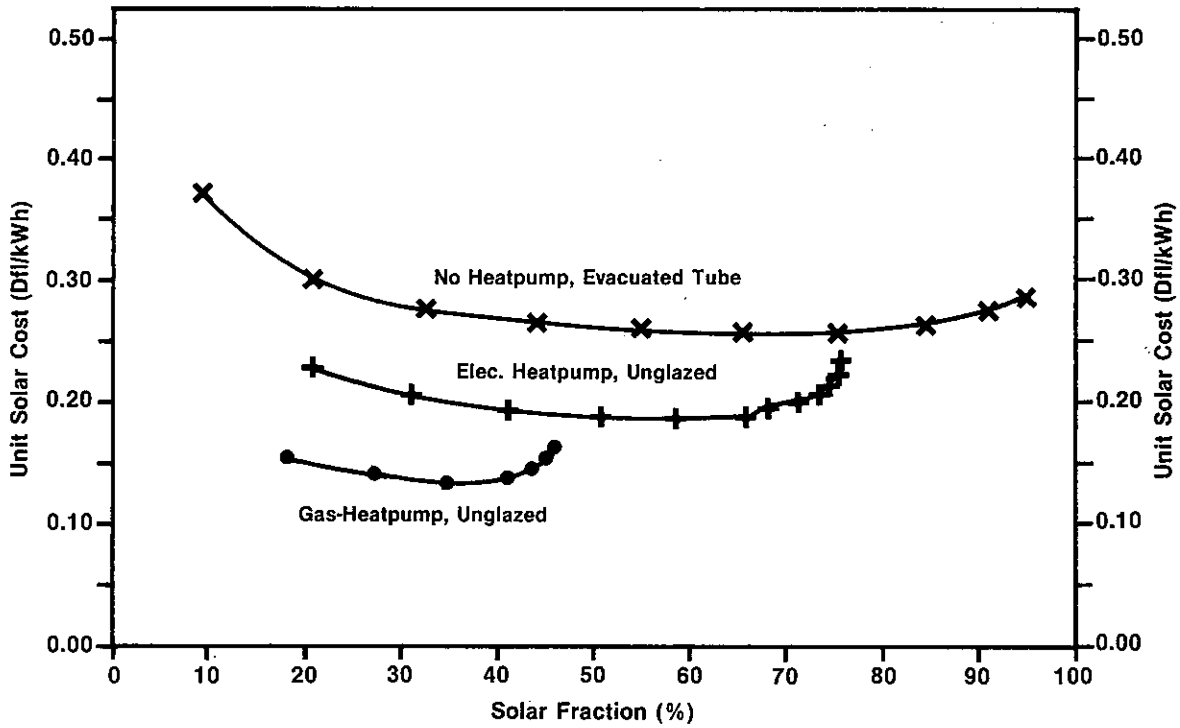


Figure 4-24. Comparison Between the Three System Concepts

Ranking of the systems:

- o Gas-driven heat pump, unglazed solar collectors for solar fractions up to 45%
- o Electric heat pump, unglazed collectors for solar fractions between 45-70%
- o No heat pump, evacuated tubular collectors for solar fractions higher than 70% .

4.4.5 Discussion

The solar costs of three system concepts have been minimized. Since these minima all result in unit solar costs greater than current gas prices, the economic optimum is not a solar system. However, it is desired eventually to displace the use of non-renewable resources of natural gas. Therefore, all three options should be considered.

The gas-driven heat pump with unglazed solar collectors has its cost minimum of 0.13 Dfl/kWh (0.037 $/KWh) at a solar fraction of about 40%. The solar unit costs for the system are 0.13 Dfl/kWh (0.037 $/kWh). In these costs there is only 0.018 Dfl/kWh (0.005 $/kWh) for solar collectors. This implies that the solar unit costs are not very sensitive to the solar collector costs; an increase from the assumed 300 Dfl/m^2 to 500 Dfl/m^2 gives a solar unit cost increase of 0.012 Dfl/kWh (0.003 $/kWh).

The electric heat pump with unglazed solar collectors has its minimum cost of 0.19 Dfl/kWh (0.05 $/kWh) at a solar fraction of 65%. Again the total solar unit costs are not very sensitive to solar collector costs; an increase from 300 Dfl/m^2 to 500 Dfl/m^2 gives a solar unit cost increase of 0.018 Dfl/kWh. It should be noted that the fossil energy savings for the optimum system are only 13% (in the Netherlands electricity is generated by gas turbines with an efficiency of 33%). Therefore, the electric heat pump option is not very attractive from the point of view of energy saving.

The system with evacuated tubular solar collectors and no heat pump has its minimum unit solar cost of 0.26 Dfl/kWh (0.07 $/kWh) at a solar fraction of about 75%. In these costs there are 0.20 Dfl/kWh for solar collectors. This means that the solar unit costs are highly sensitive to solar collector costs; a decrease from the assumed 800 Dfl/m^2 to a future cost of 500 Dfl/m^2 reduces the solar unit costs to 0.19 Dfl/kWh (the same as for the system with electric heat pump). The same solar unit costs as for the system with the gas-driven heat pump can be reached provided the evacuated tubular collectors cost the same as unglazed solar collectors (300 Dfl/m^2).

The storage system contribution to the solar unit costs is about the same for all systems--0.04 - 0.06 Dfl/kWh.

This study was carried out with the MINSUN model with its many limitations. Although an attempt was made to improve some of the assumptions used in the MINSUN model, it should be noted that the results are to be regarded as rough figures. The results can be used to indicate trends.

The component performance data are well-known for the solar collectors and the seasonal heat store and less well-known for heat pumps. The component cost data are estimates for solar collectors and seasonal heat store based on the actual Groningen system costs; however, for heat pumps the cost data are very rough. Better cost data should be obtained from a tendering of the three optimum system concepts.

4.4.6 Conclusions

Three system concepts with a duct system as the seasonal heat store were investigated. Under Dutch conditions the system consisting of a gas-driven heat pump and unglazed solar collectors gives the best results; a gas energy saving of 40% can be obtained with unit costs of 0.13 Dfl/kWh (0.03 $/kWh). This should be compared with unit costs of 0.08 Dfl/kWh for heating with a gas-fired boiler.

The system with an electric heat pump comes second economically [solar unit costs 0.19 Dfl/kWh (0.05 \$/kWh), solar fraction 65%] but is very unattractive because of the low fossil energy saving (13%).

The system without heat pump is the least attractive economically; however, an energy saving of 75% can be reached with solar unit costs of 0.26 Dfl/kWh (0.07 \$/kWh). In this system the solar unit costs are highly sensitive to solar collector unit costs as compared to the heat pump systems. Solar collector unit costs should go down to 400-500 Dfl/m^2 to make this system concept economically attractive.

4.5 SWEDEN*

4.5.1 Introduction

The development of CSHPSS systems in Sweden since 1977 has resulted in a continuous improvement of solar collector and storage technology. Today, storage systems are considered to be accepted parts of effective district heating load management systems, and the cost effectiveness of solar collector arrays is nearly competitive with the costs of oil or household electricity.

With this background in mind, the scope of the national system study was to find optimal CSHPSS configurations based on performance and cost data reflecting the state of the art in Sweden. At the same time, this study can give valuable guidance for future projects in the concept or design phases.

The study included evaluation of three different water storage systems; caverns, pits, and tanks. Since many solar installations are planned for South Sweden, the well-established weather tape for Copenhagen was used.

4.5.2 Conditions and Systems

To facilitate a comparison with the reference cases, most of the system parameters from Appendix A were retained. Deviations reflecting Swedish experiences or cost levels are summarized in Table 4-8. Only LTDS were considered. For systems with heat pumps, conventional unglazed collectors were used. For systems without a heat pump, the recently developed Swedish high-efficiency, flat-plate collectors were assumed. The collector costs are representative of array costs from recently installed systems. Storage costs are based either on experience (tank, caverns) or design projects (pits).

4.5.3 Methodology

The optimization procedure was similar to that of the water team reference case studies. The six different base systems (cavern, pit, and tank storage with and without heat pumps respectively) were optimized by finding minimum solar cost points for different volume/storage combinations. Sensitivity studies were used to determine the influence of collector, heat pump, and storage costs. The total system unit energy costs were determined as a function of auxiliary energy cost. Tank systems with heat pumps were calculated with both the tank model and the stratified temperature storage model.

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*[1] Material for this section was provided by Heimo Zinko, Studsvik Energiteknik AB, S-611 82 Nykoping, Sweden. Please contact Dr. Zinko directly for additional details.

Table 4-8. SWEDISH CONDITIONS AND SYSTEMS*

Climate: Copenhagen

Solar Collectors:

High-efficiency flat plat (for systems without heat pump)		Unglazed flat plate (for systems with heat pump)	
η_o	= 0.75	η_o	= 0.75
UL1	= 2.7 W/m^2K	UL1	= 18 W/m^2K
UL2	= 0	UL2	= 0
b_o	= 0.1	b_o	= 0
TILT	= 42°	TILT	= 35°
Array effect	0.88	Array effect	0.88

Low Temperature Distribution System: 60/50/30°C

Load: 500 houses - DHW 20%

Costs: SEK Swedish crowns, 1984 if not otherwise specified

Solar Collectors

FP (SEK/m^2)	1200
UG (SEK/m^2)	600
Collector Pipes	

Storage

	Tank	Cavern	Pit
Asymptotic costs (SEK/m^3)	250	100	
Small store cost (SEK/m^3)	450	240	250
Size small (m^3)	10000	50000	5000
Beta**	0.4	0.7	0.4
Land (SEK/m^2)	10		
Insulation (SEK/m^3)	800		

Heat Pump:

Condenser (SEK K/W)	1.8
Evaporator (SEK K/W)	1.8
Motor-installation (SEK /W)	1.8
Reference power MW	0.6
Exponent	-0.3

Auxiliary Boiler (SEK /W) 0.8/W

Fuel and Electricity Costs (SEK/kWh) 0.3/kWh

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*Only those conditions deviating from II(b) reference case conditions are cited.

**Volume ratio exponent in storage system cost equation.

4.5.4 Results

Unit solar cost vs solar fraction is plotted in Figure 4-25 a & b for pit, cavern, and tank systems with and without electric heat pumps. In Figure 4-26 a & b the solar costs are combined with the auxiliary energy cost of 300 SEK/MWh to produce the total cost curves for each of the options.

In heat pump systems there is a remarkable difference found for tank storages on one hand and pits and caverns on the other hand. Because of high tank costs, the optimal tank systems favor small volumes which force the collectors to work ineffectively at high temperatures. The other lower cost water storage systems favor larger storage volumes instead of larger collector arrays and hence operate at relatively low temperatures, as can be seen from Figure 4-27a.

In systems without heat pumps, tank storage also shows the highest costs, and the cost curve increases steadily with the solar fraction, i.e., increasing the ratio of storage volume/collector area.

Figure 4-27b shows the stratification achieved in the water pit simulation and Figures 4-28 a & b indicates the sensitivity of the non-heat pump system cost to the the cost of collectors, storage, and auxiliary energy. From these results, a pit system without a heat pump, designed to deliver a solar fraction of about 80%, appears to be the economic optimum, with a delivered energy cost of about 380 SEK/MWh. Note, however, that either the pit or the cavern system can deliver 100 percent of the load with a delivered energy cost of slightly more than 400. Thus, systems with heat pumps show essentially the same economy in Sweden as those without, i.e., around 400 SEK/MWh, which is a remarkable difference from the findings of the reference case studies in which heat pump systems were more favorable than non-heat pump systems.

Figure 4-29 shows the comparison of the total costs of the various systems as a function of the auxiliary energy cost. In this plot, each system is the optimum system for which the marginal cost of the solar energy is equal to or less than the cost of auxiliary energy. This plot shows that the non-heat pump systems with high solar fraction become the systems of choice when energy costs exceed 360 SEK/MWh, i.e. 20 percent more than today's electricity cost in Sweden, and that 100 percent solar systems would be preferred if the energy cost were expected to exceed 430 SEK/MWh (annualized over the plant life).

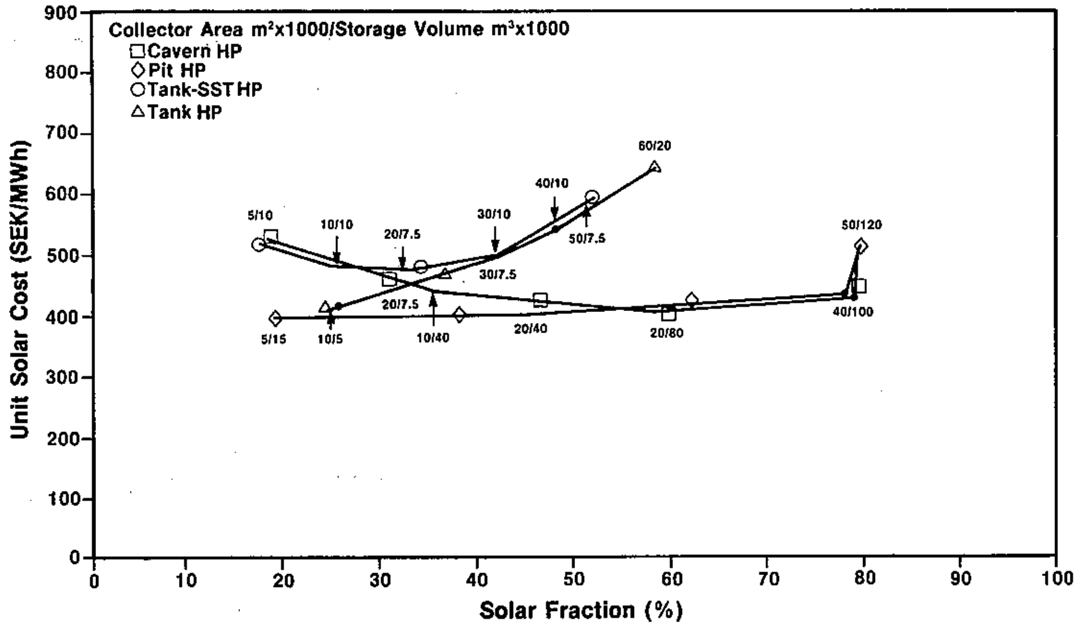

Figure 4-25a. Swedish National Study, Solar Cost vs Solar Fraction. Copenhagen, Low-Temp, DHW 20%. Systems With Heat Pumps

