MASTER

A SECOND GENERATION GROUND COUPLED
SOLAR ASSISTED HEAT PUMP SYSTEM

SIX MONTH PROGRESS REPORT

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FIGURE 2-2: RELATIONSHIP BETWEEN GLAZING TO ABSORBER SPACING AND HORIZONTAL INTERCEPT

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ABSTRACT

The objective of this contract is to investigate the technical and commercial viability of a novel ground coupled, solar assisted heat pump system for residential space heating and cooling applications.

The system consists of a space conditioning load met by a water source heat pump. Sources of heating energy are flat plate solar collectors and flat plate earth heat exchangers. For cooling, the same equipment is used to reject heat to the nighttime ambient and to the earth.

Specific areas of study are solar collector/heat rejector performance, flat plate earth heat exchanger performance, system performance simulations, and commercialization and marketing analysis. Collector/rejector performance, determined by various thermal experiments, is discussed. The design and construction of an experimental site to study ground coupling is discussed. Theoretical analysis is also presented. The performance of the GCSAHP system and conventional alternatives, as determined by simple computer models, is presented and discussed. Finally, the commercial viability of this unique space conditioning system is examined.

This document is a six month progress report on collector/rejector research, ground coupling research, system thermal analysis and commercialization and marketing analysis.

TABLE OF CONTENTS

<u>Section</u>		<u>Page</u>
1.0	INTRODUCTION	1-1
2.0	COLLECTOR/REJECTOR RESEARCH	2-1
	2.1 Introduction	2-1
	2.2 Design Concepts	2-1
	2.3 Experimental Methods	2-2
	2.4 Thermal Performance Results	2-2
	2.4.1 Solar Collection	2-2
	2.4.2 Stall Control	2-3
	2.4.3 Nocturnal Heat Rejection	2-7
	2.5 Conclusions	2-13
3.0	GROUND COUPLING RESEARCH	3-1
	3.1 Introduction	3-1
	3.2 Experiment Design & Construction	3-2
	3.2.1 Objective Of The Field Experiments	3-2
	3.2.2 Description Of Heat Pump And Earth Heat Exchangers	3-3
	3.2.3 Description Of The Site And Soil	3-3
	3.2.4 Instrumentation	3-5
	3.2.5 Plumbing	3-7
	3.2.6 Experimental Plan	3-7
	3.3 Computer Modeling	3-9
4.0	SYSTEM THERMAL ANALYSIS	4-1
	4.1 Introduction	4-1
	4.2 Analytical Methods	4-1
	4.2.1 General	4-1
	4.2.2 Heating Systems Simulation	4-1
	4.2.3 Cooling Systems Simulation	4-4
	4.3 Element Descriptions	4-5
	4.4 Outputs	4-15
	4.5 Conclusions	4-15
	4.6 Recommendations	4-19

TABLE OF CONTENTS

<u>Section</u>		<u>Page</u>
5.0	MARKETING AND COMMERCIALIZATION	5-1
5.1	Existing Products and Energy Sources	5-1
5.2	Market Segmentation	5-6
5.2.1	Geography and Climate	5-6
5.2.2	Type of Residence	5-6
5.2.3	Energy Prices and Availability	5-7
5.3	Emerging Products and Energy Sources	5-7
5.4	FAFCO SAHP Concept	5-8
5.5	Evaluating New Technologies - Economics	5-10
5.6	Customers and Decision Criteria	5-13
5.7	FAFCO SAHP System Simulations	5-14
5.8	SAHP Source Energy Requirements	5-17
5.9	Conclusion	5-22
REFERENCES		R-1

LIST OF FIGURES

<u>Figure</u>		<u>Page</u>
1-1	Ground Coupled Solar Assisted Heat System Schematic	1-3
2-1	Typical Stall Sample Cross Section.	2-4
2-2	Relationship Between Glazing To Absorber Spacing And Horizontal Intercept	2-5
2-3	Hottel-Whillier Thermal Performance of Collector/Rejector With Glazing Spacing Stall Control.	2-6
2-4	Test Apparatus For Forced Air Flow Stall Control Experiments	2-8
2-5	Relationship Between Cooling Air Velocity And Horizontal Intercept.	2-9
2-6	Hottel-Whillier Thermal Performance of Collector/Rejector With Forced Air Flow Stall Control	2-10
2-7	Approximate Relationship Between Heat Removed By Forced Air Flow And Exhauster Power	2-11
2-8	Test Apparatus For Natural Air Flow Stall Control Experiments	2-12
2-9	Heat Loss Factors For Nocturnal Operation Of Glazing Spacing And Forced Air Collector/Rejector Designs	2-14
2-10	Summary Of Heat Loss Factors For Candidate Collector/Rejector Designs	2-15
3-1	Site Plan And Trench Cross Sections Of Ground Coupling Test Field	3-4
3-2	Placement Of Earth Temperature And Moisture Sensors	3-6
3-3	Schematic Piping And Instrument Diagram For Ground Coupling Experiments	3-8

LIST OF FIGURES
(CONT.)

<u>Figure</u>		<u>Page</u>
3-4	Actual And Predicted Earth Temperatures From First Generation Ground Coupling Experiments, Aug, 1979.	3-11
3-5	Soil Temperature 3" From Sources During 14 Day Cyclic Heat Rejection	3-12
3-6	Soil Temperature 3" From Sources During Days 1 To 4 Of 14 Day Cyclic Heat Rejection	3-14
3-7	Soil Temperature Contours After 14 Day Cyclic Heat Rejection	3-15
4-1	System Schematics For Thermal Analysis Of Ground Coupled Solar Assisted Heat Pumps And Conventional Alternatives	4-2
4-2	Cooling COP, 3 Ton Heat Pumps	4-6
4-3	Cooling COP, 4 Ton Heat Pumps	4-7
4-4	Cooling Capacity, 3 Ton Heat Pumps	4-8
4-5	Cooling Capacity, 4 Ton Heat Pumps	4-9
4-6	Heating COP, 3 And 4 Ton Heat Pumps	4-10
4-7	Heating Capacity, 3 Ton Heat Pumps	4-11
4-8	Heating Capacity, 4 Ton Heat Pumps	4-12
4-9	Collector Performance	4-13
4-10	Rejector Performance	4-14
4-11	Storage Volume Sensitivity, Heating Mode	4-17
4-12	Collector Area Sensitivity, Heating Mode	4-18
5-1	Type Of Heating In Occupied Housing Units	5-2
5-2	Percentage Of New Homes In U.S. Using Different Types Of Heating Fuels	5-3
5-3	Residential Unitary Air Conditioning Shipments Not Including Heat Pumps	5-4
5-4	Residential Heat Pump Shipments	5-5

LIST OF FIGURES
(CONT.)

<u>Figure</u>		<u>Page</u>
5-5	Cumulative Cash Flow Profiles	5-12
5-6	California Customer Segments Acceptance Criteria	5-15
5-7	Economic Assumptions For SAHP System Evaluations	5-18
5-8	Typical Installed Capital Costs For SAHP Systems In Fresno, Ca. (Before Tax Credits)	5-19
5-9	Cost Estimate Worksheet Example	5-20
5-10	Source Energy Requirements	5-21
5-11	Market Acceptance Issues	5-23

LIST OF TABLES

<u>Table</u>		<u>Page</u>
4-1	Comparison Of Heating Load Calculation Methods	4-3
4-2	Comparison Of Cooling Load Calculation Methods	4-3
4-3	Results Summary Of Systems Thermal Analysis	4-16

The objective of this contract is to investigate the technical and commercial viability of a novel ground coupled, solar assisted heat pump system for residential space heating and cooling applications.

The space conditioning equipment market in the U.S. consists of all equipment used to heat or cool occupied living spaces. It is a highly diverse market with well known technologies and well established firms competing in national and regional markets. The new product concept proposed by FAFCO represents a residential space conditioning system using existing heat pump technology supplemented by solar energy and earth heat exchange. This solar assisted heat pump (SAHP) system is one of several emerging technologies challenging the existing products and market structure. The SAHP systems proposed in this study appear to show promise as future competitive space heating systems for new residences.

The system under investigation is composed of four distinct elements interconnected and operated to meet a residential space conditioning load. See Figure 1-1. A high performance water to air heat pump, optimized for solar assist, supplies heated or cooled air to the load. A storage tank contains the mass of fluid acting as an energy source or sink for the heat pump. In the heating mode, flat plate polymeric collectors supply solar energy to storage for direct or heat pump assisted space heating. Planar polymeric earth heat exchangers function as a back up energy source when solar input is insufficient, thus avoiding electric resistance heat. In the cooling mode, heat moved by the heat pump from the conditioned space to storage is rejected either to the nighttime ambient using the solar collectors or to the earth. An important feature of this system is that the four basic elements are utilized all year, for both heating and cooling operation.