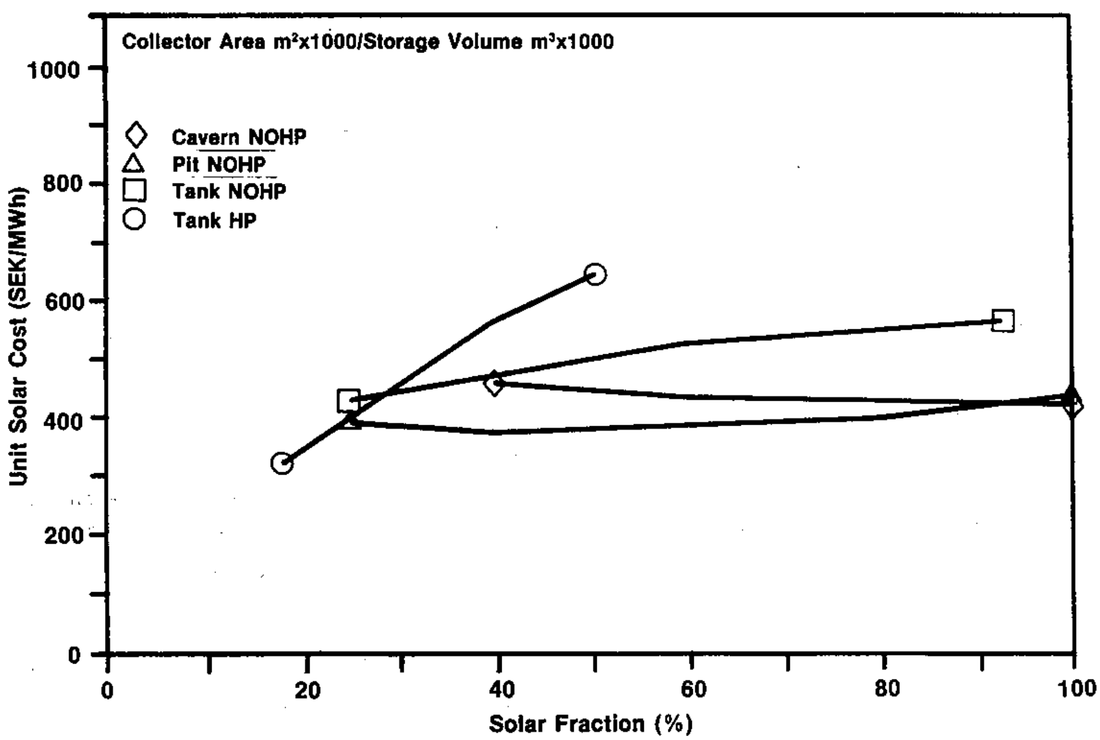

Figure 4-25b. Swedish National Study, Solar Cost Vs Solar Fraction. Copenhagen, Low-Temp, DHW 20%. Systems Without Heat Pumps

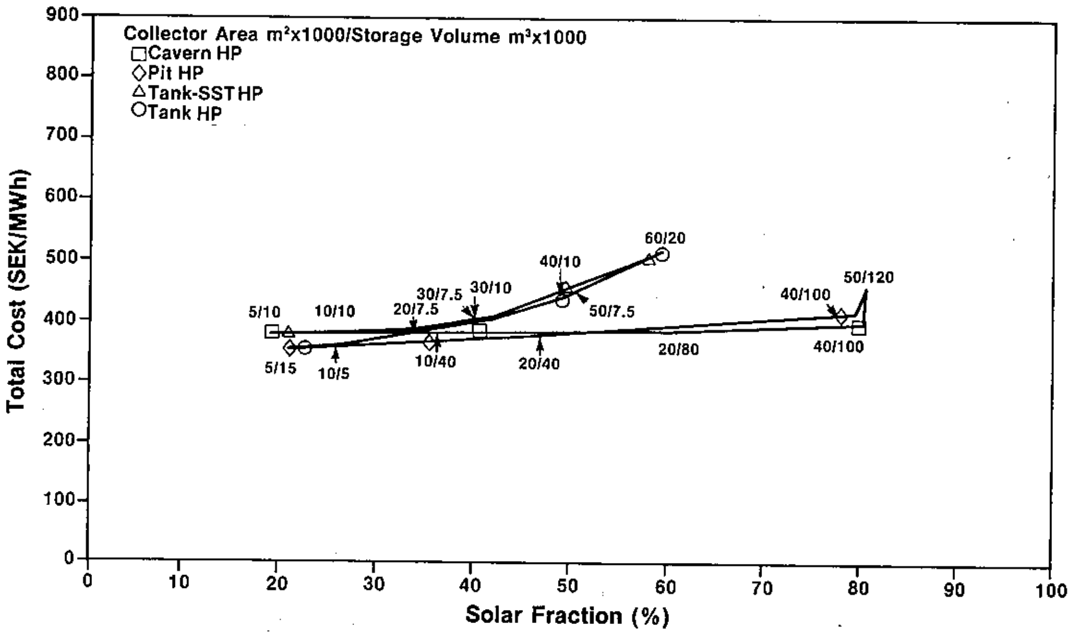

Figure 4-26a. Swedish National Study. Total Cost vs Solar Fraction. Copenhagen, Low-Temp, DHW 20% Systems With Heat Pumps

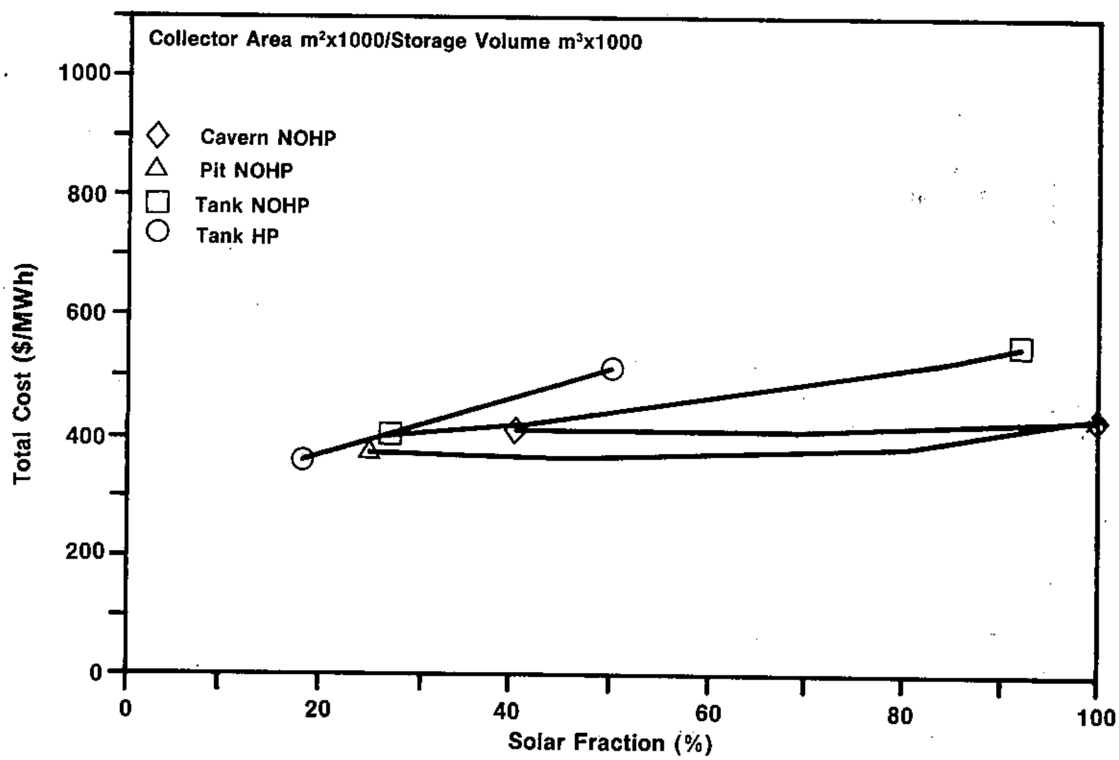

Figure 4-26b. Swedish National Study. Total Cost vs Solar Fraction. Copenhagen, Low-Temp, DHW 20%. Systems Without Heat Pumps

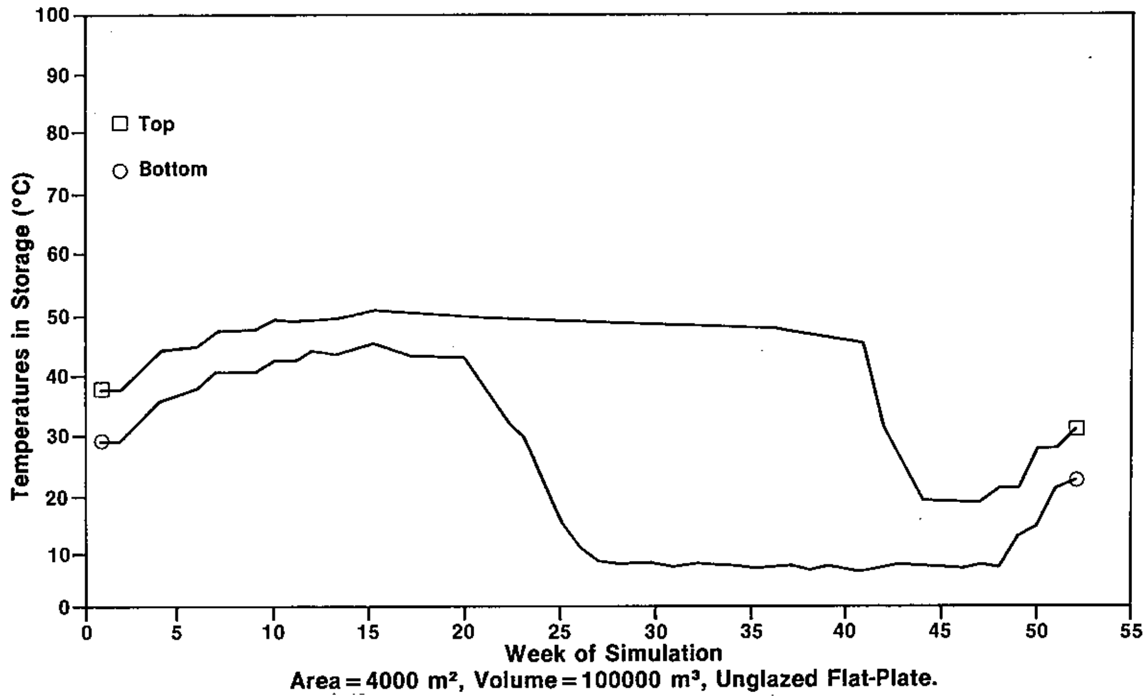

Figure 4-27a. Swedish National Study. Storage Temperature vs Week of Simulation, Copenhagen, Low-Temp. HP, DHW 20%, Pit

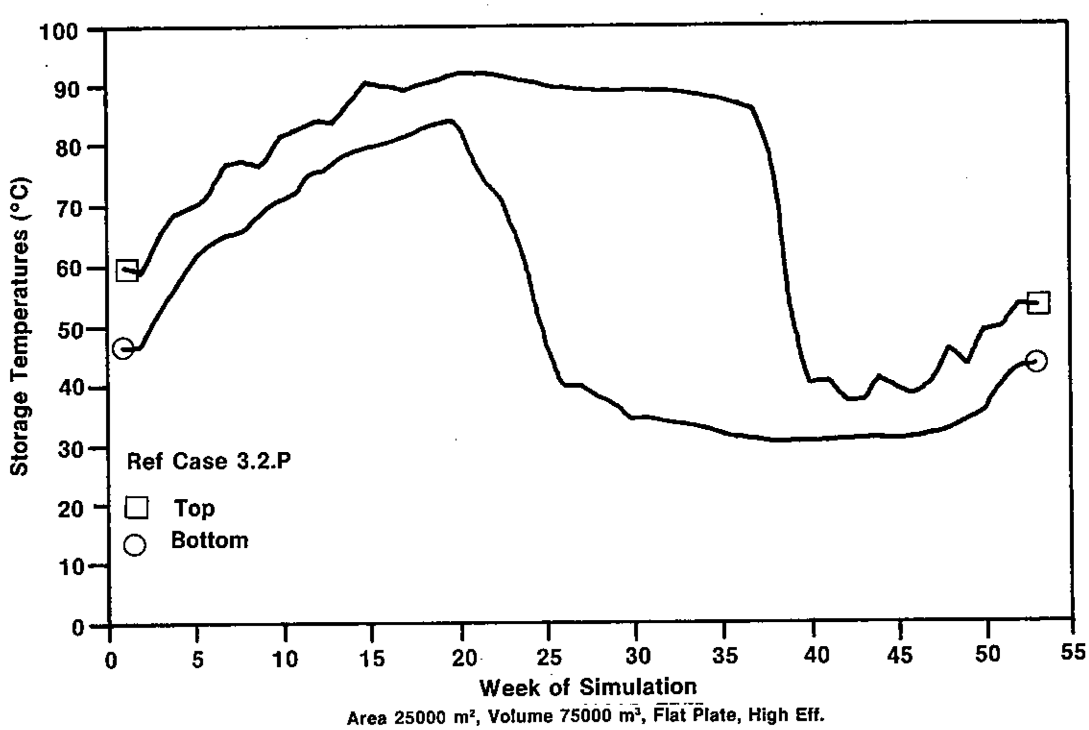

Figure 4-27b. Swedish National Study. Storage Temperature vs Week of Simulation, Copenhagen, Low-Temp. NOHP, DHW 20%, Pit

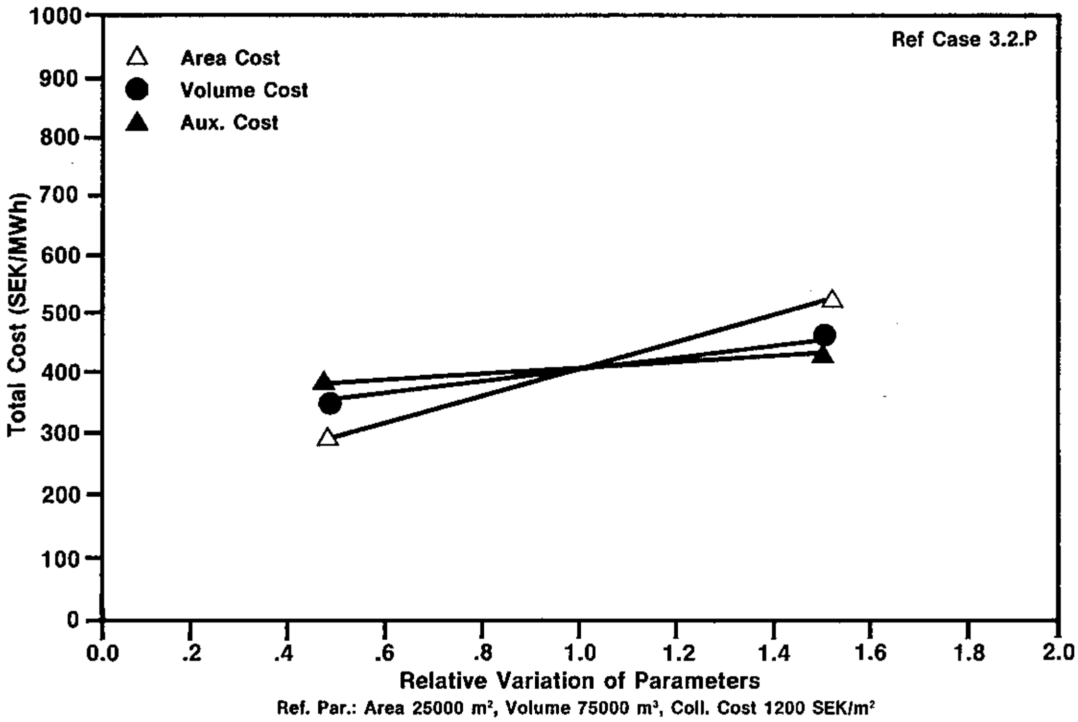

Figure 4-28. Swedish National Study. Total Cost vs Relative Variation of Parameters, Copenhagen, Low-Temp, NOHP, DHW 20%, Pit

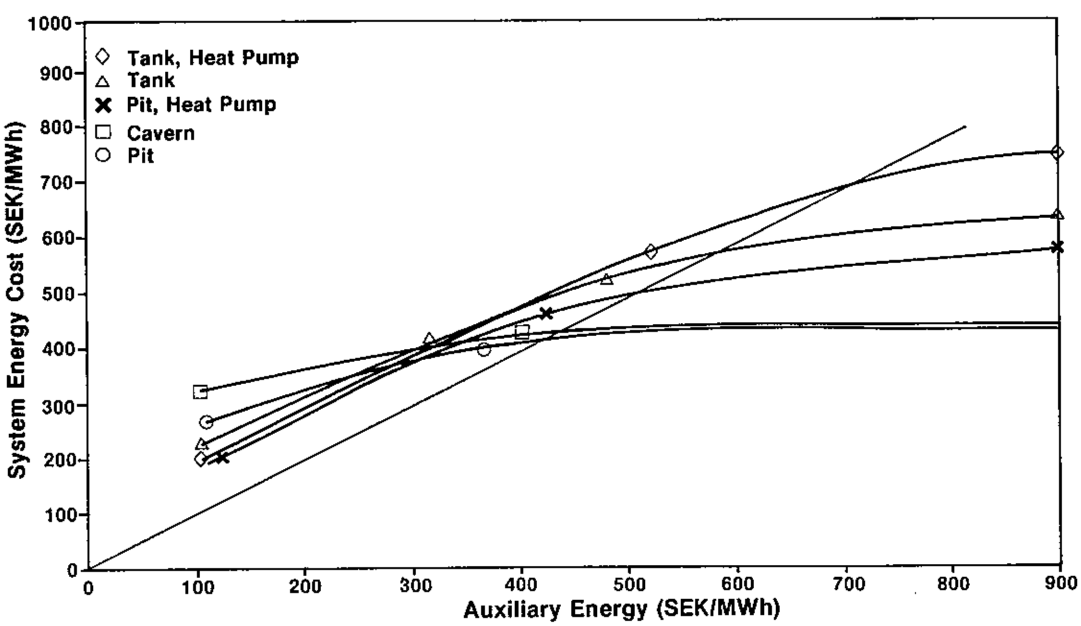

Figure 4-29. Comparison of Total Costs of Various Systems as a Function of Auxiliary Energy Cost

4.5.5 Discussion

The results of the Swedish National Study for Solar Central Heating Systems with Seasonal Storages confirm, by and large, the results from the international working groups.

The tank/heat pump systems show the highest costs of all water storage, steadily increasing from an initial cost point at about 25-30% solar fraction. The tank system experiences a clear penalty because of high storage costs, rather small storage volumes, and high temperatures resulting in less efficient use of the solar collectors. The lowest solar system costs were found for non heat pump pit systems with highly efficient flat plate collectors. The minimum cost of about 380 SEK/kWh [5 cents/kWh (1984)] makes such systems rather interesting. For caverns, the costs are about 50 SEK/kWh higher with and without heat pumps.

Heat pumps can be used advantageously in systems with unglazed collectors up to solar fractions of about 80%. Above that limit a steep rise in systems costs occurs.

For auxiliary costs higher than 360 SEK/kWh, i.e., about 20% above today's costs, the CSHPSS systems without heat pumps appear to be an interesting alternative for district heating. Both pit and cavern systems can then deliver solar fractions close to 100 percent on an economically competitive basis.

When optimizing systems, the collector area and collector costs are the most important cost factors and hence should be analyzed more carefully.

4.6 UNITED STATES OF AMERICA*

4.6.1 Introduction

This National Evaluation focuses on the New England region of the United States. It employs CSHPSS systems with duct storage in rock and a heat pump. Adequate bedrock for large-scale storage is available in the New England region, and the climate, which includes cold, relatively cloudy winters and summers with a substantial amount of solar radiation makes seasonal storage technology of particular interest. The analysis performed for the U. S. assessment is similar to that presented by the II(b) Subtask, although system parameters and economic variables are representative of the local conditions.

The general results of the analysis indicate that the optimized systems provide 75-80% of the low-temperature demand load and 60-65% of the high-temperature demand load with solar energy, with the remaining portion coming primarily from the heat pump electrical input energy. The cost of the energy supplied by the system to the distribution network on an annual basis and using the base case economic scenario is 44-52 $/MWh and 58-62 $/MWh for the low- and high-temperature loads respectively. Significant reductions in cost are found when systems are financed with present incentives provided by the federal government. These results are encouraging in terms of the system cost-effectiveness relative to energy costs in the New England area.

The New England climate is characterized by four distinct seasons. During the summer, the insolation is comparable with most of the United States, and the winters are cold with low insolation compared with other winter U. S. climates. The resulting large space heating loads are, therefore, particularly difficult to displace with diurnal solar systems or passive architecture. New England's ambient winter temperatures do not exhibit the extreme lows of the northern-central regions of the country.

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*Material for this section was provided by Mr. Dwayne Breger, Charles A. Bankston, Inc., 5039 Cathedral Avenue, NW, Washington, DC 20016. Please contact him directly for further details.

4.6.2 Conditions and Systems

Hartford, Connecticut was selected as the most representative location with available TMY data. Average monthly temperatures and total incident insolation (from UMSORT) are shown below.

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	ANNUAL
TEMP, °C	-3.4	-2.7	1.7	9.5	14.8	20.7	23.1	21.3	17.1	10.8	5.5	-1.7	9.7
INSOL, MJ	261	334	383	444	533	498	535	482	422	389	227	183	4690

The New England region includes long coastal regions and mountain ranges in western Massachusetts, central New Hampshire, and Vermont. Bedrock close to the earth's surface is prevalent throughout most of the area. The bedrock composition includes granite, felsite, metamorphized shale, and sandstone. As one approaches the mountainous regions, the bedrock can be quite folded and twisted from the glacial movements. The extent of fracturing in the rock varies considerably and decreases with depth. Water movement through the paper-thin fractures is often slight and not a serious source of heat loss though very few site-specific data are available.

Despite a main trend of population shift to the south and southwestern areas of the United States, the economic health of the Northeast remains strong, and many new development areas are present. Most of the new development is outside the main cities where less restrained land area is available for potential CSHPSS systems. Many new office and multi-family housing units have been built and more are in the planning stage. Both retrofit and new building applications are of importance. The cost of energy in the Northeast is the highest in the country, and all oil and natural gas products are imported into the local New England economy.

The main system parameters and economic/cost data used in this analysis are summarized below.

- Collector Parameters

Collector Type	Intercept	Slope W/m^2K	Incident Angle Parameter, b_o	Tilt	Array Effect
Flat Plate	0.800	4.5	0.1	lati-	0.66
Unglazed	0.778	15.0	0.1	tude	0.70

- Duct Storage Parameters

Surficial Layer (thickness)	6.1 m
Borehole Depth (in bedrock)	100 m
Borehole Radius	0.15 m
Thermal Resistance (between fluid and rock)	0.01 m/WK
Bedrock Thermal Conductivity	3.0 W/mK
Bedrock Heat Capacity	2.1 MJ/m^3K

- Component Costs and Economic Assumptions

Component Cost or Economic Factor		Base Case	Low Cost	High Cost
Collector				
Flat Plate	(\$/m^2)	250	200	325
Unglazed	(\$/m^2)	140	100	175
Piping	(\$/m)	250	200	300
Storage				
Borehole	(\$/m)	20	15	35
Fixed Cost	(k\$)	35	20	50
Heat Pump				
Evaporator	(\$K/W)	0.2	0.15	0.3
Condenser	(\$K/W)	0.2	0.15	0.3
Electric Motor	(\$/W)	0.2	0.15	0.3
HP Operating Cost				
Elec. Rate	(\$/kWh)	0.06	0.03	0.09
Escalation	(%)	1.5	0.0	3.0
(AFUEL)		(1.13)	(1.00)	(1.25)
Discount Rate*	(%)	5	2	8
(AKAP)		(.076)	(.060)	(.094)
System Lifetime (yrs.)		20	20	20

*Additional calculations were performed using the most favorable depreciation schedules and investment tax credits allowed under U.S. laws. These incentives were applied to the base case costs and their impacts are shown in Figure 4-31.

o Load (Distribution) Demand Temperature

	Ambient Temperature	Low-Temperature Distribution	High-Temperature Distribution
Delivery	0°C	55°C	75°C
	-20°C	65°C	105°C
Return		35°C	55°C

4.6.3 Methodology

The New England evaluation employed the same methods used in Subtask II(b) case studies except that the solar cost function was redefined to be the annualized solar system capital cost plus the heat pump operation cost, divided by the solar energy supplied plus the heat pump input energy. This value is referred to as the System Energy Cost (SEC) but does not include cost of auxiliary energy not supplied through the electric heat pump, e.g., from an oil burner. Solar fraction, however, still refers only to the portion of the load displaced by solar energy, i.e., the solar fraction does not include the heat pump electrical energy input.

4.6.4 Results

The base case economic assumptions are applied to the base case system configuration with flat plate collectors. Each parameter combination produces a value of solar fraction and system energy cost (SEC). The points are plotted in Figure 4-30 with different symbols for each collector area. The connecting lines denote the variation in performance and economics for each collector area because of changes in storage volume and number of boreholes. The minimum cost systems at each solar fraction combine to form the system expansion path.

The expansion paths for this base case flat plate collector system have been derived for the alternative economic assumptions and these results are plotted in Figure 4-31. Generally, the high-cost scenario increases the base case SEC by about 70 percent, and the low-cost scenario reduces the SEC by about 45 percent. Financing also provides significant energy cost reductions, which substantiate their importance in the economic analyses of these systems.

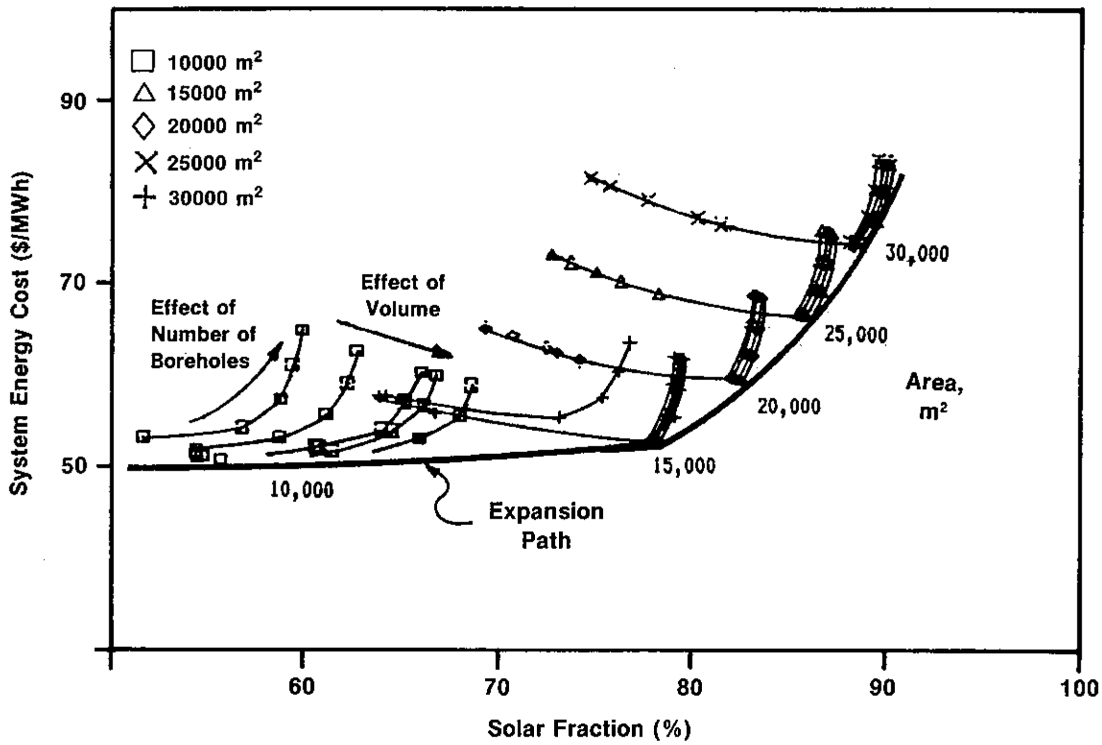


Figure 4-30. Hartford Base-Case Results — Expansion Path Diagrams

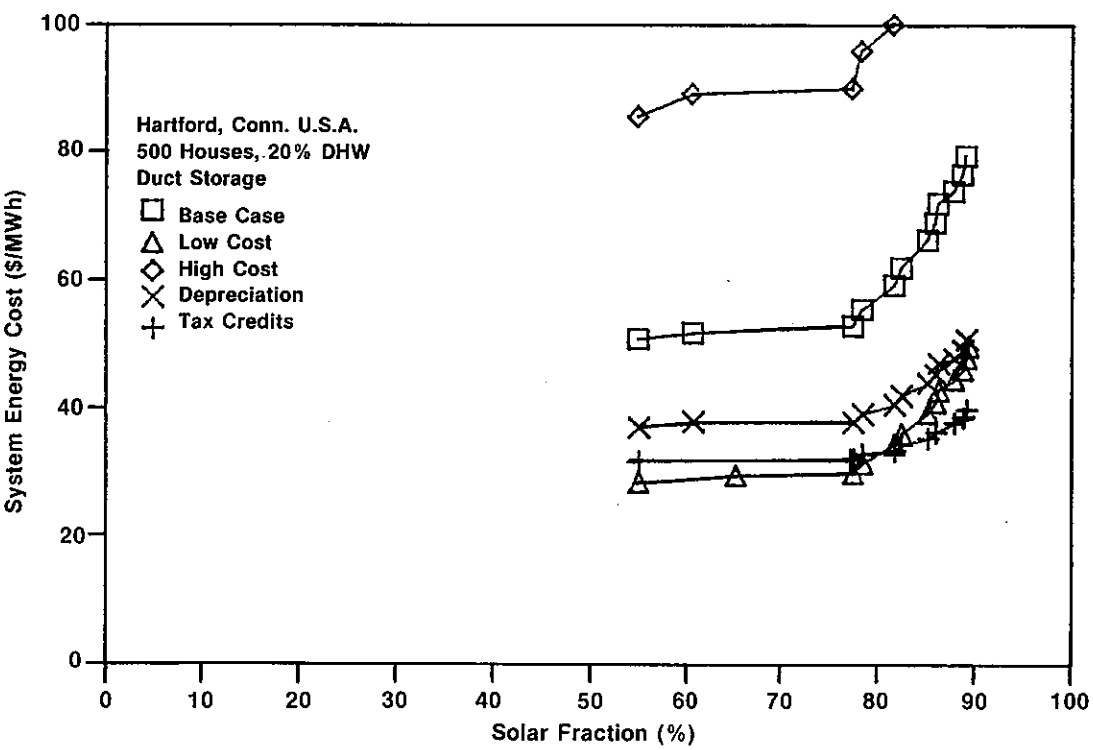


Figure 4-31. Hartford Results for Different Economic Assumptions - Expansion Path Diagram