The heating performance of solar assisted heat pump (SAHP) systems has been the subject of many investigations. This research

is distinguished by its emphasis on efficient cooling mode operation. Unique features of the approach to optimized cooling performance are the use of inexpensive, high heat loss polymeric solar collectors and planar earth heat exchangers for efficient heat rejection. Specific thermal experiments and detailed computer models have been designed to determine the potential performance of these system components. System simulations have been performed to establish desired performance.

This document is a six month progress report on collector/rejector research, ground coupling research, system thermal analysis and commercialization and marketing analysis.

FIGURE 1-1: GROUND COUPLED SOLAR ASSISTED HEAT PUMP SYSTEM SCHEMATIC

2.0 COLLECTOR/REJECTOR RESEARCH

2.1 Introduction

The flat plate polymeric solar collector/heat rejector must perform two functions in the space conditioning system. Efficient solar energy collection is required for heating, and effective nocturnal heat rejection is necessary for cooling. Depending upon the thermal efficiency of the candidate design, a third function, stall control, may also be required. A method of stall control may be needed to prevent excessive absorber plate temperatures when solar energy is available but not utilized. Simplicity, reliability, and low cost are also essential attributes of the collector/rejector.

2.2 Design Concepts

Three basic design concepts for a collector/rejector with stall control have been investigated. These are:

1. Glazing Spacing Control: Transparent glazing is in contact with absorber plate for stall control and heat rejection. Glazing is spaced off absorber for solar collection.
2. Forced Air Flow Control: Heated air between glazing and absorber is removed continuously by a small electric powered exhaust fan for stall control and heat rejection. Air flow is stopped for solar collection.
3. Natural Air Flow Control: Transverse slots in the glazing are opened to allow heat removal by natural convection. The slots are closed by a transparent cover for solar collection.

2.3 Experimental Methods

Research has been directed principally toward the determination of the thermal performance of the candidate designs in the solar collection, stall control and heat rejection modes. Three methods have been used to determine thermal performance. The simplest method consists of building a "stall sample." A stall sample is a 12" square panel body section of a candidate design with insulated edges. The X-intercept of a Hottel-Whillier performance curve can be estimated by allowing the sample to "stall" (no heat removed by working fluid) and measuring absorber plate temperature, ambient temperature, and normal insolation. The Y-intercept can be estimated by the known $Fr \tau \alpha$ product. A straight line drawn between these points is a good approximation of the performance curve. The slope of the curve is the heat loss factor, U_L .

A second method consists of building full scale prototypes and generating performance curves from stall data as described above. This method is appropriate for the air flow control methods where average performance over the entire absorber surface cannot be estimated by data from one point on the surface.

The third method is applicable to nocturnal heat rejection performance. Full scale prototypes are operated at night with working fluids at different temperatures. Knowing the flow rate and temperature drop for the fluid, and ambient conditions, instantaneous U_L can be calculated at several points of operation. Another less precise method is to cool a large tank over several hours via an array of rejectors. Knowing the initial and final tank temperature and average ambient conditions, U_L can be extracted from the governing first order differential equation.

2.4 Thermal Performance Results

2.4.1 Solar Collection

For all candidate designs, predicted solar collection per-

formance is similar and typical of a well-insulated, single glazed flat plate collector with a non-selective surface absorber. A Y-intercept of about 0.80 would result from the $Fr \tau \alpha$ product of the candidate materials. An X-intercept of about 0.55 is achieved by using back insulation with an R value of 7 and minimizing heat loss by air leakage.

2.4.2 Stall Control

Glazing Spacing Stall Control

It is possible, by varying the spacing between glazing and absorber surface, to affect solar collector performance and achieve effective stall control. The purpose of this testing was to determine the relationship between glazing spacing and stall intercept.

Six glazed, insulated stall samples were constructed with glazing spacings of 0", 1/32", 1/8", 1/4", 1/2" and 1". See Figure 2-1 for a typical cross section. One unglazed sample was also constructed. Under good solar conditions, temperature readings were taken. Four sets of data were recorded.

It can be seen in Figure 2-2 that the stall intercept increases very rapidly between 0" and 1/4" spacing and rises more slowly to a peak of about 0.55 at 1". It is apparent that the glazing must maintain good contact with the absorber surface to achieve effective stall control. Any gaps will cause undesirable hot spots. Figure 2-3 is the Hottel-Whillier representation of this collector's thermal performance.

Forced Air Flow Stall Control

Heated air between the glazing and the absorber surface can be effectively removed with an exhauster, thus effecting stall control or heat rejection. The purpose of this testing was to 1) determine the relationship between air flow volume, velocity and stall intercept and 2) determine approximate power requirements.

FIGURE 2-1: TYPICAL STALL SAMPLE CROSS SECTION

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FIGURE 2-3: HOTTEL-WHILLIER THERMAL PERFORMANCE OF COLLECTOR/REJECTOR WITH GLAZING SPACING STALL CONTROL

Figure 2-4 shows the test apparatus. From the test data, Figures 2-5 and 2-6 were plotted. The stall control range in Figure 2-6 is based on heat removal at air velocities of 5-15 ft/sec. It can be seen that substantial heat removal can be accomplished at low velocities (5 ft/sec) and hence low exhaustor power consumption. A very approximate power consumption curve has been included as Figure 2-7. A 12 watt exhaustor generated the 5 ft/sec velocity over a 4'x10' absorber plate.

Natural Air Flow Stall Control

A marginally effective method for natural convection stall control is to open several transverse slots in the glazing to vent the air space. Opening only the ends of the air space on the top and bottom of the collectors was found to be insufficient. Solar heat gain exceeds heat removal by an amount such that the top half of the collector reaches temperatures very close to those that occur when convection is suppressed. Opening vents at several locations along the collector body, as shown in Figure 2-8 was found to be necessary.

2.4.3 Nocturnal Heat Rejection

The purpose of performing nocturnal heat rejection experiments is to verify that the heat loss factor from the Hottel-Whillier performance curve is applicable to a simple heat rejection equation of the form $Q = U_L A(T_f - T_a)$ in the absence of solar radiation. It was found that this is a valid concept.

Two types of experiments were done. In the first set of experiments, collector/rejectors of the variable glazing spacing type and forced air flow type were each used to cool a small tank of heated water. A FAFCO IV collector ($F_r \tau \alpha = 0.75$, $U_L = 2.0$) was also tested. At different tank temperatures, instantaneous heat rejection values were measured using a flow meter and inlet and outlet temper-

FIGURE 2-4: TEST APPARATUS FOR FORCED AIR FLOW STALL CONTROL EXPERIMENTS

FIGURE 2-6: HOTTEL-WHILLIER THERMAL PERFORMANCE OF COLLECTOR/REJECTOR WITH FORCED AIR FLOW STALL CONTROL

FIGURE 2-7: APPROXIMATE RELATIONSHIP BETWEEN HEAT REMOVED BY FORCED AIR FLOW AND EXHAUSTER POWER

FIGURE 2-8: TEST APPARATUS FOR NATURAL AIR
FLOW STALL CONTROL EXPERIMENTS

ature sensors. The results for the three collector/rejector types are shown in Figure 2-9. Agreement with U_L determined by other methods is good.

In a second type of experiment, a large tank was cooled over several hours with a bank of 10 FAFCO IV collectors. Fitting the governing first order differential equation to the plot of (tank temp/ambient temp) vs. time, a U_L of 1.75 was obtained. A similar experiment performed at Oklahoma State yielded a U_L of 2.14.

2.5 Conclusions

1. All three experimental methods (daytime operation of stall samples and full scale prototypes and nighttime operation of full scale prototypes) yield consistent values for U_L for all candidate designs.
2. The thermal performance of the air flow control design is best, the design is straight forward, but it requires electrical power and it is not failsafe.
3. Glazing spacing control design gives satisfactory thermal performance, but it is very sensitive to small gaps and is a more difficult design to execute.
4. Heat removal by natural convection between absorber and glazing is only marginally effective.

Figure 2-10 is a summary of the heat rejection factors for the candidate designs. It is seen that, unless electrical energy is consumed, the unglazed panel is the best heat rejector. This is not very surprising. The system thermal analysis section of this report will address the question of how much heat rejection via the collector/rejector is needed.