The high-temperature distribution system is a very likely design option for retrofit. Figure 4-32 shows a comparison of expansion paths for both collectors and both distribution temperatures under the base case economic assumptions. The results show a significant cost increase for high-temperature energy and a reduction in the maximum economic solar fraction. The effect is more pronounced for the unglazed collectors. The reduction in solar fraction is mainly caused by the lower heat pump COP, which is reduced from approximately 5 to 3 for the higher load temperature.

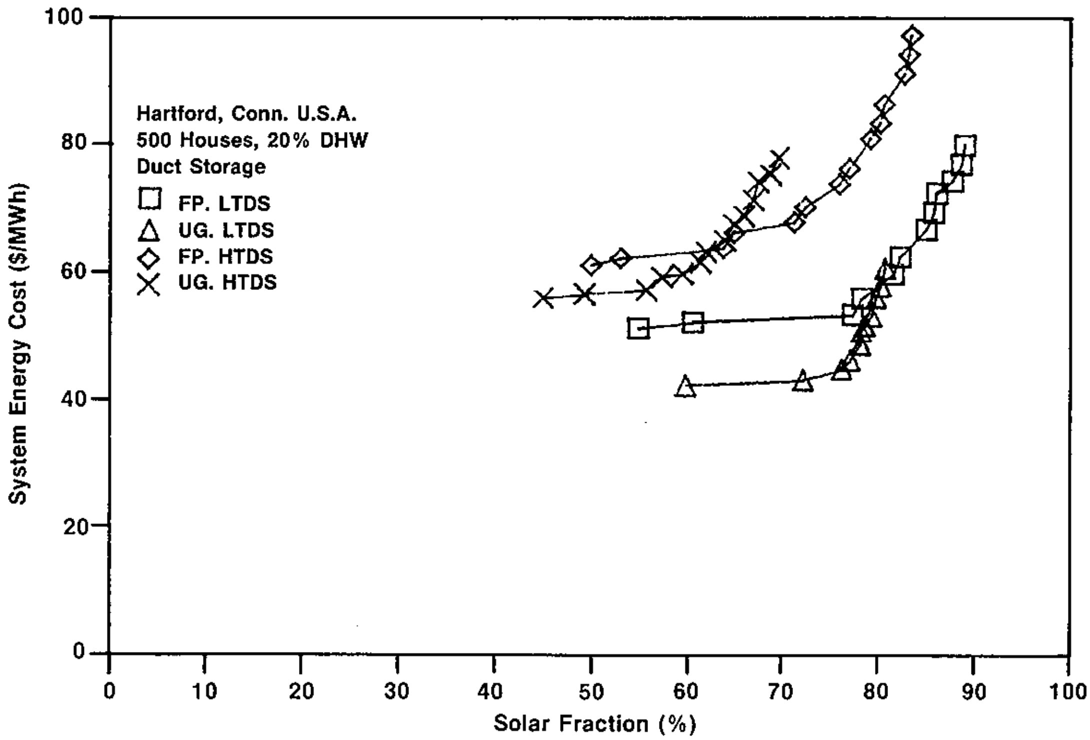


Figure 4-32. Hartford Results for System and Component Variations - Expansion Path Diagram

An annual simulation of a system is shown in Figure 4-33. This system is characterized by 15,000 m^2 flat plate collectors, a storage volume of 300,000 m^3 with 400 boreholes (100 m deep). Average storage temperature is not shown but exhibits a sinusoidal shape with a maximum of 33°C in October and a minimum of 11°C in March.

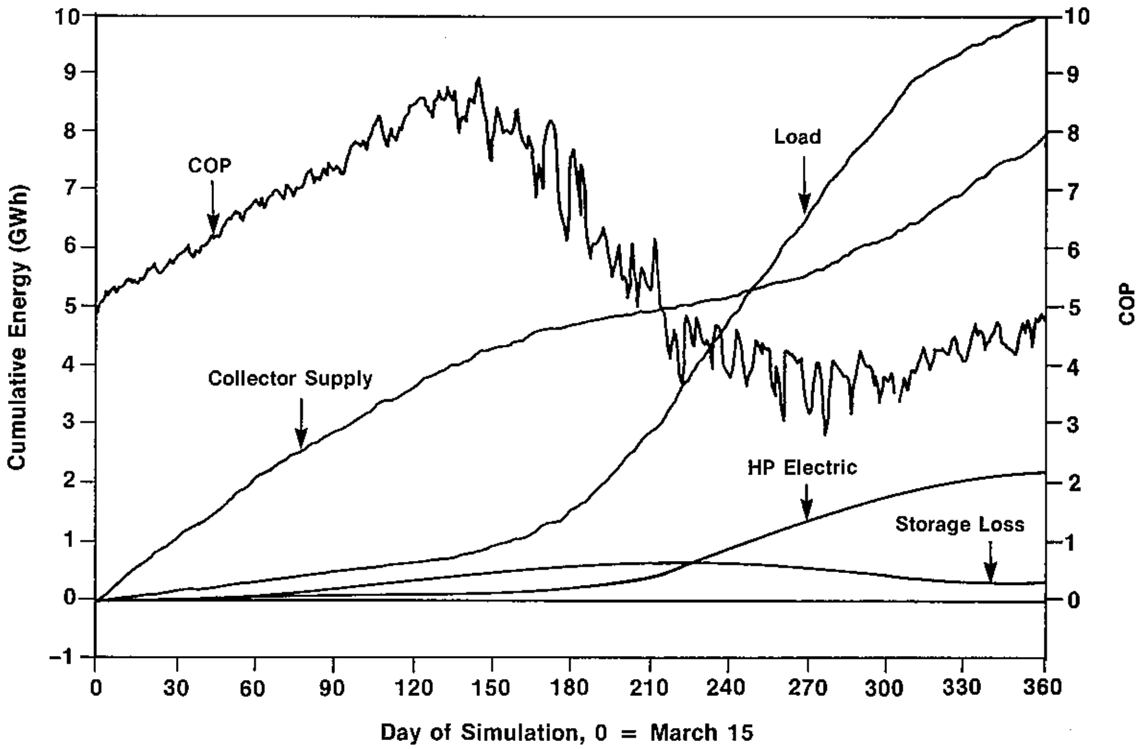


Figure 4-33. System Simulation Over Annual Cycle--Energy Demand and Supply Profiles

4.6.5 Discussion

The results of this National Evaluation are very encouraging in terms of the cost-effectiveness of CSHPSS using duct storage relative to present energy costs in New England. However, a great amount of uncertainty remains, making the implementation of this technology by the private sector difficult. A primary source of uncertainty for any site is the bedrock geology and particularly the extent of water movement. Smaller scale systems may be best to consider an earth pit or tank storage if sufficient surficial layer exists, but again ground water flow must be assessed.

The Hartford results for the low-temperature distribution system are very close to the Madison reference case study results. The system expansion paths are similar and show a familiar transition between optimum collector types, from unglazed to flat plate at higher solar fractions. The high-temperature distribution system results also show similar trends though the expansion paths systems costs tend to be lower in this New England evaluation. This is explained by the slightly lower demand temperatures assumed and the difference in the economic criteria used in this assessment; both would tend to reduce solar cost (in this case where heat pump electrical energy cost is less than the solar energy cost), and the lower demand temperature would provide an increase in the solar fraction.

4.7 SUMMARY OF NATIONAL EVALUATIONS

4.7.1. Scope of National Studies

Although six participating countries completed national evaluations, the scope of the task was not extended greatly beyond that of the reference cases. The latitudes and climates of the national sites were generally similar to one of the two reference sites, the configurations were still restricted to those allowed by MINSUN, and most of the component models were either identical or very similar to those used in the reference studies. One new component--a gas-fired heat pump--was introduced.

The most significant differences between the national evaluations and the reference studies were in the component costs and the economic conditions. These parameters were detailed in the preceding sections in terms of the national currencies, circa 1984. A few of the more important cost figures and economic parameters have been extracted from the national evaluations, converted to U.S. currency, and presented in Table 4-9. Because of the extreme volatility of the U.S. dollar in the past two years and the spread of times for the national evaluations, it was difficult to specify a meaningful set of conversion rates. The rather arbitrary choice of conversion rates used in Table 4-9 is adequate to illustrate some of the important trends however.

It is interesting to note that all the national evaluations used collector prices that were as low as or lower than the reference values. Performance parameter data (not shown in the table) were generally as good or even better than the reference models; thus, it appears that there has been a real international advance in collector cost-effectiveness. Storage costs were generally in line with the reference set or based on very limited data. This reflects the lack of experience with large energy storage technology that still prevails in most participating countries. Similar remarks apply to the heat pump costs.

The cost of competitive energy is the most important variable since it determines the current economic viability of CSHPSS in many nations. The cost of thermal energy from fossil fuels varies by a factor of <u>four</u>, and the cost of electrical power by a factor of five. In fact, differences nearly this large could exist within a single country, e.g., the U.S., or within the rate structure of a particular utility company.

4.7.2. <u>Tabulation of Results</u>

An abridged table of some of the significant performance and economic results from the national evaluation is given in consistent units in Table 4-10. The cost and performance of the optimal systems included in this group do not differ greatly from those found in the reference study. Heat pump systems with low cost collectors were generally found most economical, and the solar fractions and unit solar costs were similar to the reference results--except for Germany where cost experience with pit storage and heat pumps leads to high cost systems. Again, the most important differences are in the cost of competitive energy forms. On the basis of the ability to meet the conventional energy costs, the U.S. and Sweden appear to be the most attractive market for CSHPSS.

Table 4-9 NATIONAL EVALUATIONS - SUMMARY OF COST AND ECONOMIC DATA

	Reference Study	Canada	CEC	Germany	Netherlands	Sweden	USA
Conversion Rate		0.75	1	2.67	3.5	8	1
Collector Costs							
Unglazed ($/m^2)	140		140	93	86	75	140
Flat Plate ($/m^2)	245	225	245			150	250
Evacuated ($/m^2)	350		350	228	228		
Storage Costs ($/m^3)							
Aquifer, total	154000	154000					
Cavern, C_b,C_s	10, 48					6, 30	
Duct, C_b,C_s,C_{bh}	0.1, 0.2, 30		0.1, 0.2, 30		1.7,3.4,4.3		, , 20
Tank, C_b,C_s	50, 90					31, 56	
Pit, C_b,C_s	20, 30			34, 83		12, 31	
Heat Pump	LT HT	LT	LT HT				
Condenser ($K/kW)	200, 300	150	200, 300	300	250	225	200
Evaporator ($K/kW)	200, 300	150	200, 300	300	250	225	200
Motor/Comp ($/kW)	200, 300	150	200, 300	300	250	225	200
Auxiliary							
Boiler ($/kW)	100	75	100	100	57	100	
Fuel ($/MWh)	NA	12 - 15	NA	27	23	38	40
Electricity($/MWh)	NA	14 - 31	NA	38	69	38	57
Economics							
Lifetime (yr)	20	20	20	15	20	20	20
Discount Rate (%)	5	5	5	5	4	5	5
Fuel Escalation(%)	NA	2	NA	2	2	2	1.5
Fuel = Elect	Y	N	Y	N	N	Y	N

Table 4-10. NATIONAL EVALUATION RESULTS

Participant/Site	System	Collector Area m^2	Storage Volume m^3	Heat Pump Capacity kW/K (COP)	Distrib. System	Solar Fraction	Solar Cost \$/MWh	Auxiliary Cost \$/MWh	Cost Ratio
Canada									
- Toronto	Aquifer,FP,HP	12500	6812[a]	1794 (3.9)	LTDS	0.68	39	15o[c]	2.6
- Fredericton	Aquifer,FP,HP	15000	7526[a]	2150 (3.7)	LTDS	0.69	41	19o	2.2
- Winnipeg	Aquifer,FP,HP	15000	8901[a]	3342 (3.2)	LTDS	0.66	35	12g	2.9
CEC									
- Ispra, Italy	Duct,FP,HP	10000	325000	NA (6.5)	LTDS	0.70	50	NA	NA
Germany									
- Hamburg	Water Pit,UG,HP	17000	35000	NA (NA)	NA	0.50	92	38o	2.4
- Hamburg	Water Pit,EC	20000	53000	-----	NA	0.70	104	38o	2.7
Sweden									
- Gothenburg	Water Pit, FP	25000	75000	-----	LTDS	0.85	50	38o, 38e	1.3
the Netherlands									
- De Bilt	Duct,FP,GHP	1150	38000	NA (1.7)	LTDS[d]	0.40	37	23g, 69e	1.6g
	Duct,UG,EHP	1750	58000	NA (3.5)	LTDS[d]	0.65	54[b]	23g, 69e	2.3g
	Duct,EC	3500	16800	-----	LTDS[d]	0.75	74	23g, 69e	3.2g
U.S.A.									
- Hartford	Rock Duct,FP,HP	15000	300000	NA (4.5)	LTDS[d]	0.77	52[b]	40o, 57e	1.3o

a - Volume of water injected

b - Includes the cost of heat pump operation

c - o = oil, g = gas, e = electricity

d - Temperatures modified from II(b) LTDS

5.0 SPECIAL STUDIES -- RESULTS

5.1 MINSUN VALIDATION

One of the objectives of the Subtask II(b) effort was to validate the analysis methods and models used to perform the system studies. Since MINSUN was the basic simulation program used for the task, and since it is of relatively recent origin, most of the concern about validation centered on MINSUN and the models it uses for storage, collectors and heat pumps.