FIGURE 2-9: HEAT LOSS FACTORS FOR NOCTURNAL OPERATION OF GLAZING SPACING, FORCED AIR, AND FAFCO IV COLLECTOR/REJECTOR DESIGNS

FIGURE 2-10: SUMMARY OF HEAT LOSS FACTORS FOR CANDIDATE COLLECTOR/REJECTOR DESIGNS

3.0 GROUND COUPLING RESEARCH

3.1 Introduction

An integral component of the space conditioning system is the earth heat exchanger, also known as ground coupling. In the heating mode, the earth is a back up source to the heat pump when solar input to storage is insufficient. A ground coupled heat pump can supply heating energy at a COPH of 2 to 2.5, rather than at 1.0 with electric resistance heat. The ground is, in effect, a solar collection and storage device. In the cooling mode, the earth acts as a heat sink for some of the energy rejected by the heat pump to storage. (The remainder is rejected nocturnally by the collector/rejectors). COPC improvement over air/air heat pumps or air conditioners is possible, since the ground is usually cooler than ambient when there is a cooling load.

Conduction is the heat transfer mechanism between a closed loop heat exchanger and earth. Because soil is a relatively poor conductor, heat transfer rates are limited, and thus large heat exchanger areas are needed. The most usual heat exchanger geometry is a very long (more than 1,000 ft., typically) 1-1/2" plastic pipe buried at a depth of 3 to 4 ft. An alternative method, discussed here, is to bury a much shorter length, perhaps 200 to 300 ft., of 3 ft. wide flat plate heat exchanger at the same centerline depth. Using this flat plate heat exchanger concept, it should be possible to obtain substantially more energy exchange per unit length of trench, due to the much greater surface area of the plate in contact with the soil. Cost reductions may result by a decrease in trenching cost more than offsetting the more costly heat exchanger. Further, less land area will be needed with the planar approach.

The goals of this research are to:

1. Design, construct, and run a set of ground coupling experiments in which the heat exchange performance of pipes and plates can be compared directly.

2. Compare experimental results to results of computer models, and make adjustments to the models where appropriate.
3. Use these validated computer models to predict short term and long term ground coupling performance.
4. Determine if plate heat exchangers offer a cost and/or a performance advantage over pipes.
5. Establish sizing guidelines for ground coupling arrays.

The status of the ground coupling research program is as follows:

1. An experimental site has been selected, all equipment has been purchased and received, construction is complete, and equipment installation and check-out is complete. An experimental plan is written. Experiments will start as soon as a utility power connection is made.
2. Steady state and transient finite element heat transfer analysis has been completed for both the pipe and plate geometries.

3.2 Experiment Design & Construction

3.2.1 Objective of the Field Experiments

The primary objective of the ground coupling field experiments is to obtain an in-depth understanding of steady state and transient heat flow around buried pipes and plates. Data from the experiments will be compared to the results of a two-dimensional finite element conduction model (HEAT) and a TRNSYS compatible ground coupling model (GROCS). It is hoped that agreement will be suf-

ficiently good so that, in most cases, relatively fast and inexpensive computer simulations can replace costly and time consuming field experiments.

The instrumentation and experimental plan are designed to yield accurate measures of energy flows to and from the ground coupling fields and also to yield detailed soil temperature and moisture profiles at representative cross-sections of each field. Such information will allow straight forward comparison with HEAT and GROCS.

3.2.2 Description of Heat Pump and Earth Heat Exchangers

The two independent ground coupling test fields are driven by energy inputs and extractions from a one ton (nominal) water to air heat pump. Rated heating capacity at ARI Standard Conditions is 17,800 BTUH at a COPH of 3.0. ARI rated cooling capacity is 12,000 BTUH at a COPC of 2.3. For the purposes of the experiments, about 11,000 BTUH can be extracted from the earth, and about 16,000 BTUH can be rejected to the earth.

The heat exchanger configurations to be studied consist of 1) six planar polymeric heat exchangers, each 3' wide x 20' long, plumbed in series, and 2) approximately 360 lineal feet of 1" plastic pipe in a serpentine arrangement. Both types of heat exchangers are buried in 1 ft. wide trenches at a centerline depth of 3-1/2 ft. A sand backfill was used around the pipes and plates. Figure 3-1 shows the field geometry and trench cross sections.

3.2.3 Description of the Site and Soil

The site is located near Stanford University, at the base of the foothills between San Francisco Bay and the Pacific Ocean. The elevation is approximately 50 feet. The earth surface is flat, with no trees or underbrush, and slopes very gently to a 10' deep ravine 200 yards away. The ravine is dry except during rainy periods when it is filled with runoff from the foothills. The soil is typical of the area.

FIGURE 3-1: SITE PLAN & TRENCH CROSS SECTIONS OF GROUND COUPLING TEST FIELD

Five 10 ft. deep borings were made at the site in August. The first 2 ft. were hard, brown, dessicated clay. From 2 ft. to 10 ft., the soil transitions from a brownish green moist medium plasticity silty clay to a greenish yellow non-plastic silty sand. Laboratory soil classification tests have determined the soil liquid limit, plastic limit, natural water content, and particle diameter distribution. Sand core density tests were performed to obtain in situ density. Conductivity measurements are presently in progress. For the sand backfill, a grain size analysis has been performed.

3.2.4 Instrumentation

The measurements of primary interest are soil temperatures, soil moisture content, instantaneous and integrated energy flows to and from the earth, and heat exchanger fluid temperatures. All data except soil moisture content will be read onto magnetic tape by an automatic data acquisition system. The data will be reduced by a microcomputer.

Soil temperature measurements are made by thermocouples buried in plastic pipes packed with sand. The placement of the thermocouple arrays around the pipe and plates is shown in Figure 3-2. The purpose of such arrays is to track the spacial temperature distribution in a single representative plane perpendicular to the heat exchanger axis. This allows very detailed heat transfer studies.

Energy flows are measured by two independent systems. The first system is simply a commercially available BTU meter, which integrates BTU's and gallons. The second system is composed of a target type flow meter and two platinum RTD's. The outputs of these instruments are logged on tape, and the computation of energy flows is done by the microcomputer. Inline flow and temperature indicators are included for operator convenience and manual data audits.

Much care was taken to select a soil moisture sensing system that would yield data of sufficient accuracy for engineering purposes. Most systems use moisture absorbing sensors composed of

FIGURE 3-2: PLACEMENT OF EARTH TEMPERATURE & MOISTURE SENSORS

nylon, gypsum, or fiberglass that exhibit a change in resistance or capacitance with moisture content. Such systems are sensitive to soil properties and placement, and exhibit large hysteresis during moisture changes. They are not considered to be very accurate. The selected system utilizes sensors which incorporate the surrounding soil as part of a resonant electrical circuit. Frequency shifts due to changes in moisture content, in combination with a simply generated calibration curve, yield a reportedly stable and accurate measure of volumetric soil moisture.¹

Figure 3-3 is a schematic diagram of the piping and instrumentation for the ground coupling experiments.

3.2.5 Plumbing

As shown in Figure 3-3, the plumbing has been designed for maximum flexibility. A 350 gallon tank has been included to provide thermal inertia and a buffer between the heat pump and the earth. This arrangement allows good field inlet water temperature control. The tank can be bypassed, if desired, to connect the heat pump directly to either field. In both the pipe and plate fields, isolation and bypass valves have been included. This allows active pipe lengths of 120, 240 or 360 feet, and either four or six active plates to be used. The pipe and plate fields cannot be run simultaneously.

3.2.6 Experimental Plan

These experiments are designed to directly compare the heat exchange effectiveness of the plate and pipe geometries, and to validate HEAT and GROCS as accurate models for these two ground coupling methods.

Initially, identical experiments of fairly short duration will be run sequentially in the plate and pipe fields. The storage tank will be cooled to 45°F, then held there for one week while heat is extracted from the earth and rejected to ambient by the heat pump.

¹ Selig, E.T., Wobscholl, D.C., et. al., "Capacitance Sensor For Soil Moisture Measurement," Transportation Research Board Record 532, Wash., D.C., pp. 64-76.

FIGURE 3-3: SCHEMATIC PIPING & INSTRUMENT DIAGRAM FOR GROUND COUPLING EXPERIMENTS

The earth temperature history resulting from these two experiments will be compared to transient HEAT simulations driven by the measured heat extractions at suitable time steps.

Second, experiments of longer duration designed to more closely mimic a real load profile will be run. In these experiments the heat pump will be cycled on and off every 12 hours, for a two week period. Again, the resulting earth temperature history will be compared to transient HEAT simulations.

A special GROCS configuration for flat plate ground coupling has been integrated with TRNSYS as part of the Thermal Analysis task area. This model will be driven by heat flows from the field experimental data to determine the ability of GROCS to model the planar configuration.

During the cooling season, a similar set of heat rejection experiments will be run. After baseline data is collected for both fields as above, the effect of adding moisture to the plate field via buried sprinkler pipes will be studied. Enhanced heat transfer may be possible by restoring soil moisture driven from the vicinity of the planar heat source by temperature gradients.

3.3 Computer Modeling

In order to understand the relative heat exchange effectiveness of pipes and plates in some detail, simulations using the HEAT model were performed.

HEAT is a finite element program able to solve two-dimensional heat conduction problems. A variety of problems are easily modeled using the many capabilities of the program. Steady state or transient analysis is possible. Boundary conditions can be specified as any combination of temperatures or heat flows applied at nodes or through heat generating elements, and can be varied with time. Any number of material types (e.g. sand, clay and concrete) can be

specified. HEAT has been used to solve earth heat transfer problems such as heat generation of underground utility cables.

An example of the ability of HEAT to simulate a transient heat transfer problem is shown in Figure 3-4. The experimental data was generated at the first generation FAFCO ground coupling experiment. For four days, seven hours per day for four consecutive days, heat was rejected to the earth at the rate of 400 BTUH/lineal ft. of trench. On the first attempt, HEAT generated the dashed lines. No effort was made to adjust the model inputs to obtain a better match.

To compare pipes to plates, a steady state heat rejection analysis was initially performed. In the case of a 3 ft. wide plate and pipe at the same centerline depth and same temperature, the continuous steady state heat transfer rate of the plate is about twice that of the pipe. Given the much greater surface area of the plate, a factor of two is disappointingly low. A more accurate transient analysis, in which thermal capacitive as well as resistive effects are significant, was next undertaken.

The transient problem is an attempt to realistically simulate an air conditioning load met with a water source heat pump. The problem is as follows: A pipe source and plate sources 2,3 and 4 feet wide are at the same centerline depth at the same initial conditions. Heat is rejected at a constant rate for 12 hours on, 12 hours off for 14 days, at such a rate that at day 14 the soil temperature 3 inches from each source is 100°F at the end of the "heat on" period.

Figure 3-5 is a plot of soil temperature 3 inches from the source on the centerline vs. time for 14 days. The following heat generation rates yield 100°F soil at day 14.

Pipe:	75 BTUH/lineal ft (of trench)
2' Plate:	110 BTUH/lineal ft.
3' Plate:	160 BTUH/lineal ft.
4' Plate:	195 BTUH/lineal ft.

FIGURE 3-4: ACTUAL & PREDICTED EARTH TEMPERATURES FROM FIRST GENERATION
GROUND COUPLING EXPERIMENTS, AUG., 1979.

LEGEND

POINT SOURCE, 75 BTU/HR-FT
 2 FT PLATE, 110 BTU/HR-FT
 3 FT PLATE, 160 BTU/HR-FT
 4 FT PLATE, 195 BTU/HR-FT

Figure 3-5: SOIL TEMPERATURE 3" FROM SOURCE DURING 14 DAY CYCLIC HEAT REJECTION

An interesting feature of this solution is apparent in Figure 3-6, which shows the first four days of Figure 3-5 in more detail. There is a much lower average soil temperature around the plates during the heat rejection period, which would yield higher cooling COP's. Alternatively, to achieve equal soil temperatures in a shorter time period, the heat flux from the plates must be increased further. For example, if the goal were equal soil temperatures after only four days of cyclic operation, the pipe heat rate would be about 85 BTUH/lineal ft., while the 3 ft. plate could discharge energy at the rate of about 250 BTUH/lineal ft.

Soil temperature profiles at the end of the "on" time on day 14 are shown in Figure 3-7.

LEGEND

POINT SOURCE, 75 BTU/HR-FT
 2 FT PLATE, 110 BTU/HR-FT
 3 FT PLATE, 160 BTU/HR-FT
 4 FT PLATE, 195 BTU/HR-FT

Figure 3-6: SOIL TEMPERATURE 3" FROM SOURCES DURING DAYS 1 TO 4 OF 14 DAY CYCLIC HEAT REJECTION

Figure 3-7: SOIL TEMPERATURE CONTOURS AFTER 14 DAY CYCLIC HEAT REJECTION

4.0 SYSTEM THERMAL ANALYSIS

4.1 Introduction

The purpose of this study is to estimate and compare the thermal performance of ground coupled solar assisted heat pump (GCSAHP) space conditioning systems to more conventional alternatives in three geographic areas. The alternatives are 1) Fossil Fuel Heating/Electric Cooling, 2) Air/Air Heat Pump and 3) Solar Heating/Electric Cooling. System 4, the FAFCO GCSAHP system, is analyzed using medium and high efficiency heat pumps, and using glazed and unglazed solar collectors/ heat rejectors. System schematics are shown in Figure 4-1.

The results of this study yield approximate system sizing and reasonably accurate annual energy usage information. It should be recognized that this is a "first cut" analysis using simple modeling methods. The TRNSYS model will be used to perform a similar thermal analysis in much greater detail.

4.2 Analytical Methods

4.2.1 General

Annual heating and cooling loads are calculated by SOLCOST and SAHP COOL, respectively. For comparison, Table 4-1 shows the results of heating load calculations by accepted ASHRAE methods. Table 4-2 contains similar information for cooling loads.

4.2.2 Heating Systems Simulation

The solar systems performance is determined using SOLCOST. Given site, collector tilt, storage mass, collector performance, heat pump performance, and house UA, SOLCOST calculates the solar heating fraction. The heating load used is based on "average" weather data and calculated on an hourly basis for one day in the month. The load for this one day is then multiplied by the number of days in the month to arrive at the total monthly load. The system performance is

System 1: "Conventional"

System 2: Air/Air Heat Pump

System 3: "Conventional" Solar

System 4: FAFCO Solar Assisted Heat Pump

System 4A: Medium COP Heat Pump, Glazed Panels

System 4B: High COP Heat Pump, Glazed Panels

System 4C: High COP Heat Pump, Unglazed Panels

Figure 4-1: SYSTEM SCHEMATICS FOR THERMAL ANALYSIS OF GROUND COUPLED SOLAR ASSISTED HEAT PUMPS AND CONVENTIONAL ALTERNATIVES

	Degree Day Method 10 ⁶ BTU	Bin Method 10 ⁶ BTU	SOLCOST Method 10 ⁶ BTU
Fresno, Ca. UA _H =720	45.2	54.5	46.2
Fort Worth, Tx UA _H =720	41.6	44.0	41.4
Wash., D.C. UA _H =720	73.2	75.5	86.5

Note: UA_H includes heat losses by conduction and infiltration, and heat gains by solar and internal sources.

Table 4-1: Comparison of Heating Load Calculation Methods

	Degree Day Method 10 ⁶ BTU	Bin Method 10 ⁶ BTU	SAHPCOOL Method 10 ⁶ BTU
Fresno, Ca. UA _C =1200	48.1	53.9	48.8
Fort Worth, Tx. UA _C =1002	61.3	70.5	62.0
Wash., D.C. UA _C =1200	27.1	37.0	37.8

Note: Cooling load in Fresno and Washington are based on 1.0 air change per hour. Loads in Fort Worth are based on 0.5 air changes per hour. UA_C includes heat gains by conduction, infiltration, solar and internal sources.

Table 4-2: Comparison of Cooling Load Calculation Methods

also calculated on an hourly basis, but for an average "sunny" day and an average "cloudy" day. These days are summed based on the percent sunshine for the month in question. This is done for every month of the year, with monthly and annual results reported.

SOLCOST does not include ground coupling, but a manual adjustment is made to the results. It is assumed that, instead of operating on resistance heat backup with a COP of 1, it operates on ground coupling with a COP of 2.5. To calculate system energy consumption, the auxiliary total is divided by 2.5 then added to the compressor energy consumption. Another 410 KWH per year is added to account for pump energy usage.

For the non-solar systems, two methods are used to calculate energy consumption. For the combustion furnace and resistance heater, the ASHRAE modified degree day method is employed (see ASHRAE Systems 1976, Sec. 43.8). For the air to air heat pump system, the ASHRAE Bin method is used (see ASHRAE Systems 1976, Sec. 43.10). Sources for the Bin weather data are Air Force Engineering Weather Data and ASHRAE Systems. The final results are within 15% of the ASHRAE methods, and are scaled to the calculated SOLCOST load. This is done so that all of the systems will operate with the same load.

For all heat pump systems, the Seasonal Performance Factor for Heating, SPFH, is defined as the heating load divided by the purchased energy. A higher number indicates greater energy savings.

4.2.3 Cooling Systems Simulation

Simulation cooling loads are generated through independent calculations of transmission, infiltration and solar gains. Transmission loads are calculated using average cooling degree days and the transmission UA calculated as 698 BTU/HR. °F. The infiltration model calculates the energy required to bring a specified amount of outside air to the inside air conditions of 75°F and 50% relative humidity. The solar contribution is based on horizontal surface

radiation transformed to radiation on a vertical surface facing south. The solar flux is assumed to act on an unshaded window area equal to 3% of the floor area. After cooling loads are generated for an "average" day in each month, the program reduces the loads and average weather data to hourly data. For the GCSAHP system, an hourly energy balance is generated for the house, heat pump, storage tank, solar collectors and ground coupling panels, if included. The hourly approach allows the tank temperature and equipment performance points to fluctuate. Important dynamics of the system are ignored if the energy balance is not done hourly.