At the beginning of the subtask in November 1983, Hadorn [20] compiled a list of MINSUN validation work that already had been accomplished. Since most of the component models used in MINSUN were developed from well-known and well-validated principles, equations, or numerical models, there is little concern about their validity. For example, the MINSUN radiation processor is based on the well-established procedures used in TRNSYS, the collector models utilize the standard procedures developed by the U.S. National Bureau of Standards, adopted as national standards in the U.S., and tested in the IEA Solar Heating and Cooling Program. The models used for energy storage were thoroughly tested in Subtask I(c) by comparison with more complex mathematical models and with experimental results [21, 22]. The heat pump model is based solely upon theoretical considerations in order that it be applicable over a broad range of conditions and a wide range of sizes. It may not represent the performance of a particular heat pump (fixed capacity) as well as a model based on actual performance maps, but it should be representative of the performance that could be achieved in any range by a suitably designed heat pump.

Validation of complete system simulation is less obvious, and few experimental or numerical results are available for comparison. Each simulation program has constraints and limitations that make direct inter-code comparison difficult. Validation of computer results on the basis of experimental results is always the ultimate goal of computer model validation, but this is especially difficult in systems with seasonal storage because it may take years of field operation to achieve a quasi-steady state. In view of these difficulties, we feel fortunate to be able to cite results from four system simulation comparisons that do add to our confidence that MINSUN provides a reasonably correct simulation of actual system behavior.

A MINSUN simulation of the Groningen CSHPSS system was reported by Wijsman [23] and compared with earlier results obtained with a more detailed code developed the Institute of Applied Physics at Delft (TNO). Although adjustment of some of the MINSUN parameters was required (especially those not used in the TNO model, e.g., array factors and specific house loads) the annual results compared quite well. There was less agreement between instantaneous values of variables such as the temperatures in storage, however.

A second comparison of MINSUN results with TRNSYS simulation was reported by Krischel [24] who modeled a system with tank storage and a heat pump with MINSUN and with TRNSYS. The greater flexibility of the TRNSYS program allowed inclusion of a collector-to-store heat exchanger, a buffer tank, and a direct collector-to-load connection that was not possible in MINSUN. Figure 5-1 shows the energy collected as predicted by the two programs. MINSUN predicted substantially greater collector outputs in most months. A part of the discrepancy is due to the heat exchanger that was included in TRNSYS but not in MINSUN, but further examination shows that the MINSUN tank model and control strategy results in more pronounced stratification of storage temperatures, which also increases the collector output. Figure 5-2 shows that MINSUN, which controls both inlet and outlet tank levels according to the temperatures required gives a much higher top layer temperature and a lower bottom temperature except in the summer months. The drop in top layer temperature is shifted from December in TRNSYS to January in MINSUN. Yearly solar fractions from MINSUN were generally about 10 percent higher for systems with heat pumps and about 15 percent higher for systems without heat pumps.

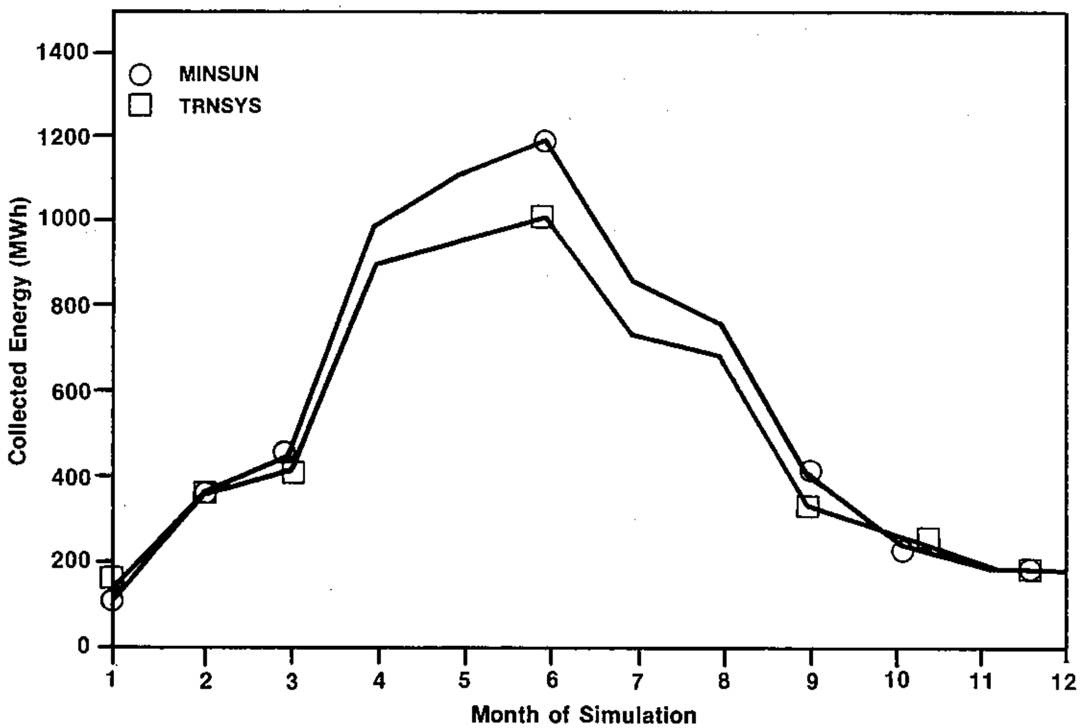


Figure 5-1. MINSUN and TRNSYS Predictions of Energy Collection

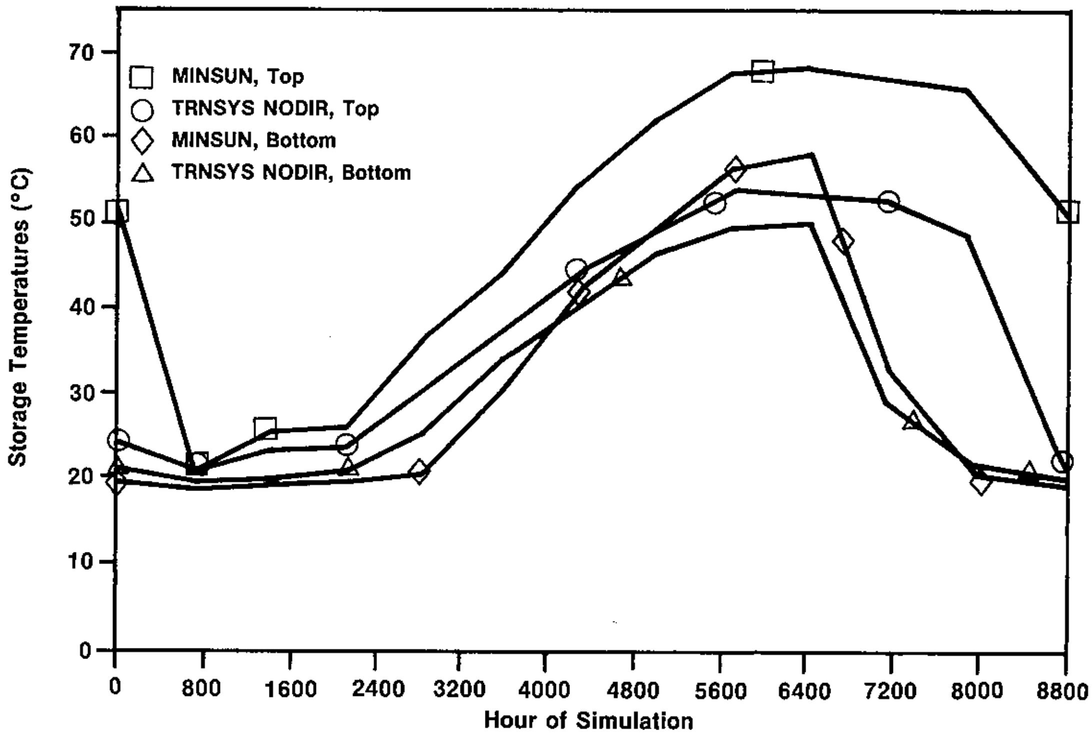


Figure 5-2. MINSUN and TRNSYS Predictions of Storage Temperatures

In general the characteristics of MINSUN and TRNSYS results are in acceptable agreement. The most remarkable difference is a more pronounced temperature stratification from the MINSUN model. This is caused by variable temperature-controlled storage inlets and outlets on both the collector <u>and</u> load side. Additionally, somewhat higher solar fractions from MINSUN result from the fact that devices are missing which might reduce the amount of usable solar energy, e.g., heat exchanger between collector array and storage tank.

Hadorn [25] reports the results of two detailed comparisons of MINSUN and TRNSYS results. In the first study the total radiation incident on the plane of the collector as predicted by the MINSUN and TRNSYS radiation processors were compared for three locations -- Madison, Copenhagen, and Zurich. On an annual basis, the Madison and Copenhagen calculations agreed within 1%. The Zurich results differed by 6 percent, but Hadorn attributes the discrepancy to pre-calculations required for the Zurich weather file.

In the second study, a complex system for 50 houses involving flat plate collectors, a large central water tank, distribution network, a pre-heat loop for domestic hot water, and all the necessary valves, pumps, and heat exchangers was modeled with TRNSYS. The MINSUN calculations necessarily involved a simplified representation of the same system. Although the requirements and limitations of the two codes are different, the input parameters were all selected to describe, as nearly as permitted, the same physical system. The collector areas were 1000 and 5000 m^2 and the storage volume ranged from 10 to 10000 l/m^2. Annual values of incident radiation, energy delivered to storage, storage losses, auxiliary for space heat, auxiliary for DHW and solar fraction were compared. The results were quite reassuring.

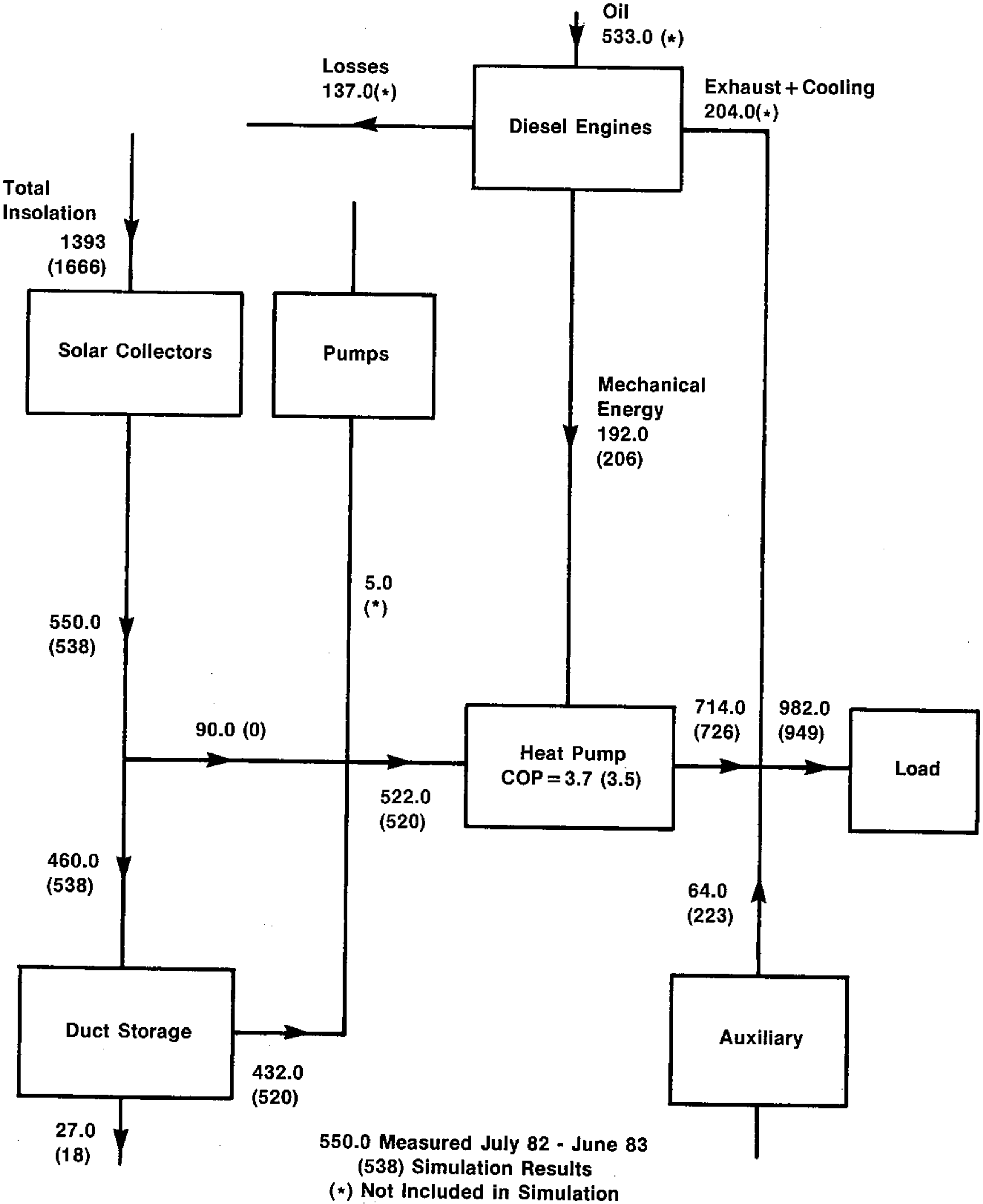
The TRNSYS and MINSUN predictions of solar fraction were always within 10%. The MINSUN predictions were generally the more conservative (lower). The other quantities were equally close. There was a tendency for better agreement when the storage volume was large which may reflect both the difference in time step (1 day for MINSUN vs 15 minutes for TRNSYS) and the more sophisticated tank inlet and outlet temperature control strategy used by MINSUN. Short term comparisons were not made, but the variations would be expected to be larger.