The non-solar cooling systems employ either an air to air heat pump or a vapor compression air-conditioner. A modified version of SAHP COOL, known as AIRCOND, computes cooling load and energy consumption using the same method as SAHP COOL, but with air source equipment. Average monthly weather data is converted to hourly information and energy consumption is determined.

For all systems, the Seasonal Performance Factor for Cooling, SPFC, is defined as the cooling load divided by the purchased energy. A higher number indicates greater energy savings.

4.3 Element Descriptions

Detailed performance data for all elements is given in Figures 4-2 to 4-10.

House: (All systems)	The house is a single story residence with a floor area of 2400 sq. ft. One wall faces due south. 30% of the 480 sq. ft. of glazing faces south. Total transmission UA is 698 BTU/HR/°F.
Fueled Furnace: (System 1)	70% efficiency, 20% oversized.
Air Conditioners: (Systems 1 & 3)	3 Ton Carrier 50YH036 4 Ton Carrier 50YH048

Figure 4-2: COOLING COP 3 TON HEAT PUMPS

LEGEND

- FEDDERS WATER/AIR HEAT PUMP
- COMMAND-AIRE WATER/AIR HEAT PUMP
- CARRIER AIR/AIR HEAT PUMP
- △ CARRIER AIR CONDITIONER

Figure 4-3: COOLING COP 4 TON HEAT PUMPS

LEGEND

- FEDDERS WATER/AIR HEAT PUMP
- COMMAND-AIRE WATER/AIR HEAT PUMP
- CARRIER AIR/AIR HEAT PUMP
- △ CARRIER AIR CONDITIONER

Figure 4-4: COOLING CAPACITY 3 TON HEAT PUMPS

LEGEND

- FEDDERS WATER/AIR HEAT PUMP
- COMMAND-AIRE WATER/AIR HEAT PUMP
- CARRIER AIR/AIR HEAT PUMP
- △ CARRIER AIR CONDITIONER

Figure 4-5: COOLING CAPACITY 4 TON HEAT PUMPS

LEGEND

- FEDDERS WATER/AIR HEAT PUMP
- COMMAND-AIRE WATER/AIR HEAT PUMP
- CARRIER AIR/AIR HEAT PUMP
- △ CARRIER AIR CONDITIONER

Figure 4-6: HEATING COP 3 AND 4 TON HEAT PUMPS

LEGEND

- FEDDERS WATER/AIR HEAT PUMP
- COMMAND-AIRE WATER/AIR HEAT PUMP
- CARRIER AIR/AIR HEAT PUMP

Figure 4-7: HEATING CAPACITY 3 TON HEAT PUMPS

LEGEND

- FEDDERS WATER/AIR HEAT PUMP
- COMMAND-AIRE WATER/AIR HEAT PUMP
- CARRIER AIR/AIR HEAT PUMP

Figure 4-8: HEATING CAPACITY 4 TON HEAT PUMPS

LEGEND

- FEDDERS WATER/AIR HEAT PUMP
- COMMAND-AIRE WATER/AIR HEAT PUMP
- CARRIER AIR/AIR HEAT PUMP

Figure 4-9: COLLECTOR PERFORMANCE

Figure 4-10: REJECTOR PERFORMANCE

Air/Air Heat Pumps (System 2)	3 Ton Carrier 38RQ034
	4 Ton Carrier 38RA046
Water Source Heat Pumps: (System 4)	3 Ton Command-Aire SWP-300
	4 Ton Command-Aire SWP-400
	3 Ton Fedders Compression Furnace
	4 Ton Fedders Compression Furnace
Solar Panels:	System 3: Revere single glazed selective surface.
	System 4: FAFCO IV or FAFCO pool panel.

4.4 Outputs

Table 4-3 is a summary of the results. For systems 1-4 in all three cities, the table shows annual heating and cooling loads, solar fraction where applicable, purchased energy, seasonal performance factors for vapor compression equipment, and components sizing. Monthly totals are available, but are not included in this report. For the GCSAHP system, the sensitivity of solar fraction to storage size is shown in Figure 4-11, and to collector area in Figure 4-12.

4.5 Conclusions

1. A large fraction of annual heating and cooling energy can be saved by the GCSAHP system over the other systems studied.
2. A high COP heat pump is essential if these savings are to be realized.
3. High efficiency solar panels (high x intercept) do not appear to offer any advantage in the heating mode, and offer none in the cooling mode. The unglazed pool

SYSTEM	CITY	HTG LOAD 10 ⁶ BTU	SOLAR ^a (SHF)	HEATING		HEAT PUMP PERF (SPFH)	COOLING		COMPONENT SIZING AND NOTES	
				PURCHASED ENERGY 10 ³ KWH (10 ⁶ BTU)			COOLING LOAD 10 ⁶ BTU	PURCHASED ENERGY 10 ³ KWH		
				ELEC	FOSSIL					
1 CONVENTIONAL	FRESNO	46.24		13.5	90.8		48.8	4.91	2.91	3 TON CARRIER AIR CONDITIONER
	FT. WORTH	41.41	N/A	12.1	81.7	N/A	62.0	6.57	2.76	3 TON CARRIER AIR CONDITIONER
	WASH. DC	86.46		29.3	162.1		37.8	3.55	3.12	3 TON CARRIER AIR CONDITIONER
2 AIR/AIR HEAT PUMP	FRESNO	46.24		4.9	2.75		48.8	5.39	2.65	3 TON CARRIER AIR/AIR H. PUMP
	FT. WORTH	41.41	N/A	4.5	2.68		62.0	6.87	2.64	3 TON CARRIER AIR/AIR H. PUMP
	WASH. DC	86.46		10.3	2.60		37.8	3.97	2.79	3 TON CARRIER AIR/AIR H. PUMP
3 50% NOM. SHF REVERE	FRESNO	46.24	.508	7.01			48.8	4.91	2.91	COOLING HEATING SEE SYSTEM 1 11 COLL. 380GAL
	FT. WORTH	41.41	.527	6.11	N/A		62.0	6.57	2.76	12 COLL. 415GAL
	WASH. DC	86.46	.516	12.70			37.8	3.55	3.12	28 COLL. 968GAL
3 75% NOM. SHF REVERE	FRESNO	46.24	.765	3.57			48.8	4.91	2.91	COOLING HEATING SEE SYSTEM 1 24 COLL. 820GAL
	FT. WORTH	41.41	.775	3.12	N/A		62.0	6.57	2.76	22 COLL. 752GAL
	WASH. DC	86.46	.754	6.62			37.8	3.55	3.12	51 COLL. 1778GAL
4A COMMAND-AIRE FAFCO IV	FRESNO	46.24	.668	3.40	3.98		48.8	6.17	2.32	3 TON HP 1000GAL 8 COLL. 1440 FT ² GC
	FT. WORTH	41.41	.672	3.47	3.50		62.0	*6.87	*2.64	4 TON HP 1000GAL 8 COLL. 1800 FT ² GC COMP OFF 217 HRS.
	WASH. DC	86.46	.640	6.88	3.68		37.8	4.36	2.54	3 TON HP 1000GAL 23 COLL. 1800 FT ² GC
4B FEDDERS FAFCO IV	FRESNO	46.24	.822	2.32	5.84		48.8	4.62	3.09	3 TON HP 1000GAL 9 COLL. 1200 FT ² GC
	FT. WORTH	41.41	.827	2.18	5.67		62.0	*6.18	*2.94	4 TON HP 1000GAL 9 COLL. 1800 FT ² GC COMP OFF 31 HRS
	WASH. DC	86.46	.817	4.04	6.27		37.8	2.68	4.13	3 TON HP 1000GAL 29 COLL. 1800 FT ² GC
4C FEDDERS POOL PANEL	FRESNO	46.24	.837	2.30	5.89		48.8	3.81	3.75	3 TON HP 1000GAL 12 COLL. 1200 FT ² GC
	FT. WORTH	41.41	.827	2.20	5.52		62.0	5.74	3.16	4 TON HP 1000GAL 12 COLL. 1440 FT ² GC
	WASH. DC	86.46	.755	5.09	4.98		37.8	2.45	4.52	3 TON HP 1000GAL 30 COLL. 1800 FT ² GC

*COOLING LOAD NOT MET
COMPRESSOR OFF TIME NOTED

TABLE 4-3: RESULTS SUMMARY OF SYSTEM THERMAL ANALYSIS

Figure 4-11: STORAGE VOLUME SENSITIVITY HEATING MODE

LEGEND

- △ FRESNO FEDDERS, FAFCO IV
- WASHINGTON, D.C., FAFCO IV, FEDDERS
- FRESNO, COMMAND-AIRE, FAFCO IV
- WASHINGTON, D.C., COMMAND-AIRE, FAFCO IV

Figure 4-12: SOLAR HEATING FRACTION VS. COLLECTOR AREA

LEGEND

- △ FRESNO, FEDDERS, FAFCO IV
- WASHINGTON, D.C., FEDDERS, FAFCO IV
- WASHINGTON, D.C., FEDDERS, UNGLAZED

panel appears to perform well with a heat pump in the climates examined, both as collector and rejector.

4. Ground coupling is an essential element, both as a minimum COP backup in heating, and as a heat sink in climates with high cooling loads.

5. It will be difficult to meet the cooling load at all times in hot humid areas such as Fort Worth with a reasonably sized GCSAHP system.

6. Air to air heat pumps are energy efficient machines with broad temperature ranges, and are the closest thermal competitor to the GCSAHP system.

4.6

Recommendations

Future efforts will be directed as follows:

1. Study the same systems in the same climates using TRNSYS, with real weather in much greater thermal detail. The use of average weather in SOLCOST and SAHP COOL is questionable.

2. A better ground coupling model, which accounts for dynamic effects of energy rate changes, is needed to correctly size the ground coupled panels. GROCS will be used.

5.0 MARKETING AND COMMERCIALIZATION

5.1 Existing Products and Energy Sources

The existing residential space conditioning market is composed of many different products, manufacturers and sources of energy. The history of the market has been marked by slow adoption of new technologies as consumers react to the changing availability and price of energy sources, and to preferences for higher comfort levels in living spaces.

The sources of energy for space heating have shifted in the past 30 years toward greater use of natural gas and electricity (Figure 5-1). In new homes, gas and electricity are the dominant fuels (Figure 5-2). The relative popularity of one fuel over another fluctuates with changes in the market for fuels. As is seen, the relative use of natural gas has declined in the 1970's, probably due to the widespread belief that gas was in short supply. Also, oil has declined as a result of large price increases. Electricity use has expanded in the expectation of abundant nuclear generating capacity, and the adoption of less costly methods of heating with electricity.

The equipment for space heating is relatively simple and of low cost. For oil and gas a furnace is used to directly burn the fuel. Electrical space heating can be of two types: either resistance heating or mechanical compression of a refrigerant gas. Heat pumps and compression furnaces operate in the latter mode and have been gaining greater market acceptance as the price of electricity has escalated.

Heat pumps have an added benefit of being able to provide cooling in the summer months. The residential space cooling market has grown for many years as home owners desired a higher level of comfort in their homes. Growth has been lower in recent years as many regional areas have reached saturation levels for cooling (Figure 5-3). Now the market, which is dominated by a few large companies, is more dependent on the growth of new housing in sunbelt areas. The increasing reliability of compressors and favorable electricity

Figure 5-1

Type of Heating in Occupied Housing Units

From 1950 to 1970 the percentage of occupied housing units heated by natural gas more than doubled, increasing from 26.6 percent to 55.2 percent. Electricity has also expanded rapidly as a heating source, increasing from 2.2 percent of total housing units surveyed in 1960 to 14.8 percent in 1977. Coal, on the other hand,

has declined as a major source of heat, from 34.6 percent in 1950 to a negligible 0.6 percent in 1977. Homes heated by fuel oil accounted for 21.3 percent of the total in 1977, only slightly changed from 22.6 percent in 1950.

Source: Energy Information Administration
Annual Report to Congress, 1979.

Figure 5-2: PERCENTAGE OF NEW HOMES IN U.S. USING DIFFERENT TYPES
OF HEATING FUELS

Source: Bureau Of The Census

Number of
Units (000's)

Figure 5-3: RESIDENTIAL UNITARY AIR-CONDITIONING SHIPMENTS
NOT INCLUDING HEAT PUMPS

Source: ARI

Figure 5-4: RESIDENTIAL HEAT PUMP SHIPMENTS

Number of Units
(000's)

Source: ARI

rates have led to a greater penetration of heat pumps for both heating and cooling. Shipments have been growing at a rapid rate since the mid 1970's (Figure 5-4). Note the growing importance of water source heat pumps which offer greater efficiency than the air to air type.

5.2 Market Segmentation

The space conditioning market can be broken into several segments so that major market forces can be traced. Three of the important segments are:

5.2.1 Geography and Climate

Regional variations in weather creates differential demand for space conditioning products. Cold climates are more sensitive to the need for heating capability than are the warmer areas which respond more to cooling needs. Climate differences affect equipment sizing and mechanical design, as well as influencing life styles and house designs.

5.2.2 Type of Residence

Two distinct segments are apparent: 1) new construction and 2) retrofit to existing housing units. Each segment presents opportunities for space conditioning equipment. The retrofit market is highly decentralized with sales directed at the individual homeowner. Motivation for retrofitting space conditioning equipment can be 1) replacement of worn out equipment, 2) adding additional capability such as air conditioning, 3) achieving lower operating costs, or 4) experimenting with new technologies. The potential for retrofit equipment is large with over 80 million housing units in the U.S. Retrofits, however, require a great deal of field labor and infinite set of site variations and unique customers.

New construction represents a smaller market at around 1 to 2 million units per year, but a more predictable set of circum-

stances. Selling can be directed at the builder/developer and can concentrate on specific regions with high housing growth. Equipment can be designed for a specific region on a mass production basis. A major uncertainty in the new housing market results from the cyclical nature of the industry and its strong dependence on favorable government monetary policy. The new construction market can be further defined by the size of house, the degree to which the units are custom vs. tract, and the price of the house.

5.2.3 Energy Prices and Availability

Differentials in price between various energy sources can cause shifts in demand for equipment as consumers try to achieve the lowest cost for space conditioning. In some cases only one energy source, such as electricity, may be available to a consumer, and this will dictate the space conditioning options. Government policy may also exist which limits the choice of energy sources to ones which have a higher value to the society as a whole.

5.3 Emerging Products and Energy Sources

The space conditioning market is currently in a state of transition due to the advent of high energy prices in the 1970's. The availability and price of various energy sources is in an almost constant state of change and uncertainty, making the choice between sources and types of equipment a very difficult issue. The clear trend is toward conservation of all energy sources as their prices increase and as society puts a higher value on all forms of energy.

This era of change and transition is a fertile arena for new concepts in space conditioning systems. These ideas fall into several major groups as listed below:

- 1) More efficient equipment
 - Furnaces - 90% vs. 60% efficient
 - Air conditioners - higher efficiency (EER)
 - Heat pumps - water source, high efficiency

- 2) Conservation
 - Insulation
 - Passive designs
- 3) Renewable Energy Sources
 - Direct solar heating
 - Geothermal heating
 - Wood burning
- 4) Hybrid Systems
 - Solar assisted heat pumps
 - Solar air conditioning

In some respects, all of these technologies are competing for the future space conditioning market. The acceptance and ultimate adoption of any of these systems will depend on a wide variety of market forces.

5.4 FAFCO SAHP Concept

The FAFCO product concept falls into the solar assisted heat pump (SAHP) category. The ultimate SAHP system under development by FAFCO consists of a fully integrated space heating and cooling system which substantially reduces the amount of energy required by current space conditioning technology. A diagram of the system is seen in Figure 1-1, and is described in the engineering discussion. Major features of the FAFCO SAHP system are:

- High efficiency water source heat pump
- Low cost efficient collectors
- Optimum storage volume
- Unique ground coupling design to utilize earth as heat source and sink
- Integrating control system

The expected benefits of this design are:

- Significant cooling savings
- Reduced energy usage for heating and cooling
- No need for auxiliary backup fuel
- Reduced energy expenses which quickly pay for the cost of the system
- Easily installed and maintained
- More efficient use of non-renewable energy resources
- Relatively low initial system cost

Beyond the technical viability of the SAHP concept, the market for such a system will be dependent on the degree to which these benefits are realized. Also, these benefits must be consistent with the needs of the consumers and the macro market.

A potential market for the FAFCO SAHP system has been identified as the residential new construction market. Sunbelt areas have been considered based on the amount of solar energy available and the future growth of the new construction industry in these areas. Also, these areas will achieve maximum benefit from potential cooling savings. The SAHP system will probably have the greatest application in the higher priced home market, at least during the first phases of market penetration. The additional costs of the SAHP system will be a lower percentage of total house value for larger, more expensive homes. New residential construction is expected to be more favorable than retrofit because the house design can be altered to incorporate the SAHP system. This will reduce the installation cost of the SAHP system and help to make it more attractive to a larger market. Also, standard designs are easier to duplicate in high volume, new construction areas.