One attempt has been made to compare MINSUN predictions with results from an operating system. Zinko and Perers [26] reported on the simulation of the "SUNCLAY" project in Kungsbacka, Sweden near Gothenburg. The system is a 15000 m^2 school building heated by 1500 m^2 array of unglazed solar collectors with the assistance of four 200 KW diesel-driven heat pumps. The storage system uses about 87000 m^3 of soft clay which is accessed by 608 U-shaped plastic pipes inserted to depth of 35 m. Auxiliary energy, required in addition to the waste heat from the diesel engines is supplied by an oil burning plant. The building has been in service and monitored since April 1981 and has been trouble free since autumn of 1982 [27].

Since the purpose of the comparison was primarily qualitative, existing weather tapes for a typical year in nearby Copenhagen were used in place of the actual on-site climatological data. This difference and the inability of MINSUN to model the engine waste heat precluded a completely satisfactory comparison. Even so, the results show that MINSUN reliably predicts the main features of the system response. Figure 5-3 shows the annual energy balances for the actual and simulated systems. The chief differences are the insolation and the required auxiliary energy. The storage and heat pump outputs are quite accurately predicted. Figure 5-4 shows the actual and predicted storage system temperature. The calculated temperature swings and phases are quite close to the measurements. It appears that the main discrepancies between the calculation and measurements is the initial condition for the simulation.

Although identical weather data could not be used for the simulations, the authors conclude that MINSUN is well suited for the purpose of systems analysis and sensitivity studies (including optimization) for all systems using one of the following generic families of storage types: tank storage, ground-coupled stratified water storage, duct storage, or aquifers.

Figure 5-3. ACTUAL AND PREDICTED ANNUAL ENERGY BALANCE FOR THE SUNCLAY PROJECT



Figured 5-3. Actual and Predicted Annual Energy Balance for the SUNCLAY Project

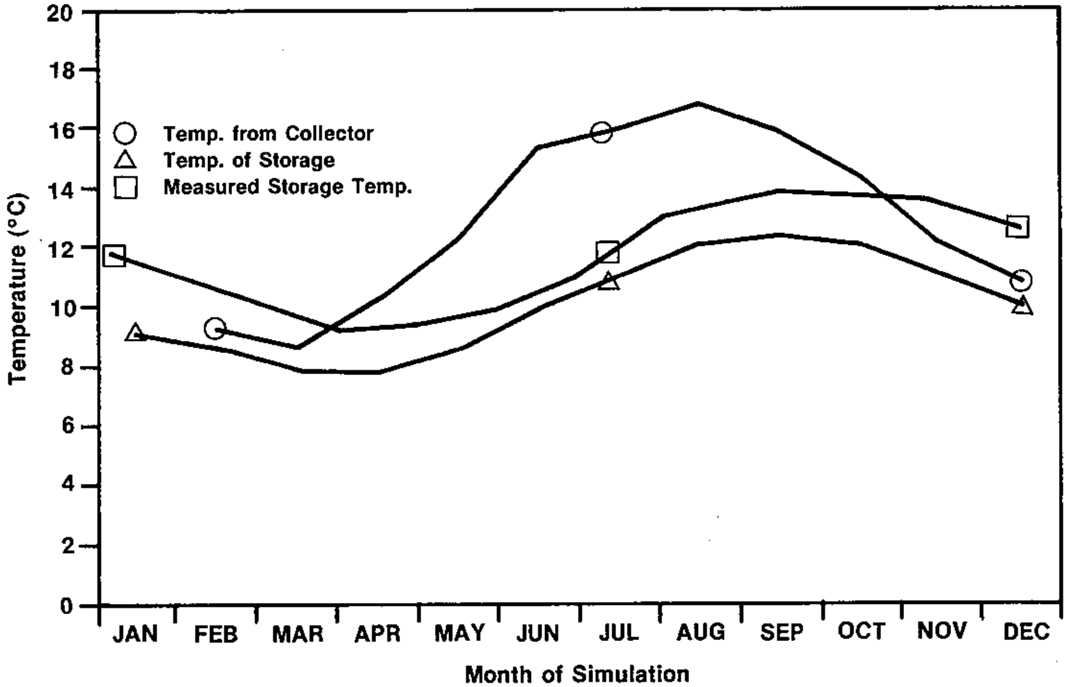


Figure 5-4. Actual and Predicted Simulation Temperatures in the Duct Storage and Solar Collectors

5.2 SYSTEM VARIATIONS

There are a number of configurations of interest that are not currently within the simulation capability of the MINSUN code. These include direct connection between the collector array and the load, buffer storage either at the collector array or at the load, distributed collectors and storage, multiple well aquifers, collector loop heat exchangers, and a few others. Since modifying MINSUN to accommodate more general designs was considered to be a major undertaking, a few studies were conducted using the TRNSYS simulation program to determine if, in fact, the concepts were advantageous and to what degree.

5.2.1 Collector-to-Load Connection

The effect of supplying load directly from the collectors when the availability and demand are in phase was analyzed by Krischel [24], using the TRNSYS version 12.1 and the same performance parameters used in the reference cases described in Section 3 for a system consisting of 24,000 to 36,000 m^2 of flat plate or evacuated collectors, a 70,000 m^3 storage tank, and the nominal 500 residence with 20% DHW load. Figure 5-5 shows the energy delivered directly to the load by the various systems on a monthly basis. During the summer months, the direct connection supplies most of the DHW demand, and, during September and October, the larger or more productive arrays meet a part of the space heating load as well. Krischel's preliminary results indicated that the direct-to-load connection increases this system's annual solar fraction by about 7.3 percent or that a direct-to-load connected system with 24,000 m^2 of collector area delivered about the same solar fraction as a 36,000 m^2 system without direct-to-load connection. Since the collector array represents about half of the cost of most systems, a 50 percent reduction in collector area would imply a 25 percent reduction in unit solar cost.

For heat pump systems, however, the direct connection is of minor importance. Since the temperature level in the bottom layer is low, even in summer, and the collectors are less efficient, the demand load temperature of 50°C cannot often be met.

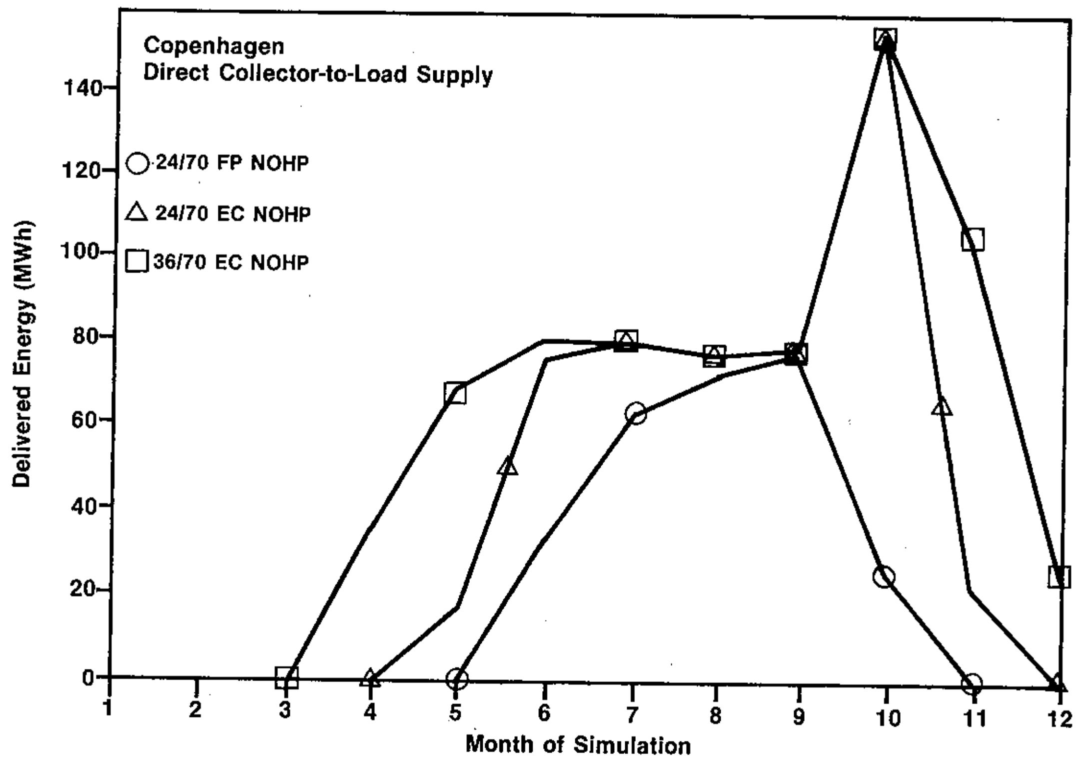


Figure 5-5. TRNSYS Predictions of Energy Supplied Directly from Collectors to Load

5.2.2 Buffer Tanks for Daily Heat Storage

Early studies by Silman [28] indicated that buffer tanks that could store the output from a good day of solar collector operation to be used to meet DHW or night-time heating load had some advantages over systems that always return the collected energy to the main storage system. Krischel [24] investigated the effect of a small buffer tank in parallel with a large stratified tank storage system.

Because the good thermal stratification in the main storage provides a high temperature level in the upper layers, an additional buffer tank gave no higher solar fractions for either the heat pump systems or for the non-heat pump systems.

The buffer configuration should be more effective with a non-stratified main storage system such as the duct system in which all of the energy collected for a period of time may be absorbed by the store at a relatively low temperature--making it unavailable to meet a DHW or direct heating load. An interesting combined buffer/duct system has been proposed by Peter Margen [29] in which deep bore holes into rock are drilled from a network of relatively shallow tunnels. The rock below the tunnels serves as the main duct storage system while the volume of the tunnels provide short term buffering.

5.3 ADDITIONAL STUDIES

5.3.1 Unglazed Collectors at Low Temperature

The solar collector routines used in MINSUN are intended only for collectors that operate during the daylight hours when the insolation is above a threshold level. In very low-temperature systems such as those employing heat pumps, the inlet temperature to the collector array may be below the ambient night temperature. Thus, it is possible for unglazed collectors to collect energy over a full 24 hour day. Under conditions of nighttime operation, the effective sky temperature is quite low and the effective heat loss coefficient may be quite different from that measured in normal daytime tests. To evaluate the importance of subambient temperature operation and cold sky radiation losses, Wijsman [30] modified the MINSUN collector routines and recalculated the energy collection as a function of collector inlet temperature for unglazed collectors operating whenever the inlet temperature is below ambient and with and without an IR- radiation heat flux of 40 w/m^2 (typical for the Netherlands). These results are shown in Figure 5-6.

Wijsman also recalculated array energy reduction factors for typical unglazed collector arrays operating at low temperatures and found them significantly higher than for flat plate collectors. He recommends an array factor of 0.83 for the unglazed collector as compared with the 0.66 value used for flat plate collectors.

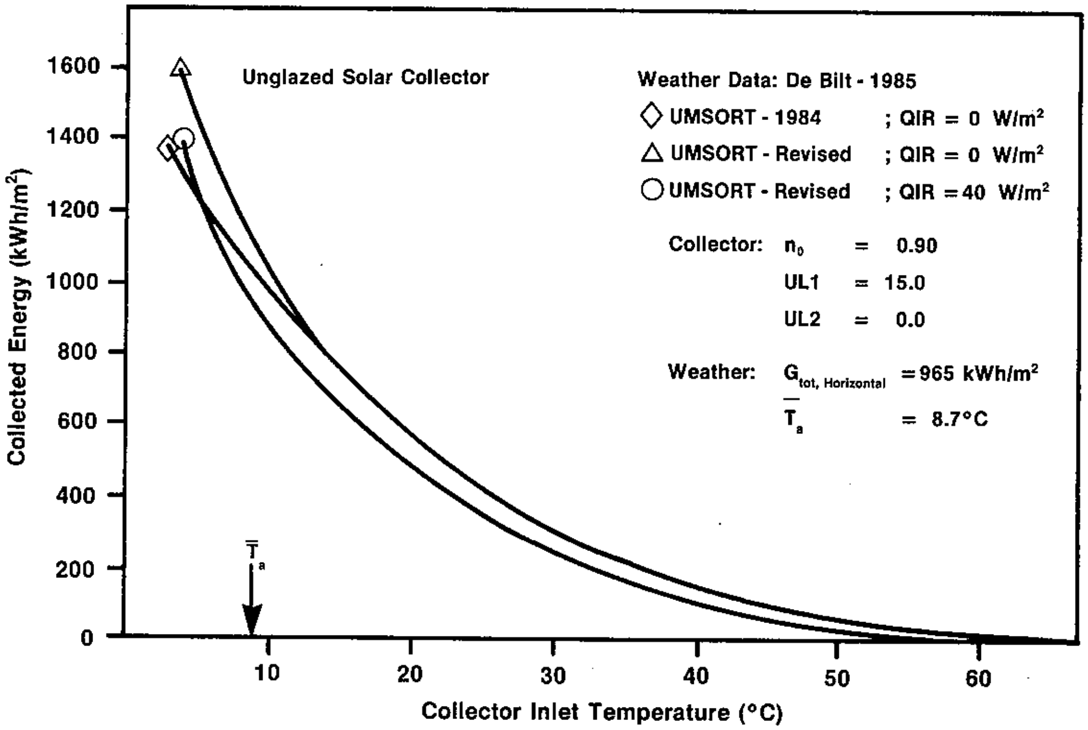


Figure 5-6. Influence of Change in Heat Gain Control and Influence of IR-Radiation Heat Losses to the Sky or Heat Gain by Unglazed Solar Collectors

5.3.2 Pumping Energy Studies

The MINSUN program does not currently contain a calculation of pressures, pressure drops, or pumping power, and these quantities were not included in the reference case calculations. For most of the systems considered, i.e. those involving tanks, caverns, pits, and most water-cooled collectors, the pumping power is known to be relatively small so these omissions are not of serious concern. The pumping power required for the distribution is usually significant (~4-5 percent of energy delivered), but since the current analysis is only comparing energy option at the source, the distribution pumping power has not been included. Aquifers and duct storage systems, however, may introduce significant pressure drops and pumping requirement. A brief preliminary study of the magnitude of these requirements was conducted by Canada as an adjunct to their Aquifer system report [9]. Rather than calculating pressure drops and pumping power directly from flow rates and the dimensions and characteristics of components, the Canadian study simply specifies the pressure drops for the major subsystems on the basis of typical systems examined or described in the literature. Three configurations involving four subsystems were considered. The subsystems and component pressure drops are shown in Figure 5-7.