The annual cash flow of the SAHP system is another factor in considering the new housing market. In a new house, the cost of the system can be included in the principal of the long term mortgage. This results in lower annual payments than for a typical short term home improvement loan available for retrofits. The final

criteria for a potentially attractive market is the existence of current high prices for energy which serve to immediately improve the savings from the SAHP product. If energy prices are already high, then the uncertainty of escalation rates will not be a significant risk.

5.5 Evaluating New Technologies - Economics

There are many potential methods of evaluating the merits of a new product or technology. To determine the marketability of a product the evaluation measures must be meaningful to the potential customers for the product. As will be seen, the SAHP product for new residential construction has at least two direct customers and several indirect customers. Each of these have a unique set of criteria to judge a new product. A few of the criteria are quantitative, but many are qualitative.

Common to most of these customers is a sensitivity to product costs and potential economic benefits to be gained from its use. The generally accepted criteria is that the cost of an energy saving product should be "paid back" by savings accruing from the use of the product within a reasonable period of time. The three interrelated variables - cost, savings and time horizon - can be interpreted differently for the same product by different customers. Typically, a homeowner is most interested in short term benefits. He will be motivated by a low cost system that gives a high level of savings within a very short time. In contrast a government agency is more interested in maximizing the long term benefits to society as a whole. In this case a product which has a high initial cost and which promises to have large benefits over the life of the product will be acceptable. While the time horizon of a government agency can easily be 20 to 30 years, a homeowner will have difficulty justifying an investment of more than 5 years due to the uncertainty of the future.

The long term benefits are usually evaluated in a Life Cycle Costing (LCC) calculation using assumptions for the future in-

flation rates and appropriate discount rates. To calculate the LCC, it is first necessary to calculate annual cash flows resulting from the purchase of the product. These cash flows should simulate all effects that the purchaser will see including financing and tax consequences. These annual cash flows are also required to calculate the short term decision criteria of the homeowner. The purchaser will be interested in seeing how fast his initial cash outlays will start saving him money.

Figure 5-5 shows several different cumulative cash flow profiles for a SAHP investment. The government agency would probably set a time horizon of possibly 20 years and then look favorably on an investment that had a positive cumulative cash flow at the end of that time span. As can be seen, for a SAHP type investment with escalating future fuel costs, if a long enough time period is chosen, then the cumulative cash flow will almost always be positive. The homeowner cannot focus on these long time periods. He is more interested in the first few years where the cash flows tend to be negative. Curves C and D in Figure 5-5 are examples of potentially favorable investments for the homeowner. In C an initial positive cash flow due to a tax credit more than offsets the first few years of negative cash flows so that the buyer is never in a negative cumulative cash position. The buyer will, however, remain sensitive to the duration of his negative annual cash flows. If the turning point comes within 5 years, the buyer will probably think that he has a good deal as long as he anticipates living in the house for more than 5 years. Profile D shows the best possible situation. Here, a buyer starts out with a positive tax credit and then the savings resulting from the product is so large that it leaves a positive cash flow from the first year even after all annual costs are accounted for. This then is the objective for any SAHP system or competing technology; a positive cumulative cash flow which grows larger with each year. This is the most meaningful measure of potential market acceptance.

In calculating the annual cash flows, the initial costs are translated into annual costs as a buyer would through debt

Figure 5-5

CUMULATIVE CASH FLOW PROFILES

financing. The initial cost is influenced by several factors. The mechanical design of the system should be optimized to give the least cost for the most benefit assuming high volume production cost. Installation labor costs can be significant especially if the product is complex with many components. A very important impact on initial cost is government sponsored tax credits and subsidies. The current Federal solar tax credit of 40% effectively reduces the system cost by that amount. Further reductions are possible by considering State tax credits and possible utility company subsidies. The tax credit status of SAHP systems is somewhat uncertain at the present time, however. In most cases the solar portion of the system is eligible for the credits and this can be significant since SAHP system costs are dominated by the solar subsystem. The heat pump part of a SAHP system is potentially eligible for tax credits if regulatory agencies are convinced that the SAHP benefits are significant enough to encourage their use.

The other element of the annual cash flows is the amount of savings expected from the SAHP system. The savings of energy must be compared to some reference system which would be an existing space heating system or an alternative future system. The savings result from the actual energy saved and the unit cost of that energy. The expected rate of energy price escalation can be a highly sensitive assumption, with the rate of escalation usually determining the timing of positive cash flows. Financing terms can also be important. Short loan periods and high interest rates will require higher savings in the first years to offset the high loan payments. For a SAHP system, savings are possible in both the heating and cooling modes. It is important to design a system which gives both so that the same capital cost can be utilized more fully throughout the year.

5.6 Customers and Decision Criteria

In any market there will be one or two prime customer groups and other additional customer groups which may influence a product selection process. The direct customers are those people

who actually purchase and use the product, while the secondary customers have some interest in the product and can create barriers against the use or attractiveness of the product. In order to market the product successfully, the product should offer benefits to all the customer groups.

Figure 5.6 lists all the potential customer groups for the FAFCO SAHP product in the new residential market. Under each group are the major criteria by which the customer will evaluate a new space conditioning product. The prime customers are the homeowner and the builder/developer. The builder will actually select and buy a particular system, and he will be heavily influenced in his decision by the needs and desire of his customer, the potential homeowner. These two groups tend to be primarily motivated by short term economic considerations and the reliable performance of the product.

The secondary customers are a variety of institutional groups such as utility companies and government agencies. For the purposes of this study, the experiences typical in California have been detailed. For other regions, the list of secondary customers and their criteria will differ. This variation in secondary customers can be a problem in itself and points toward regional marketing strategies directed at these groups. The major marketing consideration of the secondary customers is to avoid having them establish barriers to the new technology. For a SAHP system these barriers could be in the form of policies which discourage the use of heat pumps to alleviate summer electrical peaks.

5.7 FAFCO SAHP System Simulations

A preliminary evaluation of several SAHP systems has been completed using F-Chart 4.0 and other relatively simple computer simulation techniques. These evaluations have been centered on establishing the economic benefits of various SAHP configurations. These results are preliminary in that the mechanical design of the systems has not been optimized, and the ability of the models to accurately simulate

Figure 5-6

CALIFORNIA CUSTOMER SEGMENTS

Acceptance Criteria

PRIMARY CUSTOMERS

Homeowner-

- High comfort level
- Positive annual cash flow

Builder/Developer

- Able to sell house faster
- More profit potential
- Reliable operation

SECONDARY CUSTOMERS

Mechanical Contractor

- Ease of installation
- High mark-up potential

Public Utility Company

- Comply with PUC regulations
- Reduce peak load growth

Municipal Utility

- Minimize energy costs
- Comply with federal/state regulations

State Public Utility Commission

- Conservation of imported fuels
- Long term benefits to ratepayers
- Minimize utility bills in long run
- Reduce peak electric loads

State Energy Commission

- Conserve all fossil fuel energy sources
- Protect environmental quality
- Decentralize energy production

Other Entities

- Local building officials
- Unions
- Lending institutions

yearly operation is limited. A series of simulations were done for various California cities and two locations outside California. The primary measure of marketability is the short term cash flow characteristics of the investment in a SAHP system. In general the investment is probably acceptable if the cumulative cash flow turns positive in less than 5 years. Also, this will usually lead to a positive LCC using a 20 year evaluation period.

The results of these simulations are presented below where the leading systems are listed.

SAHP Simulation Results

<u>System</u>	<u>Years to Positive Cash Flow</u>
1. SAHP with swimming pool as source/sink	0
2. Dual source SAHP with \$170-\$340 of cooling savings per year	3 - 5
3. SAHP with a geothermal well and \$170-\$340 of cooling savings	3 - 5
4. SAHP with plate ground coupling, advanced heat pump, cooling savings and high utility costs.	1 - 2

In doing the simulations several variables were changed to see what sensitivities may exist. In general there were several factors that are common to the best systems. These include:

- Relatively high utility costs
- Annual cooling savings of at least \$100 per year
- Tax credits of about 40% of system cost
- Gas heat reference system
- Advanced high efficiency heat pump
- Elimination of backup fuel with a ground source

Solar direct heating systems were also evaluated and found to be very poor investments due to their very high initial cost and lack of cooling savings. The economic assumptions used in the sim-

ulations are shown in Figure 5-7. Also, installed capital costs for a variety of systems were calculated and used in the analysis. Examples of these costs are seen in Figure 5-8. A sample of a detailed cost estimate is shown in Figure 5-9.

Further simulations were performed assuming that utility costs in California were decontrolled so that natural gas prices rose to the world price. Also, electrical costs were increased to reflect higher generation costs. In this future scenerio it was assumed that there would be an incentive electricity pricing structure in the summer months to encourage the use of off peak power for cooling. Calculating cooling savings with these assumptions resulted in a savings of about \$500. This savings and a higher heating savings due to the high gas prices brought most of the SAHP systems to a first year positive cash flow level, in areas with significant cooling loads.

In conclusion, the analysis points toward the probable market acceptance of SAHP systems where energy prices are high and favorable tax credits exist. This is especially true where the SAHP system has been optimized to achieve high cooling savings and a low installed cost. These conclusions apply to a system when evaluated against a gas heating system. Where an air to air heat pump is used as a reference, the SAHP system may not be as favorable depending on the cost of electricity.

5.8 SAHP Source Energy Requirements

Another important measure of marketability for the institutional customers is the amount of source energy required by a SAHP system vs. other technologies. The goal is to reduce source energy use to benefit society as a whole. Figure 5-10 shows a comparison of source energy requirements for various heating systems. Direct solar systems have very good source energy requirements and serve as a goal to be reached. Unfortunately, as previously stated these systems tend to not be economical. The combined SAHP system has the potential to be more efficient than gas and even direct solar. This level of source energy use appears to be sufficient to satisfy the conservation mandates of local and national policies.

Figure 5-7
ECONOMIC ASSUMPTIONS
FOR SAHP SYSTEM EVALUATIONS

1. Tax Credits:

Combined California and Federal credits of
40% of system cost

2. Financing:

40% down payment (equivalent to total tax
credit)
25 year term at 11% interest

3. Discount Rate:

9%

4. Insurance and Maintenance Costs:

0.5% of original investment (assumes warranty
liability is honored by manufacturers)
Escalate at 9% per year

5. Property Tax Rate:

1.5% of original investment
Escalate at 7% per year
(Note: As of November 1980, California law
exempts solar systems from property tax)

6. Energy Costs: (Varies by location & utility company)

	1980 <u>CURRENT COSTS</u>	<u>ESCALATION RATE</u>	EXPECTED DECONTROLLED <u>COST</u>
GAS	30-45 ¢/THERM	15% per year	60 ¢/THERM
ELECTRICITY	3-6 ¢/KWH	10% per year	7 ¢/KWH

Figure 5-8
TYPICAL INSTALLED CAPITAL COSTS
FOR SAHP SYSTEMS IN FRESNO, CALIFORNIA
 (BEFORE TAX CREDITS)

<u>SYSTEM TYPE</u>	<u>COST</u>
1. Gas Heat/Air Conditioning	\$ 2,900.00
2. Air to Air Heat Pump	\$ 2,075.00
3. Solar direct heating, 50% solar fraction, electric air conditioning	\$ 10,100.00
4. Same as 3 but with 75% solar fraction	\$ 16,125.00
5. SAHP with glazed polymer collectors Ground Coupling	\$ 10,859.00
6. SAHP same as 5 but with more efficient heat pump	\$ 11,000.00
7. SAHP same as 6 but with unglazed polymer collector	\$ 10,175.00
8. SAHP using a swimming pool for source/sink, glazed polymer collectors	\$ 8,735.00
9. Dual source SAHP with glazed polymer collectors	\$ 13,883.00
10. SAHP with geothermal well and glazed polymer collectors	\$ 13,500.00

(NOTE: DUCTING COSTS NOT INCLUDED; COMMON TO ALL SYSTEMS)

Figure 5-9

COST ESTIMATE WORKSHEET EXAMPLE

SYSTEM: Dual Source SAHP with glazed polymer collector

SITE: Fresno, California

EQUIPMENT SCHEDULE / COST ESTIMATING					
1	COLLECTOR - FAFCO IV SIZE <u>4x10</u> AREA <u>400</u> SQ.FT. QTY. <u>10</u>				3410.-
					650.-
2	STORAGE TANK - NORTHCOAST MODEL <u>180°F</u> GAL <u>1000</u> WT. <u>410</u> LABOR				1226.-
					500.-
3	WATER/FREON HEAT EXCHANGER MODEL <u>X30</u> GPM GEO-SYSTEMS LABOR				413.-
					60.-
4	WATER CONTROL VALVE MODEL <u>DELTA-T</u> FLOW DEVERTER LABOR				150.-
					50.-
5	SOLAR WATER PUMP TACO 1/3-HP MODEL <u>TA117</u> A28 GPM <u>15</u> HEAD <u> </u> FT. LABOR				152.-
					125.-
6	SYSTEM WATER PUMP 1/6-HP MODEL <u>STARITE</u> GPM <u>9</u> HEAD <u> </u> FT. LABOR				160.-
					75.-
7	SYSTEM SOLID STATE CONTROLLER MODEL <u>C-100-1S</u> INDEP ENERGY LABOR				270.-
					125.-
SOLAR TOTAL					
8	HEAT PUMP FAN/COIL - CARRIER 40FS/28TQ MODEL <u>CARRIER</u> LABOR				562.-
					80.-
9	HEAT PUMP - CARRIER 38TQ MODEL <u>38HQ</u> SIZE <u>3-TON</u> LABOR				1680.-
					250.-
10	HOT WATER/HEAT RECLAIM - CARRIER 38HQ MODEL <u>CARRIER</u> LABOR				500.-
					175.-
11	AIR COOLED CONDENSER - CARRIER 38HQ MODEL <u>38HQ 940</u> 3-TON LABOR				627.-
					250.-
12	HOT WATER COIL/CABIENT MODEL <u>MAGIC-AIR</u> SIZE <u>32x22</u> GPM <u>9</u> LABOR				258.-
					80.-
13	DOMESTIC HOT WATER HEATER SIZE <u> </u> GAL <u> </u> NEW <u>X</u> EXISTING <u> </u>				
EQUIPMENT TOTAL					
1.	PIPING				1100.-
2.	ELECTRICAL				500.-
3.	SUB CONTRACTOR				200.-
4.	SITE PREPARATION				200.-
5.	PERMITS/FEES				85.-
6.					
					113,883.-
TOTAL \$					

Figure 5-10

SOURCE ENERGY REQUIREMENTS

	Seasonal Gas Efficiency (%) Or Heat Pump Performance Factor	% Direct Solar Fraction	Source Energy Units Required Per Unit Of Load
1. Natural Gas Furnace			
A. Now	70%	-	1.42
B. Future	95%	-	1.05
2. Air/Air Heat Pump			
A. Mild Climate	2.7	-	1.11
B. Cold Climate	1.7	-	1.76
3. Traditional Solar Systems			
A. Electric Resistance Back up	3.57 Equivalent	75	0.84
B. Gas Back up	70%	60	0.64
4. Water Source Heat Pump On a Pond	3.3	-	0.91
5. Solar Assisted Heat Pump Systems			
A. Ground Source Heat Pump Back Up-Present Machines	3.0	50	0.56±
B. Advanced Solar Assisted Heat Pump 90° Evaporator Temperature	5.5 40% time 2.5 10% time	50	0.40±

5.9 Conclusion

The preliminary analysis of the FAFCO SAHP system compared to other heating/cooling systems shows that the SAHP system may be a beneficial new technology. In some geographical areas the SAHP system should be able to achieve critical economic acceptance tests as well as conservation goals. Figure 5-11 shows a comparison of several space conditioning options and the degree to which they satisfy certain market acceptance criteria. While this is a somewhat subjective comparison, it points toward potential usefulness for a SAHP product. Further work will be done in the remaining portion of the contract to more precisely identify the potential energy savings benefits of various SAHP configurations, and the regulatory energy pricing structures which may act as barriers to SAHP acceptance.

Figure 5-11

MARKET ACCEPTANCE ISSUES

TECHNOLOGY	POSITIVE CASH FLOW	LOW INITIAL COST	CONSERVE SOURCE ENERGY	REDUCE PEAK POWER	COMMENTS
INSULATION, PASSIVE	RESTRAINT	RESTRAINT	PARTIAL	PARTIAL	
WOOD BURNING	-	GOOD	GOOD	-	INCONVENIENT ENVIRONMENTAL PROBLEM
A/A HEAT PUMP	EXCELLENT	GOOD	POOR	MAKES WORSE	
DIRECT SOLAR	FAILS	FAILS	GOOD	NO HELP	
FAFCO SAHP W/G.C. & OFF-PEAK COOLING	ACCEPTABLE	RESTRAINT	EXCELLENT (HEAT & COOL)	EXCELLENT (HEAT & COOL)	

APPLICATIONS: SUNBELT GEOGRAPHY
 ABOVE AVERAGE HOUSE SIZE
 INNOVATIVE ENERGY/POLITICAL CLIMATE
 REFERENCE - GAS FURNACE, ELECTRICAL A.C.
 SUMMER PEAKING UTILITY

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