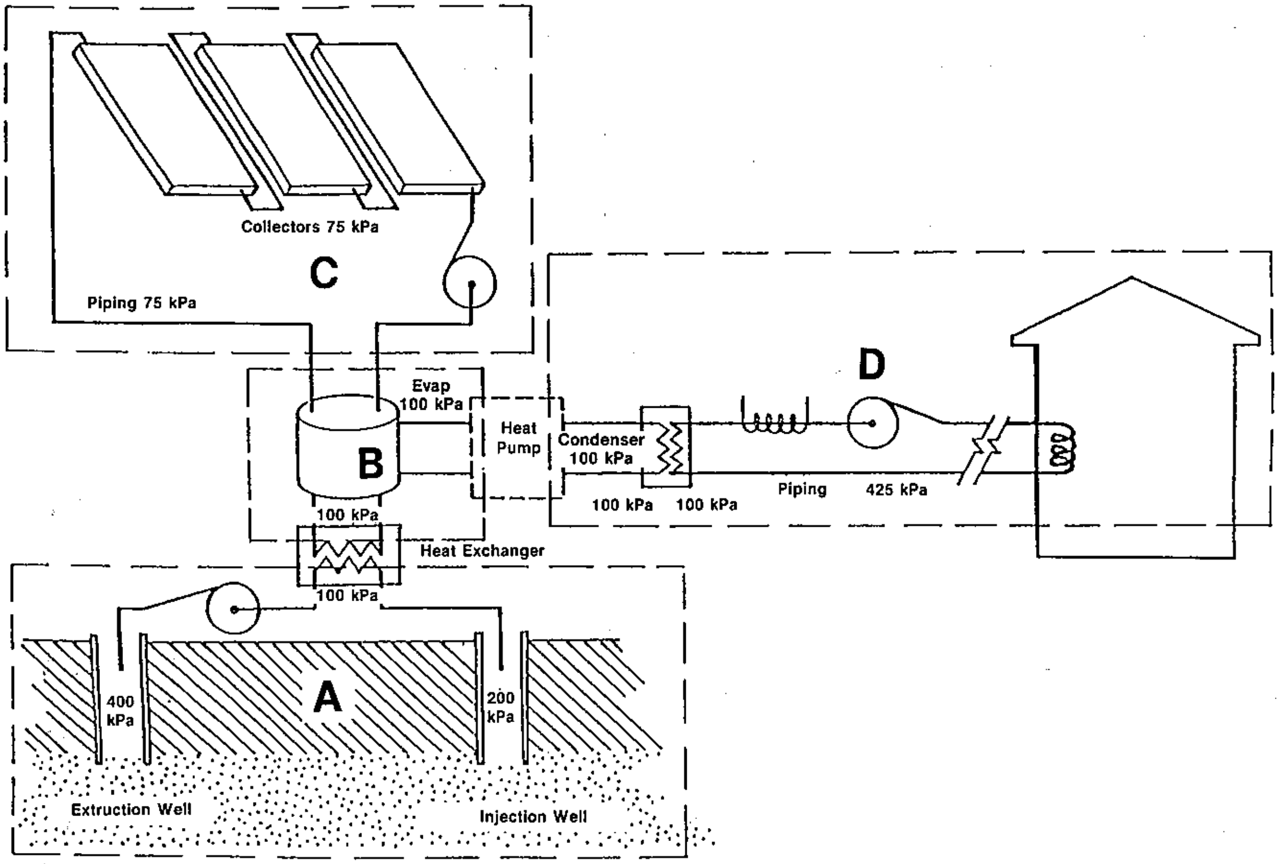


Figure 5-7. Subsystem Schematic With Typical Pressure Drops

Energy requirements for three configurations have been calculated. Each configuration provides a different flow situation. These configurations are:

1. Aquifer storage with heat pump
2. Aquifer storage without heat pump
3. Tank storage without heat pump.

Configuration 1 represents the heat pump base case of this study. Large flows of low-temperature water are transferred through subsystems A, B, and C. Configuration 2 represents the non-heat pump base case of this study. Small flows of high-temperature water are transferred through subsystems C and A. Because the heat pump does not exist, subsystems B and D are combined; the pressure drop across the new subsystem D is the same as the old one.

Configuration 3 represents a simple tank storage system for comparison with the higher pressure losses of aquifer storage. Again subsystems B and D are combined. Subsystem A now has only the pressure drop associated with the one side of the heat exchanger.

In all three configurations, the flows through subsystem D are equal because the loads are identical (10,000 MWh/year).

Table 5-1 presents the results. Storage has been further separated into charging and discharging the storage volume. These results are illustrated in Figure 5-8.

Table 5-1. PUMPING ENERGY REQUIREMENTS

	Aquifer Storage With Heat Pump Configuration 1	Aquifer Storage Without Heat Pump Configuration 2	Tank Storage Without Heat Pump Configuration 3
lection			
Flow (000 m^3/year)	1263	330	330
Pressure Drop (kPa)	150	150	150
Energy* (MWh/year)	88	23	23
rging			
Flow (0003m^3/year)	1263	330	330
Pressure Drop (kPa)	700	700	330
Energy (MWh/year)	408	107	15
charging			
Flow (000m^3/year)	1048	293	293
Pressure Drop (kPa)	700	700	100
Energy (MWh/year)	340	95	14
t Pump			
Flow (000m^3/year)	1048	NA	NA
Pressure Drop (kPa)	200	NA	NA
Energy (MWh/year)	97	NA	NA
tribution			
Flow (000m^3/year)	293	293	293
Pressure Drop (kPa)	725	725	725
Energy (MWh/year)	98	98	98
al Energy (MWh/year)	1031	323	150
f 10,000 MWh load	10.3%	3.2%	1.5%

ergy (MWh) = $\frac{\text{Flow } (m^3) \times \text{Pressure Drop (MPa)}}{0.6 \text{ (efficiency)} \times 3600}$

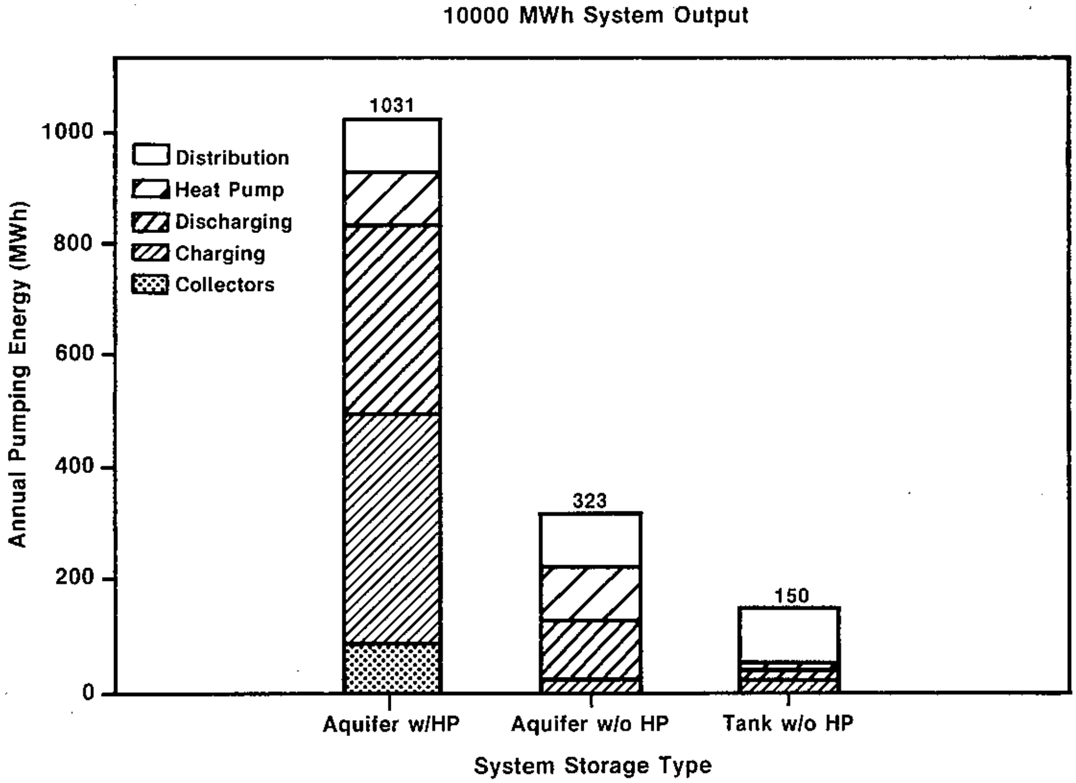


Figure 5-8. Pumping Energy Requirements

The pumping energy requirement varies from a maximum of 10.3 percent to a minimum of 1.5 percent of the total energy delivered to the load. These figures likely represent the upper and lower bounds for probable systems of this size.

The aquifer storage subsystem requires by far the largest pumping energy. The amount of pumping energy required by systems of the size represented here is not sufficient to affect significantly the unit price of solar energy. However, consideration of pumping power would alter the design of optimum systems. After accounting for pumping requirements, optimum systems would use higher distribution temperatures and lower flow rates, and be smaller and more centralized with components of low restriction.

5.3.3 Distribution System Costs

Although detailed information for the design and cost estimation of heating distribution has been compiled [7], these data were not used in the reference case analyses. To do so would require specification of a great many variables characterizing the load. Since the primary purpose of the reference case analyses was to evaluate the different solar/storage system options, it was decided to omit distribution systems from the analyses. However, the inevitable question arises in comparing central solar plant options with distributed systems--either solar or conventional. As a first estimate, a figure based on Swedish experience of $2000 per unit was used. This figure applies to a "normal Swedish distribution network in a medium density residential development." [31].

To give the full picture of the costs for heating, the in-house installations should be added. These costs are, however, omitted in this report. The reason for this is that all heating systems need installations of this kind and the costs are not very dependent upon type of system. In other words, the costs of an individual boiler are not drastically different from those of an in-house installation (i.e., heat exchanges, values, meters) in connection with a central heating plant. Furthermore, the heating system of a building is not optimized for an individual building. It is normally determined by a national code. It is thus not a part of the optimization of the external heating system.

When comparing the central system costs with distributed system, the distribution network adds about 10 $/MWh to the net cost of energy delivered to the home.

6.0 SUMMARY OF FINDINGS

6.1 REFERENCE SYSTEM STUDY

The objective of Task VII is to determine the technical feasibility and cost-effectiveness of central solar heating plants with seasonal storage (CSHPSS). The results of a broad analytical study of the performance and the economics of CSHPSS systems using a common set of parameters to evaluate various system configurations and components for a number of reference conditions are summarized below. Although the scope of the study was broad, it was necessary to introduce a number of limitations in order to accomplish the study within the resources of the task. The most important restrictions are:

- A limited number of configurations were analyzed.

- Control strategies for each storage type were fixed.

- Cost and performance data were standardized in order to make the task feasible and the results broadly relevant. Note, however, that due to different states of development of technology, the cost performance data can differ considerably from country to country.

- The annualized equivalent cost of auxiliary energy is variable in the study, but all forms of auxiliary were assumed to have the same effective cost.

- The system cost and performance results exclude the distribution network since this influence will be the same for all configurations. In a comparison with conventional, distributed heating systems, however, the distribution network cost must be included.

The findings of the general study are enumerated below.

1. The economic rankings of system configurations depend primarily upon the required distribution temperature and the cost of auxiliary energy and are less sensitive to the climate, total load, and DHW fraction.

2. For low-temperature distribution networks, the economic rankings show a prevalence of systems with unglazed solar collectors and heat pumps, but for high-temperature distribution networks, evacuated collectors without heat pumps predominate.

3. Low-temperature distribution systems with heat pumps offer the lowest unit solar costs--less than 20 $/MWh when a suitable aquifer is available--and can meet about 75 percent of the load from solar.

4. Low-temperature distribution systems without heat pumps are more costly--about 60-70 \$/MWh--but can meet 100% of the load and thus offer greater energy security from curtailment and cost escalation.

5. The most cost-effective plants for use with high-temperature distribution systems employ temperature stratification of the storage volume and use evacuated collectors. The minimum solar cost of these systems is 90-100 \$/MWh.

6. All systems show economies-of-scale because of diminishing unit storage costs and reduced storage heat losses, and show improved cost effectiveness for increasing domestic hot water (DHW) fractions due to the more uniform temporal load. Rock cavern systems exhibit the greatest size dependence and duct systems the least.

7. Collector costs dominate all but the low-solar-fraction, low-temperature systems with heat pumps. The overall collector cost is often twice as large as the storage cost.

6.2 NATIONAL EVALUATIONS

Country and site-specific studies were performed by six of the participants using the system configuration and storage technology deemed most appropriate for the site. These studies used the same methodology employed in the reference system studies but substituted national data, where appropriate, for the performance, cost, and economic parameters. The quantitative results of these studies were presented in Section 4. Additional important findings revealed by these studies are listed below.

- CANADA

  Systems with aquifer storage, heat pumps, and flat plate collectors were evaluated in three Canadian locations. Although the results confirm the reference system findings, the solar systems are not yet competitive because of the low cost of conventional energy in Canada.

- COMMISSION OF EUROPEAN COMMUNITIES (CEC)

  The CEC study employed the reference case parameter set for duct storage except for weather data which was taken for a site near Ispra in northern Italy. The results are very similar to those obtained in the reference system for Copenhagen--except that performance is somewhat higher and costs are somewhat lower because of the 17 percent greater annual insolation at Ispra.

- GERMANY

German experience in thermal energy storage and heat pump technology indicates that costs for those subsystems are substantially higher than the reference case values. These costs, together with the lower insolation in Germany, make solar heating rather unattractive in the current economic climate.

Technically, the study found that the most cost effective pit storage could be designed with minimum insulation.

- THE NETHERLANDS

Duct storage systems were studied. The CSHPSS systems with heat pumps, especially the gas-driven heat pumps, were found the most cost-effective. With a cost of 130 Dfl/MWh (37 $/MWh) including heat pump fuel cost, this is close to the cost of conventional gas boiler heating of 80 Dfl/MWh (23 $/MWh).

The solar cost of the electrical heat pump system comes to 190 Dfl/MWh (54 $/MWh) and non-heat pump system to 260 Dfl/MWh (74 $/MWh). It was found that the cost for a system with heat pumps including heat pump fuel cost is not very sensitive to collector unit cost; for the system without heat pumps, however, the systems cost is very sensitive to the collector unit cost.

- SWEDEN

Results in Sweden show that the development of thermal energy storage in water and of collector technology has reached a level at which systems without heat pumps are competitive with heat pump systems for all solar fractions. Those systems are already nearly competitive with conventional energy systems.

- UNITED STATES OF AMERICA

The performance and cost of the optimized drilled rock storage systems analyzed in the U.S. study are not much different from those of the reference studies or other national evaluations. However, because of the high cost of oil and electricity in the New England region of the U.S., CSHPSS systems offer system cost that are already attractive.

The U.S. study showed that, even without the tax incentives which are currently available, the system unit energy costs for optimized CSHPSS systems are below electricity prices and on a par with oil.

6.3 VARIATION AND VALIDATION STUDIES

- Very little field data from solar systems employing seasonal energy storage are available for validation of computer simulations. However, comparisons of MINSUN results with the limited data available have been encouraging.

- Four years of experience with the MINSUN program and comparisons of its results with the results of more detailed simulation codes such as TRNSYS have led to a high level of confidence in the MINSUN results.

- Some limitations of MINSUN have been noted and their implications examined using other codes (TRNSYS). The ability to model the direct collector-to-load options could increase the system performance prediction by 5 to 10 percent in some systems. Daily storage buffering seems to offer little advantage in stratified storage units, but could be important in duct systems. Pumping power can be significant in aquifer systems. Similarly, collector arrays using modules that are not designed for large scale applications may result in large pumping energy requirements.

- The collector models used in MINSUN for unglazed collectors are inadequate for accurate prediction of energy gains from ambient where sky and wind conditions vary widely.

- Detailed estimates of distribution system costs have not been made, but rough estimates based on the experience in Sweden indicate that distribution will add about 10 $/MWh to the cost of energy delivered to the load.

7.0 GENERAL CONCLUSIONS AND RECOMMENDATIONS FOR TASK VII

7.1 CONCLUSIONS

- CSHPSS can meet a large fraction of the space and water heating load for buildings even in harsh northern climates, and they are already cost-competitive in some locations.

- Solar costs of less than 20 \$/MWh are possible where appropriate aquifers are available and low temperature distribution systems can be used. Solar fractions of as much as 75% can be achieved using unglazed collectors with heat pumps.

- Large solar fractions, more than 80%, can be achieved by systems without heat pumps using stratified energy storage and high performance collectors. Minimum costs for these systems are about 40 \$/MWh (Sweden) for low temperature distribution systems and 70-100 \$/MWh for high temperature distribution systems.

- Systems with heat pumps are generally more economical for solar fractions below about 70 percent. However, results from Sweden show that non-heat pump systems using collector modules designed and developed for large scale applications can match or exceed the cost effectiveness of heat pump systems over the entire range of solar fraction.

7.2 RECOMMENDATIONS FOR FUTURE TASK VII ACTIVITIES

Based on the findings of Subtask II(b), we recommend the following activities be included in the continuation of the IEA work on CSHPSS.

- The generally favorable findings of the study for the economic viability of CSHPSS should be widely reported within the IEA and the solar community.

- Existing and planned CSHPSS systems should be instrumented, monitored, analyzed, and evaluated to verify the method, models, data, and findings of the analyses and to provide a foundation for extension and improvement of the analyses.

- It is also important to continue to obtain more reliable cost data for the various components in the CSHPSS. Costs for the various storage concepts, which are very new at the moment, may decrease drastically in the future. Furthermore, the collector costs are expected to decrease because of increasing production volume and lower costs for installation of large panels.

- The analytical work should continue to explore promising new configurations, to support further design and system development, to verify the preliminary findings, and to validate the analytical methods.

- The analytical tools and procedures developed for the system analysis and parametric study should be used in the evaluation of operating systems. Validated methods and models should be used to re-optimized the design of existing plants using updated knowledge and data, and to simulate the performance and economics of these plants in other locations and under other economic conditions.

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APPENDIX A

COMPILATION OF PERFORMANCE AND ECONOMIC PARAMETERS USED IN REFERENCE STUDIES

The first five pages of Appendix A list the parameters used in the analysis of duct storage systems. Storage-specific parameters for the aquifer and water storage studies follow.

PARAMETERS	VALUES
UMSORT	
Number of Days in each month J, F, M, A, M, J, J, A, S, O, N, D.	31, 28, 31, 30 31, 30, 31, 31 30, 31, 30, 31
Collector Type	0 or 1
Flat Plate (includes Unglazed)	
Evacuated Tube (includes CPC)	
Boes Model	
Madison	0
Copenhagen	0
Collector $\tau\alpha$ Product	
Flat Plate	0.808
Unglazed	0.90
CPC	0.700
Collector Linear Heat Loss Coefficient $W/m^2 K$	
Flat Plate	4.4
Unglazed	15.0
CPC	1.0
Collector Quadratic Heat Loss Coefficient $(W/m^2 K)^2$	
Flat Plate	0.0
Unglazed	0.0
CPC	0.0
Incident Angle Modifier	
Flat Plate	0.1
Unglazed	0.1
CPC	0.05
Tilt Angle °	45 for all collectors
Direction (South = 0)	0
Ground Diffuse Reflectance	0.0
Latitude	
Madison	43
Copenhagen	56
Shift in Solar Time Hour Angle	0
5 Collector Inlet Temperatures (°C)	
Flat Plate	5, 20, 35, 50, 65
Unglazed	0, 10, 20, 30, 40
CPC	20, 40, 60, 80, 100

Outside Temperature below which Space Heating is required (°C)	10
Desired Indoor Temperature	18
ADVANCE	
Start Day Number	136
Number of Days	365
Array Effect Factor	
Flat Plate	0.66
Unglazed	0.70
CPC	0.70
MINSUN	
Options	Case Dependant
Insulation (of pipes)	
Conductivity W/m K	0.04
Earth Temperature °C	10
Diameter Dependent Insulation Coeff.	0.1
Fixed Insulation Thickness	0.02
COLLECTOR	
Option	3
Area	to be optimized

Collector Temperature Control	
- Temperature Increased (°C)	10
- Temperature at which Flow in Collectors increases	10
Maximum Outlet Temperature	97
Normal Flow $kg/s/m^2$	0.005
Maximum Flow $kg/s/m^2$	0.03
Collectors to Storage Network	Load Dependent
Length "	1 GWh 100 m
	10 GWh 315 m
	100 GWh 1000 m
Pipe Diameter m	Load Dependent
	1 GWh 0.2 m
	10 GWh 0.3 m
	100 GWh 0.4 m

DUCT STORAGE	
Volume m^3	to be optimized
Depth of Storage m	"
Number of Boreholes	"
Outer Radius of Boreholes m	0.10
Thickness of Insulation m	to be optimized
Thermal Conductivity of Insulation W/m K	0.05
Option for Insulation Location	2
Side Cover (% of height)	0.100
Distance Surface/Storage m	1
Thermal Conductivity in Storage W/m K	2
Fluid Soil Heat Resistance m^2 K/W	0.05
Maximum Duration - years	2
Maximum Storage Temperature °C	100
Number of Preheating Cycles	2
Maximum Storage Temperature during Preheating °C	30 with heat pump 70 without
Maximum Difference allowed in Preheating °C	2
Start Surface Temperature °C	10
Temperature Gradiant in Ground °C/m	0.03

Number of Soil Layers	1
Thermal Conductivity W/m K	2.0
Heat Capacity J/m^3 K	2.5 10^6
Layer Thickness m	300
Starting Temperatures °C	10 with heat pump 40 without
Maximum Temperature Change allowed between 2 cycles to converge °C	1
PERIOD OF HEATING	
First Day, Last Day	274, 135
HOUSELOAD AND DISTRIBUTION	
Number of Houses	Load Dependent 1 GWH, 20% DHW : 50 10 GWH, 20% DHW : 500 100 GWH, 20 DHW : 5000 % DHW Dependent 0% DHW : 625 20% DHW : 500 50% DHW : 310

Heat Loss Skin Area m^2	350	
K Value W/m^2 K		
Madison	0.583	
Copenhagen	0.707	
Indoor Temperature	18	
Distribution Option 3 way / 4 way	3	
Distribution Control Option		
Return Temperature fixed (1)	1	
Temperature Decreased fixed (2)		
	HT	LT
Minimum Delivery Temperature °C	80	50
Slope Temperature Curve	1.75	0.50
Breakpoint Temperature °C	0	0
Return Temperature °C	60	30
Other Heat Sources w/House	400	
Tapwater Demand w/House	% DHW dependent	
	0% :	0
	20% :	460
	50% :	1840
Distribtion Network Length	0	
Distribution Network Insulation m	0.099	
Distribution Pipe Diameter m	0.099	

HEAT PUMP	
Option - Constant Flow (1)	1
Constant Temperature (2)	
HP Efficiency	Load Dependent
	1 GWH : 0.55
	10 GWH : 0.60
	100 GWH : 0.65
Breakpoint in Efficiency Curve °C	50
Stagnation Point °C	100
Minimum Evaporation Outlet Temperature °C	-5
Minimum COP	1.0
Evaporator Heat Transfer kw/k	to be optimized
Condensor " " "	" (300)
COST	
Collector Cost (1980 US $/m^2$)	
Flat Plate	245
Unglazed	140
CPC	350
Storage Cost Parameter (1980 US $)	
Asymptotic Storage Cost $/m^3$	0.1
Specific Cost of Small Storage $/m^3$	0.2
Small Storage Volume m^3	10,000

Beta	0.1	
Depth Cost Exponent 1/m	1.0	
Number of Ducts Cost Exp.	1.0	
Borehole Cost \$/m	30	
Ground Cost \$/m^2	0	
Insulation Cost \$/m^3	100	
HEAT PUMP COST	HT	LT
Cost of Condensor \$. K/W	0.3	0.2
Cost of Evaporator \$. K/W	0.3	0.2
Cost of HP Motor \$. K/W	0.3	0.2
Reference Condensor Power MW	0.6	
Scale Factor for HP Costs	- 0.3	
DISTRIBUTION COSTS		
Specific Pipe Cost \$/m^2	0	
Linear Pipe Cost \$/m	250	
Cost of Pipe Insulation \$/m^3	0	

AUXILIARY COSTS	
Auxiliary Heat \$/W	0.1
Auxiliary Delivered Heat Cost \$/KWH	0.07
HP Electricity Cost \$/KWH	0.07
Amortisation Period - Years	20
Discounting Rate (real) % per annum	5
Fuel Escalation Rate (real) % per annum	2

STORAGE PARAMETERS: AQUIFER STORAGE

Collector Parameters

Collector Temperature Control	
- Temperature increase (°C)	0
- Temperature at which flow increases (°C)	Optimized
- Maximum outlet temperature (°C)	Equal to temperature at which flow increases
- Normal flow ($kg/s/m^2$)	0.0001
- Maximum flow ($kg/s/m^2$)	0.1

Storage Parameters

Aquifer Size	
- Height (m)	20
- Estimated Maximum Thermal Radius (m)	100
Thermal Conductivity (W/mK)	
- Horizontal	2.0
- Vertical	2.75
Aquifer Heat Capacity (J/m^3K)	$2.5x10^6$
Aquifer Supply Temperature (°C)	Varied (see text)
Maximum Simulation Time (years)	2
Initial Ground Surface Temp. (°C)	10
Vertical Temperature Gradient (°C/m)	0
Number of Layers Above Aquifer	1
Layer Thermal Conductivity (W/mK)	2.75
Layer Heat Capacity (J/m^3K	$1.5x10^6$
Layer Thickness (m)	20
Number of Layers Below Aquifer	1
Layer Thermal Conductivity (W/mK)	2.75
Layer Heat Capacity (J/m^3K)	$3.1x10^6$
Layer Thickness (m)	40

Cost Parameters

Well cost ($/m)	100
Depth Cost Exponent	1
Number of Wells	1
Number of Wells Exponent	1
Reference Flow Rate (m^3/s)	0.03
Well Flow Exponent	0
Equipment Cost ($)	150,000
Equipment Flow Exponent	0
Ground Surface Area (m^2)	0
Ground Cost ($\$/m^2$)	0

Heat Pump Parameters

(TAPW heat pump is specified--no associated cost)

Option	TAPW	House
- Constant Flow (1) - Constant Temperature (2)	1	1
HP Efficiency	0.60	0.60
Break Point in Efficiency Curve (°C)	50	50
Stagnation Point in Efficiency Curve (°C)	100	100
Minimum Evap. Outlet Temperature (°C)	5	5
Minimum COP	1.0	1.0
Evap. Heat Transfer (kW/K)	20 (Low Temp) 10 (High Temp)	optimized
Cond. Heat Transfer (KW/K)	40 (Low Temp) 30 (High Temp)	250 (Madison) 200 (Copenhagen)

STORAGE PARAMETERS: TANK STORAGE

Storage Parameters

Volume (m^3)	optimized
Height (m)	40 (20)*
Number of Segments (minimum 2)	5
Density (kg/m^3)	1000
Ground Temperature (°C)	10
Initial Storage Temperature (°C)	40
Insulation Thickness (m)	40
- Top	0.7
- Wall	0.5
- Bottom	0.4
Thermal Conductivity (W/mK)	
- Top	0.05
- Wall	0.05
- Bottom	0.05
Storage Tank Environment Option (Only Bottom Surface of Tank at Ground Temperature)	1
Concrete Thickness (m)	0

Cost Parameters

- Asymptotic Storage Costs ($/$m^3$)	50
- Specific Cost ($/$m^3$)	90
- Small Size (m^3)	10000
- Beta	0.4
- Alpha	0
- Cost of Insulation ($/$m^3$)	100
- Cost of Ground ($/$m^2$)	1

*Parameter value used for simulations of 50 houses (1 GWh) load

APPENDIX B

CLIMATIC DATA FOR COPENHAGEN AND MADISON (Monthly Averages)

	MADISON			COPENHAGEN		
	Amb. Temp.	Direct Normal Radiation	Global Horizontal	Amb. Temp.	Direct Normal Radiation	Global Horizontal
	°C	kWh	kWh	°C	kWh	kWh
January	-8.3	76	51	-0.6	25	12
February	-6.0	81	71	-1.1	56	33
March	-1.9	130	119	2.6	60	59
April	8.7	107	131	6.6	125	119
May	14.6	139	166	10.6	143	155
June	19.6	139	177	15.7	180	185
July	22.1	155	187	16.4	138	161
August	20.0	154	171	16.7	135	135
September	16.8	124	126	13.7	88	83
October	10.5	93	85	9.2	60	44
November	2.3	67	48	5.0	36	37
December	-3.7	51	37	1.6	19	12
Year	7.9	1318	1370	8.0	1085	1018

*U.S. GOVERNMENT PRINTING OFFICE: 1986-181-180:40156